1

Fundamentals

1.1 Introduction

This textbook deals with the study of different vehicle thermal systems and components from an energy engineering point of view. It is therefore necessary to recall the fundamentals of heat transfer as well as thermodynamics and some elements of fluid mechanics for a good understanding of the content of the next Chapters 2, 3, and 4. This is the objective of the present chapter, the content of which has been largely summarized from major reference textbooks, especially those of Incropera and DeWitt (2002), Çengel and Boles (2006), Braun and Mitchell (2012), and Klein and Nellis (2016).

1.2 Fundamental Definitions in Thermodynamics

Thermodynamics is the branch of physics that studies conversions between heat and work in one or the other direction. Thermodynamics is particularly useful for the analysis of components and systems presented in this book.

Thermodynamics makes use of some important notions to which the reader should become familiar.

1.2.1 System, Surroundings, and Universe

In thermodynamics, a *system* is defined as a delimited region of space or a quantity of matter that is investigated. The concept of “investigation” may still be a little bit fuzzy and will progressively develop. Let’s say that investigating a system means quantifying its energy performance and the relation between this performance and operating conditions. The system is delimited by a *boundary* (Figure 1.1). A boundary has neither mass nor thickness. The *surroundings* of the system are the region of space or the quantity of matter that is outside the system. Hence, the boundary is the surface that separates the system from its surroundings. The system and its surroundings constitute the *universe*.

Among the systems, one can distinguish the *closed systems* and the *open systems*. A closed system does not exchange any mass with its surroundings. Consequently, its mass is constant. An open system, also called *control volume* (CV), exchanges mass with its surroundings. Such a system is represented in Figure 1.2. The system consists of the region in space delimited partially by plain lines and partially by two dashed lines. Some fluid can enter or leave the system through physical connections to the surroundings. Consequently, the mass of the system may vary.

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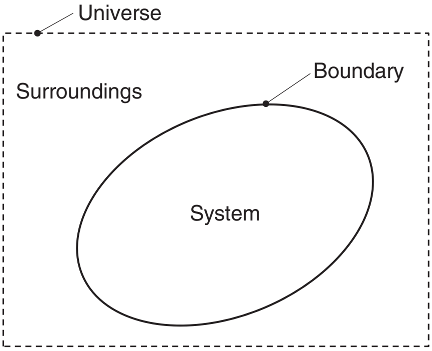

Figure 1.1 System, boundary, surroundings, and universe.

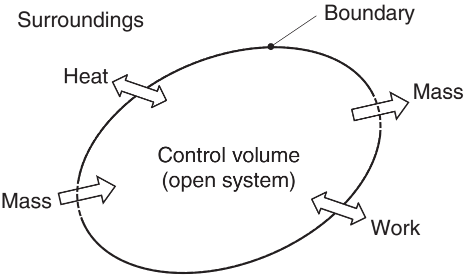

Figure 1.2 Mechanisms of energy transfer between an open system and its surroundings.

In Figure 1.2, part of the boundary is real. It is represented by solid lines and can correspond to the physical envelope of the system. Contrarily, dashed lines represent imaginary boundaries. They are the openings of the system that allow for mass exchange with the surroundings.

As depicted in Figure 1.2, an open system can exchange energy with its surroundings through three mechanisms: heat transfer, work transfer, and mass transfer. It will be shown later that the energy transfer associated with the mass transfer is computed based on the *enthalpy* of the flow.

In the specific case where the open system does not exchange mass with its surroundings, it becomes a closed system. If the system does not exchange heat with its surroundings, it is said to be *adiabatic*. If the system exchanges neither heat nor work nor mass, it is said to be *isolated*. It will be shown later that engineering applications of thermodynamics are particularly interested by (useful) energy transfers between a system and its surroundings.

A very common example of an open system, largely described in textbooks, is the cylinder-piston assembly equipped with valves represented in Figure 1.3. Since one desires to describe the state of the fluid inside the cylinder, the content of the cylinder is defined as the thermodynamic system. The dashed line represents the boundary of the system. It consists of the cylinder wall, cylinder

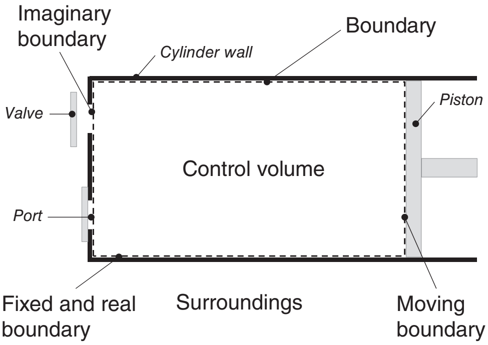

Figure 1.3 Example of an open system with a moving boundary.

head (comprising the ports), and the inner surface of the piston. The latter portion of the boundary is moving. Some fluid can enter or leave the control volume through the ports, provided they are not covered by the valves.

In this book, many open and closed systems will be investigated. Among the major open systems, one can mention heat exchangers, compressors, turbines, and pumps. All these components have inlet and outlet ports that allow for mass exchange with their surroundings. The vehicle cabin can also be considered as an open system, since it exchanges air with the vehicle outdoor through the ventilation system. When the ventilation is in the recirculation mode (which will be described in Chapter 3) and if infiltrations and exfiltrations of air are neglected, the vehicle cabin can be seen as a closed system. Another example of a closed system could be the whole engine coolant loop. During regular operation, the mass of coolant in the loop is constant.

An appropriate choice of the boundary will simplify the thermodynamic description of the system under investigation.

1.2.2 Properties

When investigating vehicle thermal management, one has to describe numerous thermodynamic systems.

A system can be described by its characteristics, which are named as thermodynamic *properties*. Describing a system means describing its thermodynamic *state*. The state of a system is defined when the latter is in equilibrium. The most known properties are pressure P [Pa], temperature T [K], mass m [kg], and volume V [m^3]. These properties are *internal* properties. Speed C [m s^{-1}] and elevation z [m] are *external* properties and do not depend on the molecular structure of the matter (Klein and Nellis, 2016). Note that these properties are measurable properties. It will be shown later that other properties that cannot be directly measured are also very useful for the description of a system, such as internal energy U [J], enthalpy H [J], or entropy S [J K^{-1}]. Such nonmeasurable properties can be calculated based on measurable ones and thermodynamic relations.

A *specific property* is a property expressed per unit of mass of the system. Specific properties are usually denoted with lowercase letters. For instance, specific volume v [m^3kg^{-1}] is volume V [m^3] divided by mass m [kg]. Other properties that will be used in this book are specific internal energy u [J kg^{-1}], specific entropy s [$\text{J kg}^{-1}\text{K}^{-1}$], specific enthalpy h [J kg^{-1}], and specific heat at constant pressure c_p [$\text{J kg}^{-1}\text{K}^{-1}$].

Among the properties, the distinction can also be done between the *extensive* and *intensive* properties. The extensive properties depend on the size of the system and vary linearly with its mass. Examples are mass m [kg], volume V [m^3], and internal energy U [J]. On the contrary, the intensive properties do not depend on the system size and mass. Examples are pressure P [Pa] and temperature T [°C] and also any specific properties.

As it will be explained later, to specify the state of a system in internal equilibrium (and in the absence of electrical, magnetic, or other effects), two independent intensive properties are needed.

1.2.3 Process

A *process* is a transformation that brings the system from a given state A to another state B. The initial and final states are described by two independent intensive properties. Therefore, the state of a system is a "picture" of the system and does not depend on its "history" (the way this state was obtained). Indeed, different paths, involving different heat and work transfers, can bring the system from state A to state B. Hence, the properties of the system are called *point functions*.

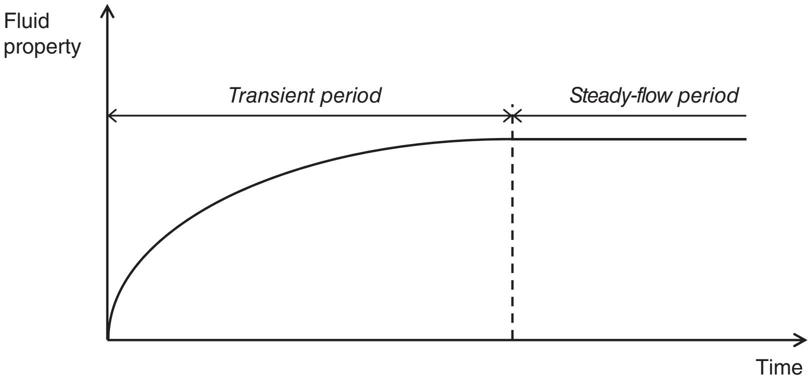

Figure 1.4 Steady-flow period following a transient period.

In contrary, heat and work are called *path functions*. Examples of processes involved in vehicle thermal management are fluid compression, expansion, cooling, and heating.

A process that brings the system back to its initial state is a *cycle*. This type of process is encountered in thermal machines where a fluid is circulating in a cyclic manner. One major example of the thermodynamic cycle is the refrigeration cycle that is exploited by the vehicle air-conditioning loop. It will be described in detail in Chapter 3.

There is one particular operation of open systems that largely simplifies their analysis. This is the steady-state regime. A fluid flows steadily through a control volume if the properties inside the control volume do not vary with time. The fluid properties can vary from one point to another inside the control volume, but at a given location inside the control volume, the fluid properties do not vary with time (Çengel and Boles, 2006). In such a situation, the control volume is said to undergo a *steady-flow* (or steady-state) process. During such a process, the energy and the mass contained in the system are constant in time. Steady-flow processes are approximated in practice when a system is operating for a long period of time with no variation in operating conditions.

For instance, when an internal combustion engine is switched on, it will first undergo a transient period during which the temperature of the metal and internal fluids will increase. After this period, the engine temperature and consequently the temperature of the fluids leaving the engine will stabilize. This is represented in Figure 1.4. When acceptable, the assumption of the steady-flow process simplifies the analysis of the performance of the systems.

1.2.4 Energy

This book will extensively make use of the concept of energy. Actually, it will describe different components and systems of components used in vehicles that transfer energy or convert it from one form to another. Defining the energy is not an easy task. Our everyday experience teaches us that energy can appear under different forms, among which thermal energy, mechanical energy, kinetic energy, potential energy, electric energy, or nuclear energy. By summing all the quantities of energy contained in a system under its different forms, one obtains the total energy E [J] of the system. The specific total energy is defined as $e = E/m$ [J kg^{-1}].

It should also be mentioned that the energy contained in a system depends on the reference state for which this energy is null. This is not of primary importance, since the description of thermal systems mainly implies the quantification of *energy variations*.

Among the different forms of energy, one can also distinguish the *microscopic* forms and the *macroscopic* forms of energy. The microscopic forms of energy are sensible energy (energy associated with the movement of molecules, atoms, and nucleons), latent energy (energy associated with the binding forces between the molecules; these forces decrease from the solid phase to the liquid phase and to the gaseous phase), chemical energy (energy associated with the atomic bonds in a molecule), and nuclear energy (energy associated with the bonds between the nucleons inside the nucleus of the atom). The macroscopic forms of energy of a system are associated with its velocity C [m s^{-1}] and altitude z [m], i.e. its kinetic and potential energies.

The internal energy U [J] of a system is the sum of all microscopic forms of energy. The specific internal energy is defined as $u = U/m$ [J kg^{-1}] . The total and internal energies are related by

$$E = U + m\frac{C^2}{2} + m\,g\,z \tag{1.1}$$

where

C is the system velocity, [m s^{-1}]
g is the gravitational acceleration, [m s^{-2}]
z is the elevation of the system from a reference altitude, [m].

In the right-hand side of the previous equation, the second and third terms are the kinetic and potential energies, respectively.

1.2.5 Heat

Heat Q [J] is the form of energy that is exchanged between a system and its surroundings because of their difference in temperatures. A system does not contain heat, but thermal energy. Heat is the visualization of thermal energy transfer through a system boundary under the action of a temperature gradient. To be rigorous, one should not talk about "heat transfer," but "thermal energy transfer." However, the latter expression is commonly accepted.

The heat transfer rate $\dot{Q}$ [W] is defined as the heat exchanged between a system and its surroundings per unit of time. In the case of an adiabatic system, $\dot{Q} = 0$ [W].

1.2.6 Work

Work is the form of energy that is transferred when a force acts on the system over a distance (Çengel and Boles, 2006). The different forms of work can be categorized as mechanical forms and nonmechanical forms. Different forms of mechanical work can also be distinguished.

1.2.6.1 Mechanical Forms of Work

A mechanical work is exchanged between a system and its surroundings when a force is acting on the system boundary and when this system or its boundary is moving. If these two conditions are met, a work interaction exists between the system and its surroundings. The work is either done *by* the system or done *on* the system. In the former case, the external force acting on the system and its motion have opposite directions. In the latter case, the external force acting on the system and its motion has the same direction.

The mechanical forms of work that will be met in the rest of this book are the moving boundary work, the shaft work, the spring work, and the work necessary to raise or to accelerate a system.

1.2.6.1.1 Moving Boundary Work When a fluid is compressed or expanded, the boundary separating the fluid from its surroundings is moving. As a consequence of the displacement of the boundary, moving boundary work W_b[J] is exchanged between the fluid and its surroundings. Such a work interaction is represented in Figure 1.5.

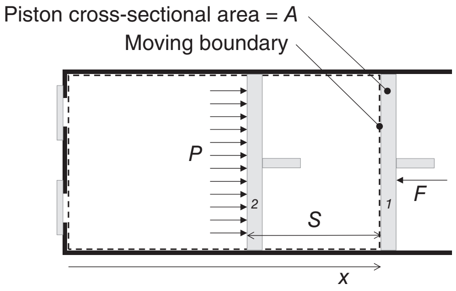

Figure 1.5 Moving boundary work interaction in a piston-cylinder apparatus.

During the compression (related to distance $S = x_1 - x_2$ travelled by the piston), moving boundary work W_b can be computed by integrating the product of the force acting on the piston and incremental displacement dx. That is,

$$W_b = -\int_1^2 F\, dx \tag{1.2}$$

In Eq. (1.2), the force can be related to the pressure acting on the inner surface of the piston, leading to Eq. (1.3). In the case of a *reversible* evolution (1-2), since the pressure is uniform within the cylinder, the pressure in Eq. (1.3) is the system pressure. The notion of reversible evolution will be further developed later.

$$W_b = -\int_1^2 P\, A\, dx = -\int_1^2 P\, dV \tag{1.3}$$

1.2.6.1.2 Shaft Work Shaft work is the work associated with a rotating shaft. This is, for instance, the work transmitted at the shaft of the engine of a car or the work absorbed at the shaft of a compressor (Figure 1.6).

Shaft work W_{sh} [J] exchanged after X revolutions of a shaft on which a torque T [N · m] of moment arm r [m] is acting is given by

$$W_{sh} = \frac{T}{r}\, 2\,\pi\, r\, X = 2\,\pi\, X\, T \tag{1.4}$$

Shaft power $\dot{W}_{sh}$ [W] (shaft work per unit of time) can be calculated based on the rotational speed N [Hz] of the shaft, which is the number of revolutions done per second.

$$\dot{W}_{sh} = 2\,\pi\, N\, T \tag{1.5}$$

1.2.6.1.3 Spring Work According to Hooke's law, the force F [N] needed to compress a linear-elastic spring of spring constant k [N m^{-1}] by a distance X [m] varies linearly with this distance (Figure 1.7).

$$F = kx \tag{1.6}$$

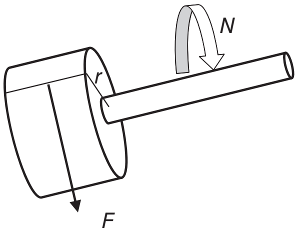

Figure 1.6 Work transmitted at a shaft. Source: Reproduced from Çengel and Boles (2006).

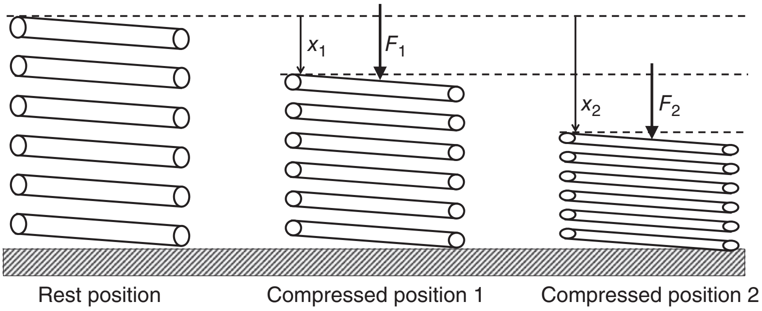

Figure 1.7 Spring work.

Hence, the work needed to compress (or extend) a spring from a position where its displacement is x_1 to a position where its displacement is x_2 is

$$W_{spring} = \frac{1}{2}k\ \left(x_2^2 - x_1^2\right) \tag{1.7}$$

Springs are commonly used in thermostatic actuators met in thermal management systems. Two examples are the engine coolant thermostat (described in Chapter 2) and the thermostatic valve of the air-conditioning loop (studied in Chapter 3).

1.2.6.1.4 Work Necessary to Raise or to Accelerate a System Work must be transferred to a system to accelerate it or to increase its altitude. Once transferred to the system, the kinetic and potential energies are increased, respectively. Similarly, a work is done by a system when it is decelerated or when its altitude is decreased. In that case, the kinetic and potential energies of the system are decreased, respectively.

Hence, work W_a transferred to a system of mass m to increase its velocity from C_1 to C_2 is equal to

$$W_a = \frac{1}{2}m\left(C_2^2 - C_1^2\right) \tag{1.8}$$

Similarly, work W_g transferred to a system of mass m to increase its altitude from z_1 to z_2 is equal to

$$W_g = m\,g\,(z_2 - z_1) \tag{1.9}$$

Such works are illustrated in Figure 1.8, depicting a car of mass m [kg] accelerating from velocities C_1 to C_2 and climbing a hill from altitudes z_1 to z_2.

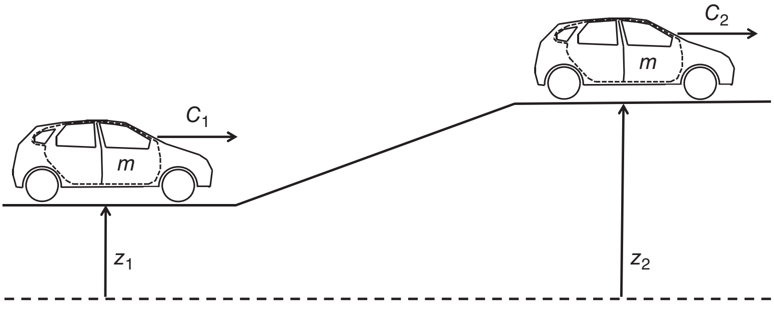

Figure 1.8 Work necessary to raise and accelerate a system.

1.2.6.2 Nonmechanical Forms of Work

Two usual nonmechanical forms of work are magnetic work and electrical work. Transfers of electrical work are common in automotive applications (for instance, positive temperature coefficient [PTC] heaters introduced in Chapter 3). The rate transfer of electrical work is electrical power. It is associated with current I [A] flowing through resistance R [Ω] under the action of a difference of potential V [V].

$$\dot{W}_{el} = V\,I = R\,I^2 \tag{1.10}$$

1.2.7 Enthalpy

When a fluid flows into the control volume, it does not bring only the total energy e [J kg^{-1}] but also an additional energy named *flow energy*. Flow energy, or flow work, is the work necessary to push a fluid element into the control volume. This work is done by the fluid upstream the flow element. Per unit of mass of fluid element, flow energy is equal to $P\,v$ [J kg^{-1}]. Hence, the total energy of a flowing fluid is equal to

$$\theta = e + Pv = u + Pv + ke + pe = h + ke + pe \tag{1.11}$$

As indicated in Eq. (1.11), internal energy u and flow energy Pv have been merged into a variable h [J kg^{-1}], which is named *specific enthalpy*. This variable is very convenient when investigating open systems, since it includes both the internal energy and energy involved when pushing the fluid into or out of the control volume.

1.3 Fluids

Vehicle thermal management systems involve different fluids, the properties of which make them suitable for specific functions.

This book focuses on thermodynamic properties, which are properties that allow describing the thermodynamic state of the fluid. Among fluids, the distinction can be done between pure fluids and mixtures. Some mixtures can be treated as pseudo-pure fluids. We will see hereunder that if the fluid is a pure fluid or pseudo-pure fluid, its thermodynamic state can be easily described.

1.3.1 Pure and Pseudo-Pure Fluids

A pure fluid (also named pure substance or chemical substance) is a substance that has a homogeneous and stable chemical composition. Homogeneous mixtures can be considered as pure fluids. This is, for instance, the case of air, which contains nitrogen and oxygen. Heterogeneous mixtures cannot be considered as pure fluids.

A pure fluid can exist under different phases: solid, liquid, and gas. Each phase has a different molecular structure. The strength of the intermolecular bonds decreases from solid to liquid and from liquid to gas. The molecular spacing increases from solid to liquid and from liquid to gas (phase changes). A mixture of several phases of a pure fluid is still a pure fluid as long as all phases have the same chemical composition (Çengel and Boles, 2006).

1.3.2 Liquid–Vapor Phase Change for a Pure or Pseudo-Pure Fluid

Some phase changes are largely exploited in vehicle thermal management systems. This is the case of liquid–vapor phase changes (exploited in air conditioning (A/C) loops or to explain the phenomenon of mist formation on the windshield) and to a lesser extent solid–liquid phase

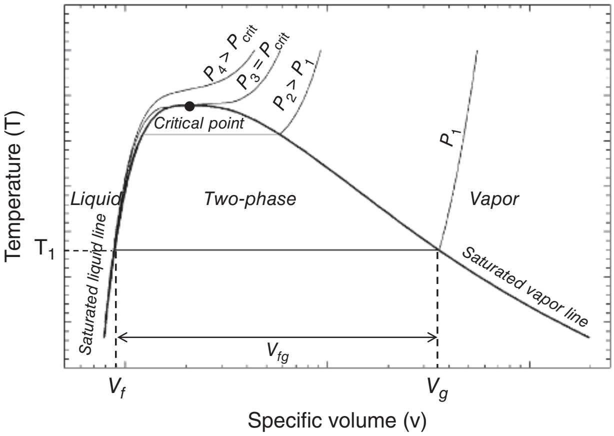

Figure 1.9 Temperature–volume diagram of a pure fluid (different lines of constant pressure are represented).

changes (exploited in wax-type thermostats or in phase change material [PCM] storage). This section emphasizes the liquid–vapor phase change.

A pure fluid can be in different equilibrium states such as liquid, two-phase, and vapor states. The temperature–volume and pressure–enthalpy diagrams of Figures 1.9 and 1.10 are commonly used to represent the state of a fluid that can possibly undergo a phase change. At a given pressure P_1, if the temperature of the fluid is low enough (lower than temperature T_1 in Figures 1.9 and 1.10), the fluid is in *subcooled liquid state*. If the fluid is heated at a constant pressure P_1, its temperature and specific volume increase. When this temperature reaches T_1, the liquid is about to vaporize. In this particular state, the fluid is in *saturated liquid state*. Any slight heat addition to the fluid would result in the vaporization of some liquid. Temperature T_1 is the saturation temperature at pressure P_1. From the saturated liquid state, as heat is provided to the fluid at constant pressure P_1, its temperature remains constant (and equal to T_1), and its specific volume increases. The quantity of liquid decreases, and the quantity of vapor increases. This is the liquid–vapor phase change process. The phase change ends when the last particle of liquid turns into vapor. In this particular state, the fluid is in *saturated vapor state*. Actually, any slight cooling of the fluid would result into

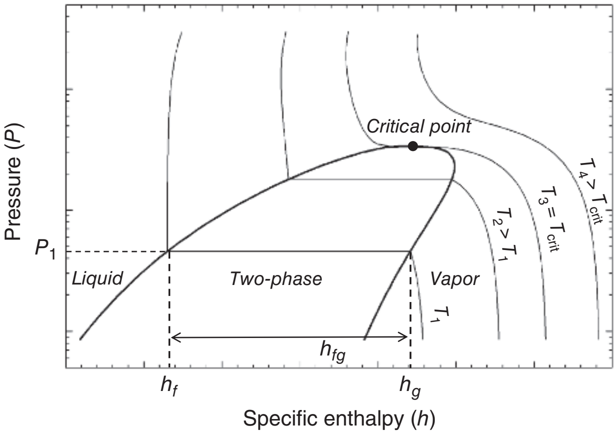

Figure 1.10 Pressure–enthalpy diagram of a pure fluid (different lines of constant temperature are represented).

the condensation of some vapor. During the phase change, the fluid is in *two-phase state*, which corresponds to an equilibrium state between a saturated liquid and a saturated vapor. During the phase change, the mass fractions of saturated liquid and saturated vapor change, but the intensive properties of each phase remain constant. If the saturated vapor is heated at constant pressure P_1, its temperature and specific volume increase. The fluid is in *superheated vapor state*.

The transition from subcooled liquid to superheated vapor at a constant pressure P_1 can be visualized in the temperature–volume diagram given in Figure 1.9. The constant pressure curve associated with pressure P_1 is represented. The specific volume of the saturated liquid is v_f, and the specific volume of the saturated vapor is v_g. The variation of the specific volume during the phase change is $v_{fg} = v_g - v_f$.

The transitions from the subcooled liquid state to the two-phase state and from the two-phase state to the superheat vapor state can be visualized for pressure P_2 higher than P_1 in Figure 1.9. It can be observed that the variation of the specific volume v_{fg} during the phase change is smaller than that for pressure P_1. The same transitions can be drawn for a large range of pressures. The larger the pressure, the shorter is the saturation line. By connecting the saturated liquid states corresponding to the different pressures, the *saturated liquid line* is obtained. Similarly, the *saturated vapor line* is obtained by connecting the saturated vapor states corresponding to the different pressures. The saturated liquid and vapor lines can be visualized in Figure 1.9. The two lines intersect at the *critical point* and form a dome.

The fluid inside the dome is in the two-phase state, which is a mixture of saturated liquid and saturated vapor. The zones on the left-hand and right-hand sides of the dome correspond to the subcooled liquid state and superheated vapor state, respectively.

At the critical point, the variation v_{fg} of specific volume is null. The saturated liquid state and the saturated vapor state are identical. At the critical point, the pressure is equal to the critical pressure and the temperature is equal to the critical temperature. For pressures equal to or larger than the critical pressure, there is no clear transition between the subcooled liquid state and the superheated vapor state. For those pressures, as the temperature increases, at constant pressure, from temperatures lower than the critical temperature to temperatures higher than the critical temperature, the specific volume increases, but there is one single phase all along the process. It is not possible to distinguish a phase change, even if the fluid finally looks like vapor.

Figure 1.10 shows the pressure–enthalpy diagram of a pure fluid. Similar to the temperature–volume diagram, this diagram makes appear a dome that delimits liquid, two-phase, and vapor regions. In this diagram, some constant-temperature curves are overlaid. It can be observed that the curve associated with the critical temperature goes through the critical point. As explained in Chapter 3, this diagram will be used to describe the vapor–compression refrigeration cycles.

To undergo a liquid–vapor phase change, heat must be provided to the fluid. The amount of heat necessary to vaporize 1 kg of a substance at a given pressure is *the latent heat of vaporization* (or *condensation*). Figure 1.10 indicates the latent heat of vaporization h_{fg} [J/kg] at pressure P_1. It corresponds to

$$h_{fg}(P_1) = h_g(P = P_1) - h_f(P = P_1) \tag{1.12}$$

The saturated vapor enthalpy and the saturated liquid enthalpy are given by

$$h_g = h(P = P_1, x = 1) \tag{1.13}$$

$$h_f = h(P = P_1, x = 0) \tag{1.14}$$

The latent heat of vaporization at a temperature of 0°C for 3 fluids typically used in vehicles is provided in Table 1.1.

Table 1.1 Heat of vaporization of three different fluids at 0°C.

	Water	R1234yf	CO_2
$h_{fg}(T=0°\mathrm{C})$ [J/kg]	2,501E+06	163283	230884

The latent heat of vaporization can also be defined for another temperature T_1. The enthalpy of vaporization at 0°C is generally written as $h_{fg,0}$.

Similarly, the latent heat of fusion (or solidification) is the heat necessary to melt 1 kg of a substance at the normal melting temperature or at the triple-point temperature.

1.3.3 Computing the Properties of Pure and Pseudo-Pure Fluids

1.3.3.1 Phase Rule

To specify the state of system in internal equilibrium, a given number of internal intensive properties must be known.

The Phase Rule states that for a nonreacting thermodynamic system, the number F of internal intensive properties required to fix the state of the system is given by

$$F = C - \Pi + 2 \tag{1.15}$$

where

C is the number of distinguishable chemical species, [−]
Π is the number of phases, [−]

For a pure fluid or pseudo-pure fluid, $C = 1$. A pseudo-pure fluid is a mixture characterized by a constant composition, which allows to treat it as a pure fluid (Klein and Nellis, 2016).

1.3.3.1.1 Single-Phase State For a pure or pseudo-pure fluid in the single-phase state, $F = 2$. This means that two internal intensive properties must be specified to fix the thermodynamic state of the system. It is not possible to specify the value of a third variable.

For a pure or pseudo-pure fluid in two-phase state, $F = 1$. This means that only 1 internal intensive property must be specified to fix the intensive state of each phase. For instance, if the pressure P of the system is specified, the temperature of both phases is fixed.

Coming back to the case of pure or pseudo-pure fluid in single-phase state, the number of internal intensive properties needed to fix the system is 2. Hence, any intensive property can be expressed as a function of two other independent properties.

These two properties could be temperature T and volume v or pressure P and volume v. For instance, the specific enthalpy can be computed as

$$h = f_1(T, v) \, or \, h = f_2(P, v) \tag{1.16}$$

For single-phase fluids (f.i. subcooled liquids and superheated vapors), the pair P–T could also be used, since these two properties are independent. However, the pressure and temperature are not independent for saturated liquids, saturated vapors, and mixtures of both. Figure 1.11 shows the evolution of the saturation pressure with the saturation temperature for different working fluids that are typically met in automotive applications. Temperature ranges extend from the triple point to the critical temperature (the latter being 30.98°C, 94.7°C, and 373.9°C for CO_2, R1234yf, and water, respectively).

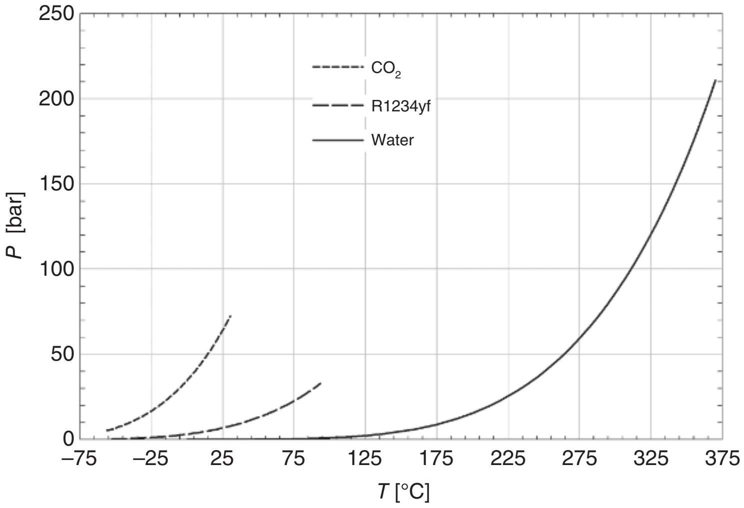

Figure 1.11 Liquid–vapor saturation curves for different fluids commonly used in automotive applications.

1.3.3.1.2 Two-Phase State In the two-phase state or mixed phase, (i.e. a mixture of saturated liquid and saturated vapor), temperature T and volume v can be, for instance, used to describe other properties. Another property that is typically used to describe mixtures of saturated liquid and vapor is the quality. The latter is defined as the ratio of the mass of vapor to the total mass of the mixture.

$$x = \frac{m_g}{m_f + m_g} \tag{1.17}$$

where

m_g is the mass of saturated vapor, [kg]
m_f is the mass of saturated liquid, [kg].

In practice, it is not usual to distinguish the mass of liquid and the mass of vapor inside the mixture. The mixture is considered as a fictitious homogeneous mixture showing average properties. For instance, the average specific volume of the mixture can be defined as

$$v_{avg} = (1 - x)v_f + x\, v_g \tag{1.18}$$

The average specific internal energy and enthalpy of the mixture can be computed in a similar way.

$$u_{avg} = (1 - x)u_f + x\, u_g \tag{1.19}$$

$$h_{avg} = (1 - x)h_f + x\, h_g \tag{1.20}$$

1.3.3.2 The Equations of State Relating *P*, *T*, and *v* (Relation Between Measurable Properties)

Relations between intensive properties have been established for fluids used in engineering applications and are available in the forms of tables, equations, or graphs (Klein and Nellis, 2016).

Among these relations, the equations of state for pressure, temperature, and volume are largely used. They are expressed in the form $P = f(T,v)$. The interest in such relations is that they involve measurable internal intensive variables.

The equation of state of the ideal gas is well known and very convenient to use. When a gas is at low density, it can be accurately described by the ideal gas law given by Eq. (1.21).

$$P\,v = r\,T = \frac{R_u}{MM}T \tag{1.21}$$

where

R_u is the gas constant, [8314 J kmol^{-1} K^{-1}]
MM is the molar mass of the gas, [].

Respecting this equation is actually the condition for a gas to be considered as ideal. Air in cabin and also its water vapor content can be considered as ideal gases.

At high pressures or low temperatures, more accurate equations of state can be used. Among the most famous equations of state, one can mention the Peng–Robinson Equation, which has the following form

$$P = \frac{r\,T}{v-b} - \frac{a}{v(v+b)+b(v-b)} \tag{1.22}$$

Coefficients a and b can be expressed as function of the critical temperature and pressure, reduced temperature, and acentric factor of the fluid. The interested reader is invited to refer to the book of Klein and Nellis (2016) for more information.

Equations of state such as Peng–Robinson can also be used to describe the liquid phase. However, the accuracy of the prediction of the saturated liquid specific volume is limited. Specific correlations are proposed in the literature for the prediction of the saturated liquid specific volume v_f (see Klein and Nellis, 2016).

1.3.3.3 Computing Non-Measurable Properties (*u*, *h*, and *s*) in General Case of Real Pure Fluids

Pressure, temperature, and specific volume are the properties that can be measured. However, first and second laws of thermodynamics involve other properties that cannot be directly measured, such as specific internal energy, specific enthalpy, and specific entropy. The meaning and use of entropy will be presented later.

1.3.3.3.1 Computing Thermodynamic Properties Based on an Equation for P, T, and v These nonmeasurable properties can be calculated provided that an equation of state of the working fluid is known as well as the specific heat capacity at a pressure low enough for the working fluid to behave as an ideal gas (Klein and Nellis, 2016). The latter condition means that the specific heat capacity is a function of only temperature and not of pressure.

The previous equations of state relate the pressure to the specific volume and temperature. If we want to determine other properties of pure fluids, such as specific internal energy u, specific enthalpy h, or specific entropy s, we need complete equations of state. The latter equations are, for instance,

$$\begin{aligned} u &= u(s,v) \\ h &= h(s,P) \\ a &= a(T,v) \\ g &= g(T,v) \end{aligned} \tag{1.23}$$

In the latter equations, a is the specific Helmoltz free energy and g is the specific Gibbs free energy. The meaning of the entropy will be explained in Section 1.6.5.

If one of the complete equations of state is known, all thermodynamic properties can be determined using the fundamental properties relations for pure fluids (Klein and Nellis, 2016). These four fundamental properties equations (Eq. (1.24)) are also called Gibbs equations.

$$
\begin{aligned}
du &= Tds - Pdv \\
dh &= Tds + vdP \\
da &= -sdT - Pdv \\
dg &= -sdT + vdP
\end{aligned}
\tag{1.24}
$$

The Maxwell relations express that the properties u,h,a, and g are state properties, and therefore, the fundamental properties relations du,dh,da, and dg are exact differentials of the complete equations of state. Therefore,

$$
\begin{aligned}
\left(\frac{\partial T}{\partial v}\right)_s &= -\left(\frac{\partial P}{\partial s}\right)_v \\
\left(\frac{\partial T}{\partial P}\right)_s &= \left(\frac{\partial v}{\partial s}\right)_P \\
\left(\frac{\partial s}{\partial v}\right)_T &= \left(\frac{\partial P}{\partial T}\right)_v \\
\left(\frac{\partial s}{\partial P}\right)_T &= -\left(\frac{\partial v}{\partial T}\right)_P
\end{aligned}
\tag{1.25}
$$

Actually, one can note that the fundamental properties relations are the total differentials of the complete equations of state. For instance, the temperature can be obtained by the following partial derivative

$$T = \left(\frac{\partial h}{\partial s}\right)_P \tag{1.26}$$

Using the fundamental property relations and Maxwell's relations, one can express specific internal energy u, specific enthalpy h, and specific entropy s as functions of measurable variables: $h(T,P)$, $u(T,v)$, and $s(T,v)$, respectively. For instance, the differential of $h(T,P)$ is given by

$$dh = c_p\, dT + \left[v - T\left(\frac{\partial v}{\partial T}\right)_P\right] dP \tag{1.27}$$

In this equation, all quantities can be measured. The derivative of the specific volume with respect to the temperature at constant pressure can also be assessed by means of an equation of state, such as Peng–Robinson (Klein and Nellis, 2016).

Enthalpy $h(T,P)$ at a given temperature and pressure can be determined by integrating the previous differential. The methodology to conduct this integration is detailed in Klein and Nellis (2016).

With a similar procedure as that for the specific enthalpy, differentials of u and s involving only measurable quantities can be developed.

$$du = c_v dT + \left[T\left(\frac{\partial P}{\partial T}\right)_v - P\right] dv \tag{1.28}$$

$$ds = \frac{c_v}{T} dT + \left(\frac{\partial P}{\partial T}\right)_v dv = \frac{c_p}{T}\, dT - \left(\frac{\partial v}{\partial T}\right)_P dP \tag{1.29}$$

1.3.3.3.2 Computing Thermodynamic Properties Based on Helmoltz-Energy-Explicit EOS Another way to derive thermodynamic properties is to use the EOS based on the Helmoltz energy a and consider temperature T and density ρ as independent properties. The nondimensionalized Helmoltz energy is obtained by dividing Helmoltz energy a by rT. The nondimensionalized Helmoltz energy can be represented as the sum of an ideal gas contribution (denoted by superscript 0 in Eq. (1.30)) and a residual contribution (denoted by superscript r), leading to the following nondimensionalized Helmoltz energy equation of state:

$$\alpha(\tau,\delta)=\frac{a}{r\,T}=\frac{u-Ts}{r\,T}=\alpha^0(\tau,\delta)+\alpha^r(\tau,\delta) \tag{1.30}$$

with

$$\delta=\frac{\rho}{\rho_{crit}}$$
$$\tau=\frac{T_{crit}}{T} \tag{1.31}$$

The ideal gas reduced Helmoltz energy $\alpha^0(\tau,\delta)$ and the residual reduced Helmoltz energy $\alpha^r(\tau,\delta)$ can be expressed by correlations built from experimental data.

High-accuracy equations of states are based on this formalism of nondimensionalized Helmoltz energy, since all other thermodynamic properties can be obtained by analytical derivatives of the terms $\alpha^0(\tau,\delta)$ and $\alpha^r(\tau,\delta)$. The analytical expressions are derived using the fundamental property relations. More details can be found in Bell et al. (2014), and a detailed example is provided in the thermodynamics textbook of Klein and Nellis (2016). Also, Bell et al. (2014) explain how to compute saturated liquid and vapor densities.

$$\frac{P}{\rho\,r\,T}=1+\delta\left(\frac{\partial\alpha^r}{\partial\delta}\right)_\tau \tag{1.32}$$

$$\frac{h}{r\,T}=\tau\left[\left(\frac{\partial\alpha^0}{\partial\tau}\right)_\delta+\left(\frac{\partial\alpha^r}{\partial\tau}\right)_\delta\right]+\delta\left(\frac{\partial\alpha^r}{\partial\delta}\right)_\tau+1 \tag{1.33}$$

$$\frac{s}{r}=\tau\left[\left(\frac{\partial\alpha^0}{\partial\tau}\right)_\delta+\left(\frac{\partial\alpha^r}{\partial\tau}\right)_\delta\right]-\alpha^0-\alpha^r \tag{1.34}$$

1.3.3.4 Computing Non-measurable Properties (*u*, *h,* and *s*) in the Specific Case of Ideal Fluids

The equations developed in Section 1.3.3.3.1 simplify in the case of ideal behavior.

1.3.3.4.1 Ideal Gas An ideal gas is a gas that obeys the equation of state given in Eq. (1.21).

The differential of $h(T,P)$, $u(T,v)$, and $s(T,P)$ reduces to

$$dh=c_p dT \tag{1.35}$$

$$du=c_v dT \tag{1.36}$$

$$ds=\frac{c_p}{T}\,dT-\frac{r}{P}dP \tag{1.37}$$

The two first equations indicate that the enthalpy and internal energy depend only on the temperature. This is true since, for ideal gases, c_p and c_v are function of only T. This can be demonstrated by expressing the exactness of both expressions of the total differential ds (Eq. (1.29)) developed previously (Chapter 12 of Çengel and Boles (2006)).

Therefore, the variations of specific enthalpy and internal energy between two temperatures T_1 and T_2 are expressed by the relations:

$$\Delta h=h_2-h_1=\int_{T_1}^{T_2} c_p(T)\,dT \tag{1.38}$$

$$\Delta u=u_2-u_1=\int_{T_1}^{T_2} c_v(T)\,dT \tag{1.39}$$

If the specific heat at constant pressure and the specific heat at constant volume are assumed constant (possibly as a first approximation, assuming a linear dependency with respect to temperature, the values calculated at the average temperature between T_1 and T_2 can be considered), the previous relations reduce to

$$\Delta h = h_2 - h_1 = c_p\,(T_2 - T_1) \tag{1.40}$$

$$\Delta u = u_2 - u_1 = c_v\,(T_2 - T_1) \tag{1.41}$$

Regarding the variation of the specific entropic, still considering a constant value of c_p, the integration of the differential ds yields

$$\Delta s = c_p \ln \frac{T_2}{T_1} - r \ln \frac{P_2}{P_1} \tag{1.42}$$

Previous equation is particularly useful to derive relations between pressure and temperature in the case of an isentropic process ($\Delta s = 0$). Assuming that the specific heats are constant (the average values can be considered) and recognizing that $r = c_p - c_v$, it yields

$$\frac{T_{2,\mathrm{s}}\,[\mathrm{K}]}{T_1\,[\mathrm{K}]} = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} \tag{1.43}$$

where $\gamma = c_p/c_v$ is the specific heat ratio.

1.3.3.4.2 Incompressible Fluid

Liquids (but also solids) often behave as incompressible substances, meaning that a very large pressure difference is required to observe a variation of the specific volume. In that case, the specific volume is a function of only temperature ($v = v(T)$). For incompressible substances, the specific heats at constant pressure and volume are equal ($c_v(T) = c_p(T) = c(T)$). This can be demonstrated using the Mayer relationship (Çengel and Boles, 2006). Similar to ideal gases, the dependence of c only on T can be demonstrated by expressing the exactness of the differential ds.

In the ideal case, the specific volume is constant. An average specific volume can also be considered in the range of temperature variations considered. In both cases, the specific volume is constant and assuming a constant specific heat, the differentials of specific enthalpy, specific internal energy, and specific entropy reduce to

$$dh = c\,dT + v\,dP \tag{1.44}$$

$$du = c\,dT \tag{1.45}$$

$$ds = \frac{c}{T}dT \tag{1.46}$$

As for ideal gases, the specific internal energy is a function of only temperature ($u = u(T)$). That is,

$$\Delta u = c\,(T_2 - T_1) \tag{1.47}$$

However, the variation of the enthalpy depends also on the variation of pressure, even though the second term can be often neglected

$$\Delta h = c\,(T_2 - T_1) + v\,(P_2 - P_1) \tag{1.48}$$

Also,

$$\Delta s = c \ln \frac{T_2[K]}{T_1[K]} \tag{1.49}$$

Therefore, in the case of an incompressible liquid undergoing an isentropic evolution, the temperature remains constant. For instance, assuming an isentropic compression from P_1 to P_2 of an ideal liquid in a pump, the temperature would not increase, and the specific work would be equal to

$$w_s = \Delta h_s = v\,(P_2 - P_1) \tag{1.50}$$

1.3.4 Fluids Commonly Used in Automotive Applications

1.3.4.1 Oil

Lubricating oil is mainly composed of hydrocarbons plus some additives. It can be considered as an incompressible liquid with density and specific heat, which are functions of only temperature. The following correlations can be used in the range of temperatures [−20; 200 °C]

$$\rho\,[\mathrm{kg\,m^{-3}}] = 895.75 - 0.65 \cdot T\,[^{\circ}\mathrm{C}] \tag{1.51}$$

$$c[\mathrm{J\,kg^{-3}K^{-1}}] = 1787.33 + 3.60125 \cdot T[^{\circ}\mathrm{C}] \tag{1.52}$$

1.3.4.2 Coolant

As described later in this book, many components inside the vehicle must be cooled down, and liquid water is a convenient fluid. However, it has the drawback to freeze if its temperature is decreased below 0°C. This is likely to happen if the vehicle is at a standstill under negative outdoor temperatures. To decrease the freezing point of the coolant, an antifreeze agent is mixed with water. Antifreeze agents are typically ethylene glycol or propylene glycol.

Figure 1.12 shows the evolution of the freezing point of an aqueous solution of ethylene glycol with the mass concentration of glycol in the solution. For instance, the freezing point of a 50% in mass aqueous solution of ethylene glycol is −36°C.

Adding ethylene glycol in water also increases the boiling point. For instance, the boiling point of a 50% in mass aqueous solution of ethylene glycol is close to 107°C.

However, adding glycol in water decreases the thermal capacity and the thermal conductivity of the coolant and increases its viscosity. This increases pump consumption and decreases heat exchanger performance (lower convective heat transfer coefficient than water).

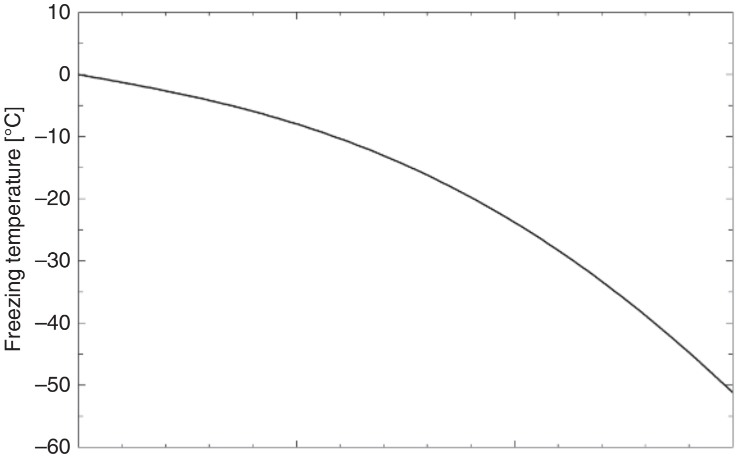

Figure 1.12 Evolution of the freezing temperature of an aqueous solution of ethylene glycol with the mass concentration of glycol.

The following correlations can be used to assess the density and specific heat of an aqueous solution of ethylene glycol 50% in mass as a function of temperature in the range [−20; 100 °C]

$$\rho\,[\mathrm{kg\,m^{-3}}] = 1074.91 - 0.597356 \cdot T\,[^\circ\mathrm{C}] \tag{1.53}$$

$$c[\mathrm{J\,kg^{-3}K^{-1}}] = 3204.02 + 5.72974 \cdot T[^\circ\mathrm{C}] - 0.012805 \cdot T^2[^\circ\mathrm{C}] \tag{1.54}$$

1.3.4.3 Refrigerant

In 1989, the Montreal Protocol banned fluids using chlorine, such as CFC (chlorofluorocarbon) R12 used in automotive applications. R12 was replaced by HFC (hydrofluorocarbon) R134a. Until recently, refrigerant R134a has been largely used in automotive air conditioners. However, this refrigerant shows a large global warming potential (GWP) of 1430. The GWP is a measure of heat trapped in the atmosphere by a given mass of greenhouse gas over a given time horizon relative to the heat trapped by the same mass of carbon dioxide. Recent European regulations have forbidden the use of refrigerants with GWP larger than 150 in all vehicles. Consequently, HFC R134a has been phased out and is replaced by HFO (hydrofluoroolefin) R1234yf in most of the vehicles. This fluid shows a very low GWP of 4. It has been classified as mildly flammable.

Carbondioxide (R744) is another replacement fluid for R134a. This natural fluid has a GWP of 1. Its major thermophysical properties are given in Table 1.2. R744 shows the major advantages of being nontoxic and nonflammable. The use of CO_2 in refrigerators is not new: it has already been used in the early times of artificial refrigeration. However, there is renewed interest for R744 in automotive air-conditioners since the 1990s. R744 is currently used in domestic and industrial vapor compression machines. Because of its low critical temperature, R744 cycles used in A/C loops operate in transcritical regime. In such a regime, the compressor discharge pressure is larger than the critical pressure. This will be discussed more in detail in Chapter 3.

1.3.4.4 Humid Air

Air is involved in many components encountered in automotive thermal management (air-cooled condenser, air-conditioning evaporator, radiator, air circulation inside the cabin, etc.). This air is atmospheric air and contains not only oxygen and nitrogen but also water in vapor state. It is called humid air. The mixture of air and water vapor is a gas–vapor mixture, meaning that the water vapor is close to saturation and may condense during processes. For instance, water condensation may occur in an air-conditioning evaporator when the air temperature is decreased. It may also occur on the glazing surface when the glazing surface temperature is lower than that of the air. The condition for condensation is that the air temperature is decreased down to the dew point temperature, defined hereunder.

To describe the properties of humid air, it is assumed that the air and water behave as two ideal gases. As a consequence, the total pressure is the sum of the partial pressure of air and the partial pressure of water vapor. The latter is also called the *vapor pressure*. That is,

$$P = P_a + P_w \tag{1.55}$$

Table 1.2 Major characteristics of mobile A/C refrigerants.

	HFC-134a	**HFO-1234yf**	**R744**
Molecule	$C_2F_4H_2$	$C_3F_4H_2$	CO_2
Critical temperature [°C]	101	94.7	30.98
Critical pressure [bar]	40.59	33.82	73.77
Molar mass [$\mathrm{g\,mol^{-1}}$]	102	114	44.01
100-year GWP	1430	4	1

Note that the total pressure is the atmospheric pressure if the humid air is at the atmospheric pressure. It could be higher if humid air is compressed.

Being ideal gases, both the air and water vapor can be described by the equation of state (1.21), paying attention that quantity r is not the same for air and water.

In engineering problems involving humid air, it is useful to quantify the content of water in the mixture as well as to express energy and mass balances across components. The quantities introduced in the following sections will be of primary importance.

1.3.4.4.1 Specific Humidity *Specific humidity* ω (also called *absolute humidity* or *humidity ratio*) is the mass of water vapor m_w [kg] per kilogram of dry air m_a [kg]. It can be expressed as a function of the vapor pressure P_w and total pressure P by

$$\omega = \frac{m_w}{m_a} = \frac{0.622\,P_w}{P - P_w} \tag{1.56}$$

Humidifying the air increases its specific humidity, while drying the air decreases its specific humidity.

1.3.4.4.2 Relative Humidity Dry air contains no moisture. Hence, its specific humidity is equal to 0. As long as the air absorbs water vapor, its specific humidity increases. Simultaneously, its vapor pressure P_w increases until it reaches the saturation pressure of water $P_g(T)$ corresponding to atmospheric air temperature T. When this pressure is reached, the air cannot absorb any additional moisture, and it is said to be saturated. Any additional water vapor injection in the air would result in condensation. If m_g is the maximum mass of water vapor that the atmospheric air at temperature T can hold, the relative humidity is defined as the ratio of the actual mass of water vapor m_w to m_g.

$$RH = \frac{m_w}{m_g} = \frac{P_w}{P_g(T)} \tag{1.57}$$

Hence, the relative humidity of dry air is equal to 0, and the relative humidity of saturated air is equal to 1. It is important to observe that the relative humidity depends on the air temperature. Increasing the air temperature decreases its relative humidity, while its specific humidity is constant. Decreasing the air temperature increases its relative humidity, while its specific humidity is constant (as long as no water condensation occurs). This observation is of primary importance when investigating windshield misting and demisting mechanisms.

It should also be noted that the thermal comfort is affected by the relative humidity. The latter can be controlled by adjusting the air temperature or by adding or removing moisture in the air.

By combining previous equations, the specific humidity can be related to the relative humidity by

$$\omega = \frac{0.622\,RH\,P_g(T)}{P - RH\,P_g(T)} \tag{1.58}$$

The relative humidity can be directly measured by relative humidity sensors. This measurement must be combined with the measurement of the temperature and total pressure to determine the specific humidity.

1.3.4.4.3 Dew-Point Temperature As explained previously, when atmospheric air is progressively cooled down at a constant pressure, its relative humidity increases. The temperature at which air becomes saturated is the dew-point temperature. The dew-point can be directly measured by means of a dew-point meter. This delicate measurement consists of observing the apparition of condensates on the surface of a mirror that is progressively cooled down.

At the dew-point temperature, observing that RH = 1, Eq. (1.58) becomes

$$\omega = \frac{0.622\ P_g(T_{dp})}{P - P_g(T_{dp})} \tag{1.59}$$

Equation (1.59) indicates that only the measurement of the total pressure P must be associated with the measurement of the dew-point temperature to determine the specific humidity of the air. The dew-point measurement is hence a highly accurate measurement technique of the moisture content of the air.

1.3.4.4.4 Wet-Bulb Temperature The wet-bulb temperature is another indicator of the moisture content of the air. The wet-bulb temperature is the temperature read by a thermometer the bulb of which is covered by a wetted wick. The wet-bulb thermometer is often part of the psychrometer, which consists of two thermometers whose bulbs are in contact with the same air stream. Such an apparatus is illustrated in Figure 1.13. The first thermometer measures the dry-bulb temperature, which is the atmospheric air temperature. The second thermometer measures the wet-bulb temperature. The bulb of such a thermometer is covered by a wick wetted with water. As the air flows through the wick, water from the wick vaporizes, leading to saturated air leaving the wick (if original air is unsaturated). The heat of vaporization of water is taken from the water, yielding a decrease in water temperature inside the wick. Ultimately, because of the temperature gradient, the air temperature decreases down to the wet-bulb temperature, which is read by the thermometer. The larger the difference between the dry-bulb and the wet-bulb temperature, the drier the air is. If the air were saturated, both thermometers would read the same temperature. In that specific case, the dry-bulb temperature, the wet-bulb temperature, and the dew-point temperature are equal.

This measurement is close to the adiabatic saturation temperature (or thermodynamic wet-bulb). The latter is the temperature of air that would be obtained if the air was brought to saturation by vaporizing water at a temperature equal to that of the air after saturation.

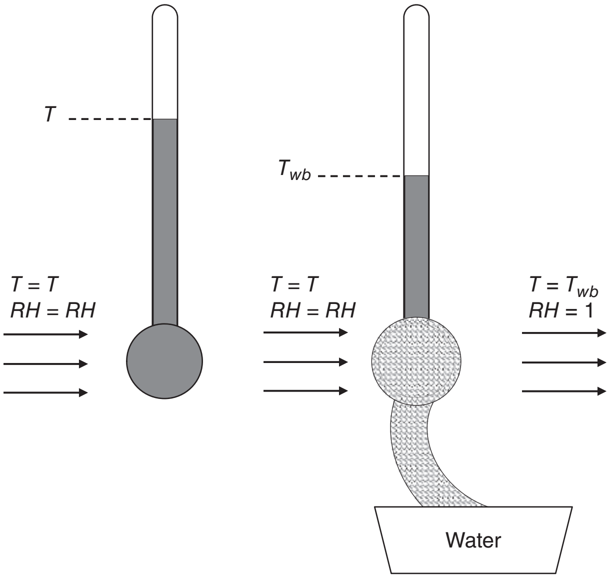

Figure 1.13 Working principle of a psychrometer.

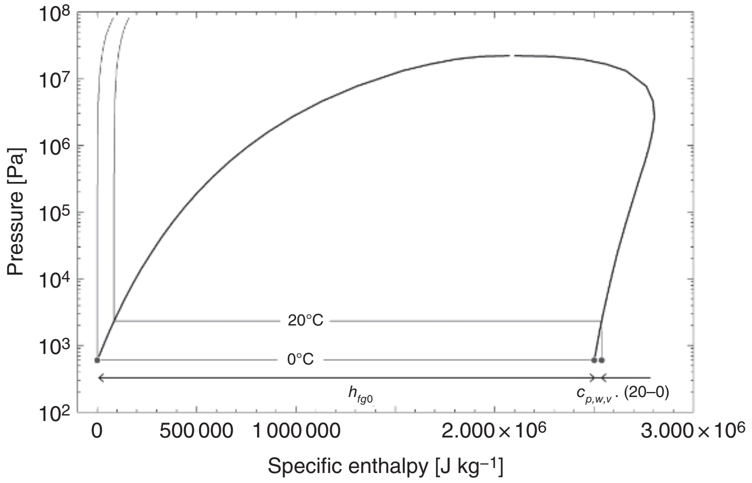

Figure 1.14 Visualization of the specific enthalpy of water vapor in the pressure–enthalpy diagram.

1.3.4.4.5 *Specific Enthalpy*

Having an expression of enthalpy of humid air is very convenient for expressing energy balances across open components involving humid air. The specific enthalpy h [J kg^{-1}] of humid air is given by

$$h = h_a + \omega\, h_w \tag{1.60}$$

Taking 0°C as the reference temperature, the enthalpy of dry air is given by

$$h_a = c_{p,a} T = 1005\, T \tag{1.61}$$

Taking saturated liquid at 0°C as the reference state, the specific enthalpy of water vapor is given by

$$h_w = h_{fg,0} + c_{p,w,v}\, T = 2500561 + 1820\, T \tag{1.62}$$

This equation assumes that water vapor behaves as an ideal gas, and hence, its enthalpy is function of only temperature. Figure 1.14 illustrates the computation of the water vapor enthalpy for a temperature of 20°C. It can be observed that the iso-thermal line associated with 20°C is quasi vertical in the vapor phase, indicating that the enthalpy is not a function of pressure.

1.3.4.4.6 *Psychrometric Diagram*

The thermodynamic state of humid air can be conveniently visualized in the psychrometric diagram. Such a diagram is represented in Figure 1.15 for a given total pressure. The x- and y-axes indicate the dry-bulb and the specific humidity, respectively. Lines of constant relative humidity and lines of constant wet-bulb temperature are drawn. The diagram is delimited on the top by the thicker line associated with $RH = 1$ (saturation state).

1.3.4.4.7 *Typical Processes Encountered by Moist Air*

Among others, the psychometric chart is commonly used to represent air-conditioning processes and any other processes characterized by moisture content variation. As shown in Figure 1.16, these processes include heating, cooling, humidification, and dehumidification processes or combinations of some of them. Examples of applications to vehicles are as follows: a is the humidity increase by water vapor produced by occupants (if their sensible heat is neglected), c represents air heating at windshield position, f represents the air evolution in the A/C loop evaporator (if water condensation occurs), and h is the adiabatic cooling of the air produced by water pulverization in the air.

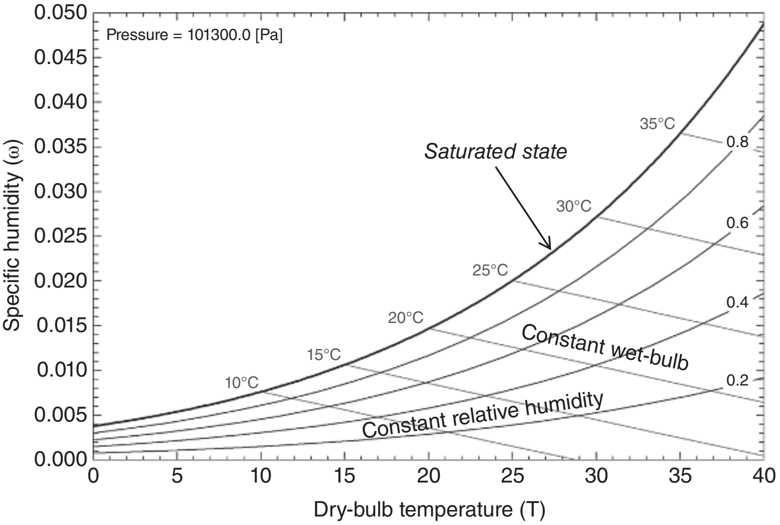

Figure 1.15 Example of a psychrometric diagram.

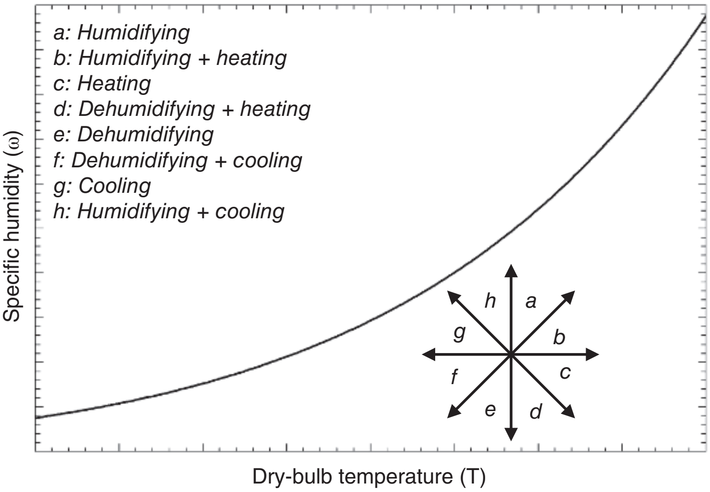

Figure 1.16 Basic processes in the psychrometric diagram. Source: Reproduced and adapted from Çengel and Boles (2006).

The energy analysis of such processes involves expressing the energy balance as well as mass balance (in the case of an open system). The mass balance is split into both water mass balance and dry air mass balance.

1.4 Heat Transfers

This textbook will largely illustrate heat transfers between bodies. The purpose of this section is to better understand the physical mechanisms associated with heat transfers.

The thermal energy of a body seen as a thermodynamic system corresponds to the kinetic energy at the microscopic scale (molecules, atoms, free electrons, and nucleons) plus the latent energy. These microscopic particles can exchange part of their kinetic energy with the system surroundings, leading to an increase or a decrease in the thermal energy of the body.

When two bodies are at temperatures that are different, the hottest body exchanges thermal energy with the coldest one. A heat transfer process occurs between the two bodies. This heat transfer process can be characterized by its intensity and direction.

Three mechanisms, corresponding to different physical phenomena, can explain the heat transfer processes described in this book:

- Conduction
- Convection
- Radiation

In this section, the main mechanisms of heat transfer are described. Heat exchangers, which ensure the function of heat transfer, will be described in a future Section 1.8 of this chapter.

1.4.1 Conduction

Heat transfer by conduction occurs within the body and is characterized by its thermal conductivity k, [W/m · K]. The thermal conductivity is the capacity of the body material to conduct heat.

Conduction of heat results from the *“transfer of energy from the more energetic to the less energetic particles of a substance due to interactions between the particles”* (Incropera and DeWitt, 2002), without any bulk motion of the body. At the microscopic scale, moving or vibrating particles transfer their kinetic and potential energies to the neighboring particles. In gases and liquids, the transfer of energy results from the collision between the molecules. In solids, this transfer results from the oscillation of the lattice due to the movement of atoms and, for solid conductors, from the translational motion of free electrons (Incropera and DeWitt, 2002).

Conduction is the main mechanism of heat transfer within solid bodies. In solid conduction, as mentioned above, thermal energy is transferred from one molecule to another by vibration. A similar mechanism occurs in liquids and gases. However, in the latter phases, heat transfer by conduction is often negligible with respect to convection.

Conduction within a body only occurs when a temperature gradient is imposed within the body under an external action.

The inverse of thermal conductivity is thermal resistivity. For the same temperature gradient, materials characterized by higher thermal conductivity (i.e. low thermal resistivity) conduct more heat than materials characterized by lower thermal conductivity (i.e. high thermal resistivity). Materials characterized by large thermal resistivity are insulating materials.

In the steady-state regime, the heat conduction can be described by Fourier’s law, which relates the heat flux through a surface to the temperature gradient. The differential and one-dimensional form of Fourier’s law can be expressed as

$$q_x = -k\frac{dT}{dx} \tag{1.63}$$

where

q_x is the heat flux in the x-direction (normal to the cross-sectional surface), [W m^{-2}]
k is the material thermal conductivity, [W m^{-1} K^{-1}]
dT/dx is the temperature gradient, [K m^{-1}].

The minus sign indicates that the heat flux and temperature gradient have opposite directions. Assuming that the temperature is constant over the surface (as explained in Incropera and DeWitt

(2002), the heat flow direction is perpendicular to an isothermal surface), and integrating the previous equation over the surface gives

$$\dot{Q} = -k\,A\frac{dT}{dx} \tag{1.64}$$

where

$\dot{Q}$ is the heat flow rate across surface A, [W]
A is the cross-sectional area normal to the heat flow, [m^2]

Applying Fourier's law of conduction to a homogeneous 1-D wall, whose surface area is A [m^2] and thickness is e [m], gives the following relation to express the heat flow rate that is transferred through the wall

$$\dot{Q} = A \cdot \frac{k}{e} \cdot (T_1 - T_2) \tag{1.65}$$

where

$\dot{Q}$ is the heat transfer rate, [W]
A is the wall surface area, [m^2]
e is the wall thickness, [m]
k is the thermal conductivity, [W m^{-1} K^{-1}]
$\frac{e}{k}$ is the thermal resistivity of the material for a thickness e, [m^2 K W^{-1}]
T_1 is the temperature of the hottest side of the wall, [°C]
T_2 is the temperature of the coldest side of the wall, [°C]

Expression (1.65) will be used, for instance, in Chapter 4, when describing the heat transfer through the casing of a battery module.

1.4.2 Convection

This mechanism of heat transfer is associated with mass transfer. More exactly, there are two superposed mechanisms that explain the energy transfer: the random molecular motion (also called diffusion) and the bulk motion of the fluid (also called advection). Convection heat transfer between a fluid in motion and a surface occurs when the fluid is at temperature T_f different from surface temperature T_s.

1.4.2.1 Forced Convection

In the forced convection regime, the fluid motion results from a pressure difference created by an external mechanism. This mechanism can be a fan or a pump, which typically forces fluid (such as air, coolant, or oil) to flow through a pipe or a heat exchanger. The motion of a body in a fluid also creates a fluid flow around the body surfaces. This is, for instance, the case of a moving vehicle whose surfaces are in contact with airflows.

1.4.2.2 Natural Convection

In the natural convection regime (also called free convection), the fluid motion is induced by buoyancy forces caused by density differences resulting from temperature differences within the fluid. Actually, as a liquid or gas is heated, its density decreases. Due to buoyancy forces, the heated fluid moves up, transferring its heat to colder regions. As the fluid is cooled down, its density increases, and the fluid moves down. A convection cell is created. The fluid velocity within the cell is quite low.

1.4.2.3 Mixed Forced and Natural Convection

In mixed forced and natural convection regime, the flow induced by pressure difference and the flow induced by temperature difference are of similar magnitudes.

1.4.2.4 Sensible and Latent Heat Transfer by Convection

A phase change can create fluid motion. For instance, if a fluid is heated until it vaporizes, bubbles are formed and move upwards due to buoyancy forces. In that case, latent heat is transferred by convection. Condensation of a vapor also involves latent heat transfer. Without phase change, only the sensible heat is transferred.

1.4.2.5 Convection Heat Transfer Rates

Regardless of whether sensible or latent heat is transferred, the general equation to compute the heat transfer by convection has the following form:

$$q = h(T_s - T_f) \tag{1.66}$$

where

q is the heat flux, [W m^{-2}]
h is the convection heat transfer coefficient, [W m^{-2} K^{-1}]
T_s is the surface temperature, [°C]
T_f is the fluid temperature, [°C].

The heat transfer rate by convection $\dot{Q}[W]$ can be obtained by multiplying the heat flux by surface area A. Therefore,

$$\dot{Q} = A\,h\,(T_s - T_f) \tag{1.67}$$

1.4.2.6 Laminar and Turbulent Regimes

Irrespective of the type of convection, the distinction can be done between:

- The laminar regime in which the fluid motion is ordered. The fluid follows in a streamline manner.
- The turbulent regime where the fluid motion is irregular. Any direction can be taken by the fluid.

The Reynolds number, which is a dimensionless number, is used to determine whether the flow is laminar or turbulent. At low Re ($Re < 2300$ for fully developed flows in pipes), the flow is laminar. At high Re ($Re > 2900$ for fully developed flows in pipes), the flow is turbulent. The Reynolds number is defined as

$$Re = \frac{\rho\,C\,L}{\mu} \tag{1.68}$$

where

ρ is the density of the fluid, [kg m^{-3}]
C is the speed of the fluid, [m s^{-1}]
L is a characteristic dimension, [m]
μ is the dynamic viscosity of the fluid, [kg m^{-1} s^{-1}].

In the particular case of a flow in a pipe, the characteristic dimension is the hydraulic diameter of the pipe. Often, the natural convection is laminar and the forced convection is turbulent.

1.4.2.7 Convection Heat Transfer Coefficients

Convection heat transfer coefficient h depends on many parameters: the characteristics of the fluid, the nature and velocity of flow, the temperature of the fluid and surface, and the heat transfer surface geometry. There is an exhaustive literature providing correlations for assessing convective heat transfer coefficients involving dimensionless numbers (Reynolds, Prandtl, Grashof, etc.).

Order of magnitude of convection heat transfer coefficients is given in Table 1.3.

1.4.3 Radiation

Radiation is the third mechanism of heat transfer. Atoms and molecules of matter emit energy as electromagnetic waves transmitted at the speed of around 300 000 km s^{-1}. Unlike conduction and convection, energy transfer by radiation does not require any medium. Hence, a body in vacuum emits energy by radiation, while it can transfer energy neither by conduction nor convection.

The distinction must be done between the radiation that is emitted by the surface of a body and the radiation that is incident to a body.

1.4.3.1 Emitted Radiation

The rate of energy emitted by a surface per unit area of surface is called the *emissive power* E [W m^{-2}]. For a blackbody, which is not only a perfect emitter but also absorber (it absorbs any incident radiation), the emissive power is given by the Stefan–Boltzmann law

$$E_b = \sigma\, T_s^4 \tag{1.69}$$

where

σ is the Stefan–Boltzmann constant, ($\sigma = 5.67 \times 10^{-8}$ W m^{-2} K^{-4})
T_s is the blackbody absolute temperature, [K]

A real surface emits less energy than a blackbody at the same temperature. The ratio of the emissive power of a real surface to that of a blackbody at the same temperature is the surface *emissivity* ε. Therefore,

$$E = \varepsilon\, \sigma\, T_s^4 \tag{1.70}$$

Table 1.3 Typical values of convection heat transfer coefficients.

Process	Convection heat transfer coefficient h [W m^{-2} K^{-1}]
Free convection	
Gases	2–25
Liquids	50–1 000
Forced convection	
Gases	25–250
Liquids	100–20 000
Convection with phase change	
Boiling or condensation	2500–100 000

Source: Incropera and DeWitt, 2002.

The emissivity is a function of the material and of the finish of the surface (Incropera and DeWitt, 2002). In the previous equation, ε stands for the *total hemispherical emissivity*. Actually, a surface emits energy by radiation through electromagnetic waves of different wavelengths and in all possible directions. The emissivity varies with both the wavelength λ and the direction.

The *spectral hemispherical emissive power* E_λ [W m^{-2}µm^{-1}] is defined as *"the rate at which radiation of wavelength λ is emitted in all directions from a surface per unit wavelength interval $d\lambda$ about λ and per unit surface area"* (Incropera and DeWitt, 2002). Therefore,

$$E = \int_0^\infty E_\lambda \, d\lambda \tag{1.71}$$

The same equation applies for a blackbody, whose radiant energy also comprises electromagnetic waves at different wavelengths. Note that a blackbody is a diffusive emitter, which means that the radiation it emits is independent of the direction. For a blackbody, the spectral distribution of the emissive power is given by Planck's law

$$E_{\lambda,b} = \frac{3.742 \times 10^8}{\lambda^5(\exp(1.439 \times 10^4/\lambda T_s) - 1)} \ [\mathrm{W\,m^{-2}\mu m^{-1}}] \tag{1.72}$$

where

λ is the wavelength, [µm]

Wien's displacement law states that, for a blackbody at temperature T_s, there exists a wavelength λ_m for which the spectral emissive power is maximum. The relation between temperature T_s [K] and wavelength λ_m is given by

$$\lambda_m \, T_s = 2897.8 \ \mu\mathrm{m\,K} \tag{1.73}$$

Figure 1.17 shows the spectral distribution of the emissive power divided by the maximum emissive power for two blackbodies: the sun (which can be seen as a blackbody at 5760 K) and a blackbody at room temperature. It can be seen that the two spectra are clearly distinct: they go through maxima for two distinct wavelengths. The range of wavelengths associated with the room temperature blackbody corresponds to "longwave infrared" radiation. The solar radiation spectrum comprises ultraviolet, visible, and near-infrared regions. This will be discussed in Chapter 3.

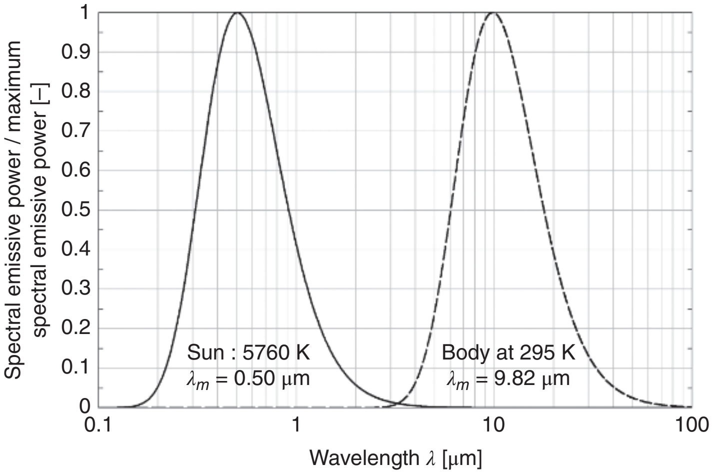

Figure 1.17 Spectral distribution of the ratio of the emissive power to the maximum emissive power for the sun (which can be viewed as a blackbody at 5760 K) and a blackbody at room temperature.

Figure 1.17 explains the greenhouse effect, which contributes to the heating of car radiated by the sun: glazing is transparent to solar radiation but opaque to the longwave infrared radiation emitted by the seats, the dashboard, and other surfaces inside the car cabin, which tends to “trap” the solar energy inside the cabin.

The emissive power of a real surface can be related to the spectral emissive power of a black body by

$$E = \int_0^\infty \varepsilon_\lambda \, E_{\lambda,b} \, d\lambda \tag{1.74}$$

This gives a relation between the total hemispherical emissivity ε [−] and the spectral hemispherical emissivity ε_λ [−]. These two quantities are related by

$$\varepsilon = \frac{1}{\sigma \, T_s^4} \int_0^\infty \varepsilon_\lambda \, E_{\lambda,b} \, d\lambda \tag{1.75}$$

1.4.3.2 Incident Radiation

The rate at which radiation is incident on a surface *“per unit area from all directions and at all wavelengths”* (Incropera and DeWitt, 2002) is called the *total irradiation* G [W m^{-2}]. The irradiation is thus the intensity of the incident radiation. *Spectral irradiation* G_λ [W m^{-2}μm^{-1}] is *“the rate at which radiation of wavelength* λ *is incident on a surface per unit area of the surface and per unit wavelength interval* $d\lambda$ *about* λ*”* (Incropera and DeWitt, 2002). Total and spectral irradiations are related to each other by

$$G = \int_0^\infty G_\lambda(\lambda) \, d\lambda \tag{1.76}$$

When radiant energy is incident to a surface, it can be absorbed, transmitted (if the surface is semitransparent), or reflected. Hence, the spectral irradiation can be split as

$$G_\lambda = G_{\lambda,\alpha} + G_{\lambda,\tau} + G_{\lambda,\rho} \tag{1.77}$$

The spectral hemispherical absorptivity α_λ is defined as the fraction of the spectral irradiation that is absorbed by the surface

$$\alpha_\lambda = \frac{G_{\lambda,\alpha}}{G_\lambda} \tag{1.78}$$

The total hemispherical absorptivity α is defined as the fraction of the total irradiation that is absorbed by the surface

$$\alpha = \frac{G_\alpha}{G} \tag{1.79}$$

Spectral and total transmissivity and reflectivity can be defined in the same way. The spectral absorptivity, transmissivity, and reflectivity are related by

$$1 = \alpha_\lambda + \tau_\lambda + \rho_\lambda \tag{1.80}$$

Averaging these properties over the full spectrum of wavelengths, it follows

$$1 = \alpha + \tau + \rho \tag{1.81}$$

Note that the latter two equations are valid for semitransparent surfaces. In the case of an opaque surface, $\tau = 0$. These thermal radiative properties will be extensively used in Chapter 3 when describing radiative exchanges with vehicle glazing and opaque body.

1.4.3.3 Kirchhoff's Law and the Gray Surfaces

The most general Kirchhoff's law states that

$$\varepsilon_{\lambda,\theta} = \alpha_{\lambda,\theta} \tag{1.82}$$

with no restriction on spectral and directional properties.

A surface is said to be *diffuse* if $\varepsilon_{\lambda,\theta}$ and $\alpha_{\lambda,\theta}$ are independent of direction θ. In that case,

$$\varepsilon_{\lambda} = \alpha_{\lambda} \tag{1.83}$$

A surface is said to be gray if ε_{λ} and α_{λ} are independent of λ. Hence (Incropera and DeWitt, 2002), for a gray and diffuse surface, Kirchhoff's law reduces to

$$\varepsilon = \alpha \tag{1.84}$$

It should be stressed that a surface could be gray in a limited spectral region, comprising irradiation and emission, where surface spectral properties are constant.

1.4.3.4 Radiation Exchange Between Surfaces

The description of vehicle thermal management systems involves radiative heat exchanges between surfaces at different temperatures, with different radiative properties, and at different orientations. A typical example is the radiative exchanges between surfaces inside the vehicle cabin, which largely impacts the air and surface temperatures and therefore the occupants' comfort. This example will be treated in Chapter 3.

This section will only treat the radiative exchange between two surfaces. To learn a more exhaustive approach to treat exchange between more than two surfaces, the reader is invited to refer to Incropera and DeWitt (2002) textbook.

Some specific cases are described in the following paragraphs and should allow to treat most of radiative heat transfer problems introduced in future chapters.

1.4.3.4.1 General Case of Radiation Exchange Between two Diffuse and Gray Surfaces Let's imagine an enclosure made of two diffuse and gray surfaces. The net rate of radiation leaving surface *1* is $\dot{Q}_1$. The latter can be seen as the heat transfer rate to provide to surface at T_1 to maintain its temperature constant (Incropera and DeWitt, 2002). Since there are only two surfaces, $\dot{Q}_1$ must be equal to $\dot{Q}_2$. The latter heat flow rate is the net radiative rate arriving on surface 2. It can also be written $\dot{Q}_{12}$ to express that the net rate of radiation leaving surface 1 is transferred to surface 2.

It can be demonstrated (Incropera and DeWitt, 2002) that the net radiative exchange between the two surfaces can be computed by

$$\dot{Q}_{12} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1-\varepsilon_1}{\varepsilon_1 A_1} + \frac{1}{A_1 F_{12}} + \frac{1-\varepsilon_2}{\varepsilon_2 A_2}} \tag{1.85}$$

Equation (1.85) introduces the view factor F_{12}. The view factor is "*the ratio of radiant energy leaving a surface i and directly incident on a surface k to the total radiant energy leaving i in all directions*" (Incropera and DeWitt, 2002). A very convenient relation, named the reciprocity relation, states that

$$F_{ik} A_i = F_{ki} A_k \tag{1.86}$$

From Eq. (1.85), simplified expressions can be derived for specific cases listed hereunder.

1.4.3.4.2 Radiation Exchange Between two Parallel Diffuse and Gray Surfaces Since $F_{12} = 1$ and $A_1 = A_2$, Eq. (1.85) reduces to

$$\dot{Q}_{12} = \frac{\varepsilon_1\, \varepsilon_2\, A_1\, \sigma\left(T_1^4 - T_2^4\right)}{\varepsilon_1 - \varepsilon_1\, \varepsilon_2 + \varepsilon_2} \tag{1.87}$$

Such heat transfer occurs typically between the two layers of a multilayer material if they are separated by a nonparticipating gas, such as the surfaces of a wall cavity (Braun and Mitchell, 2012).

1.4.3.4.3 Radiation Exchange Between a Small and Convex Diffuse and Gray Surface 1 Surrounded by a Large Diffuse and Gray Surface 2 In this case, $F_{12} = 1$ and $A_1/A_2 \ll$. Therefore, Eq. (1.85) reduces to

$$\dot{Q}_{12} = \varepsilon_1\, A_1\, \sigma\left(T_1^4 - T_2^4\right) \tag{1.88}$$

The previous equation to compute the net rate of radiation heat transfer between surface 1 surrounded by surface 2 seen as its surroundings can be linearized in the following way

$$\dot{Q}_{12} = A_1\, h_r\, (T_1 - T_2) \tag{1.89}$$

with

$$h_r[\mathrm{Wm^{-2}}] = \varepsilon_1\, \sigma\, (T_1 + T_2)\, \left(T_1^2 + T_2^2\right) \tag{1.90}$$

1.4.3.4.4 Radiation Exchange Between a Diffuse and Gray Surface 1 and a Blackbody 2 Assuming surface 2 to be a blackbody, $\varepsilon_2 = 1$. That is

$$\dot{Q}_{12} = \varepsilon_1\, A_1 F_{12}\, \sigma\left(T_1^4 - T_2^4\right) \tag{1.91}$$

If surface 2 surrounds surface 1, then $F_{12} = 1$ and it follows that

$$\dot{Q}_{12} = \varepsilon_1\, A_1 \sigma\left(T_1^4 - T_2^4\right) \tag{1.92}$$

1.4.3.4.5 Radiation Exchange Between Two Blackbodies In this case, ε_1 and ε_2 are equal to unity. It follows that

$$\dot{Q}_{12} = A_1 F_{12}\, \sigma\left(T_1^4 - T_2^4\right) \tag{1.93}$$

1.5 First Law of Thermodynamics

The First Law of Thermodynamics states that energy can be neither created nor destroyed, but only converted from one form to another. Therefore, the First Law of Thermodynamics is the principle of conservation of energy. In its more general form, it states that the variation of the total energy of a system ΔE_{system} over a process is equal to the total energy E_{su} entering the system minus the total energy E_{ex} leaving the system during the process.

$$E_{su} - E_{ex} = \Delta E_{system} = \Delta U_{system} + \Delta KE_{system} + \Delta PE_{system} \tag{1.94}$$

Often, the variation of the kinetic energy ΔKE_{system} and the variation of potential energy ΔPE_{system} of the system can be neglected. The former assumption is valid if the system is at rest or if it evolves at a constant speed. The latter assumption is valid if the altitude of the system is constant. The variation of the total energy reduces to the variation of internal energy ΔU_{system}.

A system can exchange energy with its surroundings under different forms:
Heat Q

- Work W: it could be, among others, a mechanical work or an electrical work.
- Mass m: any mass stream entering or leaving the system contains energy equal to the enthalpy of the flow plus its kinetic energy and potential energy.

Therefore, the previous equation can be written as

$$(Q_{su} - Q_{ex}) + (W_{su} - W_{ex}) + \sum_{su} m\left(h + \frac{C^2}{2} + gz\right) - \sum_{ex} m\left(h + \frac{C^2}{2} + gz\right) = \Delta U_{system} \quad (1.95)$$

where

Q_{su} is the heat transferred to the system from its surroundings, [J]
Q_{ex} is the heat transferred from the system to its surroundings, [J]
W_{su} is the work transferred to the system from its surroundings, [J]
W_{ex} is the work transferred from the system to its surroundings, [J]
h is specific enthalpy of each mass stream entering/leaving the system, [J kg^{-1}]
$C^2/2$ is the kinetic energy of each mass stream entering/leaving the system, [J kg^{-1}]
gz is the potential energy of each mass stream entering/leaving the system, [J kg^{-1}]
$h + C^2/2 + gz$ is the total energy of each mass stream entering/leaving the system, [J kg^{-1}]
ΔU_{system} is the variation of internal energy of the system, [J]

The previous equation can be written in its rate form:

$$(\dot{Q}_{su} - \dot{Q}_{ex}) + (\dot{W}_{su} - \dot{W}_{ex}) + \sum_{su} \dot{m}\left(h + \frac{C^2}{2} + gz\right) - \sum_{ex} \dot{m}\left(h + \frac{C^2}{2} + gz\right) = \frac{dU_{system}}{dt} \quad (1.96)$$

The rate of work can be, among others, a mechanical power (for instance, a shaft power) or an electrical power.

Equations (1.95) and (1.96) are of paramount importance for evaluating the performance of any component or system in vehicle thermal management.

1.5.1 Closed System

As explained previously, a closed system (or "control mass") is a system that does not exchange mass with its surrounding. As a consequence, the mass of such a system is constant. In closed systems, the only forms of energy transfer are heat and work. The First Law of Thermodynamics for closed systems is therefore:

$$(\dot{Q}_{su} - \dot{Q}_{ex}) + (\dot{W}_{su} - \dot{W}_{ex}) = \frac{dU_{system}}{dt} \quad (1.97)$$

In the particular case of a closed system undergoing a cycle, the initial and final states of the system are equal. Hence,

$$(\dot{Q}_{su} - \dot{Q}_{ex}) + (\dot{W}_{su} - \dot{W}_{ex}) = 0 \quad (1.98)$$

In the particular case of a closed system that exchanges neither heat nor work with its surroundings, the system is "isolated." Then,

$$\frac{dU_{system}}{dt} = 0 \quad (1.99)$$

1.5.2 Open System

An open system (or "control volume," indicated by "CV" hereafter) can exchange mass with its surroundings. Therefore, besides the conservation of energy, the conservation of mass must also be expressed. Note that for closed systems, the conservation of mass is trivial and states that the mass of the system is constant.

1.5.2.1 Mass Balance

Let's assume an open system with multiple supply ports and exhaust ports. The rate of variation of the mass of the control volume is equal to the total mass flow rate entering the system minus the total mass flow rate leaving the system.

$$\frac{dm_{CV}}{dt} = \sum_{su} \dot{m}_{su} - \sum_{ex} \dot{m}_{ex} \tag{1.100}$$

An open system can undergo a steady-flow process when the fluid flows steadily through the control volume. As shown previously, the opposite process is the transient process. The steady-flow process is approached when a system operates for a long time under the same conditions. We will see that this assumption is very useful to describe components such as compressors and heat exchangers. In the particular case of a steady-flow process, the properties at a given point of the control volume remain constant during the process, even if these properties can vary from point to point inside the control volume (for instance, the temperature of one stream at the inlet of a heat exchanger may be different to that at the outlet). Since the properties at each point of the control volume do not vary with time, the mass contained in the control volume is constant. Therefore, in the steady-state regime, the total mass flow rate entering the system is equal to the total mass flow rate leaving the system.

$$\sum_{su} \dot{m}_{su} = \sum_{ex} \dot{m}_{ex} \tag{1.101}$$

In the specific case of a system with one single inlet and one single outlet, the previous equation reduces to

$$\dot{m}_{su} = \dot{m}_{ex} \tag{1.102}$$

If the working fluid is incompressible (and hence its specific volume is constant), the previous equation can be expressed in terms of volume flow rate and speed, if the cross-sectional area A of the pipe is known.

$$\dot{V}_{su} = \dot{V}_{ex} \Longleftrightarrow A_{su} C_{su} = A_{ex} C_{ex} \tag{1.103}$$

where

$\dot{V}$ is a volume flow rate, [m^{3}s^{-1}]
A is a cross sectional area, [m^2]
C is a velocity, [m s^{-1}]

1.5.2.2 Energy Balance

The most general form of the energy balance applied to an open system is Eq. (1.96).

As explained for the mass balance, an open system can undergo either a steady flow process or a transient process. An example of a system that undergoes a transient process is a thermal storage system during charging or discharging processes. This will be addressed in Chapter 4 when modeling PCM energy storage. Another example of transient process is the heating-up or cooling-down phases of a car cabin. During these processes, the internal energy of the system is increasing or decreasing.

In the case of a steady-flow process, if the properties at each point of the control volume are constant in time, it means that the internal energy content is constant.

Consequently, in the steady-flow regime, the energy balance for a control volume can be expressed as:

$$(\dot{Q}_{su} - \dot{Q}_{ex}) + (\dot{W}_{su} - \dot{W}_{ex}) + \sum_{su} \dot{m}\left(h + \frac{C^2}{2} + gz\right) - \sum_{ex} \dot{m}\left(h + \frac{C^2}{2} + gz\right) = 0 \qquad (1.104)$$

As it will be shown in the next chapters, for many thermodynamic systems, the kinetic and potential energies, or their variations, can be neglected. Also, many systems are characterized by one single flow supply and exhaust. For such systems, the previous equation reduces to

$$\dot{m}(h_{ex} - h_{su}) = (\dot{Q}_{su} - \dot{Q}_{ex}) + (\dot{W}_{su} - \dot{W}_{ex}) \qquad (1.105)$$

The latter equation is of particular use to describe heat exchangers, compressors, pumps, and turbines.

1.6 Second Law of Thermodynamics

The First Law of Thermodynamics alone cannot explain all evolutions. Let's imagine, for instance, a car with a cabin indoor temperature of 15°C in surroundings with a temperature of 5°C (instead of the example of the hot coffee cup described by Çengel and Boles (2006)). Let's also assume the solar radiation to be null and there is no heat source inside the cabin (no occupant and heating system switched off). Our everyday experience teaches us that the cabin indoor temperature cannot spontaneously rise, since the cabin outdoor temperature is lower than the cabin indoor temperature. Actually, the cabin indoor temperature is expected to decrease because of heat losses from the cabin to the surroundings through the cabin envelope. Heat cannot flow spontaneously from the surroundings to a cabin. To achieve such a heat transfer, a *heat pump* is needed. However, the First Law does not prevent heat from flowing from the surroundings to the cabin, resulting in an increase of its temperature. The First Law applied to the cabin, as a thermodynamic system, will be satisfied: when the quantity of thermal energy lost by the surroundings is equal to the one received by the cabin and energy is conserved.

Another example of impossible thermodynamic evolution from our everyday life is the spontaneous transformation of heat into work. If a body is sliding along a surface, there is a friction force that is opposed to the force applied to the body to move. This friction force leads to an additional friction work to be provided to move the body. Ultimately, this friction work is converted into heat released to the body and to the surroundings. This heat cannot be spontaneously transformed into work that would contribute to move the body. To transform heat into work, a *heat engine* is required.

Hence, another law should explain why some evolutions are possible (heat flowing spontaneously from hot bodies to cold bodies, work transformed directly and entirely into heat) and not possible (heat flowing spontaneously from cold bodies to hot bodies, heat transformed directly and entirely into work). This other law is named the Second Law of Thermodynamics.

While the First Law of Thermodynamics can be "simply" summarized as the conservation of energy, the Second Law can be expressed through different statements that are equivalent. These statements will be progressively introduced in this section.

1.6.1 Concepts and Definitions

To understand and express the Second Law of Thermodynamics, it is necessary to introduce some concepts and definitions.

1.6.1.1 Heat Reservoir, Source, and Sink

A *heat reservoir* is a body that can exchange heat with its surroundings without any variation of its temperature. An example of a heat reservoir is the ocean. Our everyday experience teaches us that swimming in the ocean does not yield any noticeable variation of the ocean temperature, while the human body transfers thermal power to the water. Another example of thermal reservoir is a mixture of ice and liquid water or any two-phase mixture of a pure fluid.

A *heat source* is a thermal reservoir that gives heat. A *heat sink* is a thermal reservoir that receives heat.

1.6.1.2 Heat Engines

A heat engine is an apparatus that receives heat from a heat source at a high temperature T_{high}, converts part of it into work, and releases the remaining part to a heat sink at a low temperature T_{low}.

As depicted in Figure 1.18a, a heat engine comprises a working fluid that undergoes a cycle. During this evolution, the fluid receives and gives heat and work from/to its surroundings. The First Law expressed over one entire cyclic process can be written as

$$Q_{su} - Q_{ex} = W_{ex} - W_{su} = W_{net,ex} = -W_{net,su} \tag{1.106}$$

The thermal efficiency of a heat engine is defined as the ratio of the net work produced by the system to the heat it receives from the heat source.

$$\eta = \frac{W_{net,ex}}{Q_{su}} \tag{1.107}$$

The major example of heat engines covered in this book is the internal combustion engine. In the case of an internal combustion engine, the heat received by the engine corresponds to its fuel consumption m_{fuel} multiplied by the low heating value LHV_{fuel} of the fuel. In terms of rate of heat transfer and power, the thermal efficiency can be written as

$$\eta = \frac{\dot{W}_{net,ex}}{\dot{m}_{fuel} LHV_{fuel}} \tag{1.108}$$

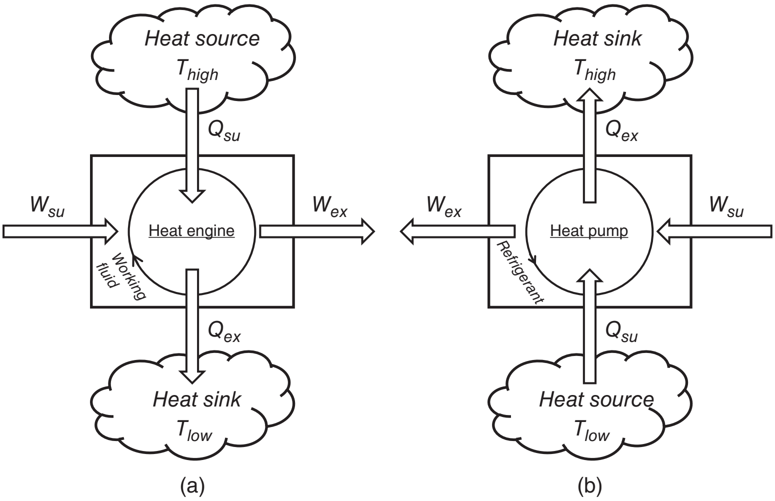

Figure 1.18 Coupling between a heat engine (a)/heat pump (b) and the heat source/heat sink.

The low heating value LHV [J kg^{-1}] is defined as the thermal energy released by the combustion of 1 kg of fuel if the water in the combustion gases stays in vapor phase. The high heating value HHV [J kg^{-1}] is defined as the heat that is released by the combustion of 1 kg of fuel if the water in the combustion gases is condensed to liquid state. Hence, the high heating value of a fuel is higher than its low heating value. For instance, for gasoline, $LHV = 44$ [MJ kg^{-1}].

1.6.1.3 Refrigerators and Heat Pumps

As explained previously, heat cannot flow spontaneously from a cold body to a hot body. To achieve such energy transfer, a refrigerator or heat pump is needed. A refrigerator or heat pump is an apparatus that absorbs a quantity of heat Q_{su} from a heat source at low temperature T_{low}, absorbs a net quantity of work $W_{net,su}$, and rejects a quantity of heat Q_{ex} into a heat sink at a high temperature T_{high}. Such a system is represented in Figure 1.18b. In a similar way to a heat engine, a heat pump or refrigerator comprises a working fluid that undergoes a cycle. This working fluid is called refrigerant. If the useful effect of the system is to produce a cooling effect, it is called a refrigerator. If the useful effect of the system is to produce a heating effect, it is called a heat pump. Because the system undergoes a cycle, Eq. (1.106) also applies.

The coefficient of performance (COP) of the refrigerator or heat pump is defined as the thermal energy exchanged (useful effect) divided by the work consumed (required input). In the refrigerator mode, this yields

$$COP_{cool} = \frac{Q_{su}}{W_{net,su}} \tag{1.109}$$

In the heat pump mode, the COP is defined as

$$COP_{heat} = \frac{Q_{ex}}{W_{net,su}} \tag{1.110}$$

Considering Eq. (1.106), the COPs in refrigerator and heat pump modes can be related by

$$COP_{heat} = COP_{cool} + 1 \tag{1.111}$$

The latter equation means that, in the absence of heat transfer with other thermal reservoirs than the heat source/sink (for instance, heat losses to the ambient), the heating COP is at least equal to unity. This equation is of primary importance to understand why it is more relevant to heat an electric vehicle with a heat pump rather than a resistive heater ($COP = 1$). This will be discussed in Chapter 4. Note that there exist many technologies of heat pump/refrigerator. However, the most used is the vapor compression one. It will be described later.

1.6.2 Kelvin Planck and Clausius Statements of the Second Law

The Second Law of Thermodynamics can be expressed in the form of two statements that are equivalent.

(1) The Kelvin–Planck Statement of the Second Law of Thermodynamics can be written as (Çengel and Boles, 2006)

> *It is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work*

As a consequence to this statement, it is not possible to build a heat engine that would have an efficiency of 100%, since part of the heat absorbed from the heat source must be rejected to the heat sink (see Figure 1.18a).

(2) The Clausius Statement of the Second Law of Thermodynamics states that (Çengel and Boles, 2006)

> *It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a lower-temperature body to a higher-temperature body*

As a consequence of this statement, it is not possible to build a refrigerator or heat pump that would allow for the thermal energy transfer from a low temperature heat source to a high temperature heat sink without any work consumption (which would correspond to a COP of infinity).

1.6.3 Reversible Processes

Summarizing the two previous statements, the Second Law of Thermodynamics imposes that the thermal efficiency of a heat engine cannot be 100%, and the coefficient of performance of a refrigerator/heat pump cannot be infinity. Hence, we would like to know what the performance of an ideal heat engine/refrigerator/heat pump is. To address this question, the definition of an ideal machine and of a reversible process should be expressed.

A *reversible process* is a process that can be reversed by bringing back the system and its surroundings to their initial states. In contrary, a process that is not reversible is said to be *irreversible*. Among reversible processes, the distinction can be done between the internally and externally reversible processes.

Internally reversible processes are processes occurring without any irreversibilities within the system boundaries. Examples of internal irreversibilities are friction between moving elements in contact, sudden compressions and expansions, and pressure drops. A process is internally reversible if it can be reversed and undergo the same intermediate equilibrium states.

Externally reversible processes are processes without any irreversibilities outside the system boundaries. In such systems, heat transfer between the system and its surroundings is achieved without any temperature difference between the heat source/heat sink and the working fluid.

Both internally and externally reversible processes are theoretical processes that cannot be achieved in practice. A process that is both internally and externally reversible is said to be *totally reversible*.

1.6.4 Ideal Heat Engines, Refrigerators, and Heat Pumps

An ideal machine is a machine whose working fluid follows internally and externally reversible processes. Consequently, the cycle described by the working fluid is totally reversible. A machine describing a totally reversible cycle is said to be reversible. In such machines, work consumption W_{su} is minimal, work production W_{ex} is maximal, and the heat exchanges with the heat source and heat sink are achieved with no temperature difference between the heat sink/source and the working fluid. A reversible heat engine produces a maximum of work and a reversible heat pump consumes a minimum of work.

An example of a totally reversible cycle is the Carnot cycle (Figure 1.19). A Carnot heat engine, which is a reversible heat engine, describes such a cycle. In this engine, a gas undergoes successively four reversible processes:

Processes 1-2: Isothermal expansion. To maintain constant the temperature of the gas during the expansion, a quantity of heat Q_{su} is transferred from the heat source to the gas. During this heat transfer, both the heat source and the gas are at temperature T_{high}.

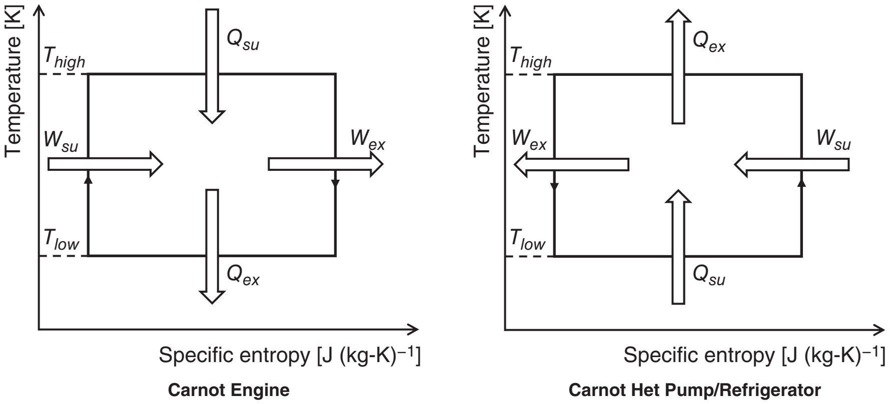

Figure 1.19 Carnot heat engine and refrigerator.

Processes 2-3: Adiabatic expansion. The expansion process continues, but without heat exchange between the gas and its surroundings. Since the expansion is adiabatic, the gas temperature decreases during the expansion from T_{high} to T_{low}.

Processes 3-4: Isothermal compression. To maintain constant the temperature of the gas during the compression, a quantity of heat Q_{ex} is transferred from the gas to the heat sink. During this heat transfer, both the heat sink and the gas are at temperature T_{low}.

Processes 4-1: Adiabatic compression. The compression process continues, but adiabatically. As a consequence, the gas temperature increases from T_{low} to T_{high}.

Other totally reversible heat engines are the Stirling and the Ericsson heat engines, which describe the Stirling and Ericsson cycles, respectively. Those cycles won't be described here, because of their limited use in vehicle applications at the present time.

Knowing what a reversible heat engine is, the two Carnot statements can be expressed as follows (Çengel and Boles 2006)

> "*The efficiency of an irreversible heat engine is always less than the efficiency of a reversible one operating between the same two reservoirs*"
>
> "*The efficiencies of all reversible heat engines operating between the same two reservoirs are the same*"

It could be demonstrated that violating these principles would violate the Kelvin–Planck Statement of the Second Law of Thermodynamics, which is the meaning of "equivalence." Therefore, up to now, we have expressed the Second Law of Thermodynamics under the form of 4 equivalent statements.

The Second Carnot principle states that the efficiency of a reversible heat engine is independent of the working fluid and of the type of heat engine and of the cycle described by the fluid (Çengel and Boles 2006). Hence, the efficiency of a reversible cycle is only function of the temperatures of the heat source and the heat sink. It could be demonstrated that the thermal efficiency of a reversible heat engine is equal to the *Carnot efficiency* expressed as

$$\eta_{rev} = 1 - \frac{T_{low}\ [K]}{T_{high}\ [K]} \tag{1.112}$$

Being a reversible engine, a Carnot heat engine, a Stirling heat engine, and an Ericsson heat engine operating between a heat source at temperature T_{high} and a heat sink at temperature T_{low} show the same efficiency that is equal to the Carnot efficiency (Eq. (1.112)).

Because it is reversible, a Carnot heat engine can be inversed and operated as a Carnot refrigerator or a heat pump between a heat source at temperature T_{low} and a heat sink at temperature T_{high}. The coefficients of performance of a Carnot refrigerator and of a Carnot heat pump are given by

$$COP_{cool,rev} = \frac{T_{low}\ [K]}{T_{high}\ [K] - T_{low}[K]} \tag{1.113}$$

$$COP_{heat,rev} = \frac{T_{high}\ [K]}{T_{high}\ [K] - T_{low}\ [K]} \tag{1.114}$$

An irreversible heat engine and an irreversible refrigerator/heat pump will show a thermal efficiency and a COP that are fractions of the Carnot efficiency and Carnot COP. These fractions are called the second law efficiencies of heat engines, refrigerators, and heat pumps.

For a heat engine, we have

$$\eta = \varepsilon_{II} \cdot \eta_{rev} \tag{1.115}$$

For a refrigerator, we have

$$COP_{cool} = \varepsilon_{II} \cdot COP_{cool,rev} \tag{1.116}$$

Typical values of second law efficiencies of around 50% can be met in practice. However, less and more efficient machines could show values lower and higher than 50%. To some extent, the value of the second law efficiency depends on a trade-off between cost and performance, which is the object of thermoeconomics.

The previous equations indicate that the thermal efficiency of a heat engine and the COP of a refrigerator/heat pump could be increased by

- decreasing irreversibilities associated with the machine, thus increasing the second law efficiency
- increasing the heat source temperature and/or decreasing the heat sink temperature, thus increasing the Carnot efficiency and Carnot COP.

For a refrigerator, this means increasing the temperature at which the cooling effect is produced and rejecting the heat into a heat sink at a lower temperature. For a heat pump, this means reducing the temperature at which the heating effect is produced and absorbing the heat from a heat source at a higher temperature.

1.6.5 Entropy

Another inequality associated with the Second Law of Thermodynamics is the Clausius inequality. It can be demonstrated based on the Kelvin–Planck Statement. The Clausius inequality states (and this is our fifth statement) that

$$\oint \frac{\delta Q}{T} \leq 0 \tag{1.117}$$

The symbol on the integral sign means that the integration is conducted over one entire cycle. In this expression, T [K] is the system boundary temperature, and δQ is the differential amount of heat exchanged between the system and its surroundings.

The equality occurs in the specific case of an internally reversible process:

$$\oint \left(\frac{\delta Q}{T}\right)_{int,rev} = 0 \tag{1.118}$$

By definition, the expression $(\delta Q/T)_{int,rev}$ must be the variation of a state variable, since this variation on a cycle is equal to zero. This state variable is called *entropy* and is denoted by the symbol S [J K^{-1}].

Since the entropy is a state variable, its variation can be calculated by considering a hypothetical process that is internally reversible. This gives

$$\Delta S = S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T}\right)_{int,rev} \tag{1.119}$$

According to the Clausius inequality, we have

$$\Delta S = S_2 - S_1 \geq \int_1^2 \frac{\delta Q}{T} \tag{1.120}$$

The equality is valid for an internally reversible process. In the case of an irreversible process, the variation of entropy has two contributions: the entropy transferred associated with the heat transfer and the entropy generation due to irreversibilities. The term S_{gen} can only be positive or equal to zero, but never negative.

$$\Delta S = S_2 - S_1 = \int_1^2 \frac{\delta Q}{T} + S_{gen} \tag{1.121}$$

If the system is isolated, the variation of its entropy corresponds to the internal production of entropy. In this case, the previous equation simplifies to

$$\Delta S = S_2 - S_1 = S_{gen} \geq 0 \tag{1.122}$$

This means that the entropy of an isolated system can only be constant or increase, but never decrease.

In the case of an adiabatic and reversible evolution, the variation of entropy is null, and the process is said to be isentropic. Indeed, the two terms of the right-hand side of Eq. (1.121) are null. The isentropic process is often considered as an ideal reference process to evaluate the performance of pumps, fans, compressors, and turbines. Actually, in the absence of cooling or heating mechanisms, these components ideally operate adiabatically.

The isentropic effectiveness of a compressor is defined as the ratio of the power that would be consumed if the fluid were compressed isentropically to the actual power that is consumed by the compressor.

$$\varepsilon_{s,cp} = \frac{\dot{W}_{cp,s}}{\dot{W}_{cp}} \tag{1.123}$$

If the variations of kinetic and potential energies can be neglected, the previous equation becomes

$$\varepsilon_{s,cp} = \frac{\dot{m}\,(h_{ex,s} - h_{su})}{\dot{W}_{cp}} \tag{1.124}$$

where

$\dot{m}$ is the fluid mass flow rate flowing through the compressor, [kg s^{-1}]

$h_{ex,s}$ is the enthalpy of the fluid at the exhaust of the compressor if the compression was isentropic, [J kg^{-1}]

h_{su} is the enthalpy of the fluid at the supply of the compressor, [J kg^{-1}]

$\dot{W}_{cp}$ is the actual power consumed by the compressor, [W].

The same definition applies for fans and pumps. For a pump, using Eq. (1.50), the isentropic effectiveness can be expressed as

$$\varepsilon_{s,pp} = \frac{\dot{m}\,v\,(P_{ex} - P_{su})}{\dot{W}_{cp}} = \frac{\dot{V}\,(P_{ex} - P_{su})}{\dot{W}_{cp}} \tag{1.125}$$

For a turbine, the isentropic effectiveness is defined as the ratio of the actual power produced by the turbine to the power that would be produced if the fluid were expanded isentropically. If the variations of kinetic and potential energies can be neglected, the turbine isentropic effectiveness is given by

$$\varepsilon_{s,turb} = \frac{\dot{W}_{turb}}{\dot{m}(h_{su} - h_{ex,s})} \tag{1.126}$$

1.7 Flows in Hydraulic Circuits

Automotive thermal management systems comprise many different hydraulic circuits, with air, coolant (glycol water), or oil as major fluids. Circulation of the fluids through these circuits is generally made possible by pumps and fans. It will be shown in Chapter 2 that the vehicle ram effect also contributes to create the motion of air through the front-end module. The energy consumption of fans and pumps depends not only on their efficiencies, but also on the mechanical energy losses to overcome. The following section introduces basic relationships to describe flows in hydraulic and aeraulic circuits and assess the consumption of fans and pumps.

The flow in a hydraulic circuit can be described using the Bernoulli equation and the First Law of Thermodynamics. Let's assume a portion of hydraulic circuit represented in Figure 1.20. The fluid enters the portion of circuit through section 1 at pressure P_1 [Pa], velocity C_1[m s^{-1}], and elevation z_1 [m] and leaves it through section 2 at pressure P_2 [Pa], velocity C_2[m s^{-1}], and elevation z_2 [m]. A pump is placed on the circuit to increase the mechanical energy of the flow. Mechanical energy losses occur on the circuit due to two types of pressure losses: a friction pressure drop and a local pressure drop (the latter can be due to a section reduction, a valve, an elbow, etc.).

The fluid is assumed to be incompressible and characterized by a specific volume v [m^{3}kg^{-1}]. Furthermore, the portion of the circuit is assumed to be in the steady-state regime.

The conservation of mass between sections 1 and 2 expresses that the mass flow rate entering the portion is equal to that leaving the portion and is simply written as $\dot{m}$ [kg s^{-1}].

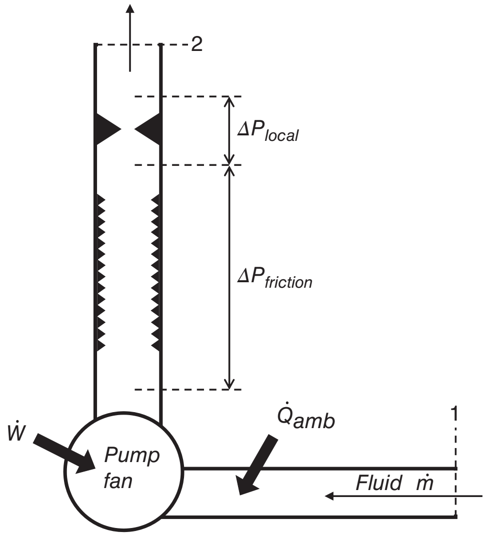

Figure 1.20 Description of the flow in a hydraulic circuit.

To determine the power of fan/pump necessary to move the fluid in this portion of hydraulic circuit, the generalized Bernoulli equation is used. It relates the rate of mechanical energy at the inlet of the portion of circuit to that at the outlet. That is,

$$\dot{m}\left(v\,P_1 + \frac{C_1^2}{2} + g\,z_1\right) + \eta\,\dot{W} = \dot{m}\left(v\,P_2 + \frac{C_2^2}{2} + g\,z_2\right) + \dot{m}\,e_{mech,loss} \tag{1.127}$$

In the latter equation, $C^2/2$ [J kg^{-1}] represents the kinetic energy per kilogram of fluid and $g\,z$ [J kg^{-1}] the potential energy per kilogram of fluid. $\dot{W}$ [W] is the power consumed by the fan or fan/pump. It is mechanical power if the fan/pump is mechanically driven or electrical power if it is driven by an electrical motor. The efficiency of the fan/pump is written as η (for electrically driven fans and pumps, it includes the electric motor efficiency). Therefore, $\eta\,\dot{W}$ represents the minimum power required to move the fluid. Pressure losses yield a rate of mechanical energy loss $\dot{E}_{mech,loss} = \dot{m}\,e_{mech,loss}$ [W].

The conservation of energy for the control volume represented in Figure 1.20 can be expressed to compute the heating of the fluid between sections 1 and 2. It follows that

$$\dot{m}\left(h_1 + \frac{C_1^2}{2} + g\,z_1\right) + \dot{W} + \dot{Q}_{amb} = \dot{m}\left(h_2 + \frac{C_2^2}{2} + g\,z_2\right) \tag{1.128}$$

Substituting h by $u + pv$ and expressing the variation of internal energy as function of the variation of temperature for an ideal liquid, it follows that

$$\dot{m}\,c\,(T_2 - T_1) = \dot{m}\left(v\,P_1 + \frac{C_1^2}{2} + g\,z_1\right) - \dot{m}\left(v\,P_2 + \frac{C_2^2}{2} + g\,z_2\right) + \eta\dot{W} + (1-\eta)\dot{W} + \dot{Q}_{amb} \tag{1.129}$$

Combining this equation with Bernoulli's equation yields

$$\dot{m}\,c\,(T_2 - T_1) = \dot{E}_{mech,loss} + (1-\eta)\dot{W} + \dot{Q}_{amb} \tag{1.130}$$

In the particular case of a closed circuit (such as the coolant loop of an engine), having section 1 corresponding to section 2, the generalized Bernoulli's equation reduces to Eq. (1.131), meaning that the minimum power of the fan/pump overcomes the rate of mechanical energy loss through the circuit.

$$\eta\,\dot{W} = \dot{E}_{mech,loss} \tag{1.131}$$

Also, if the closed circuit is in the steady-state regime, the ambient losses must balance the fluid heating-up due to the rate of mechanical energy loss and fan/pump inefficiency.

$$\dot{E}_{mech,loss} + (1-\eta)\dot{W} = -\dot{Q}_{amb} \tag{1.132}$$

As mentioned earlier, mechanical energy can be lost through two types of irreversibilities: frictional losses along the walls of the pipe and losses in valves, elbows, etc. The former losses correspond to friction pressure losses ($\Delta P_{friction}$) and the latter ones to local pressure losses (ΔP_{local}).

The friction pressure losses can be calculated by

$$\Delta P_{friction} = f\,\frac{L}{D_h}\,\frac{C^2}{2\,v} \tag{1.133}$$

In the latter equation, f is the friction factor. For laminar flows, $f = Constant/Re$. For turbulent flows, the friction factor also depends on the Reynolds number and whether the walls are smooth or rough. In the latter case, the friction factor is function of the relative roughness: $f = f(\varepsilon/D_h)$. Such relations can be easily found in fluid mechanics textbooks.

The local pressure losses can be assessed by the following relationship involving a local loss coefficient K_{local} that is function of the type of piping singularities:

$$\Delta P_{local} = K_{local} \frac{C^2}{2\,v} \tag{1.134}$$

All the previous developments will be useful when investigating the aeraulics associated with a vehicle front-end module (Chapter 2).

1.8 Heat Exchangers

A heat exchanger is a device that transfers thermal energy from a hot fluid to a cold fluid. There could be some variants to this general definition:

- The heat transfer can occur between a solid surface and a fluid or between solid particulates and a fluid (Shah and Sekulić, 2003).
- Mass transfer can occur simultaneously with heat transfer.
- The heat exchanger can involve more than two fluids.
- The heat exchanger can involve internal heat production or combustion or chemical reactions (Shah and Sekulić, 2003).

Heat exchangers are key components in the vehicle overall thermal management system. Actually, the latter comprises several fluid loops that exchange heat among themselves (e.g. oil to coolant, coolant to oil, coolant to air, refrigerant to air, and air to refrigerant). Different technologies of heat exchanger are used, depending on the nature of the working fluids, operating conditions, and weight/packaging constraints. The performance of the heat exchangers is of paramount importance to optimize the performance of the whole thermal management system. It will be shown later that this performance is expressed in terms of both thermal and hydraulic performance.

1.8.1 Classification of Heat Exchangers

There exist many technologies of heat exchangers. Several classifications can be used to emphasize the major characteristics of each of them.

1.8.1.1 Classification According to the Mechanism of Energy Transfer

1.8.1.1.1 Heat Exchangers Without Mass Transfer Most of the heat exchangers only exchange thermal energy ("heat"). In such heat exchangers, both fluids are generally separated by a wall, which is called "primary surface." This wall prevents any mixing of both fluids.

1.8.1.1.2 Heat Exchangers with Mass Transfer Heat exchangers that also allow for mass transfer between both fluids do not comprise any separating wall (or the wall is porous). This is the case of direct contact cooling towers, which find a lot of applications in stationary HVAC and industry. In such cooling towers, water to be cooled down is in direct contact with air. Heat and mass are exchanged between both fluids. Some liquid water is actually vaporized to increase the vapor content of humid air. In a cooling tower, as the temperature of water decreases, the wet-bulb temperature of the air increases. Example of porous heat exchangers are "enthalpy" heat exchangers that allow for the heat and water vapor transfer between a stream of hot and humid air and a stream of cold and dry air.

1.8.1.1.3 Heat Exchangers with Intermediate Energy Storage (Regenerators) Another distinction can be done within all mechanisms of energy transfer. In most heat exchangers, heat (and possibly mass) is directly transferred from one fluid to the other. There exist heat exchangers, named "regenerators," where heat (and possibly mass) is stored in an intermediate way in a medium during its transfer from one fluid to the other. This medium is usually a solid matrix. In this type of heat exchanger, the hot fluid flows through the matrix that stores the heat. Later in time, the cold fluid flows through the same matrix that releases its heat. Hence, both fluids flow in the same space, but at different positions in time. Regenerators combine the function of heat transfer and heat storage. Such heat exchangers are widely used for ventilation heat recovery in buildings.

1.8.1.2 Classification According to the Phases of Both Fluids

In conventional heat exchangers, where fluids are separated by a wall, only thermal energy is transferred from a fluid to another. The fluids can be in liquid or gas phase, or they can experience a phase change when flowing through the heat exchanger. Examples of heat exchangers with different fluid phases applied to automotive thermal management are provided in Table 1.4.

In the specific case of air-heated evaporators, the heat exchanger is fed by humid air on one side and refrigerant on the other side. If the temperature of the separating wall on the air-side (called "contact temperature") is decreased underneath the air dew-point, water vapor in the air condenses. If the contact temperature is negative, liquid water solidifies. This yields to the formation of frost, a mixture of ice and air.

1.8.1.3 Classification According to the Flow Arrangement

Heat exchangers can show counter-flow, parallel-flow, and cross-flow configurations.

In *counter-flow* heat exchangers, illustrated in Figure 1.21, the hot and cold fluids enter the heat exchanger through opposite ends. The fluids flow parallel to each other and in opposite directions. The exhaust temperature of the cold fluid can be larger than that of the hot fluid (Incropera and DeWitt, 2002).

Table 1.4 Classification of the heat exchangers according to the phases of the fluids – Example of applications in the automotive domain.

Phase of fluid 1–phase of fluid 2	Example of applications
Gas–gas	Air-cooled charge air cooler (Chapter 2)
Liquid–gas	Air-cooled oil cooler (Chapter 2)
	Water-cooled charge air cooler (Chapter 2)
	Water-cooled exhaust gas recirculation cooler (Chapter 2)
Liquid–liquid	Water-cooled oil cooler (Chapter 2)
Two-phase–liquid	Water-cooled condenser (Chapter 3)
	Water-heated evaporator ("chiller") (Chapter 4)
Two-phase–gas	Air-heated evaporator (Chapter 3)
	Air-cooled condenser (Chapter 3)
Two-phase–two-phase	Air-heated evaporator with water condensation on the air side (Chapter 3)
	Internal heat exchanger if condensation and evaporation occur on both sides (Chapter 3)

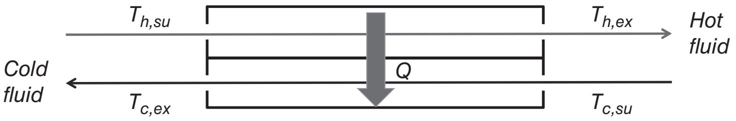

Figure 1.21 Flow arrangement in a counter-flow heat exchanger.

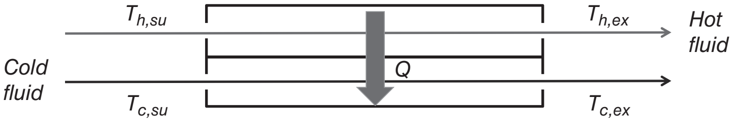

Figure 1.22 Flow arrangements in a parallel-flow heat exchanger.

In *parallel-flow* heat exchangers, illustrated in Figure 1.22, the hot and cold fluids enter and leave the heat exchanger at the same ends. They also flow parallel to each other, but in the same direction. Such heat exchanger allows for “quickly” decreasing the temperature of the hot fluid in the vicinity of the heat exchanger inlet, but also allows to bring the exhaust temperatures of both fluids close to each other (Incropera and DeWitt, 2002).

In *cross-flow* heat exchangers, both fluids flow perpendicular to each other. Within cross-flow configurations, one can also distinguish the cases where both fluids are mixed (a), un-mixed (c), or only one of them is mixed (b). Un-mixed flow means that the overall flow is divided into a number of separate and parallel flows. This can, for instance, represent the air flow on the fin-side of a tube-fin heat exchanger. Mixed and un-mixed flows are two ideal cases. In practice, no flow fits exactly in one or another category (Figure 1.23).

The performance of counter-flow heat exchangers is usually better than that of parallel-flow. Indeed, for the same supply and exhaust temperatures, the mean temperature difference between both fluids is higher. Therefore, for the same heat transfer rate $\dot{Q}$, a lower heat transfer area A is needed assuming the same overall heat transfer coefficient U (Incropera and DeWitt, 2002). The performance of a cross-flow heat exchanger lies between that of parallel-flow and counter-flow heat exchangers.

1.8.1.4 Classification According to the Pass Arrangement

A single-pass flow indicates that the fluid flows only once along the length of the heat exchanger. The fluid can flow several times along the length of the heat exchanger and show a multi-pass arrangement. With a large number of passes, the cross-flow configuration tends to be a counter-flow configuration.

1.8.1.5 Classification According to the Type of Construction

Another classification is based on the type of construction of heat exchangers. One can distinguish prime-surface heat exchangers and extended-surface heat exchangers.

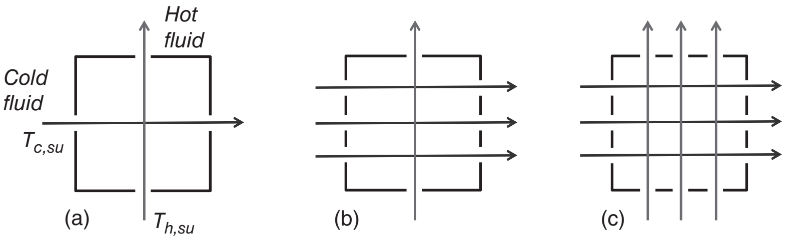

Figure 1.23 Flow arrangements in a cross-flow heat exchanger.

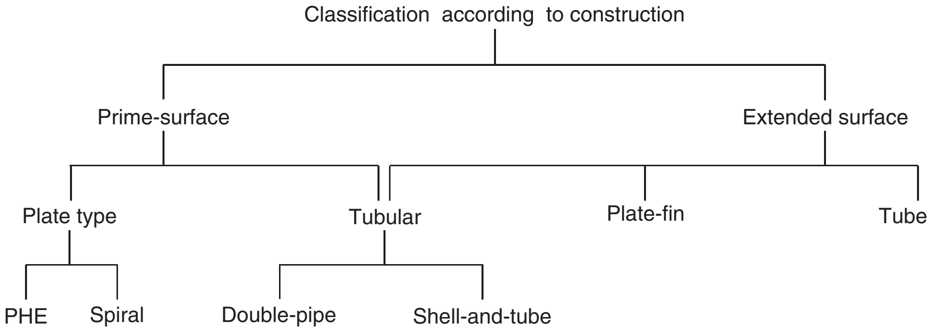

Figure 1.24 Classification of heat exchangers according to the type of construction. Source: Adapted from Shah and Sekulić (2003).

Prime-surface heat exchangers only use the primary surface, i.e. the surface of the dividing wall that separates the fluids passages, as the heat transfer surface.

Extended-surface heat exchangers use fins, also called secondary surface, to increase the heat transfer area. The increase of the heat transfer area can compensate for the decrease of the heat transfer coefficient on one side or both sides of the heat exchanger. Hence, the use of fins reduces the thermal resistance on the side of the heat exchanger where they are employed. It should be mentioned that the presence of fins can either increase or decrease the heat transfer coefficient. Louvered fins, which will be introduced later, create flow interruption and increase the heat transfer coefficient by 2–4 in comparison to plain fins (Shah and Sekulić, 2003).

Figure 1.24 proposes a classification of the heat exchangers most commonly used in vehicle thermal management. It is inspired from Shah and Sekulić (2003) who proposed a much wider classification of heat exchangers according to their construction. Prime-surface heat exchangers include plate type and tubular heat exchangers. Extended surface heat exchangers include plate-fin and tube-fin heat exchangers. Tubular heat exchangers having fins on the inner or outer surfaces of the tubes are classified as extended surface heat exchangers.

The heat exchangers mentioned in the tree structure of Figure 1.24 are described in detail hereunder, with examples of realizations dedicated to vehicle thermal management

1.8.1.5.1 Tubular Heat Exchangers Among the most commonly used tubular heat exchangers, one can mention:

- The double pipe (or “concentric tubes”) heat exchanger. In such configuration, two tubes are nested into each other concentrically. One fluid flows through the inner tube, while the other fluid flows in the annulus section between both tubes.
- The shell-and-tube heat exchanger. It is composed of a bank of tubes bundled in a shell. One of the fluids flows through the tubes, while the other one flows outside the tubes. Baffles can be used to enhance the convective coefficient of the fluid flowing outside the tubes and to support the bank of tubes. Examples of different architectures of shell-and-tube heat exchangers used as exhaust gas recirculation (EGR) coolers (the role of which is explained in Chapter 2) are given in Figure 1.25.

1.8.1.5.2 Plate Heat Exchangers Plate heat exchangers are made of a stack of thin plates. The assembly of plates creates parallel thin channels for the working fluids. Usually, each plate has four ports at the corners. All these ports form distribution headers for the two fluids (Shah and Sekulić, 2003). These headers allow for two adjacent channels to be fed by different fluids.

Cylindrical housing

Rectangular housing

Figure 1.25 Shell-and-tube heat exchangers used as EGR coolers. (a) Cutaway photograph of a heat exchanger with a cylindrical housing and round tubes. (b) Components of a heat exchanger with rectangular housing and flat tubes. Source: Courtesy of Valeo.

Plates are smoothed, corrugated, or embossed (i.e. with protrusions). The corrugations on the successive plates ensure the support of the plate stack through a large series of contact points (Shah and Sekulić, 2003). Also, corrugations create highly interrupted and tortuous channels, which yield high level of turbulence (Shah and Sekulić, 2003). This high turbulence reduces fouling of the heat exchanger. Also, high turbulence combined with low hydraulic diameters yields high heat transfer coefficients. An example of plate heat exchanger used as an oil cooler (the role of which is explained in Chapter 2) is given in Figure 1.26.

Plates are sealed around their edges by gaskets, welding, or brazing. Gasketed plate heat exchangers are not adapted to corrosive fluids. Moreover, the use of gaskets limits the operating pressures

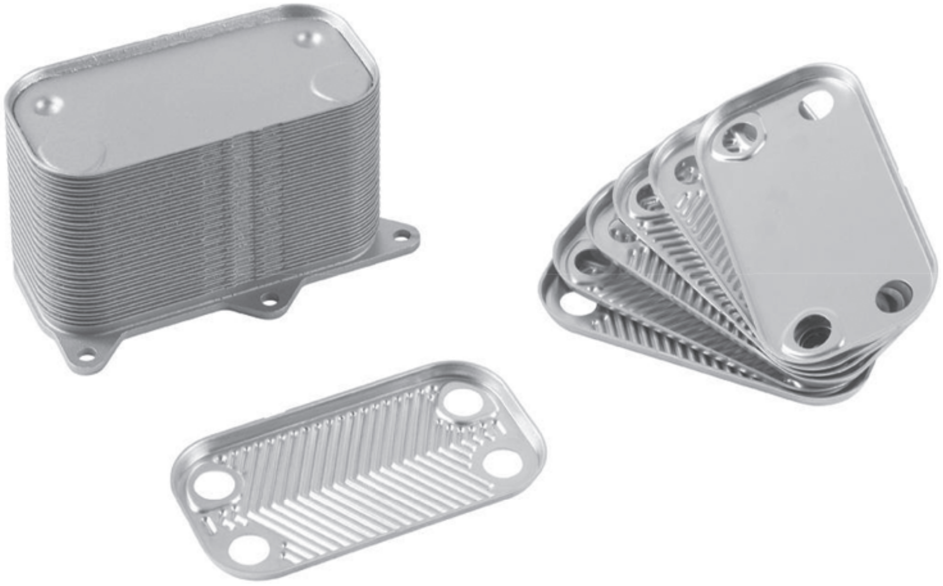

Figure 1.26 Plates from a plate heat exchanger used as coolant-to-oil cooler. Source: Courtesy of Valeo.

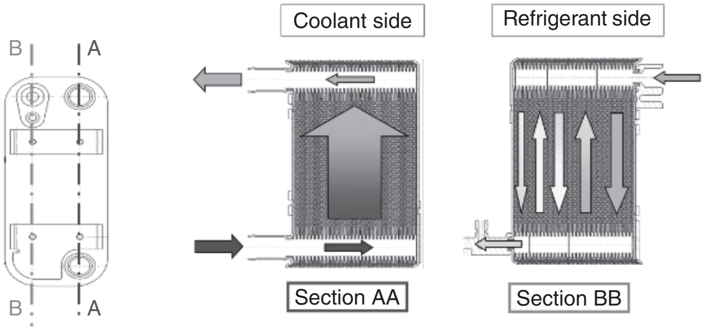

Figure 1.27 Multi-pass configuration of a BPHEX. Source: Adapted from Valeo.

and temperatures. Such limitations are partly overcome with welded or brazed plate heat exchangers. Such heat exchangers are perfectly tight but cannot be opened for servicing.

Usually, in plate heat exchangers, each fluid shows one single pass, and the flow configuration is purely counterflow. Reduced fouling combined with pure counterflow configuration and large heat transfer coefficients allow for a reduction of the heat transfer area with respect to shell-and-tube heat exchangers. Also plate heat exchangers are lighter and more compact.

It should be mentioned that some plate heat exchangers are characterized by hybrid flow configurations. This is illustrated in Figure 1.27, which represents a water-cooled condenser with one single pass on the coolant side and five passes on the refrigerant side. Regarding the latter passes, the first, third, and fifth ones are in the counter-flow configuration. The second and fourth refrigerant passes are in the parallel-flow configuration. In general, increasing the number of passes increases the heat exchanger compactness.

1.8.1.5.3 Plate-Fin Heat Exchangers Plate-fin heat exchangers are made of a stack of plates with corrugated fins sandwiched in between. There are different types of fins: they could have triangular or rectangular cross-sections, and they could be plain or multilouvered or wavy (Figure 1.28).

Among the plate-fin heat exchangers, one can mention the bar and plate heat exchangers. Such heat exchangers are composed of parting sheets (plates), corrugated fins, and side bars. The fins act as spacers between the plates. Side bars are used to form a flow channel between two plates. Such heat exchangers are mechanically robust. An example of bar and plate heat exchanger used as charge air cooler (CAC) (the role of which is explained in Chapter 2) is given in Figure 1.29.

On one fluid side, the combinations of two plates and two side bars can be replaced by flat extruded tubes. In such configuration of plate-fin heat exchangers, fins are sandwiched between the flat tubes and attached to them by brazing. In automotive applications, liquid or two-phase fluid flows inside the tubes, while air flows outside the tubes on the finned side. The presence of fins compensates for the low convection heat transfer coefficient on the air side by increasing the heat transfer area.

1.8.1.5.4 Tube-Fin Heat Exchangers In this configuration, one of the fluids flows through the tubes, while the other one flows at the outside of the tubes and is in contact with fins. The latter fluid is usually a gas such as air.

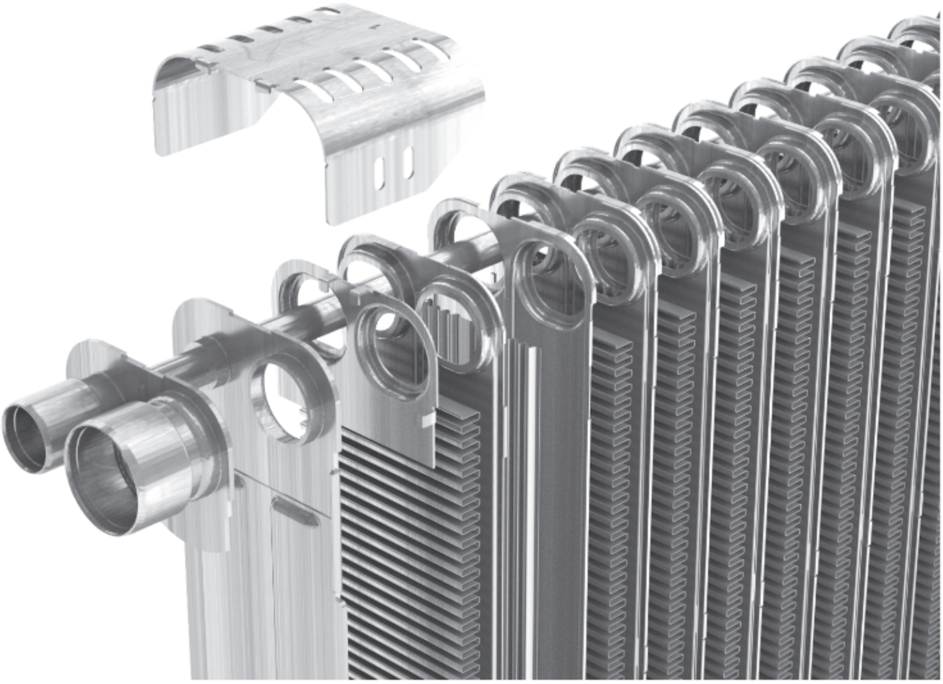

Figure 1.28 Components of a plate-fin heat exchanger used as an evaporator. Source: Courtesy of Valeo.

Different shapes of tubes are used, such as round, oval, or peanut. Either the tubes have individual fins or a matrix of tubes share continuous fins that can be plain, wavy, multilouvered, etc. This will be illustrated in Chapter 2 for radiators. Also the inner surface of the tubes can have fins or corrugations to decrease the thermal resistance of the fluid flowing inside the tubes. This is illustrated in Figure 1.30.

The different technologies of heat exchangers show different compactness. The compactness is defined in terms of heat transfer area per unit envelope volume of the heat exchanger. Compact heat exchangers show a compactness factor larger than 700 $m^2\ m^{-3}$ (Incropera and De Witt, 2002).

1.8.2 Energy Balance Across a Heat Exchanger

Before introducing the indicators of the performance of a heat exchanger, it is relevant to establish the energy balance on a heat exchanger. For all types of heat exchangers, the energy balances can be derived from the First Law of Thermodynamics applied to the control volume (Eq. (1.96)). Several assumptions are introduced:

- The heat exchanger is in the steady-flow regime.
- There is no heat transfer between the heat exchanger and the ambient.
- The variations of the kinetic and potential energies of both the cold and hot fluids between the supply and the exhaust of the heat exchanger are negligible.
- Only two fluids travel through the heat exchanger, even though there exist heat exchangers with more than two fluids.
- No work is produced or consumed by the heat exchanger. Note that thermoelectric generators can be seen as heat exchangers producing electricity.

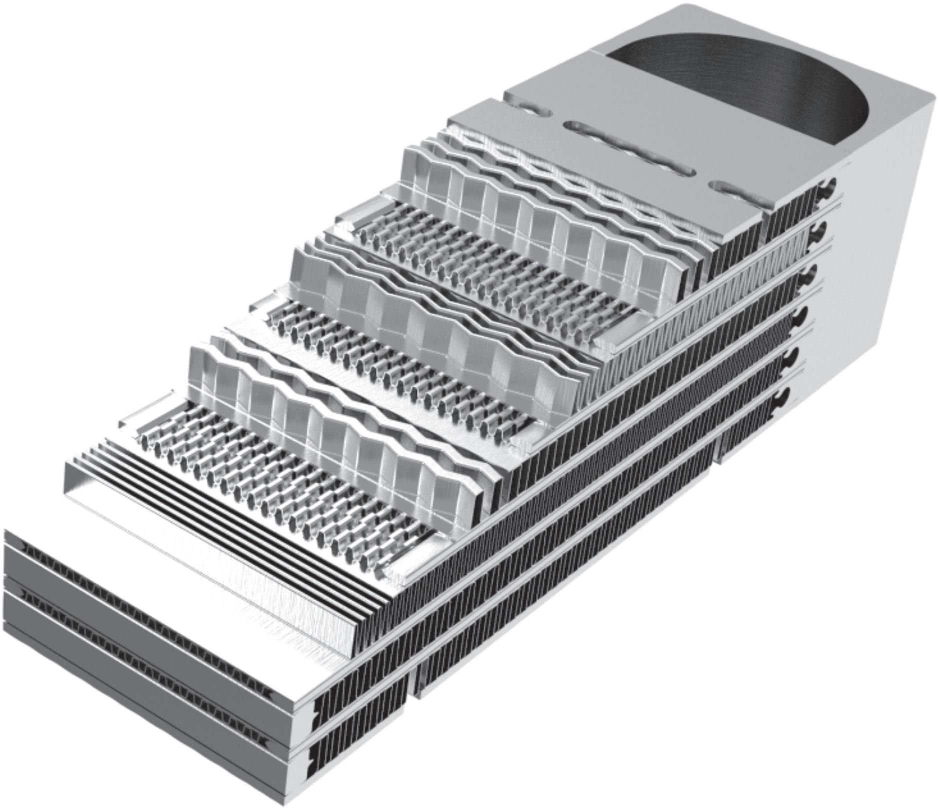

Figure 1.29 Cutaway photograph of a bar and plate heat exchanger used as a charge air cooler. Source: Courtesy of AKG.

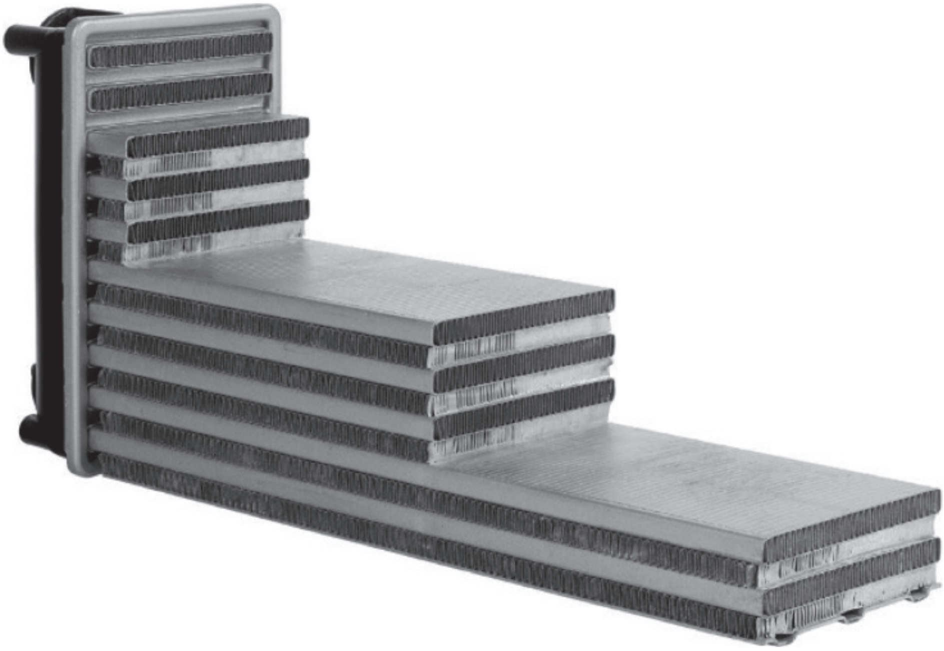

Figure 1.30 Cutaway photograph of a (flat) tube-fin heat exchanger used as charge air cooler. Source: Courtesy of Valeo.

Under all these assumptions, Eq. (1.96) applied to the control volume comprising both fluids reduces to

$$\dot{m}_h \cdot (h_{h,su} - h_{h,ex}) = \dot{m}_c \cdot (h_{c,ex} - h_{c,su}) \tag{1.135}$$

where

$\dot{m}_h$ is the hot fluid mass flow rate, [kg s^{-1}]
$\dot{m}_c$ is the cold fluid mass flow rate, [kg s^{-1}]
$h_{h,su}$ is the specific enthalpy of the hot fluid at the heat exchanger supply, [J kg^{-1}]
$h_{h,ex}$ is the specific enthalpy of the hot fluid at the heat exchanger exhaust, [J kg^{-1}]
$h_{c,su}$ is the specific enthalpy of the cold fluid at the heat exchanger supply, [J kg^{-1}]
$h_{c,ex}$ is the specific enthalpy of the cold fluid at the heat exchanger exhaust, [J kg^{-1}].

To express heat transfer rate $\dot{Q}$ $[W]$ between the hot and the cold fluids, Eq. (1.96) must be applied to control volumes comprising one single fluid. This gives

$$\dot{Q} = \dot{m}_h \cdot (h_{h,su} - h_{h,ex}) \tag{1.136}$$

$$\dot{Q} = \dot{m}_c \cdot (h_{c,ex} - h_{c,su}) \tag{1.137}$$

In the absence of phase change and assuming that the specific heats on both fluid sides are constant (and equal to the average value between the supply and exhaust temperatures) during heat transfer, one gets:

$$\dot{Q} = \dot{m}_h \cdot c_{p,h} \cdot (T_{h,su} - T_{h,ex}) = \dot{m}_c \cdot c_{p,c} \cdot (T_{c,ex} - T_{c,su}) \tag{1.138}$$

This equation can also be written in the following way by introducing the capacity flow rate $\dot{C}$, which is the product of the mass flow rate times the specific heat.

$$\dot{Q} = \dot{C}_h \cdot (T_{h,su} - T_{h,ex}) = \dot{C}_c \cdot (T_{c,ex} - T_{c,su}) \tag{1.139}$$

1.8.3 Performance

Performance of heat exchangers can be quantified in terms of energy-related indicators (thermal performance and hydraulic performance), economic indicators (e.g. the cost), compactness indicators (compactness factor, weight, etc.), or reliability. This section only focuses on the energy-related indicators owing to their impact on the energy performance of the thermal management systems.

1.8.3.1 Thermal Performance

The energy balance across the heat exchanger (Eq. (1.135)–(1.137)) does not bring information about its thermal performance. More specifically, the previous energy balance equations cannot predict the heat transfer rate $\dot{Q}$ knowing the capacity flow rates and the supply temperatures of both fluids. Another equation involving an intrinsic characteristic of the heat exchanger is necessary. This characteristic is overall heat transfer conductance AU [W K^{-1}], which is the product of heat transfer area A [m^2] and of the overall heat transfer coefficient U [W $m^{-2}K^{-1}$]. The conductance is the reciprocal of the overall heat transfer resistance R [K W^{-1}].

The overall heat transfer conductance AU [W K^{-1}] of the heat exchanger can be defined independent of the side of the heat exchanger, which is not the case of the overall heat transfer coefficient U [W $m^{-2}K^{-1}$]. Considering Figure 1.31, one can write

$$\frac{1}{AU} = \frac{1}{A_h U_h} = \frac{1}{A_c U_c} = R \tag{1.140}$$

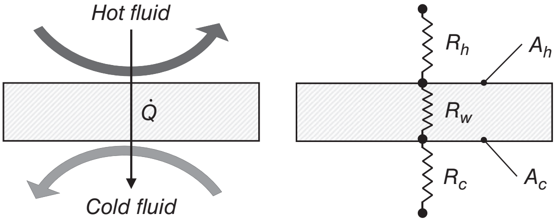

Figure 1.31 Desegregation of the heat exchanger conductance into thermal resistances.

where

A_h is the area of the hot surface of the heat exchanger, [m^2]
A_c is the area of the cold surface of the heat exchanger, [m^2]
U_h is the overall heat transfer coefficient defined on the hot fluid side, [W m^{-2}K^{-1}]
U_c is the overall heat transfer coefficient defined on the cold fluid side, [W m^{-2}K^{-1}].

The overall thermal resistance between the hot and cold fluids can be seen as the association of several resistances in series:

- the convection resistance R_h between the hot fluid and the heat exchanger hot surface,
- the conduction resistance R_w between the hot and cold surfaces through the heat exchanger wall,
- the convection resistance R_c between the cold fluid and the heat exchanger cold surface.

Associating these resistances yields

$$\frac{1}{AU} = R_h + R_w + R_c \tag{1.141}$$

Assuming that both the hot and the cold surfaces of the heat exchanger are finned, Eq. (1.141) can be developed as (Incropera and DeWitt, 2002)

$$\frac{1}{AU} = \frac{1}{(\eta_0\, A\, h)_h} + R_w + \frac{1}{(\eta_0\, A\, h)_c} \tag{1.142}$$

where

η_0 is the overall surface efficiency, [–]
h is the convective heat transfer coefficient, [Wm^{-2}K^{-1}].

The overall surface efficiency η_0 can be related to the efficiency of a single fin η_f by

$$\eta_0 = 1 - \frac{A_f}{A}(1 - \eta_f) \tag{1.143}$$

where

A_f is the fin surface area, [m^2]
A is the total surface area (exposed base and fin), [m^2].

Hence, enlarging the heat transfer area (and thus the size of the heat exchanger), increasing the convective heat transfer coefficients on both fluid sides, and limiting the conduction resistance of the wall separating the two fluids can increase the heat exchanger conductance.

Two methods are commonly used for computing the heat transfer rate $\dot{Q}$ as a function of the heat exchanger conductance AU: the log-mean temperature difference method and the epsilon-NTU method. It can be shown that both methods are algebraically equivalent.

1.8.3.1.1 Log-Mean Temperature Difference (LMTD) Method In this method, the heat transfer rate $\dot{Q}$ [W] is expressed by the product of the conductance AU [W] and the log-mean temperature difference ΔT_{lm} [K]. The log-mean temperature difference is the average temperature difference between the hot and cold fluids across the heat exchanger. Therefore,

$$\dot{Q} = A \cdot U \cdot \Delta T_{lm} \tag{1.144}$$

For a counter-flow heat exchanger, the log-mean temperature difference is

$$\Delta T_{lm} = \frac{(T_{h,su} - T_{c,ex}) - (T_{h,ex} - T_{c,su})}{\ln\left(\frac{(T_{h,su} - T_{c,ex})}{(T_{h,ex} - T_{c,su})}\right)} \tag{1.145}$$

For a parallel-flow heat exchanger, the log-mean temperature difference becomes

$$\Delta T_{lm} = \frac{(T_{h,su} - T_{c,su}) - (T_{h,ex} - T_{c,ex})}{\ln\left(\frac{(T_{h,su} - T_{c,su})}{(T_{h,ex} - T_{c,ex})}\right)} \tag{1.146}$$

For a cross-flow heat exchanger or a shell-and-tube heat exchanger with multiple passes, the log-mean temperature difference is computed by applying a correction factor to the log-mean temperature difference obtained as if the heat exchanger was counter-flow (the reader is invited to refer to Incropera and DeWitt (2002) for more information).

1.8.3.1.2 Epsilon-NTU ($\varepsilon -$ NTU) Method

The *LMTD* method is well adapted to problems where the 4 temperatures are known, for instance, to identify the necessary conductance for a given heat transfer rate. This is the case during the design phase of a heat exchanger. If only two temperatures are known (at least one of each fluid; the supply temperatures for instance), an iterative procedure must be implemented. In that situation, the $\varepsilon - NTU$ method is more adapted. Here, the heat transfer rate is computed as a fraction ε of the maximal heat transfer rate. The latter is equal to the minimum capacity flow rate multiplied by the maximal temperature difference.

$$\dot{Q} = \varepsilon \cdot \dot{Q}_{max} = \varepsilon \cdot \dot{C}_{min} \cdot (T_{h,su} - T_{c,su}) \tag{1.147}$$

$$\dot{C}_{min} = \min(\dot{C}_h, \dot{C}_c) \tag{1.148}$$

The effectiveness ε of the heat exchanger can be expressed as a function of the NTU, which is the ratio of the overall thermal conductance and the minimal capacity flow rate. It gives

$$NTU = \frac{AU}{\dot{C}_{min}} \tag{1.149}$$

The effectiveness ε can be expressed as a function of the number of transfer units NTU [−] and the capacity ratio C_r[−] for different flow configurations. Expressions are given in Table 1.5.

Table 1.5 Relation between epsilon, NTU, and Cr for different flow configurations.

Counter-flow	$C_r < 1$	$\varepsilon = \frac{1 - \exp(-NTU(1 - C_r))}{1 - C_r \cdot \exp(-NTU(1 - C_r))}$
	$C_r = 1$	$\varepsilon = \frac{NTU}{1 + NTU}$
Parallel flow		$\varepsilon = \frac{1 - \exp(-NTU(1 + C_r))}{1 + C_r}$
Cross-flow	Both fluids mixed	$\varepsilon = \frac{1}{\frac{1}{1-\exp(-NTU)} + \frac{C_r}{1-\exp(-C_r \cdot NTU)} - \frac{1}{NTU}}$
	Both fluids unmixed	$\varepsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right) NTU^{0.22} \cdot (\exp(-C_r \cdot NTU^{0.78}) - 1)\right]$
	$\dot{C}_{\mathrm{max}}$ mixed, $\dot{C}_{\mathrm{min}}$ unmixed	$\varepsilon = \left(\frac{1}{C_r}\right) \cdot [1 - \exp(-C_r(1 - \exp(-NTU)))]$
	$\dot{C}_{min}$ mixed, $\dot{C}_{max}$ unmixed	$\varepsilon = 1 - \exp\left(-\frac{1}{C_r}(1 - \exp(-C_r \cdot NTU))\right)$

The relation $\varepsilon = f(NTU, C_r, flow\ configuration)$ is used to compute the heat transfer rate knowing the capacity flow rates and supply temperatures ($T_{h,su}\ and\ T_{c,su}$) as well as the conductance AU. Similar relations but under the form of $NTU = g(\varepsilon, C_r, flow\ configuration)$ can be used in the frame of heat exchanger design problems.

With the heat capacity ratio C_r defined as:

$$C_r = \frac{\dot{C}_{min}}{\dot{C}_{max}} \tag{1.150}$$

Independent of the construction type of heat exchanger, if $C_r = 0$, then the effectiveness reduces to

$$\varepsilon = 1 - \exp(-NTU) \tag{1.151}$$

The previous relation applies to semi-isothermal heat exchangers such as an air-conditioning evaporator if the vapor superheat zone is neglected (this will be described in Chapter 3). In the case of $C_r = 0$, it can be observed that ε tends to 100% when NTU tends to infinity. For other values of C_r, the limiting value of ε depends on flow configuration and C_r. However, independent of flow configuration and C_r, the pinch point temperature (which is the minimal temperature difference between the two fluids at the same location in the heat exchanger) tends to 0 K when NTU tends to infinity.

It must also be stressed that both the *LMTD* and *epsilon-NTU* methods rely on some assumptions, the major ones of which are: the heat exchanger is insulated, the specific heats of the two fluids are constant, and the overall heat transfer coefficient U is constant along the heat exchanger.

1.8.3.2 Hydraulic Performance

Besides the energy performance, the hydraulic performance of the heat exchanger is another performance indicator to consider. The hydraulic performance can be expressed in terms of pressure drops encountered by the different fluids flowing through the heat exchanger. Limiting these pressure drops allows for limiting the energy consumptions of fans and pumps displacing the fluids.

References

Bell, I.H., Wronski, J., Quoilin, S., and Lemort, V. (2014). Pure and pseudo-pure fluid thermophysical property evaluation and the open-source thermophysical property library coolprop. *Industrial and Engineering Chemistry Research* 53 (6): 2498–2508. https://doi.org/10.1021/ie4033999.

Braun, J.E. and Mitchell, J.W. (2012). *Principles of Heating, Ventilation, and Air Conditioning in Buildings*. Wiley.

Çengel, Y.A. and Boles, M.A. (2006). *Thermodynamics, An Engineering Approach*, 5e. McGraw-Hill Higher Education.

Incropera, F.P. and DeWitt, D.P. (2002). *Fundamentals of Heat and Mass Transfer*, 5e. Wiley.

Klein, S. and Nellis, G. (2016). *Thermodynamics*. Cambridge University Press.

Shah, R.K. and Sekulić, D.P. (2003). *Fundamentals of Heat Exchanger Design*. Wiley.