# 1

# General Aspects of Thermodynamics, Fluid Flow and Heat Transfer

## 1.1 Introduction

Refrigeration is a diverse field and covers a large number of processes ranging from cooling to air conditioning and from food refrigeration to human comfort. Refrigeration as a whole, therefore, appears complicated because of the fact that thermodynamics, fluid mechanics, and heat transfer are always encountered in every refrigeration process or application. For a good understanding of the operation of the refrigeration systems and applications, an extensive knowledge of such topics is indispensable.

When an engineer or an engineering student undertakes the analysis of a refrigeration system and/or its application, he or she should deal with several basic aspects first, depending upon the type of the problem being studied, which may be of thermodynamics, fluid mechanics, or heat transfer. In conjunction with this, there is a need to introduce several definitions and concepts before moving into refrigeration systems and applications in depth. Furthermore, the units are of importance in the analysis of such systems and applications. One should make sure that the units used are consistent to reach the correct result. This means that there are several introductory factors to be taken into consideration to avoid getting lost inside. While the information in some situations is limited, it is desirable that the reader comprehends these processes. Despite assuming that the reader, if he or she is a student, has completed necessary courses in thermodynamics, fluid mechanics, and heat transfer, there is still a need for him or her to review, and for those who are practicing refrigeration engineers, the need is much stronger to understand the physical phenomena and practical aspects, along with a knowledge of the basic laws, principles, governing equations, and related boundary conditions. In addition, this introductory chapter reviews the essentials of such principles, laws, and so on, discusses the relationships between different aspects, and provides some key examples for better understanding.

We now begin with a summary of the fundamental definitions, physical quantities and their units, dimensions, and interrelations. We then proceed directly to the consideration of fundamental topics of thermodynamics, fluid mechanics, and heat transfer.

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### 1.1.1 Systems of Units

Units are accepted as the currency of science. There are two systems: the *International System of Units* (*Le Système International d'Unitès*), which is always referred to as the SI units, and the *English System of Units* (the *English Engineering System*). The SI units are most widely used throughout the world, although the English System is the traditional system of North America. In this book, the SI units are primarily employed.

## 1.2 Thermodynamic Properties

### 1.2.1 Mass, Length and Force

Mass is defined as a quantity of matter forming a body of indefinite shape and size. The fundamental unit of mass is the kilogram (kg) in the SI and its unit in the English System is the pound mass (lbm). The basic unit of time for both unit systems is the second (s). The following relationships exist between the two unit systems:

$1\,\text{kg} = 2.2046\,\text{lbm}$ or $1\,\text{lbm} = 0.4536\,\text{kg}$

$1\,\text{kg/s} = 7936.6\,\text{lbm/h} = 2.2046\,\text{lbm/s}$

$1\,\text{lbm/h} = 0.000126\,\text{kg/s}$

$1\,\text{lbm/s} = 0.4536\,\text{kg/s}$

In thermodynamics the unit *mole* (mol) is commonly used and defined as a certain amount of substance containing all the components. The related equation is

$$n = \frac{m}{M} \tag{1.1}$$

if $m$ and $M$ are given in grams and gram/mol, we get $n$ in mol. If the units are in kilogram and kilogram/kilomole, $n$ is given in kilomole (kmol). For example, 1 mol of water, having a molecular weight of 18 (compared to 12 for carbon-12), has a mass of 0.018 kg and for 1 kmol, it becomes 18 kg.

The basic unit of length is the meter (m) in the SI and the foot (ft) in the English System, which additionally includes the inch (in.) in the English System and the centimeter (cm) in the SI. The interrelations are

$1\,\text{m} = 3.2808\,\text{ft} = 39.370\,\text{in.}$

$1\,\text{ft} = 0.3048\,\text{m}$

$1\,\text{in.} = 2.54\,\text{cm} = 0.0254\,\text{m}$

Force is a kind of action that brings a body to rest or changes the direction of motion (e.g., a push or a pull). The fundamental unit of force is the newton (N).

$1\,\text{N} = 0.22481\,\text{lbf}$ or $1\,\text{lbf} = 4.448\,\text{N}$

The four aspects (i.e., mass, time, length, and force) are interrelated by Newton's second law of motion, which states that the force acting on a body is proportional to the mass and the acceleration in the direction of the force, as given in Equation 1.2:

$$F = ma \tag{1.2}$$

Equation 1.2 shows the force required to accelerate a mass of 1 kg at a rate of $1\ \text{m/s}^2$ as $1\,\text{N} = 1\,\text{kg}\,\text{m/s}^2$.

It is important to note that the value of the earth's gravitational acceleration is 9.80665 m/s$^2$ in the SI system and 32.174 ft/s$^2$ in the English System, and it indicates that a body falling freely toward the surface of the earth is subject to the action of gravity alone.

### *1.2.2 Specific Volume and Density*

Specific volume is the volume per unit mass of a substance, usually expressed in cubic meters per kilogram (m$^3$/kg) in the SI system and in cubic feet per pound (ft$^3$/lbm) in the English System. The *density* of a substance is defined as the mass per unit volume and is therefore the inverse of the specific volume:

$$\rho = \frac{1}{v} \tag{1.3}$$

and its units are kg/m$^3$ in the SI system and lbm/ft$^3$ in the English System. Specific volume is also defined as the volume per unit mass, and density as the mass per unit volume, that is

$$v = \frac{V}{m} \tag{1.4}$$

$$\rho = \frac{m}{V} \tag{1.5}$$

Both specific volume and density are intensive properties and are affected by temperature and pressure. The related interconversions are

$1\,\text{kg/m}^3 = 0.06243\,\text{lbm/ft}^3$ or $1\,\text{lbm/ft}^3 = 16.018\,\text{kg/m}^3$

$1\,\text{slug/ft}^3 = 515.379\,\text{kg/m}^3$

### *1.2.3 Mass and Volumetric Flow Rates*

Mass flow rate is defined as the mass flowing per unit time (kg/s in the SI system and lbm/s in the English system). Volumetric flow rates are given in m$^3$/s in the SI system and ft$^3$/s in the English system. The following expressions can be written for the flow rates in terms of mass, specific volume, and density:

$$\dot{m} = \dot{V}\rho = \frac{\dot{V}}{v} \tag{1.6}$$

$$\dot{V} = \dot{m}v = \frac{\dot{m}}{\rho} \tag{1.7}$$

### *1.2.4 Pressure*

When we deal with liquids and gases, pressure becomes one of the most important components. Pressure is the force exerted on a surface per unit area and is expressed in bar or Pascal (Pa). 1 bar is equal to 10$^5$ Pa. The related expression is

$$P = \frac{F}{A} \tag{1.8}$$

The unit for pressure in the SI denotes the force of 1 N acting on 1 m$^2$ area (so-called *Pascal*) as follows:

$1\,\text{Pascal (Pa)} = 1\,\text{N/m}^2$

The unit for pressure in the English System is pounds force per square foot, $\text{lbf/ft}^2$. The following are some of the pressure conversions:

$$1\ \text{Pa} = 0.020886\ \text{lbf/ft}^2 = 1.4504 \times 10^{-4}\ \text{lbf/in.}^2 = 4.015 \times 10^{-3}\ \text{in water} = 2.953 \times 10^{-4}\ \text{in Hg}$$

$$1\ \text{lbf/ft}^2 = 47.88\ \text{Pa}$$

$$1\ \text{lbf/in.}^2 = 1\ \text{psi} = 6894.8\ \text{Pa}$$

$$1\ \text{bar} = 1 \times 10^5\ \text{Pa}$$

Here, we introduce the basic pressure definitions, and a summary of basic pressure measurement relationships is shown in Figure 1.1.

#### 1.2.4.1 Atmospheric Pressure

The atmosphere that surrounds the earth can be considered a reservoir of low-pressure air. Its weight exerts a pressure which varies with temperature, humidity, and altitude. Atmospheric pressure also varies from time to time at a single location, because of the movement of weather patterns. While these changes in barometric pressure are usually less than one-half inch of mercury, they need to be taken into account when precise measurements are essential.

$$1\ \text{standard atmosphere} = 1.0133\ \text{bar} = 1.0133 \times 10^5\ \text{Pa} = 101.33\ \text{kPa} = 0.10133\ \text{MPa}$$
$$= 14.7\ \text{psi} = 29.92\ \text{in Hg} = 760\ \text{mmHg} = 760\ \text{Torr}.$$

#### 1.2.4.2 Gauge Pressure

The *gauge pressure* is any pressure for which the base for measurement is atmospheric pressure expressed as kPa as gauge. Atmospheric pressure serves as reference level for other types of pressure measurements, for example, gauge pressure. As shown in Figure 1.1, the gauge pressure is either positive or negative, depending on its level above or below the atmospheric pressure level. At the level of atmospheric pressure, the gauge pressure becomes zero.

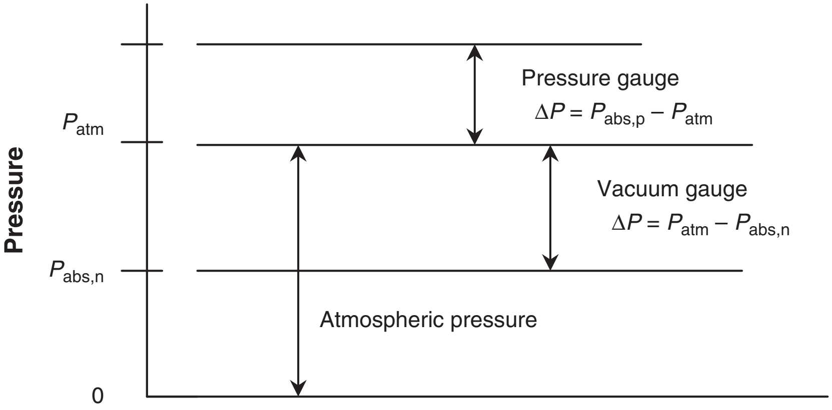

**Figure 1.1** Illustration of pressures for measurement.

### 1.2.4.3 Absolute Pressure

A different reference level is utilized to obtain a value for absolute pressure. The absolute pressure can be any pressure for which the base for measurement is full vacuum, being expressed in kPa as absolute. In fact, it is composed of the sum of the gauge pressure (positive or negative) and the atmospheric pressure as follows:

$$\text{kPa (gauge)} + \text{atmospheric pressure} = \text{kPa (absolute)} \tag{1.9}$$

For example, to obtain the absolute pressure, we simply add the value of atmospheric pressure of 101.33 kPa at sea level. The absolute pressure is the most common one used in thermodynamic calculations despite the pressure difference between the absolute pressure and the atmospheric pressure existing in the gauge being read by most pressure gauges and indicators.

### 1.2.4.4 Vacuum

A vacuum is a pressure lower than the atmospheric one and occurs only in closed systems, except in outer space. It is also called the *negative gauge pressure*. As a matter of fact, vacuum is the pressure differential produced by evacuating air from the closed system. Vacuum is usually divided into four levels: (i) low vacuum representing pressures above 1 Torr absolute (a large number of mechanical pumps in industry are used for this purpose; flow is viscous), (ii) medium vacuum varying between 1 and $10^{-3}$ Torr absolute (most pumps serving in this range are mechanical; fluid is in transition between viscous and molecular), (iii) high vacuum ranging between $10^{-3}$ and $10^{-6}$ Torr absolute (nonmechanical ejector or cryogenic pumps are used; flow is molecular or Newtonian), and (iv) very high vacuum representing absolute pressure below $10^{-6}$ Torr (primarily for laboratory applications and space simulation).

A number of devices are available to measure fluid (gaseous or liquid) pressure and vacuum values in a closed system and require the fluid pressure to be steady for a reasonable length of time. In practice, the most common types of such gauges are the following:

- **Absolute pressure gauge.** This is used to measure the pressure above a theoretical perfect vacuum condition and the pressure value is equal to ($P_{\text{abs,p}} - P_{\text{atm}}$) in Figure 1.1. The most basic type of such gauges is the barometer. Another type of gauge used for vacuum measurements is the U-shaped gauge. The pressure value read is equal to ($P_{\text{atm}} - P_{\text{abs,n}}$) in Figure 1.1.
- **Mercury U-tube manometer.** These manometers use a column of liquid to measure the difference between two pressures. If one is atmospheric pressure, the result is a direct reading of positive or negative gauge pressure.
- **Plunger gauge.** This gauge consists of a plunger connected to system pressure, a bias spring, and a calibrated indicator. An auto tire gauge would be an example.
- **Bourdon gauge.** This is the most widely utilized instrument for measuring positive pressure and vacuum. Measurements are based on the determination of an elastic element (a curved tube) by the pressure being measured. The radius of curvature increases with increasing positive pressure and decreases with increasing vacuum. The resulting deflection is indicated by a pointer on a calibrated dial through a ratchet linkage. Similar gauges may be based on the deformation of diaphragms or other flexible barriers.
- **McLeod gauge.** This is the most widely used vacuum-measuring device, particularly for extremely accurate measurements of high vacuums.

Among these devices, two principal types of measuring devices for refrigeration applications are manometers and Bourdon gauges. However, in many cases manometers are not preferred because of the excessive length of tube needed, inconvenience at pressures much in excess of 1 atm, and less accuracy.

There are also pressure transducers available, based on the effects of capacitance, rates of change of strain, voltage effects in a piezoelectric crystal, and magnetic properties (Marquand and Croft, 1997). All have to be calibrated and the only calibration possible is against a manometer under steady conditions, even though they are most likely to be used under dynamic conditions.

It is important to note at another additional level that the *saturation pressure* is the pressure of a liquid or vapor at saturation conditions.

### *1.2.5 Temperature*

Temperature is an indication of the thermal energy stored in a substance. In other words, we can identify hotness and coldness with the concept of temperature. The temperature of a substance may be expressed in either relative or absolute units. The two most common temperature scales are the Celsius (°C) and the Fahrenheit (°F). As a matter of fact, the Celsius scale is used with the SI unit system and the Fahrenheit scale with the English Engineering system of units. There are also two more scales: the Kelvin scale (K) and the Rankine scale (R) that is sometimes employed in thermodynamic applications. The relations between these scales are summarized as follows:

$$T_{(^\circ\mathrm{C})} = \frac{T_{(^\circ\mathrm{F})} - 32}{1.8} \tag{1.10}$$

$$T_{(\mathrm{K})} = T_{(^\circ\mathrm{C})} + 273.15 = \frac{T_{(\mathrm{R})}}{1.8} = \frac{T_{(^\circ\mathrm{F})} + 459.67}{1.8} \tag{1.11}$$

$$T_{(^\circ\mathrm{F})} = 1.8T_{(^\circ\mathrm{C})} + 32 = 1.8(T_{(\mathrm{K})} - 273.15) + 32 \tag{1.12}$$

$$T_{(\mathrm{R})} = 1.8T_{(\mathrm{K})} = T_{(^\circ\mathrm{F})} + 459.67 \tag{1.13}$$

Furthermore, the temperature differences result in

$$1\,\mathrm{K} = 1\,^\circ\mathrm{C} = 1.8\,\mathrm{R} = 1.8\,^\circ\mathrm{F}$$

$$1\,\mathrm{R} = 1\,^\circ\mathrm{F} = 1\,\mathrm{K}/1.8 = 1\,^\circ\mathrm{C}/1.8$$

Kelvin is a unit of temperature measurement; zero Kelvin (0 K) is absolute zero and is equal to −273.15 °C. The K and °C are equal increments of temperature. For instance, when the temperature of a product is decreased to −273.15 °C (or 0 K), known as *absolute zero*, the substance contains no heat energy and supposedly all molecular movement stops. The saturation temperature is the temperature of a liquid or vapor at saturation conditions.

Temperature can be measured in many ways by devices. In general, the following devices are in common use:

- **Liquid-in-glass thermometers.** It is known that in these thermometers the fluid expands when subjected to heat, thereby raising its temperature. It is important to note that in practice all thermometers including mercury ones only work over a certain range of temperature. For example, mercury becomes solid at −38.8 °C and its properties change dramatically.
- **Resistance thermometers.** A resistance thermometer (or detector) is made of resistance wire wound on a suitable former. The wire used has to be of known, repeatable, electrical characteristics so that the relationship between the temperature and resistance value can be predicted precisely. The measured value of the resistance of the detector can then be used to determine the value of an unknown temperature. Among metallic conductors, pure metals exhibit the greatest change of resistance with temperature. For applications requiring higher accuracy,

especially where the temperature measurement is between −200 and +800 °C, the majority of such thermometers are made of platinum. In industry, in addition to platinum, nickel (−60 to +180 °C), and copper (−30 to +220 °C) are frequently used to manufacture resistance thermometers. Resistance thermometers can be provided with 2, 3, or 4 wire connections and for higher accuracy at least 3 wires are required.

- **Averaging thermometers.** An averaging thermometer is designed to measure the average temperature of bulk stored liquids. The sheath contains a number of elements of different lengths, all starting from the bottom of the sheath. The longest element which is fully immersed is connected to the measuring circuit to allow a true average temperature to be obtained. There are some significant parameters, namely, sheath material (stainless steel for the temperature range from −50 to +200 °C or nylon for the temperature range from −50 to +90 °C), sheath length (to suit the application), termination (flying leads or terminal box), element length, element calibration (to copper or platinum curves), and operating temperature ranges. In many applications where a multielement thermometer is not required, such as in air ducts, cooling water, and gas outlets, a single element thermometer stretched across the duct or pipework will provide a true average temperature reading. Despite the working range from 0 to 100 °C, the maximum temperature may reach 200 °C. To keep high accuracy these units are normally supplied with 3-wire connections. However, up to 10 elements can be mounted in the averaging bulb fittings and they can be made of platinum, nickel or copper, and fixed at any required position.
- **Thermocouples.** A thermocouple consists of two electrical conductors of different materials connected together at one end (so-called *measuring junction*). The two free ends are connected to a measuring instrument, for example, an indicator, a controller, or a signal conditioner, by a reference junction (so-called *cold junction*). The thermoelectric voltage appearing at the indicator depends on the materials of which the thermocouple wires are made and on the temperature difference between the measuring junction and the reference junction. For accurate measurements, the temperature of the reference junction must be kept constant. Modern instruments usually incorporate a cold junction reference circuit and are supplied ready for operation in a protective sheath, to prevent damage to the thermocouple by any mechanical or chemical means. Table 1.1 gives several types of thermocouples along with their maximum absolute temperature ranges. As can be seen in Table 1.1, copper–constantan thermocouples have an accuracy of ±1 °C and are often employed for control systems in refrigeration and food-processing applications. The iron–constantan thermocouple with its maximum temperature of 850 °C is used in applications in

**Table 1.1** Some of the most common thermocouples.

| Type | Common Names | Temperature Range (°C) |
|---|---|---|
| T | Copper–Constantan (C/C) | −250 to 400 |
| J | Iron–Constantan (I/C) | −200–850 |
| E | Nickel Chromium–Constantan or Chromel–Constantan | −200–850 |
| K | Nickel Chromium–Nickel Aluminum or Chromel–Alumel (C/A) | −180–1100 |
| – | Nickel 18% Molybdenum–Nickel | 0–1300 |
| N | Nicrosil–Nisil | 0–1300 |
| S | Platinum 10% Rhodium–Platinum | 0–1500 |
| R | Platinum 13% Rhodium–Platinum | 0–1500 |
| B | Platinum 30% Rhodium–Platinum 6% Rhodium | 0 to 1600 |

the plastics industry. The chromel–alumel type thermocouples, with a maximum temperature of about 1100 °C, are suitable for combustion applications in ovens and furnaces. In addition, it is possible to reach about 1600 or 1700 °C using platinum and rhodium–platinum thermocouples, particularly in steel manufacture. It is worth noting that one advantage thermocouples have over most other temperature sensors is that they have a small thermal capacity and thus a prompt response to temperature changes. Furthermore, their small thermal capacity rarely affects the temperature of the body under examination.

- **Thermistors.** These devices are semiconductors and act as thermal resistors with a high (usually negative) temperature coefficient. Thermistors are either self-heated or externally heated. Self-heated units employ the heating effect of the current flowing through them to raise and control their temperature and thus their resistance. This operating mode is useful in such devices as voltage regulators, microwave power meters, gas analyzers, flow meters, and automatic volume and power level controls. Externally heated thermistors are well suited for precision temperature measurement, temperature control, and temperature compensation due to large changes in resistance versus temperature. These are generally used for applications in the range $-100$ to $+300$ °C. Despite early thermistors having tolerances of ±20% or ±10%, modern precision thermistors are of higher accuracy, for example, ±0.1 °C (less than ±1%).
- **Digital display thermometers.** A wide range of digital display thermometers, for example, handheld battery powered displays and panel mounted mains or battery units, are available in the market. Figure 1.2 shows a handheld digital thermometer with protective boot (with a high accuracy, e.g., ±0.3% reading ± 1.0 °C). Displays can be provided for use with all standard thermocouples or BS/DIN platinum resistance thermometers with several digits and 0.1 °C resolution.

It is very important to emphasize that before temperature can be controlled, it must be sensed and measured accurately. For temperature measurement devices, there are several potential sources of error such as sensor properties, contamination effects, lead lengths, immersion, heat transfer, and controller interfacing. In temperature control there are many sources of error which can be

**Figure 1.2** Handheld digital thermometers (*Courtesy of* Brighton Electronics, Inc.).

**Figure 1.3** A data acquisition system for temperature measurements in cooling.

minimized by careful consideration of the type of sensor, its working environment, the sheath or housing, extension leads, and the instrumentation. An awareness of potential errors is vital in the applications dealt with. Selection of temperature measurement devices is a complex task and has been discussed briefly here. It is extremely important to remember to "choose the right tool for the right job." Data acquisition devices are commonly preferred for experimental measurements. Figure 1.3 shows a data acquisition system set-up for measuring temperatures during heating and cooling applications.

### *1.2.6 Thermodynamic Systems*

These are devices or combination of devices that contain a certain quantity of matter being studied. It is important to carefully define the term *system* as that portion of all matter under consideration. There are three systems that we can define as follows:

- **Closed system.** This is defined as a system across the boundaries of which no material crosses. In other words, it is a system that has a fixed quantity of matter, so that no mass can escape or enter. In some books, it is also called a *control mass*.
- **Open system.** This is defined as a system in which material (mass) is allowed to cross its boundaries. It is also called a *control volume*.
- **Isolated system.** This is a closed system that is not affected by the surroundings at all, in which no mass, heat, or work crosses its boundary.

### *1.2.7 Process and Cycle*

A process is a physical or chemical change in the properties of matter or the conversion of energy from one form to another. Several processes are described by the fact that one property remains constant. The prefix iso- is employed to describe a process, such as an isothermal process (a constant-temperature process), an isobaric process (a constant-pressure process), and an isochoric process (a constant-volume process). A refrigeration process is generally expressed by the conditions or properties of the refrigerant at the beginning and end of the process.

A cycle is a series of thermodynamic processes in which the endpoint conditions or properties of the matter are identical to the initial conditions. In refrigeration, the processes required to produce a cooling effect are arranged to operate in a cyclic manner so that the refrigerant can be reused.

### *1.2.8 Property and State Postulate*

This is a physical characteristic of a substance used to describe its state. Any two properties usually define the state or condition of the substance, from which all other properties can be derived. This is called *state postulate*. Some examples are temperature, pressure, enthalpy, and entropy. Thermodynamic properties are classified as intensive properties (independent of the mass, e.g., pressure, temperature, and density) and extensive properties (dependent on the mass, e.g., mass and total volume). Extensive properties per unit mass become intensive properties such as specific volume. Property diagrams of substances are generally presented in graphical form and summarize the main properties listed in the refrigerant tables.

### *1.2.9 Sensible Heat, Latent Heat and Latent Heat of Fusion*

It is known that all substances can hold a certain amount of heat; this property is their thermal capacity. When a liquid is heated, the temperature of the liquid rises to the boiling point. This is the highest temperature that the liquid can reach at the measured pressure. The heat absorbed by the liquid in raising the temperature to the boiling point is called *sensible heat*. The heat required to convert the liquid to vapor at the same temperature and pressure is called *latent heat*. In fact, it is the change in enthalpy during a state change (the amount of heat absorbed or rejected at constant temperature at any pressure, or the difference in enthalpies of a pure condensable fluid between its dry saturated state and its saturated liquid state at the same pressure).

Fusion is the melting of a material. For most pure substances there is a specific melting/freezing temperature, relatively independent of the pressure. For example, ice begins to melt at 0 °C. The amount of heat required to melt 1 kg of ice at 0 °C to 1 kg of water at 0 °C is called the *latent heat of fusion of water* and equals 334.92 kJ/kg. The removal of the same amount of heat from 1 kg of water at 0 °C changes it back to ice.

### *1.2.10 Vapor States*

A vapor is a gas at or near equilibrium with the liquid phase – a gas under the saturation curve or only slightly beyond the saturated vapor line. *Vapor quality* is theoretically assumed; that is, when vapor leaves the surface of a liquid it is pure and saturated at the particular temperature and pressure. In actuality, tiny liquid droplets escape with the vapor. When a mixture of liquid and vapor exists, the ratio of the mass of the liquid to the total mass of the liquid and vapor mixture is called the *quality* and is expressed as a percentage or decimal fraction.

*Superheated vapor* is the saturated vapor to which additional heat has been added, raising the temperature above the boiling point. Let us consider a mass ($m$) with a quality ($x$). The volume is the sum of those of the liquid and the vapor as defined below:

$$V = V_{\mathrm{liq}} + V_{\mathrm{vap}} \tag{1.14}$$

It can also be written in terms of specific volumes as

$$mv = m_{\text{liq}}\, v_{\text{liq}} + m_{\text{vap}}\, v_{\text{vap}} \tag{1.15}$$

Dividing all terms by the total mass results in

$$v = (1 - x)v_{\text{liq}} + x v_{\text{vap}} \tag{1.16}$$

and

$$v = v_{\text{liq}} + x v_{\text{liq,vap}} \tag{1.17}$$

where $v_{\text{liq,vap}} = v_{\text{vap}} - v_{\text{liq}}$.

### *1.2.11 Thermodynamic Tables*

Thermodynamic tables were first published in 1936 as steam tables by Keenan and Keyes, and later were revised and republished in 1969 and 1978. The use of thermodynamic tables of many substances ranging from water to several refrigerants is very common in process design calculations. In literature they are also called either steam tables or vapor tables. In this book, we refer to them as thermodynamic tables. These tables are normally given in different distinct phases (parts), for example, four different parts for water such as saturated water, superheated vapor water, compressed liquid water, and saturated solid–saturated vapor water; and two distinct parts for R-134a such as saturated and superheated. Each table is listed according to the values of temperature and pressure and the rest contains the values of four other thermodynamic parameters such as specific volume, internal energy, enthalpy, and entropy. When we normally have two variables, we may obtain the other data from the respective table. In learning how to use these tables, the most important point is to specify the state by any two of the parameters. In some design calculations, if we do not have the exact values of the parameters, we should make an interpolation to find the necessary values. Some people find this disturbing. However, further practice will provide sufficient confidence to do so. Beside these thermodynamic tables, recently, much attention has been paid to the computerized tables for such design calculations. Of course, despite the fact that this eliminates several reading problems, the students may not well understand the concepts and comprehend the subject. That is why in thermodynamics courses it is a must for the students to know how to obtain the thermodynamic data from the respective thermodynamic tables. The *Handbook of Thermodynamic Tables* by Raznjevic (1995) is one of the most valuable sources for several solids, liquids, and gaseous substances.

### *1.2.12 State and Change of State*

The state of a system or substance is defined as the condition of the system or substance characterized by certain observable macroscopic values of its properties such as temperature and pressure. The term *state* is often used interchangeably with the term *phase*, for example, solid phase or gaseous phase of a substance. Each of the properties of a substance in a given state has only one definite value, regardless of how the substance reached the state. For example, when sufficient heat is added or removed, most substances undergo a state change. The temperature remains constant until the state change is complete. This can be from solid to liquid, liquid to vapor, or vice versa. Figure 1.4 shows the typical examples of ice melting and water boiling.

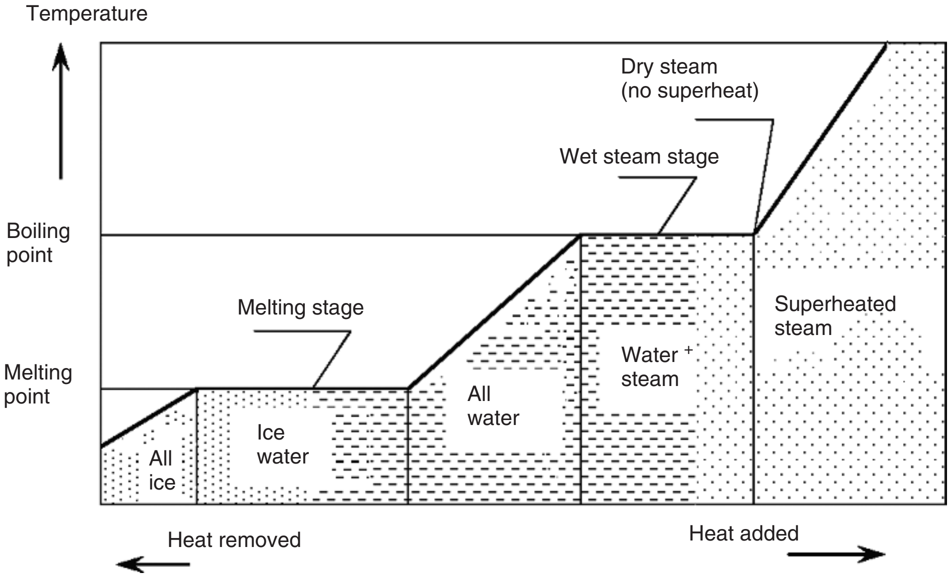

**Figure 1.4** The state-change diagram of water.

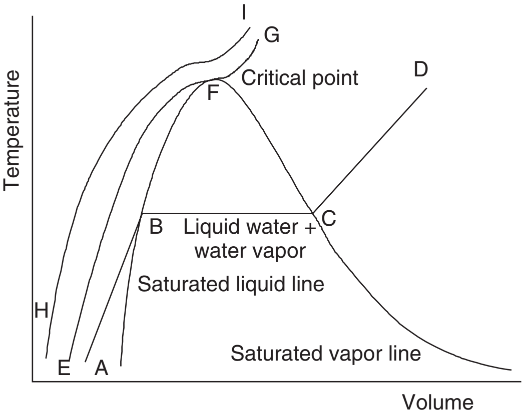

**Figure 1.5** Temperature–volume diagram for the phase change of water.

A clearer presentation of solid, liquid, and vapor phases of water is exhibited on a temperature–volume ($T-v$) diagram in Figure 1.5. The constant-pressure line ABCD represents the states which water passes through as follows:

- **A–B.** This represents the process where water is heated from the initial temperature to the saturation temperature (liquid) at constant pressure. At point B it is fully saturated liquid water with a quality $x = 0$, with zero quantity of water vapor.
- **B–C.** This is the constant-temperature vaporization process in which there is only phase change from saturated liquid to saturated vapor, referring to the fact that the quality varies from 0 to

100%. Within this zone, the water is a mixture of liquid water and water vapor. At point C it is completely saturated vapor and the quality is 100%.

- **C–D.** This represents the constant-pressure process in which the saturated water vapor is superheated with increasing temperature.
- **E–F–G.** In this line there is no constant-temperature vaporization process. The point F is called the *critical point* where the saturated liquid and saturated vapor states are identical. The thermodynamic properties at this point are called *critical thermodynamic properties*, for example, critical temperature, critical pressure, and critical specific volume.
- **H–I.** This is a constant-pressure heating process in which there is no phase change from one phase to another (only one is present); however, there is a continuous change in density.

The other process which may occur during melting of water is *sublimation* in which the ice directly passes from the solid phase to vapor phase. Another important point that needs to be emphasized is that the solid, liquid, and vapor phases of water may be present together in equilibrium, leading to the *triple point*.

### 1.2.13 Pure Substance

This is defined as a substance which has a homogeneous and invariable chemical composition. Despite having the same chemical composition, it may be in more than one phase, namely, liquid water, a mixture of liquid water and water vapor (steam), and a mixture of ice and liquid water. Each one has the same chemical composition. However, a mixture of liquid air and gaseous air cannot be considered a pure substance because of the fact that the composition of each phase differs. A thorough understanding of the pure substance is of significance, particularly for air-conditioning applications. Thermodynamic properties of water and steam can be taken from tables and charts, in almost all thermodynamic books, based on the experimental data or real-gas equations of state through computer calculations. It is important to note that the properties of low-pressure water are of great significance in air conditioning, since water vapor existing in the atmosphere typically exerts a pressure less than 1 psi (6.9 kPa). At such low pressures, it is known that water vapor shows ideal gas behavior.

### 1.2.14 Specific Heats

The energy required to change (to raise or to drop) the temperature of a unit mass of a substance by a unit temperature difference is called the *specific heat* $c$. Its unit is kJ/kg · K or kJ/kg · °C. The specific heat is called the constant-pressure specific heat ($c_p$) if the process takes place at constant pressure (e.g., heating or cooling a gas in a piston-cylinder device). It is called the constant-volume specific heat ($c_v$) if the process takes place at constant volume (e.g., heating or cooling a gas in a rigid tank).

### 1.2.15 Specific Internal Energy

This represents the molecular state type of energy and is a measure of the energy of a simple system in equilibrium as a function of $c_v\,dT$. In fact, for many thermodynamic processes in closed systems the only significant energy changes are internal energy changes, and the significant work done by the system in the absence of friction is the work of pressure–volume expansion such as in a piston–cylinder mechanism. The specific internal energy of a mixture of liquid and vapor can be written in a form similar to Equations 1.16 and 1.17:

$$u = (1 - x)u_{\text{liq}} + xu_{\text{vap}} \tag{1.18}$$

and

$$u = u_{\text{liq}} + xu_{\text{liq,vap}} \tag{1.19}$$

where $u_{\text{liq,vap}} = u_{\text{vap}} - u_{\text{liq}}$.

### *1.2.16 Specific Enthalpy*

This is a measure of the heat energy per unit mass of a substance, usually expressed in kJ/kg, as a function of $c_p\ dT$. Since enthalpy is a state function, it is necessary to measure it relative to some reference state. The usual practice is to determine the reference values which are called the *standard enthalpy of formation* (or the heat of formation), particularly in combustion thermodynamics. The specific enthalpy of a mixture of liquid and vapor components can be written as

$$h = (1 - x)h_{\text{liq}} + xh_{\text{vap}} \tag{1.20}$$

and

$$h = h_{\text{liq}} + xh_{\text{liq,vap}} \tag{1.21}$$

where $h_{\text{liq,vap}} = h_{\text{vap}} - h_{\text{liq}}$.

### *1.2.17 Specific Entropy*

Entropy is a property resulting from the second law of thermodynamics (SLT). This is the ratio of the heat added to a substance to the absolute temperature at which it was added and is a measure of the molecular disorder of a substance at a given state. The unit of entropy is kJ/K and the unit of specific entropy is kJ/kg · K.

The entropy change of a pure substance between the states 1 and 2 is expressed as

$$\Delta s = s_2 - s_1 \tag{1.22}$$

The specific entropy of a mixture of liquid and vapor components can be written as

$$s = (1 - x)s_{\text{liq}} + xs_{\text{vap}} \tag{1.23}$$

and

$$s = s_{\text{liq}} + xs_{\text{liq,vap}} \tag{1.24}$$

where $s_{\text{liq,vap}} = s_{\text{vap}} - s_{\text{liq}}$.

The entropy change of an incompressible substance (solids and liquids) is given by

$$s_2 - s_1 = c \ln \frac{T_2}{T_1} \tag{1.25}$$

where $c$ is the average specific heat of the substance.

An isentropic (i.e., constant entropy) process is defined as a reversible and adiabatic process.

$$s_2 = s_1 \tag{1.26}$$

## 1.3 Ideal Gases

In many practical thermodynamic calculations, gases such as air and hydrogen can often be treated as ideal gases, particularly for temperatures much higher than their critical temperatures and for pressures much lower than their saturation pressures at given temperatures. Such an ideal gas can be described in terms of three parameters, the volume that it occupies, the pressure that it exerts, and its temperature. As a matter of fact, all gases or vapors, including water vapor, at very low pressures show ideal gas behavior. The practical advantage of taking real gases to be ideal is that a simple equation of state with only one constant can be applied in the following form:

$$Pv = RT \tag{1.27}$$

and

$$PV = mRT \tag{1.28}$$

The ideal gas equation of state was originally established from the experimental observations and is also called the $P{-}v{-}T$ relationship for gases. It is generally considered as a concept rather than a reality. It only requires a few data to define a particular gas over a wide range of its possible thermodynamic equilibrium states.

The gas constant ($R$) is different for each gas depending on its molecular weight ($M$):

$$R = \frac{\overline{R}}{M} \tag{1.29}$$

where $\overline{R} = 8.314\,\text{kJ/kmol}\cdot\text{K}$ is the universal gas constant.

Equations 1.27 and 1.28 may be written in a mole-basis form as follows:

$$P\overline{v} = RT \tag{1.30}$$

and

$$PV = n\overline{R}T \tag{1.31}$$

The other simplification is that, if it is assumed that the constant-pressure and constant-volume specific heats are constant, changes in the specific internal energy and the specific enthalpy can be simply calculated without referring to the thermodynamic tables and graphs from the following expressions:

$$\Delta u = (u_2 - u_1) = c_v(T_2 - T_1) \tag{1.32}$$

$$\Delta h = (h_2 - h_1) = c_p(T_2 - T_1) \tag{1.33}$$

The following is another useful expression for ideal gases, obtained from the expression $h = u + Pv = u + RT$:

$$c_p - c_v = R \tag{1.34}$$

For the entire range of states, the ideal gas model may be found unsatisfactory. Therefore, the compressibility factor ($Z$) is introduced to measure the deviation of a real gas from the ideal gas equation of state, which is defined by the following relation:

$$Pv = ZRT \quad \text{or} \quad Z = \frac{Pv}{RT} \tag{1.35}$$

Figure 1.6 shows a generalized compressibility chart for simple substances. In the chart, we have two important parameters: reduced temperature ($T_\mathrm{r} = T/T_\mathrm{c}$) and reduced pressure ($P_\mathrm{r} = P/P_\mathrm{c}$).

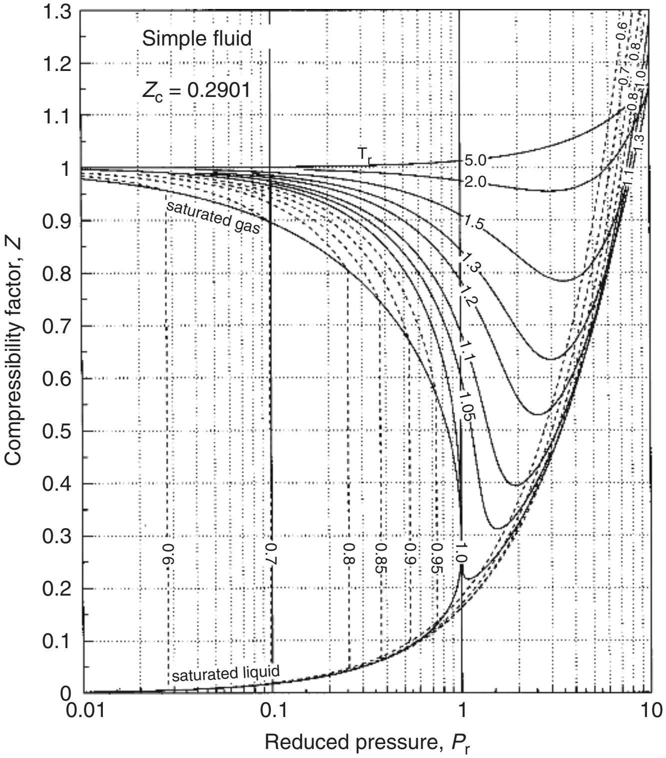

**Figure 1.6** Generalized compressibility chart for simple substances (Borgnakke, and Sonntag, 2008).

Therefore, in order to calculate the compressibility factor the values of $T_r$ and $P_r$ should be calculated using the critical temperature and pressure values of the respective substance which can easily be taken from thermodynamics books. As can be seen in Figure 1.6, at all temperatures $Z \to 1$ as $P_r \to 0$. This means that the behavior of the actual gas closely approaches the ideal gas behavior, as the pressure approaches zero. For real gases, $Z$ takes values between 0 and 1. If $Z = 1$, Equation 1.35 becomes Equation 1.27. In the literature, there are also several equations of state for accurately representing the $P-v-T$ behavior of a gas over the entire superheated vapor region, namely the Benedict–Webb–Rubin equation, van der Waals equation, and Redlich and Kwong equation. However, some of these equations of state are complicated because of the number of empirical constants, and require computer software to get the results.

There are some special cases where $P$, $v$, or $T$ is constant. At a fixed temperature, the volume of a given quantity of ideal gas varies inversely with the pressure exerted on it (in some books this is called *Boyle's law*), describing compression as

$$P_1V_1 = P_2V_2 \tag{1.36}$$

where the subscripts refer to the initial and final states.

Equation 1.36 is employed by designers in a variety of situations: when selecting an air compressor, for calculating the consumption of compressed air in reciprocating air cylinders, and for determining the length of time required for storing air. Nevertheless, it may not always be practical because of temperature changes. If temperature increases with compression at a constant pressure, the volume of a gas varies directly with its absolute temperature in $K$ as

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \tag{1.37}$$

If temperature increases at a constant volume, the pressure of a gas this time varies directly with its absolute temperature in $K$ as

$$\frac{P_1}{T_1} = \frac{P_2}{T_2} \tag{1.38}$$

Equations 1.37 and 1.38 are known as Charles' law. If both temperature and pressure change at the same time, the combined ideal gas equation can be written as follows:

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{1.39}$$

For a given mass, since $c_{v0}$ is constant, the internal energy of an ideal gas can be written as a function of temperature:

$$dU = mc_{v0}\, dT \tag{1.40}$$

and the specific internal energy becomes

$$du = c_{v0}\, dT \tag{1.41}$$

The enthalpy equation for an ideal gas, based on $h = u + Pv$, can be written as

$$dH = mc_{p0}\, dT \tag{1.42}$$

and the specific enthalpy then becomes

$$dh = c_{p0}\, dT \tag{1.43}$$

The entropy change of an ideal gas, based on the general entropy equation in terms of $T\,\mathrm{d}s = du + P\,\mathrm{d}v$ and $T\,\mathrm{d}s = dh - v\,\mathrm{d}P$ as well as the ideal gas equation $Pv = RT$, can be obtained in two ways by substituting Equations 1.41 and 1.43:

$$s_2 - s_1 = c_{v0} \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1} \tag{1.44}$$

$$s_2 - s_1 = c_{p0} \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \tag{1.45}$$

For a reversible adiabatic (i.e., isentropic) process the ideal gas equation in terms of the initial and final states under $Pv^k = \text{constant}$ is

$$Pv^k = P_1 v_1^k = P_2 v_2^k \tag{1.46}$$

where $k$ stands for the adiabatic exponent (so-called *specific heat ratio*) as a function of temperature:

$$k = \frac{c_{p0}}{c_{v0}} \tag{1.47}$$

Based on Equation 1.46 and the ideal gas equation, the following expressions can be obtained:

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{v_1}{v_2}\right)^{k-1} \tag{1.48}$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \tag{1.49}$$

$$\left(\frac{P_2}{P_1}\right) = \left(\frac{v_1}{v_2}\right)^{k} \tag{1.50}$$

Note that these equations are obtained under the assumption of constant specific heats.

Let us consider a closed system with ideal gas, undergoing an adiabatic reversible process with a constant specific heat. The work can be derived from the first law of thermodynamics (FLT) equation as follows:

$$W_{1-2} = \frac{mR(T_2 - T_1)}{1-k} = \frac{P_2V_2 - P_1V_1}{1-k} \tag{1.51}$$

Equation 1.51 can also be derived from the general work relation, $W = \int P\,\mathrm{d}V$.

For a reversible polytropic process, the only difference is the polytropic exponent $(n)$ which shows the deviation from a log $P$ and log $V$ diagram, leading to the slope. Therefore, Equations 1.46, 1.48–1.51 can be rewritten with the polytropic exponent under $Pv^n = \text{constant}$ as

$$Pv^n = P_1v_1^n = P_2v_2^n \tag{1.52}$$

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{n/(n-1)} = \left(\frac{v_1}{v_2}\right)^{n} = \left(\frac{V_1}{V_2}\right)^{n} \tag{1.53}$$

$$W_{1-2} = \frac{mR(T_2 - T_1)}{1-n} = \frac{P_2V_2 - P_1V_1}{1-n} \tag{1.54}$$

In order to give a clear idea it is important to show the values of $n$ for four different types of polytropic processes for ideal gases (Figure 1.7) as follows:

- $n = 0$ for isobaric process ($P = 0$),
- $n = 1$ for isothermal process ($T = 0$),
- $n = k$ for isentropic process ($s = 0$),
- $n = \infty$ for isochoric process ($v = 0$).

As is obvious from Figure 1.7, there are two quadrants where $n$ varies from 0 to $\infty$ and where it has a positive value. The slope of any curve drawn is an important consideration when a reciprocating engine or compressor cycle is under consideration.

In thermodynamics a number of problems involve mixtures of different pure substances (i.e., ideal gases). In this regard, it is of importance to understand the related aspects accordingly. Table 1.2 gives a summary of the relevant expressions and two ideal gas models: the Dalton model and Amagat model. In fact, in the analysis it is assumed that each gas is unaffected by the presence of other gases, and each one is treated as an ideal gas. With regard to entropy, it is important to note that increase in entropy is dependent only upon the number of moles of ideal gases and is independent of its chemical composition. Of course, whenever the gases in the mixture are distinguished, the entropy increases.

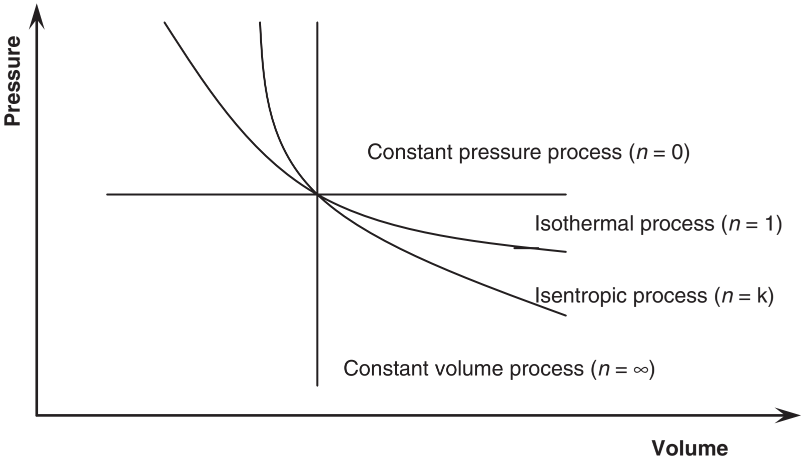

**Figure 1.7** Representation of four different polytropic processes on a pressure–volume diagram.

**Table 1.2** Equations for gas and gas mixtures and relevant models.

<table>
<tr><th>Definition</th><th>Dalton Model</th><th>Amagat Model</th></tr>
<tr><td>Total mass of a mixture of $N$ components</td><td colspan="2">$m_{\text{tot}} = m_1 + m_2 + \cdots + m_N = \sum m_i$</td></tr>
<tr><td>Total number of moles of a mixture of $N$ components</td><td colspan="2">$n_{\text{tot}} = n_1 + n_2 + \cdots + n_N = \sum n_i$</td></tr>
<tr><td>Mass fraction for each component</td><td colspan="2">$c_i = m_i/m_{\text{tot}}$</td></tr>
<tr><td>Mole fraction for each component</td><td colspan="2">$y_i = n_i/n_{\text{tot}} = P_i/P_{\text{tot}} = V_i/V_{\text{tot}}$</td></tr>
<tr><td>Molecular weight for the mixture</td><td colspan="2">$M_{\text{mix}i} = m_{\text{tot}}/n_{\text{tot}} = \sum n_i M_i/n_{\text{tot}} = \sum y_i M_i$</td></tr>
<tr><td>Internal energy for the mixture</td><td colspan="2">$U_{\text{mix}} = n_1\overline{U_1} + n_2\overline{U_2} + \cdots + n_N\overline{U_N} = \sum n_i\overline{U_i}$</td></tr>
<tr><td>Enthalpy for the mixture</td><td colspan="2">$H_{\text{mix}} = n_1\overline{H_1} + n_2\overline{H_2} + \cdots + n_N\overline{H_N} = \sum n_i\overline{H_i}$</td></tr>
<tr><td>Entropy for the mixture</td><td colspan="2">$S_{\text{mix}} = n_1\overline{S_1} + n_2\overline{S_2} + \cdots + n_N\overline{S_N} = \sum n_i\overline{S_i}$</td></tr>
<tr><td>Entropy difference for the mixture</td><td colspan="2">$S_2 - S_1 = -\overline{R}(n_1 \ln y_1 + n_2 \ln y_2 + \cdots + n_N \ln y_N)$</td></tr>
<tr><td>$P$, $V$, $T$ for the mixture</td><td>$T$ and $V$ are constant.<br>$P_{\text{tot}} = P = P_1 + P_2 + \cdots + P_N$</td><td>$T$ and $P$ are constant.<br>$V_{\text{tot}} = V = V_1 + V_2 + \cdots + V_N$</td></tr>
<tr><td>Ideal gas equation for the mixture</td><td>$PV = n\overline{R}T$</td><td></td></tr>
<tr><td>Ideal gas equations for the components</td><td>$P_1V = n_1\overline{R}T$<br>$P_2V = n_2\overline{R}T$<br>$\vdots$<br>$P_NV = n_N\overline{R}T$</td><td>$PV_1 = n_1\overline{R}T$<br>$PV_2 = n_2\overline{R}T$<br>$\vdots$<br>$PV_N = n_N\overline{R}T$</td></tr>
</table>

## 1.4 Energy Change and Energy Transfer

Energy is the capacity for doing work. Energy of a system consists of internal, kinetic, and potential energies. Internal energy consists of thermal (sensible and latent), chemical, and nuclear energies. Unless there is a chemical or nuclear reaction the internal change of a system is due to thermal energy change. The total energy change of a system is expressed as

$$\Delta E = E_2 - E_1 = \Delta U + \Delta \mathrm{KE} + \Delta \mathrm{PE} \tag{1.55}$$

For most cases, the kinetic and potential energies do not change during a process and the energy change is due to internal energy change:

$$\Delta E = \Delta U = m(u_2 - u_1) \tag{1.56}$$

Energy has the unit of kJ or Btu (1 kJ = 0.94782 Btu). Energy per unit time is the rate of energy and is expressed as

$$\dot{E} = \frac{E}{\Delta t} \text{ (kW or Btu/h)} \tag{1.57}$$

The unit of energy rate is kJ/s, which is equivalent to kW or Btu/h (1 kW = 3412.14 Btu/h). Energy per unit mass is called *specific energy*; it has the unit of kJ/kg or Btu/lbm (1 kJ/kg = 0.430 Btu/lbm).

$$e = \frac{E}{m} \text{ (kJ/kg or Btu/lbm)} \tag{1.58}$$

Energy can be transferred to or from a system in three forms: mass, heat, and work. They are briefly described in the following sections.

### 1.4.1 Mass Transfer

The mass entering a system carries energy with it and the energy of the system increases. The mass leaving a system decreases the energy content of the system. When a fluid flows into a system at a mass flow rate of $\dot{m}$ (kg/s), the rate of energy entering is equal to mass times enthalpy $\dot{m}h$ (kW).

### 1.4.2 Heat Transfer

The definitive experiment which showed that heat is a form of energy convertible into other forms was carried out by the Scottish physicist James Joule. Heat is the thermal form of energy and heat transfer takes place when a temperature difference exists within a medium or between different media. Heat always requires a difference in temperature for its transfer. Higher temperature differences provide higher heat-transfer rates.

Heat transfer has the same unit as energy. The symbol for heat transfer is $Q$ (kJ). Heat transfer per unit mass is denoted by $q$ (kJ/kg). Heat transfer per unit time is the rate of heat transfer $\dot{Q}$ (kW). If there is no heat transfer involved in a process, it is called an *adiabatic process*.

### 1.4.3 Work

Work is the energy that is transferred by a difference in pressure or force of any kind and is subdivided into shaft work and flow work. Shaft work is mechanical energy used to drive a mechanism such as a pump, compressor, or turbine. Flow work is the energy transferred into a system by fluid flowing into, or out of, the system. The rate of work transfer per unit time is called *power*. Work has the same unit as energy. Work is denoted by $W$. The direction of heat and work interactions can be expressed by sign conventions or using subscripts such as "in" and "out" (Cengel and Boles, 2008).

## 1.5 The First Law of Thermodynamics

It is simply known that thermodynamics is the science of energy and entropy and that the basis of thermodynamics is experimental observation. In thermodynamics, such observations were formed into four basic laws of thermodynamics called the zeroth, first, second, and third laws of thermodynamics. The first and second laws of thermodynamics are the most common tools in practice, because of the fact that transfers and conversions of energy are governed by these two laws, and in this chapter we focus on these two laws.

The first law of thermodynamics (FLT) can be defined as the law of conservation of energy, and it states that energy can be neither created nor destroyed. It can be expressed for a general system as the net change in the total energy of a system during a process is equal to the difference between the total energy entering and the total energy leaving the system:

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \tag{1.59}$$

In rate form,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \tag{1.60}$$

For a closed system undergoing a process between initial and final states involving heat and work interactions with the surroundings (Figure 1.8),

$$\begin{gathered} E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \\ (Q_{\text{in}} + W_{\text{in}}) - (Q_{\text{out}} + W_{\text{out}}) = \Delta U + \Delta \text{KE} + \Delta \text{PE} \end{gathered} \tag{1.61}$$

If there is no change in kinetic and potential energies,

$$(Q_{\text{in}} + W_{\text{in}}) - (Q_{\text{out}} + W_{\text{out}}) = \Delta U = m(u_2 - u_1) \tag{1.62}$$

Let us consider a control volume involving a steady-flow process. Mass is entering and leaving the system and there is heat and work interactions with the surroundings (Figure 1.9). During a

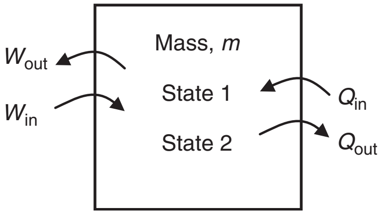

**Figure 1.8** A general closed system with heat and work interactions.

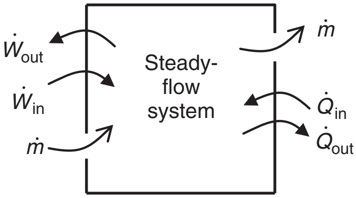

**Figure 1.9** A general steady-flow control volume with mass, heat and work interactions.

steady-flow process, the total energy content of the control volume remains constant, and thus the total energy change of the system is zero. Then the FLT can be expressed as

$$\begin{gathered}\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}} = 0 \\ \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \\ \dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \dot{m}h_{\text{in}} = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \dot{m}h_{\text{out}}\end{gathered} \tag{1.63}$$

Here, the kinetic and potential energies are neglected.

An important consequence of the first law is that the internal energy change resulting from some process will be independent of the thermodynamic path followed by the system, and of the paths followed by the processes, for example, heat transfer and work. In turn, the rate at which the internal energy content of the system changes is dependent only on the rates at which heat is added and work is done.

## 1.6 Refrigerators and Heat Pumps

A refrigerator is a device used to transfer heat from a low- to a high-temperature medium. They are cyclic devices. Figure 1.10a shows the schematic of a vapor-compression refrigeration cycle (the most common type). A working fluid (called *refrigerant*) enters the compressor as a vapor and is compressed to the condenser pressure. The high-temperature refrigerant cools in the condenser by rejecting heat to a high-temperature medium (at $T_H$). The refrigerant enters the expansion valve as liquid. It is expanded in an expansion valve and its pressure and temperature drop. The refrigerant is a mixture of vapor and liquid at the inlet of the evaporator. It absorbs heat from a low-temperature medium (at $T_L$) as it flows in the evaporator. The cycle is completed when the refrigerant leaves the evaporator as a vapor and enters the compressor. The cycle is demonstrated in a simplified form in Figure 1.10b.

An energy balance for a refrigeration cycle, based on the FLT, gives

$$Q_H = Q_L + W \tag{1.64}$$

The efficiency indicator for a refrigeration cycle is coefficient of performance (COP), which is defined as the heat absorbed from the cooled space divided by the work input in the compressor:

$$\text{COP}_{\text{R}} = \frac{Q_L}{W} \tag{1.65}$$

This can also be expressed as

$$\text{COP}_{\text{R}} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{Q_H/Q_L - 1} \tag{1.66}$$

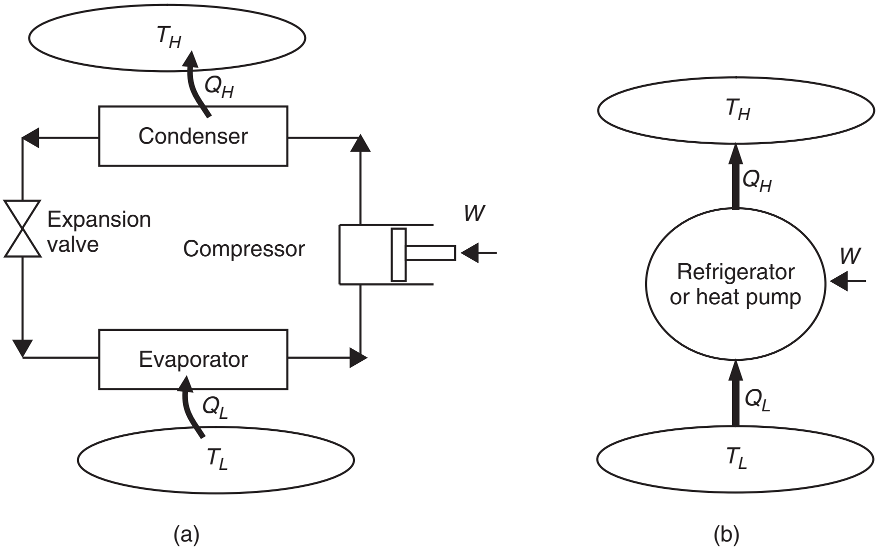

**Figure 1.10** (a) The vapor-compression refrigeration cycle. (b) Simplified schematic of refrigeration cycle.

A heat pump is basically the same device as evaporator. The difference is their purpose. The purpose of a refrigerator is to absorb heat from a cooled space to keep it at a desired low temperature ($T_L$). The purpose of a heat pump is to transfer heat to a heated space to keep it at a desired high temperature ($T_H$). Thus, the COP of a heat pump is defined as

$$\mathrm{COP_{HP}} = \frac{Q_H}{W} \tag{1.67}$$

This can also be expressed as

$$\mathrm{COP_{HP}} = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - Q_L/Q_H} \tag{1.68}$$

It can be easily shown that for given values $Q_L$ and $Q_H$ the COPs of a refrigerator and a heat pump are related to each other by

$$\mathrm{COP_{HP}} = \mathrm{COP_R} + 1 \tag{1.69}$$

This shows that the COP of a heat pump is greater than 1. The COP of a refrigerator can be less than or greater than 1.

## 1.7 The Carnot Refrigeration Cycle

The Carnot cycle is a theoretical model that is useful for understanding a refrigeration cycle. As known from thermodynamics, the Carnot cycle is a model cycle for a *heat engine* where the addition of heat energy to the engine produces work. In some applications, the Carnot refrigeration cycle is known as the *reversed Carnot cycle* (Figure 1.11). The maximum theoretical performance can be calculated, establishing criteria against which real refrigeration cycles can be compared.

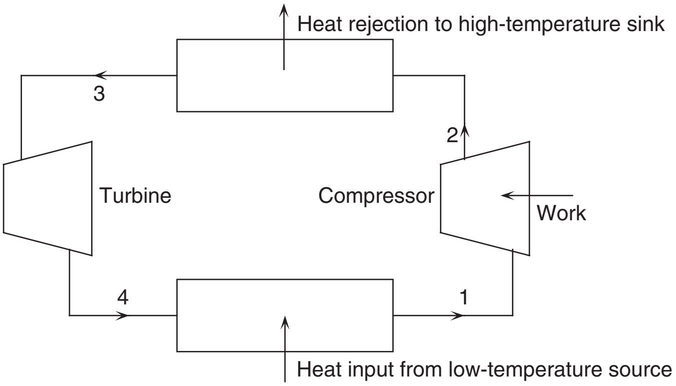

**Figure 1.11** The reversed Carnot refrigeration cycle.

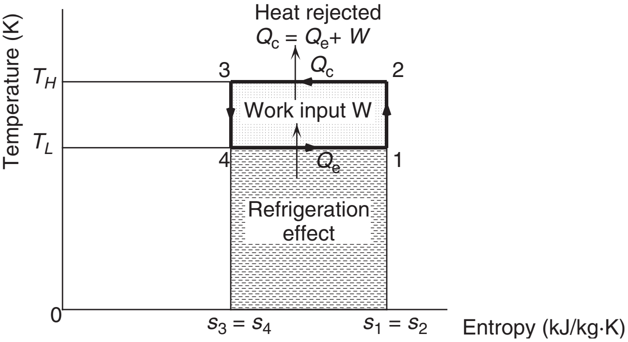

**Figure 1.12** $T-s$ diagram of the Carnot refrigeration cycle.

The following processes take place in the Carnot refrigeration cycle as shown on a temperature–entropy diagram in Figure 1.12:

- (1–2) is the ideal compression at constant entropy, and work input is required. The temperature of the refrigerant increases.
- (2–3) is the rejection of heat in the condenser at a constant condensation temperature, $T_H$.
- (3–4) is the ideal expansion at constant entropy. The temperature of the refrigerant decreases.
- (4–1) is the absorption of heat in the evaporator at a constant evaporation temperature, $T_L$.

The refrigeration effect is represented as the area under the process line 4-1, as follows:

$$Q_L = T_L(s_1 - s_4) \tag{1.70}$$

The theoretical work input (e.g., compressor work) for the cycle is represented as the area within the cycle line 1–2–3–4–1, as follows:

$$W = (T_H - T_L)(s_1 - s_4) \tag{1.71}$$

After inserting Equations 1.70 and 1.71 into Equation 1.65, we find the following equation, which is dependent on the process temperatures:

$$\mathrm{COP}_{\mathrm{R,rev}} = \frac{Q_L}{W} = \frac{Q_L}{Q_H - Q_L} = \frac{T_L}{T_H - T_L} \tag{1.72}$$

It can also be expressed as

$$\mathrm{COP}_{\mathrm{R,rev}} = \frac{1}{Q_H/Q_L - 1} = \frac{1}{T_H/T_L - 1} \tag{1.73}$$

For a reversible heat pump, the following relations apply:

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{Q_H}{W} = \frac{Q_H}{Q_H - Q_L} = \frac{T_H}{T_H - T_L} \tag{1.74}$$

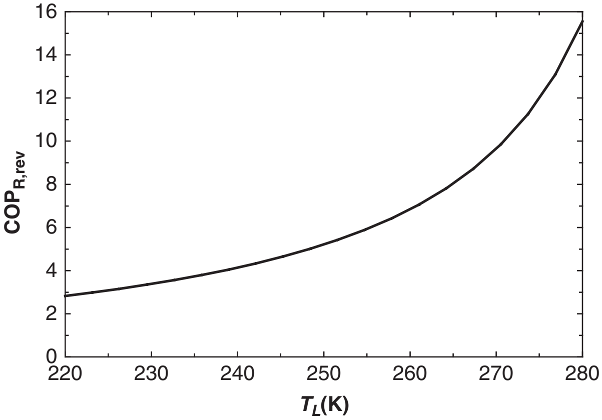

**Figure 1.13** The COP of a reversible refrigerator as a function of $T_L \cdot T_H$ is taken as 298 K.

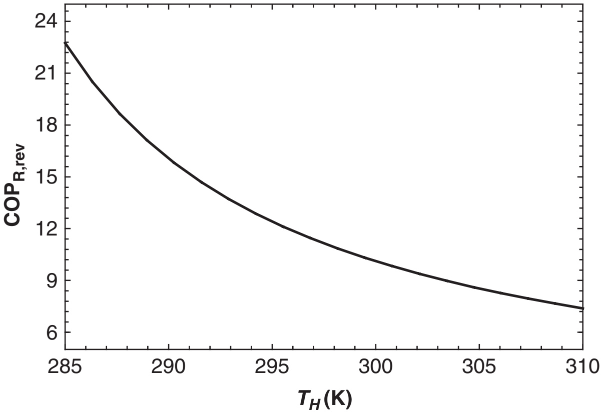

**Figure 1.14** The COP of a reversible refrigerator as a function of $T_H \cdot T_L$ is taken as 273 K.

or

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{1}{1 - Q_L/Q_H} = \frac{1}{1 - T_L/T_H} \tag{1.75}$$

The above relations provide the maximum COPs for a refrigerator or a heat pump operating between the temperature limits of $T_L$ and $T_H$. Actual refrigerators and heat pumps involve inefficiencies and thus they will have lower COPs. The COP of a Carnot refrigeration cycle can be increased by either (i) increasing $T_L$ or (ii) decreasing $T_H$. Figures 1.13 and 1.14 show that the COP of a reversible refrigerator increases with increasing $T_L$ and decreasing $T_H$.

---

## Example 1.1

A refrigeration cycle is used to keep a food department at −15 °C in an environment at 25 °C. The total heat gain to the food department is estimated to be 1500 kJ/h and the heat rejection in the condenser is 2600 kJ/h. Determine (a) the power input to the compressor in kW, (b) the COP of the refrigerator, and (c) the minimum power input to the compressor if a reversible refrigerator was used.

*Solution*

(a) The power input is determined from an energy balance on the refrigeration cycle:

$$\dot{W}_{\mathrm{in}} = \dot{Q}_H - \dot{Q}_L = 2600 - 1500 = 1100\,\mathrm{kJ/h} = (1100\,\mathrm{kJ/h})\left(\frac{1\,\mathrm{kW}}{3600\,\mathrm{kJ/h}}\right) = \mathbf{0.306\,kW}$$

(b) The COP of the refrigerator is

$$\mathrm{COP}_{\mathrm{R}} = \frac{\dot{Q}_L}{\dot{W}_{\mathrm{in}}} = \frac{(1500/3600)\,\mathrm{kW}}{0.306\,\mathrm{kW}} = \mathbf{1.36}$$

(c) The maximum COP of the cycle and the corresponding minimum power input are

$$\mathrm{COP}_{\mathrm{R,\,rev}} = \frac{T_L}{T_H - T_L} = \frac{258}{298 - 258} = 6.45$$

$$\dot{W}_{\mathrm{min}} = \frac{\dot{Q}_L}{\mathrm{COP}_{\mathrm{R,\,rev}}} = \frac{(1500/3600)\ \mathrm{kW}}{6.45} = \mathbf{0.065\,kW}$$

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## 1.8 The Second Law of Thermodynamics

As mentioned earlier, the FLT is the energy-conservation principle. The second law of thermodynamics (SLT) refers to the inefficiencies of practical thermodynamic systems and indicates that it is impossible to have 100% efficiency in heat to work conversion. The classical statements such as the Kelvin–Plank statement and the Clausius statement help us formulate the SLT:

- **The Kelvin–Plank statement.** It is impossible to construct a device, operating in a cycle (e.g., heat engine), that accomplishes only the extraction of heat energy from some source and its complete conversion to work. This simply shows the impossibility of having a heat engine with a thermal efficiency of 100%.
- **The Clausius statement.** It is impossible to construct a device, operating in a cycle (e.g., refrigerator and heat pump), that transfers heat from the low-temperature side (cooler) to the high-temperature side (hotter).

A very easy way to show the implication of both the FLT and the SLT is a desktop game that consists of several pendulums (made of metal balls) in contact with each other. When you raise the first of the balls, you give energy to the system, potential energy. Upon release, this ball gains kinetic energy at the expense of potential energy. When this ball hits the second ball, small elastic deformations transform the kinetic energy again into another form of potential energy. The energy is transferred from one ball to the other. The last one gains kinetic energy to go up again. The cycle continues but every time lower, until it finally stops. The FLT explains why the balls keep moving, but the SLT explains why they do not do it forever. In this game the energy is lost in sound and heat and is no longer useful in keeping the balls in motion.

The SLT also states that the entropy in the universe is increasing. As mentioned before, entropy is the degree of disorder and every process happening in the universe is a transformation from a lower entropy to a higher entropy. Therefore, the entropy of a state of a system is proportional to (depends on) its probability, which gives us opportunity to define the SLT in a broader manner as “the entropy of a system increases in any heat transfer or conversion of energy within a closed system.” That is why all energy transfers or conversions are irreversible. From the entropy perspective, the basis of the SLT is the statement that the sum of the entropy changes of a system and that of its surroundings must be always positive. Recently, much effort has been spent in minimizing the entropy generation (irreversibility) in thermodynamic systems and applications.

Moran and Shapiro (2007) noted that the SLT and deductions from it are useful because they provide means for

- predicting the direction of processes,
- establishing conditions for equilibrium,
- determining the best performance of thermodynamic systems and applications,
- evaluating quantitatively the factors that preclude the attainment of the best theoretical performance level,
- defining a temperature scale, independent of the properties of any thermometric substance, and
- developing tools for evaluating some thermodynamic properties, for example, internal energy and enthalpy using the experimental data available.

Consequently, the SLT is the linkage between entropy and usefulness of energy. The SLT analysis has found applications in a large variety of disciplines, for example, chemistry, economics, ecology, environment, and sociology far removed from engineering thermodynamics applications.

## 1.9 Exergy

The science of thermodynamics is built primarily on two fundamental natural laws, known as the first and the second laws. The FLT is simply an expression of the conservation of energy principle. It asserts that *energy* is a thermodynamic property, and that during an interaction, energy can change from one form to another but the total amount of energy remains constant. The SLT asserts that energy has quality as well as quantity, and actual processes occur in the direction of decreasing quality of energy. The high-temperature thermal energy is degraded as it is transferred to a lower temperature body. The attempts to quantify the quality or “work potential” of energy in the light of the SLT has resulted in the definition of the property named exergy.

Exergy analysis is a thermodynamic analysis technique based on the SLT, which provides an alternative and illuminating means of assessing and comparing processes and systems rationally and meaningfully. In particular, exergy analysis yields efficiencies which provide a true measure of how nearly actual performance approaches the ideal, and identifies more clearly than energy analysis the causes and locations of thermodynamic losses and the impact of the built environment on the natural environment. Consequently, exergy analysis can assist in improving and optimizing designs.

Performance of energy conversion systems and processes is essentially measured by efficiency, except that it becomes coefficient of performance for refrigeration and heat pump systems. There are two thermodynamic efficiencies, namely energy and exergy efficiencies. Although energy efficiency is commonly used by many for performance assessment, exergy efficiency is more beneficial, since it considers irreversibilities, and presents the actual performance of the systems. By considering both of these efficiencies, the quality and quantity of the energy used to achieve a given objective is considered and the degree to which efficient and effective use of energy resources is achieved can be understood. Improving efficiencies of energy systems is an important challenge for meeting energy policy objectives. Reductions in energy use can assist in attaining energy security objectives. Also, efficient energy utilization and the introduction of renewable energy technologies can significantly help solve environmental issues. Increased energy efficiency benefits the environment by avoiding energy use and the corresponding resource consumption and pollution generation. From an economic as well as an environmental perspective, improved energy efficiency has great potential (Dincer and Rosen, 2005).

An engineer designing a system is often expected to aim for the highest reasonable technical efficiency at the lowest cost under the prevailing technical, economic, and legal conditions and with regard to ethical, ecological, and social consequences. Exergy methods can assist in such activities and offer unique insights into possible improvements with special emphasis on environment and sustainability. Exergy analysis is a useful tool for addressing the environmental impact of energy resource utilization and for furthering the goal of more efficient energy resource use, for it enables the locations, types and true magnitudes of losses to be determined. Also, exergy analysis reveals whether and by how much it is possible to design more efficient energy systems by reducing inefficiencies. We present exergy as key tool for systems/processes analysis, design, and performance improvement.

### *1.9.1 What is Exergy?*

The useful work potential of a given amount of energy at a specified state is called *exergy*. It is also called the availability or available energy. The work potential of the energy contained in a system at a specified state, relative to a reference (dead) state, is simply the maximum useful work that can be obtained from the system (Dincer, 2002; 2003).

A system is said to be in the dead state when it is in thermodynamic equilibrium with its environment. At the dead state, a system is at the temperature and pressure of its environment (in thermal and mechanical equilibrium); it has no kinetic or potential energy relative to the environment (zero velocity and zero elevation above a reference level); and it does not react with the environment (chemically inert). Also, there are no unbalanced magnetic, electrical, and surface tension effects between the system and its surroundings, if these are relevant to the situation at hand. The properties of a system at the dead state are denoted by subscript zero, for example, $P_0$, $T_0$, $h_0$, $u_0$, and $s_0$. Unless specified otherwise, the dead-state temperature and pressure are taken to be $T_0 = 25\,^\circ\mathrm{C}$ ($77\,^\circ\mathrm{F}$) and $P_0 = 1$ atm (101.325 kPa or 14.7 psia). A system has zero exergy at the dead state.

The notion that a system must go to the dead state at the end of the process to maximize the work output can be explained as follows: if the system temperature at the final state is greater than (or less than) the temperature of the environment it is in, we can always produce additional work by running a heat engine between these two temperature levels. If the final pressure is greater than (or less than) the pressure of the environment, we can still obtain work by letting the system expand to the pressure of the environment. If the final velocity of the system is not zero, we can catch that extra kinetic energy by a turbine and convert it to rotating shaft work, and so on. No work can be produced from a system that is initially at the dead state. The atmosphere around us contains a tremendous amount of energy. However, the atmosphere is in the dead state, and the energy it contains has no work potential.

Therefore, we conclude that a system delivers the maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state. It is important to realize that exergy does not represent the amount of work that a work-producing device will actually deliver upon installation. Rather, it represents the upper limit on the amount of work a device can deliver without violating any thermodynamic laws. There will always be a difference, large or small, between exergy and the actual work delivered by a device. This difference represents the available room that engineers have for improvement, especially for greener buildings and more sustainable buildings per ASHRAE's Sustainability Roadmap.

Note that the exergy of a system at a specified state depends on the conditions of the environment (the dead state) as well as the properties of the system. Therefore, exergy is a property of the system–environment combination and not of the system alone. Altering the environment is another way of increasing exergy, but it is definitely not an easy alternative.

The work potential or exergy of the kinetic energy of a system is equal to the kinetic energy itself since it can be converted to work entirely. Similarly, exergy of potential energy is equal to the potential energy itself. On the other hand, the internal energy and enthalpy of a system are not entirely available for work, and only part of thermal energy of a system can be converted to work. In other words, exergy of thermal energy is less than the magnitude of thermal energy.

### 1.9.2 Reversibility and Irreversibility

These two concepts are highly important to thermodynamic processes and systems. The *reversibility* is defined as the statement that both the system and its surroundings can be returned to their initial states, just leading to the theoretical one. The irreversibility shows the destruction of availability and states that both the system and its surroundings cannot be returned to their initial states due to the irreversibilities occurring, for example, friction, heat rejection, and electrical and mechanical effects. For instance, as an actual system provides an amount of work that is less than the ideal reversible work, the difference between these two values gives the irreversibility of that system. In real applications, there are always such differences, and therefore, real cycles are always irreversible. For example, the entropy of the heat given off in the condenser is always greater than that of the heat taken up in the evaporator, referring to the fact that the entropy is always increased by the operation of an actual refrigeration system.

### 1.9.3 Reversible Work and Exergy Destruction

*The reversible work* $W_{\text{rev}}$ is defined as *the maximum amount of useful work output or the minimum work input for a system undergoing a process between the specified initial and final states* in a totally reversible manner.

Any difference between the reversible work $W_{\text{rev}}$ and the actual work $W_u$ is due to the irreversibilities present during the process, and this difference is called *irreversibility* or *exergy destroyed*. It is expressed as

$$Ex_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{out}} \quad \text{or} \quad Ex_{\text{destroyed}} = W_{\text{in}} - W_{\text{rev,in}} \quad \text{or} \quad Ex_{\text{destroyed}} = W_{\text{in}} - W_{\text{rev,in}} \tag{1.76}$$

Irreversibility is a *positive quantity* for all actual (irreversible) processes since $W_{\text{rev}} \geq W$ for work-producing devices and $W_{\text{rev}} \leq W$ for work-consuming devices.

Irreversibility can be viewed as the *wasted work potential* or the *lost opportunity* to do useful work. It represents the energy that could have been converted to work but was not. It is important to note that lost opportunities manifest themselves in environmental degradation and avoidable emissions. The smaller the irreversibility associated with a process, the greater the work that is produced (or the smaller the work that is consumed). The performance of a system can be improved by minimizing the irreversibility associated with it.

A heat engine (an engine that converts heat to work output, e.g., a steam power plant) that operates on the reversible Carnot cycle is called a *Carnot heat engine*. The thermal efficiency of a Carnot heat engine, as well as other reversible heat engines, is given by

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H} \tag{1.77}$$

where $T_H$ is the source temperature and $T_L$ is the sink temperature where heat is rejected (i.e., lake, ambient, and air). This is the maximum efficiency of a heat engine operating between two reservoirs at $T_H$ and $T_L$.

A refrigerator or heat pump operating on reversed Carnot cycle would supply maximum cooling (in the case of refrigerator) and maximum heating (in the case of heat pump) and the COP of such reversible cycles are

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H/T_L - 1} \tag{1.78}$$

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L/T_H} \tag{1.79}$$

### *1.9.4 Exergy Balance*

For a thermodynamic system undergoing any process, mass, energy, and entropy balances can be expressed as (see Cengel and Boles, 2008)

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \tag{1.80}$$

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \tag{1.81}$$

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}} \tag{1.82}$$

In an actual process, mass and energy are conserved while entropy is generated. Note that energy can enter or exit a system by heat, work, and mass. Energy change of a system is the sum of the changes in internal, kinetic, and potential energies. Internal energy is the energy of a unit mass of a stationary fluid while enthalpy is the energy of a unit mass of a flowing fluid. Rate of energy change of a steady-flow system is zero and the rate of total energy input to a steady-flow control volume is equal to the rate of total energy output.

The nature of exergy is opposite to that of entropy in that exergy can be *destroyed*, but it cannot be created. Therefore, the *exergy change* of a system during a process is less than the *exergy transfer* by an amount equal to the *exergy destroyed* during the process within the system boundaries. Then the *decrease of exergy principle* can be expressed as

$$Ex_{\text{in}} - Ex_{\text{out}} - Ex_{\text{destroyed}} = \Delta Ex_{\text{system}} \tag{1.83}$$

This relation can also be written in the rate form, and is referred to as the *exergy balance* and can be stated as *the exergy change of 'a system during a process is equal to the difference between the net exergy transfer through the system boundary and the exergy destroyed within the system boundaries as a result of irreversibilities.* Exergy can be transferred to or from a system by heat, work, and mass.

Irreversibilities such as friction, mixing, chemical reactions, heat transfer through a finite temperature difference, unrestrained expansion, nonquasi-equilibrium compression or expansion always *generate entropy*, and anything that generates entropy always *destroys exergy*. The *exergy destroyed* is proportional to the entropy generated, and is expressed as

$$Ex_{\text{destroyed}} = T_0 S_{\text{gen}} \tag{1.84}$$

Exergy destruction during a process can be determined from an exergy balance on the system (Equation 1.83) or from the entropy generation using Equation 1.84.

A closed system, in general, may possess kinetic and potential energies as the total energy involved. The exergy change of a closed system during a process is simply the exergy difference between the final state 2 and initial state 1 of the system. A closed system involving heat input $Q_{\text{in}}$ and boundary work output $W_{\text{out}}$ as shown in Figure 1.15, mass, energy, entropy, and exergy balances can be expressed as

Mass balance:

$$m_1 = m_2 = \text{constant} \tag{1.85}$$

Energy balance:

$$Q_{\text{in}} - W_{\text{out}} = m(u_2 - u_1) \tag{1.86}$$

Entropy balance:

$$\frac{Q_{\text{in}}}{T_s} + S_{\text{gen}} = m(s_2 - s_1) \tag{1.87}$$

Exergy balance:

$$Q_{\text{in}}\left(1 - \frac{T_0}{T_s}\right) - [W_{\text{out}} - P_0(V_2 - V_1)] - Ex_{\text{destroyed}} = Ex_2 - Ex_1 \tag{1.88}$$

where $u$ is internal energy, $s$ is entropy, $T_s$ is source temperature, $T_0$ is the dead state (environment) temperature, $S_{\text{gen}}$ is entropy generation, $P_0$ is the dead-state pressure, and $V$ is volume. For *stationary* closed systems in practice, the kinetic and potential energy terms may drop out. The exergy of a closed system is either *positive* or *zero, and* never becomes negative.

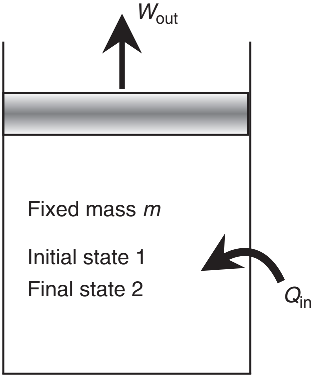

**Figure 1.15** A closed system involving heat input $Q_{\text{in}}$ and boundary work output $W_{\text{out}}$.

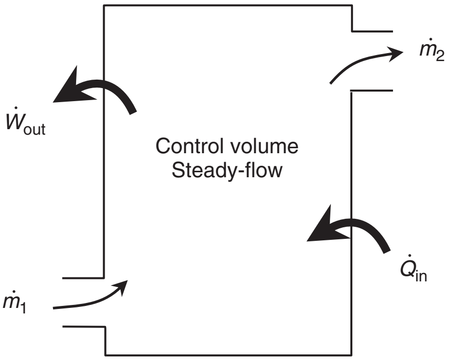

**Figure 1.16** A control volume involving heat input and power output.

A control volume involving heat input and power output as shown in Figure 1.16, mass, energy, entropy, and exergy balances can be expressed as
Mass balance:

$$\dot{m}_1 = \dot{m}_2 \tag{1.89}$$

Energy balance:

$$\dot{m}_1 h_1 + \dot{Q}_{in} = \dot{m}_2 h_2 + \dot{W}_{out} \tag{1.90}$$

Entropy balance:

$$\frac{\dot{Q}_{in}}{T_s} + \dot{m}_1 s_1 + \dot{S}_{gen} = \dot{m}_2 s_2 \tag{1.91}$$

Exergy balance:

$$\dot{Q}_{in}\left(1 - \frac{T_0}{T_s}\right) + \dot{m}_1\psi_1 = \dot{m}\psi_2 + \dot{W}_{out} + \dot{E}x_{destroyed} \tag{1.92}$$

where specific exergy of a flowing fluid (i.e., flow exergy) is given by

$$\psi = h - h_0 - T_0(s - s_0) \tag{1.93}$$

In these equations, kinetic and potential energy changes are assumed to be negligible. Most control volumes encountered in practice such as turbines, compressors, heat exchangers, pipes, and ducts operate steadily, and thus they experience no changes in their mass, energy, entropy, and exergy contents as well as their volumes. The rate of exergy entering a steady-flow system in all forms (heat, work, mass transfer) must be equal to the amount of exergy leaving plus the exergy destroyed.

### 1.9.5 Exergy or Second Law Efficiency

The first-law (i.e., energy) efficiency makes no reference to the best possible performance, and thus it may be misleading. Consider two heat engines, both having a thermal efficiency of 30%. One of the engines (engine *A*) receives heat from a source at 600 K, and the other one (engine *B*) from a source at 1000 K. After their process, both engines reject heat to a medium at 300 K. At the first glance, both engines seem to be performing equally well. When we take a second look at these engines in light of the SLT, however, we see a totally different picture. These engines, at best, can perform as reversible engines, in which case their efficiencies in terms of the Carnot Cycle become

$$\eta_{\mathrm{th,rev,A}} = \left(1 - \frac{T_0}{T_{\mathrm{source}}}\right)_{\mathrm{A}} = 1 - \frac{300\ \mathrm{K}}{600\ \mathrm{K}} = 50\%$$

$$\eta_{\mathrm{th,rev,B}} = \left(1 - \frac{T_0}{T_{\mathrm{source}}}\right)_{\mathrm{B}} = 1 - \frac{300\ \mathrm{K}}{1000\ \mathrm{K}} = 70\%$$

Engine A has a 50% useful work potential relative to the heat provided to it, and engine B has 70%. Now it is becoming apparent that engine B has a greater work potential made available to it and thus should do a lot better than engine A. Therefore, we can say that engine B is performing poorly relative to engine A even though both have the same thermal efficiency.

It is obvious from this example that the first-law efficiency alone is not a realistic measure of performance of engineering devices. To overcome this deficiency, we define an *exergy efficiency* (or *second-law efficiency*) for heat engines as the ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under the same conditions:

$$\eta_{\mathrm{ex}} = \frac{\eta_{\mathrm{th}}}{\eta_{\mathrm{th,rev}}} \tag{1.94}$$

Based on this definition, the energy efficiencies of the two heat engines discussed above become

$$\eta_{\mathrm{ex,A}} = \frac{0.30}{0.50} = 60\%$$

$$\eta_{\mathrm{ex,B}} = \frac{0.30}{0.70} = 43\%$$

That is, engine A is converting 60% of the available work potential to useful work. This ratio is only 43% for engine B. The second-law efficiency can also be expressed as the ratio of the useful work output and the maximum possible (reversible) work output:

$$\eta_{\mathrm{ex}} = \frac{W_{\mathrm{out}}}{W_{\mathrm{rev,out}}} \tag{1.95}$$

This definition is more general since it can be applied to processes (in turbines, piston–cylinder devices, and so on) and cycles. Note that the exergy efficiency cannot exceed 100%. We can also define an exergy efficiency for work-consuming noncyclic (such as compressors) and cyclic (such as refrigerators) devices as the ratio of the minimum (reversible) work input to the useful work input:

$$\eta_{\mathrm{ex}} = \frac{W_{\mathrm{rev,in}}}{W_{\mathrm{in}}} \tag{1.96}$$

For cyclic devices such as refrigerators and heat pumps, it can also be expressed in terms of the coefficients of performance as

$$\eta_{\mathrm{ex}} = \frac{\mathrm{COP}}{\mathrm{COP}_{\mathrm{rev}}} \tag{1.97}$$

In the above relations, the reversible work $W_{\mathrm{rev}}$ should be determined by using the same initial and final states as in the actual process.

For general cases where we do not produce or consume work (e.g., thermal energy storage system for a building), a general exergy efficiency can be defined as

$$\eta_{\mathrm{ex}} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy destroyed}}{\text{Exergy supplied}} \tag{1.98}$$

### *1.9.6 Illustrative Examples on Exergy*

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## Example 1.2

*A Geothermal Power Plant*

A geothermal power plant uses geothermal liquid water at 160 °C at a rate of 100 kg/s as the heat source, and produces 3500 kW of net power in an environment at 25 °C (Figure 1.17). We will conduct a thermodynamic analysis of this power plant considering both energy and exergy approaches.

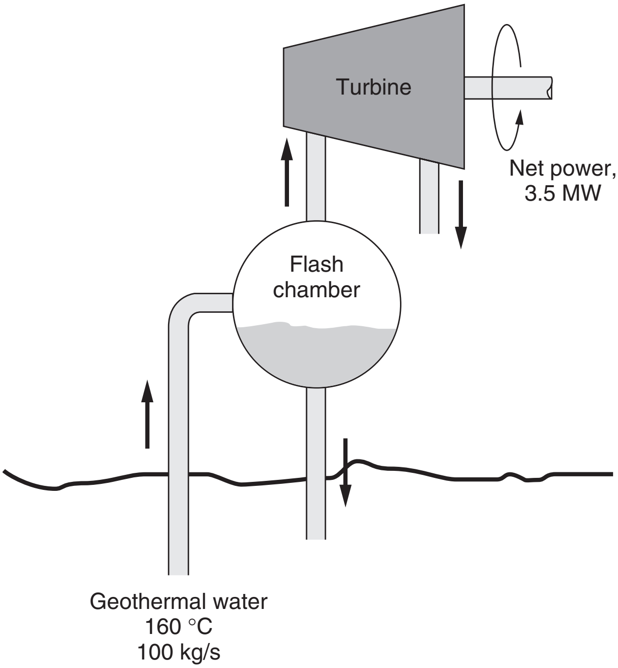

**Figure 1.17** A flash-design geothermal power plant.

The properties of geothermal water at the inlet of the plant and at the dead state are obtained from steam tables (not available in the text) to be

$$T_1 = 160\,^\circ\mathrm{C},\quad \text{liquid} \longrightarrow h_1 = 675.47\ \mathrm{kJ/kg},\quad s_1 = 1.9426\ \mathrm{kJ/kg \cdot K}$$

$$T_0 = 25\,^\circ\mathrm{C},\quad P_0 = 1\ \mathrm{atm} \longrightarrow h_0 = 104.83\ \mathrm{kJ/kg},\quad s_0 = 0.36723\ \mathrm{kJ/kg \cdot K}$$

The energy of geothermal water may be taken to be maximum heat that can be extracted from the geothermal water, and this may be expressed as the enthalpy difference between the state of geothermal water and dead state:

$$\dot{E}_{\mathrm{in}} = \dot{m}(h_1 - h_0) = (100\ \mathrm{kg/s})\left[(675.47 - 104.83)\ \mathrm{kJ/kg}\right] = 57{,}060\ \mathrm{kW}$$

The exergy of geothermal water is

$$\begin{aligned}\dot{E}x_{\text{in}} &= \dot{m}\,[(h_1 - h_0) - T_0(s_1 - s_0)]\\ &= (100\ \text{kg/s})\big[(675.47 - 104.83)\ \text{kJ/kg} - (25 + 273\ \text{K})(1.9426 - 0.36723)\text{kJ/kg}\cdot\text{K}\big]\\ &= 10{,}120\ \text{kW}\end{aligned}$$

The thermal efficiency of the power plant is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{E}_{\text{in}}} = \frac{3500\ \text{kW}}{57{,}060\ \text{kW}} = 0.0613 = 6.1\%$$

The exergy efficiency of the plant is the ratio of power produced to the exergy input to the plant:

$$\eta_{\text{ex}} = \frac{\dot{W}_{\text{net,out}}}{\dot{E}x_{\text{in}}} = \frac{3500\ \text{kW}}{10{,}120\ \text{kW}} = 0.346 = 34.6\%$$

The exergy destroyed in this power plant is determined from an exergy balance on the entire power plant to be

$$\dot{E}x_{\text{in}} - \dot{W}_{\text{net,out}} - \dot{E}x_{\text{dest}} = 0$$

$$10{,}120 - 3500 - \dot{E}x_{\text{dest}} = 0 \longrightarrow \dot{E}x_{\text{dest}} = 6620\ \text{kW}$$

Some of the results of this example are illustrated in Figures 1.18 and 1.19. The exergy of geothermal water (10,120 kW) constitutes only 17.7% of its energy (57,060), owing to its well temperature. The remaining 82.3% is not available for useful work and it cannot be converted to power by even a reversible heat engine. Only 34.6% exergy entering the plant is converted to power and the remaining 65.4% is lost. In geothermal power plants, the used geothermal water typically leaves the power plant at a temperature much greater than the environment temperature and this water is reinjected back to the ground. The total exergy destroyed (6620 kW) includes the exergy of this reinjected brine.

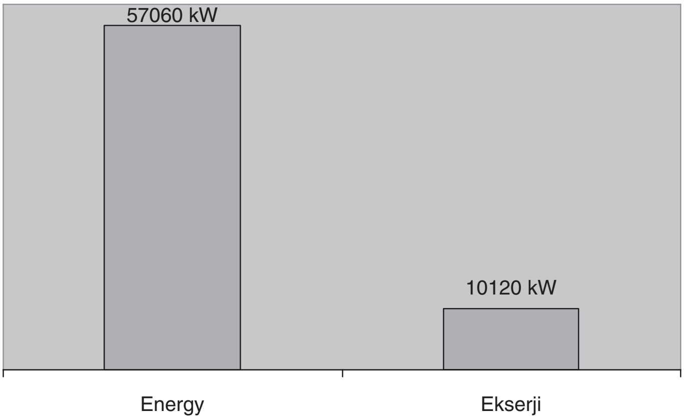

**Figure 1.18** Only 18% of the energy of geothermal water is available for converting to power.

In a typical binary-type geothermal power plant, geothermal water would be reinjected back to the ground at about 90 °C. This water can be used in a district heating system. Assuming that

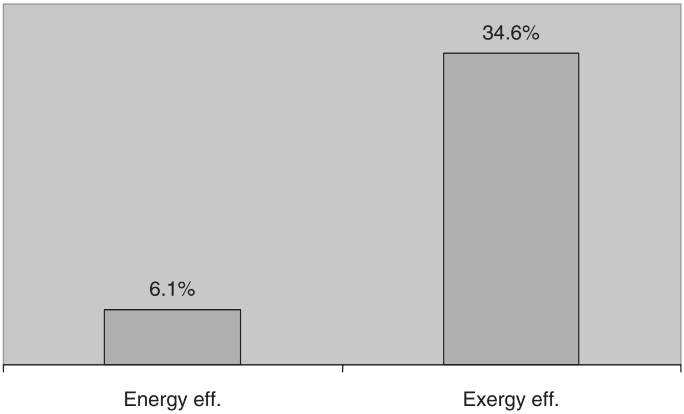

**Figure 1.19** Energy and exergy efficiencies of geothermal power plant.

geothermal water leaves the district at 70 °C with a drop of 20 °C during the heat supply, the rate of heat that could be used in the district system would be

$$\dot{Q}_{\text{heat}} = \dot{m}c\Delta T = (100\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(20\,^\circ\text{C}) = 8360\ \text{kW}$$

where $c$ is the specific heat of water. This 8360 kW heating is in addition to the 3500-kW power generated. The energy efficiency of this cogeneration system would be $(3500 + 8360)/57{,}060 = 0.208 = 20.8\%$. The energy efficiency increases from 6.1% to 20.8% as a result of incorporating a district heating system into the power plant.

The exergy of heat supplied to the district system is simply the heat supplied times the Carnot efficiency, which is determined as

$$\dot{E}x_{\text{heat}} = \dot{Q}_{\text{heat}}\left(1 - \frac{T_0}{T_{\text{source}}}\right) = (8360\ \text{kW})\left(1 - \frac{298\ \text{K}}{353\ \text{K}}\right) = 1303\ \text{kW}$$

where the source temperature is the average temperature of geothermal water (80 °C = 353 K) when supplying heat. This corresponds to 19.7% (1303/6620 = 0.197) of the exergy destruction. The exergy efficiency of this cogeneration system would be $(3500 + 1303)/10{,}120 = 0.475 = 47.5\%$. The exergy efficiency increases from 34.6% to 47.5% as a result of incorporating a district heating system into the power plant.

## Example 1.3

### *An Electric Resistance Heater*

An electric resistance heater with a power consumption of 2.0 kW is used to heat a room at 25 °C when the outdoor temperature is 0 °C (Figure 1.20). We will determine energy and exergy efficiencies and the rate of exergy destroyed for this process.

For each unit of electric work consumed, the heater will supply the house with 1 unit of heat. That is, the heater has a COP of 1. Also, the energy efficiency of the heater is 100% since the energy output (heat supply to the room) and the energy input (electric work consumed by the heater) are

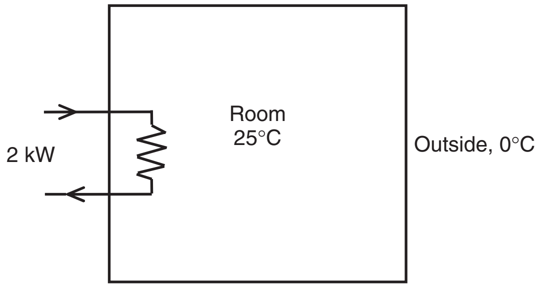

**Figure 1.20** An electric resistance heater used to heat a room.

the same. At the specified indoor and outdoor temperatures, a reversible heat pump would have a COP of

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{1}{1 - T_L/T_H} = \frac{1}{1 - (273\ \mathrm{K})/(298\ \mathrm{K})} = 11.9$$

That is, it would supply the house with 11.9 units of heat (extracted from the cold outside air) for each unit of electric energy it consumes (Figure 1.21). The exergy efficiency of this resistance heater is

$$\eta_{\mathrm{ex}} = \frac{\mathrm{COP}}{\mathrm{COP}_{\mathrm{HP,rev}}} = \frac{1}{11.9} = 0.084 = 8.4\%$$

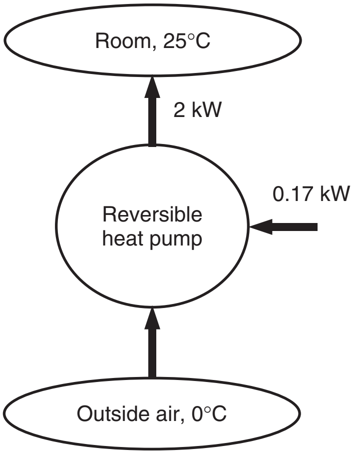

**Figure 1.21** A reversible heat pump consuming only 0.17 kW power while supplying 2-kW of heat to a room.

The minimum work requirement to the heater is determined from the COP definition for a heat pump to be

$$\dot{W}_{\mathrm{in,min}} = \frac{\dot{Q}_{\mathrm{supplied}}}{\mathrm{COP}_{\mathrm{HP,rev}}} = \frac{2\ \mathrm{kW}}{11.9} = 0.17\,\mathrm{kW}$$

That is, a reversible heat pump would consume only 0.17 kW of electrical energy to supply the room 2 kW of heat. The exergy destroyed is the difference between the actual and minimum work inputs:

$$\dot{E}x_{\text{destroyed}} = \dot{W}_{\text{in}} - \dot{W}_{\text{in,min}} = 2.0 - 0.17 = 1.83\ \text{kW}$$

The results of this example are illustrated in Figures 1.22 and 1.23. The performance looks perfect with energy efficiency but not so good with exergy efficiency. About 92% of actual work input to the resistance heater is lost during the operation of resistance heater. There must be better methods of heating this room. Using a heat pump (preferably a ground-source one) or a natural gas furnace would involve lower exergy destructions and correspondingly greater exergy efficiencies even though the energy efficiency of a natural gas furnace is lower than that of a resistance heater.

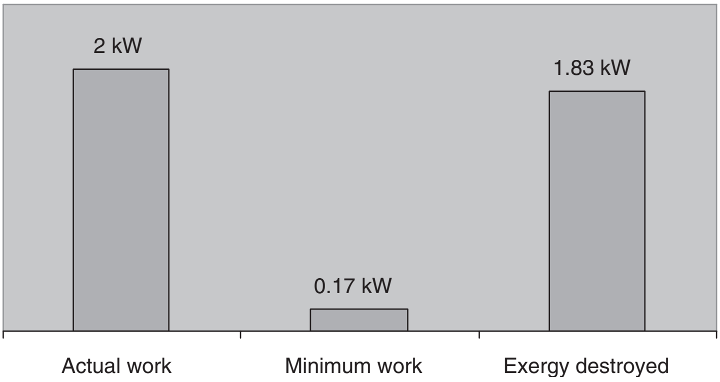

**Figure 1.22** Comparison of actual and minimum works with the exergy destroyed.

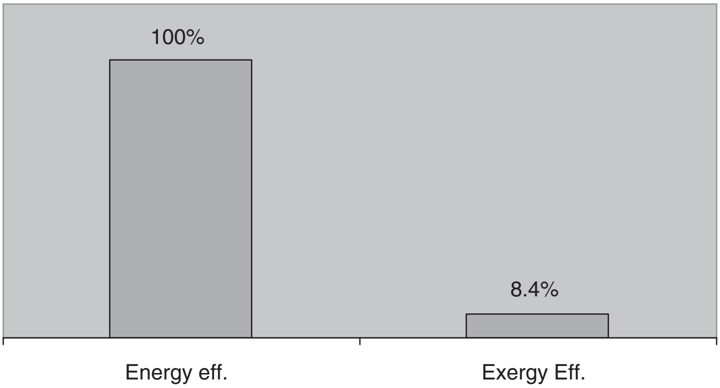

**Figure 1.23** Comparison of energy and exergy efficiencies.

Different heating systems may also be compared using primary energy ratio (PER), which is the ratio of useful heat delivered to primary energy input. Obviously, the higher the PER, the more efficient the heating system. The PER for a heat pump is defined as PER $= \eta \times$ COP where $\eta$ is the thermal efficiency with which the primary energy input is converted into work. For the resistance heater discussed in this example, the thermal efficiency $\eta$ may be taken to be 0.40 if the electricity is produced from a natural-gas-fueled steam power plant. Since the COP is 1, the PER becomes 0.40. A natural gas furnace with an efficiency of 0.80 (i.e., heat supplied over the heating value of the fuel) would have a PER value of 0.80. Furthermore, for a ground-source heat pump using electricity as the work input, the COP may be taken as 3 and with the same method of electricity production ($\eta = 0.40$), the PER becomes 1.2.

## Example 1.4

### *A Simple Heating Process*

In an air-conditioning process, air is heated by a heating coil in which hot water is flowing at an average temperature of 80 °C. Using the values given in Figure 1.24, we will determine the exergy destruction and the exergy efficiency for this process.

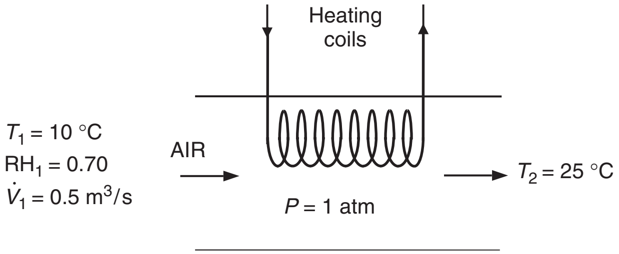

**Figure 1.24** Schematic of simple heating process.

The properties of air at various states (including dead state, denoted by the subscript 0) are determined from a software with built-in properties to be

$$v_1 = 0.810 \text{ m}^3/\text{kg}, \quad h_0 = h_1 = 25.41 \text{ kJ/kg}, \quad h_2 = 40.68 \text{ kJ/kg}, \quad s_0 = s_1 = 5.701 \text{ kJ/kg} \cdot \text{K}$$

$$s_2 = 5.754 \text{ kJ/kg} \cdot \text{K}, \quad w_1 = w_2 = 0.00609 \text{ kg water/kg air}, \quad \text{RH}_2 = 0.31$$

The dead-state temperature is taken to be the same as the inlet temperature of air. The mass flow rate of air and the rate of heat input are

$$\dot{m}_a = \frac{\dot{V}_a}{v_1} = 0.617 \text{ kg/s}$$

$$\dot{Q}_{in} = \dot{m}_a (h_2 - h_1) = 9.43 \text{ kW}$$

The exergies of air stream at the inlet and exit are

$$\dot{E}x_1 = 0 \quad \text{and} \quad \dot{E}x_2 = \dot{m}_a \left[(h_2 - h_0) - T_0(s_2 - s_0)\right] = 0.267 \, \text{kW}$$

The rates of exergy input and the exergy destroyed are

$$\dot{E}x_{\text{in}} = \dot{Q}_{\text{in}}\left(1 - \frac{T_0}{T_{\text{source}}}\right) = 1.87 \text{ kW}$$

$$\dot{E}x_{\text{destroyed}} = \dot{E}x_{\text{in}} - \dot{E}x_{\text{out}} = 1.87 - 0.267 = 1.60 \text{ kW}$$

where the temperature at which heat is transferred to the air stream is taken as the average temperature of water flowing in the heating coils (80 °C). The exergy efficiency is

$$\eta_{\text{ex}} = \frac{\dot{E}x_{\text{out}}}{\dot{E}x_{\text{in}}} = \frac{0.267 \text{ kW}}{1.87 \text{ kW}} = 0.143 = 14.3\%$$

About 86% of exergy input is destroyed owing to irreversible heat transfer in the heating section. Air-conditioning processes typically involve high rates of exergy destructions as high-temperature (i.e., high quality) heat or high-quality electricity is used to obtain a low-quality product. The irreversibilities can be minimized using lower quality energy sources and less irreversible processes. For example, if heat is supplied at an average temperature of 60 °C instead of 80 °C, the exergy destroyed would decrease from 1.60 to 1.15 kW and the exergy efficiency would increase from 14.3 to 18.8%. The exit temperature of air also affects the exergy efficiency. For example, if air is heated to 20 °C instead of 25 °C, the exergy efficiency would decrease from 14.3 to 10.1%. These two examples also show that the smaller the temperature difference between the heat source and the air being heated, the larger the exergy efficiency.

## Example 1.5

*A Heating with Humidification Process*

A heating process with humidification is considered using the values shown in Figure 1.25. Its psychrometric representation is given in Figure 1.26. Mass, energy, entropy and exergy balances, and exergy efficiency for this process can be expressed as

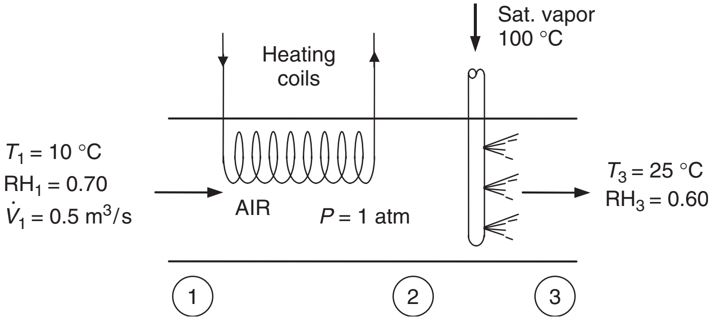

**Figure 1.25** A heating with humidification process.

Dry air mass balance:

$$\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_{a3}$$

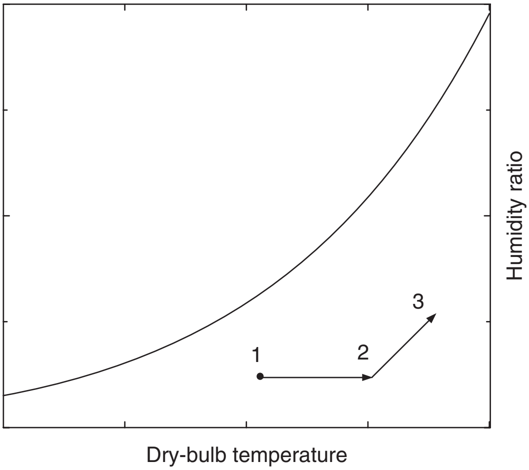

**Figure 1.26** Heating with humidification as represented on a psychrometric chart.

Water mass balance:

$$\dot{m}_{w1} = \dot{m}_{w2}$$

$$\dot{m}_{w2} + \dot{m}_{\mathrm{w}} = \dot{m}_{w3} \longrightarrow \dot{m}_{a2}\omega_2 + \dot{m}_{\mathrm{w}} = \dot{m}_{a3}\omega_3$$

Energy balance:

$$\dot{Q}_{\mathrm{in}} + \dot{m}_{a1}h_1 = \dot{m}_{a2}h_2 \text{ (process 1–2)}$$

$$\dot{m}_{a2}h_2 + \dot{m}_{\mathrm{w}}h_{\mathrm{w}} = \dot{m}_{a3}h_3 \text{ (process 2–3)}$$

$$\dot{Q}_{\mathrm{in}} + \dot{m}_{a1}h_1 + \dot{m}_{\mathrm{w}}h_{\mathrm{w}} = \dot{m}_{a3}h_3 \text{ (process 1–3)}$$

Entropy balance:

$$\dot{m}_{a1}s_1 + \dot{m}_{\mathrm{w}}s_{\mathrm{w}} + \frac{\dot{Q}_{\mathrm{in}}}{T} - \dot{m}_{a3}s_3 + \dot{S}_{\mathrm{gen}} = 0 \text{ (process 1–3)}$$

Exergy balance:

$$\dot{Q}_{\mathrm{in}}\left(1 - \frac{T_0}{T}\right) + \dot{m}_{a1}\psi_1 - \dot{m}_{a2}\psi_2 - \dot{E}x_{\mathrm{dest}} = 0 \text{ (process 1–2)}$$

$$\dot{m}_{a2}\psi_2 + \dot{m}_{\mathrm{w}}\psi_{\mathrm{w}} - \dot{m}_{a3}\psi_3 - \dot{E}x_{\mathrm{dest}} = 0 \text{ (process 2–3)}$$

$$\dot{Q}_{\mathrm{in}}\left(1 - \frac{T_0}{T}\right) + \dot{m}_{a1}\psi_1 + \dot{m}_{\mathrm{w}}\psi_{\mathrm{w}} - \dot{m}_{a3}\psi_3 - \dot{E}x_{\mathrm{dest}} = 0 \text{ (process 1–3)}$$

$$E\dot{x}_{\mathrm{dest}} = T_0\dot{S}_{\mathrm{gen}} = T_0\left(\dot{m}_{a3}s_3 - \dot{m}_{a1}s_1 - \dot{m}_{\mathrm{w}}s_{\mathrm{w}} - \frac{\dot{Q}_{\mathrm{in}}}{T}\right) \text{ (process 1–3)}$$

Exergy efficiency:

$$\eta_{\mathrm{ex}} = \frac{\dot{m}_{a3}\psi_3}{\dot{Q}_{\mathrm{in}}\left(1 - \frac{T_0}{T}\right) + \dot{m}_{a1}\psi_1 + \dot{m}_{\mathrm{w}}\psi_{\mathrm{w}}}$$

Based on these balances and using an equation solver with built-in thermodynamic functions including pschrometric properties of moist air (Klein, 2006), we obtain the following results:

$$\dot{m}_{\mathrm{a}} = 0.618 \text{ kg/s},\ \dot{m}_{\mathrm{w}} = 0.00406 \text{ kg/s},\ T_2 = 24.2\,^{\circ}\mathrm{C},\ \dot{Q}_{\mathrm{in}} = 8.90 \text{ kW}$$

$$\dot{E}x_{\mathrm{in}} = 4.565 \text{ kW},\ \dot{E}x_{\mathrm{dest}} = 4.238 \text{ kW},\ \eta_{\mathrm{ex}} = 0.0718 = 7.2\%$$

The dead-state properties of air are taken to be the same as the inlet air properties while the dead-state properties of water are obtained using the temperature of inlet air and the atmospheric pressure. The temperature at which heat transfer takes place is assumed to be equal to the temperature of the saturated water vapor used for humidification. When property data for the fluid flowing in the heating coil is available, we do not have to assume a temperature for heat transfer. For example, let us assume that a refrigerant flows in the heating coil and the properties of the refrigerant at the inlet (denoted by subscript $R1$) and exit (denoted by subscript $R2$) of the heating section are given. The balances in this case become

Energy balance:

$$\dot{Q}_{\mathrm{in}} + \dot{m}_{a1}h_1 = \dot{m}_{a2}h_2 \text{ (process 1−2)}$$

$$\dot{m}_{a1}h_1 + \dot{m}_{\mathrm{R}}h_{R1} = \dot{m}_{a2}h_2 + \dot{m}_{\mathrm{R}}h_{R2} \text{ (process 1−2)}$$

$$\dot{m}_{a1}h_1 + \dot{m}_{\mathrm{R}}h_{R1} + \dot{m}_{\mathrm{w}}h_{\mathrm{w}} = \dot{m}_{a2}h_2 + \dot{m}_{\mathrm{R}}h_{R2} \text{ (process 1−3)}$$

Entropy balance:

$$\dot{m}_{a1}s_1 + \dot{m}_{\mathrm{R}}s_{R1} + \dot{m}_{\mathrm{w}}s_{\mathrm{w}} - \dot{m}_{a3}s_3 - \dot{m}_{\mathrm{R}}s_{R2} + \dot{S}_{\mathrm{gen}} = 0 \text{ (process 1−3)}$$

Exergy balance:

$$\dot{m}_{a1}\psi_1 + \dot{m}_{\mathrm{R}}\psi_{R1} + \dot{m}_{\mathrm{w}}\psi_{\mathrm{w}} - \dot{m}_{a3}\psi_3 - \dot{m}_{\mathrm{R}}\psi_{R2} - \dot{E}x_{\mathrm{dest}} = 0 \text{ (process 1−3)}$$

$$E\dot{x}_{\mathrm{dest}} = T_0\dot{S}_{\mathrm{gen}} = T_0\left(\dot{m}_{a3}s_3 + \dot{m}_{\mathrm{R}}s_{R2} - \dot{m}_{a1}s_1 - \dot{m}_{\mathrm{R}}s_{R1} - \dot{m}_{\mathrm{w}}s_{\mathrm{w}}\right) \text{ (process 1−3)}$$

Exergy efficiency:

$$\eta_{\mathrm{ex}} = \frac{\dot{m}_{a3}\psi_3 + \dot{m}_{\mathrm{R}}\psi_{R2}}{\dot{m}_{a1}\psi_1 + \dot{m}_{\mathrm{R}}\psi_{R1} + \dot{m}_{\mathrm{w}}\psi_{\mathrm{w}}}$$

The exergy efficiency of the process is calculated to be 7.2%, which is low. This is typical of air-conditioning processes during which irreversibilities occur mainly because of heat transfer across a relatively high temperature difference and humidification.

## 1.10 Psychrometrics

Psychrometrics is the science of air and water vapor and deals with the properties of moist air. A thorough understanding of psychrometrics is of great significance, particularly to the HVAC community. It plays a key role, not only in the heating and cooling processes and the resulting comfort of the occupants, but also in building insulation, roofing properties, and the stability, deformation, and fire-resistance of the building materials. That is why understanding of the main concepts and principles involved is essential.

Actually, psychrometry also plays a crucial role in food preservation, especially in cold storage. In order to prevent the spoilage and maintain the quality of perishable products during storage, a proper arrangement of the storage conditions in terms of temperature and relative humidity is extremely important in this regard. Furthermore, the storage conditions are different for each food commodity and should be implemented accordingly.

### *1.10.1 Common Definitions in Psychrometrics*

The following definitions are the most common terms in psychrometrics:

**Dry air.** Normally, atmospheric air contains a number of constituents, as well as water vapor, along with miscellaneous components (e.g., smoke, pollen, and gaseous pollutants). When we talk about dry air, it no longer contains water vapor and other components.

**Moist air.** Moist air is the basic medium and is defined as a binary or two-component mixture of dry air and water vapor. The amount of water vapor in moist air varies from nearly zero, referring to dry air, to a maximum of 0.020 kg water vapor/kg dry air under atmospheric conditions depending on the temperature and pressure.

**Saturated air.** This is known as the saturated mixture (i.e., air and water vapor mixture) where the vapor is given at the saturation temperature and pressure.

**Dew point temperature.** This is defined as the temperature of moist air saturated at the same pressure and with the same humidity ratio as that of the given sample of moist air (i.e., temperature at state 2 in Figure 1.27). It takes place where the water vapor condenses when it is cooled at constant pressure (i.e., process 1–2).

**Relative humidity.** This is defined as the ratio of the mole fraction of water vapor in the mixture to the mole fraction of water vapor in a saturated mixture at the same temperature and pressure, based on the mole fraction equation since water vapor is considered to be an ideal gas:

$$\phi = \frac{P_v}{P_s} = \frac{\rho_v}{\rho_s} = \frac{v_s}{v_v} \tag{1.99}$$

where $P_v$ is the partial pressure of vapor, Pa or kPa, and $P_s$ is the saturation pressure of vapor at the same temperature, Pa or kPa, which can be taken directly from saturated water table. The total pressure is $P = P_a + P_v$. According to Figure 1.27, $\phi = P_1/P_3$.

**Humidity ratio.** The humidity ratio of moist air (so-called *mixing ratio*) is defined as the ratio of the mass of water vapor to the mass of dry air contained in the mixture at the same temperature and pressure:

$$\omega = \frac{m_v}{m_a} = 0.622\frac{P_v}{P_a} \tag{1.100}$$

where $m_v = P_vV/R_vT$ and $m_a = P_aV/R_aT$ since both water vapor and air, as well as their mixtures, are treated as ideal gases.

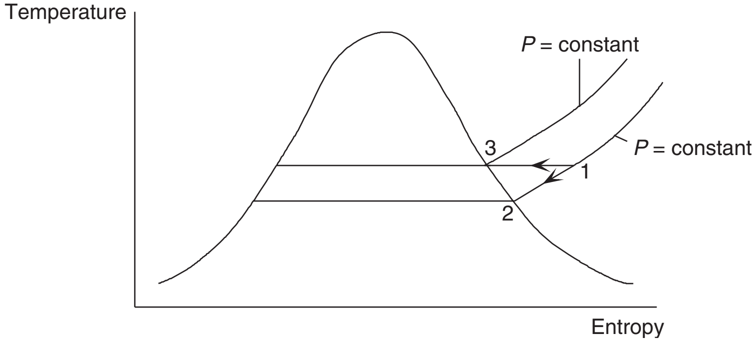

**Figure 1.27** Representation of dew point temperature on $T-s$ diagram.

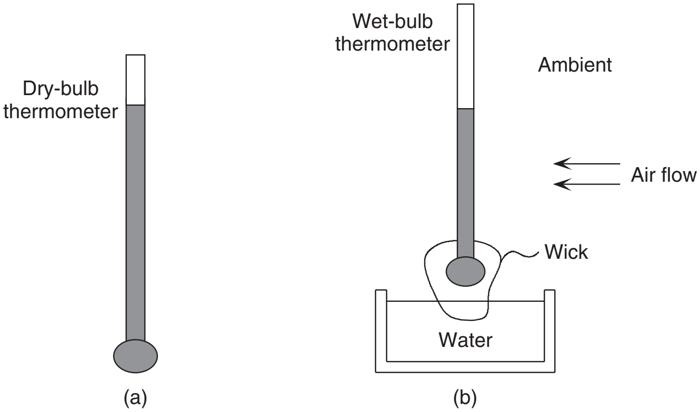

**Figure 1.28** Schematic representation of (a) dry-bulb and (b) wet-bulb thermometers.

Since we have the relative humidity and the humidity ratio in terms of the pressure ratio, it is possible to reach the following equation after making the necessary substitutions:

$$\phi = \frac{\omega P_a}{0.622 P_s} \tag{1.101}$$

**Degree of saturation.** This is defined as the ratio of the actual humidity ratio to the humidity ratio of a saturated mixture at the same temperature and pressure.

**Dry-bulb and wet-bulb temperatures.** The use of both a dry-bulb thermometer and a wet-bulb thermometer is very old practice to measure the specific humidities of moist air. The dry-bulb temperature is the temperature measured by a dry-bulb thermometer directly. The bulb of the wet-bulb thermometer is covered with a wick which is already saturated with water. When the wick is subjected to an air flow (Figure 1.28), some of the water in the wick gets evaporated into the surrounding air, thereby resulting in a temperature drop in the thermometer. This final temperature is dependent on the moisture content of the air. It is important to mention that in the past there was a convention that the wicks are boiled in distilled water first and allowed to dry before using them in wet-bulb temperature measurements. Nowadays, several new electronic devices and data loggers are preferred to measure the humidity of air due to their simplicity, accuracy, and effectiveness.

**Adiabatic saturation process.** This is the adiabatic process in which an air and water vapor mixture with a relative humidity less than 100% is subjected to liquid water addition. Some of the water evaporates into the mixture and makes it saturated, referring to the 100% relative humidity. In this respect, the temperature of the mixture exiting the system is identified as the *adiabatic saturation temperature* and the process is called the *adiabatic saturation process* (Figure 1.29).

### 1.10.2 Balance Equations for Air and Water Vapor Mixtures

As mentioned earlier, air and water vapor is considered an ideal gas mixture which makes the solution a bit easier. In terms of balance equations, we have two important aspects to deal with: the mass balance equation (i.e., the continuity equation) and the energy balance equation (i.e., the FLT). These can be written for both closed and open systems. Let us consider a cooling process, with negligible kinetic and potential energies and no work involved, that has two inputs and one

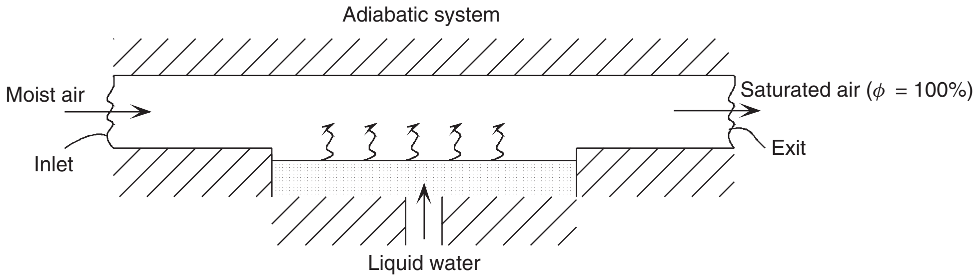

**Figure 1.29** Schematic representation of an adiabatic saturation process.

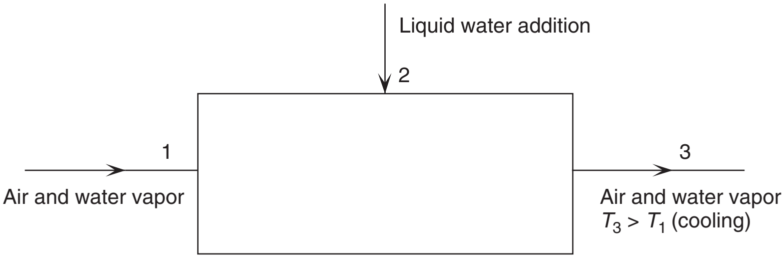

**Figure 1.30** Schematic of the system.

output as illustrated in Figure 1.30. Before going into details of analysis of this process, the general mass and energy balance equations may be written as follows:

- The mass balance equations are

$$\Sigma \dot{m}_{a,i} = \Sigma \dot{m}_{a,e} \tag{1.102}$$

$$\Sigma \dot{m}_{v,i} + \Sigma \dot{m}_{l,i} = \Sigma \dot{m}_{a,e} + \Sigma \dot{m}_{l,e} \tag{1.103}$$

- The energy balance equation is

$$\dot{Q}_i + \Sigma \dot{m}_i h_i = \Sigma \dot{m}_e h_e \tag{1.104}$$

Let us now write the respective balance equations for the subject matter system in Figure 1.30 as follows:

$$\dot{m}_{a,1} = \dot{m}_{a,3} = \dot{m}_a \tag{1.105}$$

$$\dot{m}_{v,1} + \dot{m}_{l,2} = \dot{m}_{v,3} \tag{1.106}$$

$$\dot{Q}_i + \dot{m}_a h_{a,1} + \dot{m}_{v,1} h_{v,1} + \dot{m}_{l,2} h_{l,2} = \dot{m}_a h_{a,3} + \dot{m}_{v,3} h_{v,3} \tag{1.107}$$

Equation 1.107 can be arranged in terms of the humidity ratio under $\omega = m_v/m_a$ (Equation 1.100):

$$\frac{\dot{Q}_i}{\dot{m}_a} + h_{a,1} + \omega_1 h_{v,1} + (\omega_1 - \omega_2) h_{l,2} = h_{a,3} + \omega_3 h_{v,3} \tag{1.108}$$

where $\omega_2 = \omega_3$ since there is no more water addition or removal between 2 and 3.

### *1.10.3 The Psychrometric Chart*

This chart was developed in the early 1900s by a German engineer named Richard Mollier. It is a graph (Figure 1.31) that represents the properties of moist air in terms of the dry-bulb temperature, the wet-bulb temperature, the relative humidity, the humidity ratio, and the enthalpy. Three of these properties are sufficient to identify a state of the moist air. It is important to note that the chart can only be used for atmospheric pressure (i.e., 1 atm, or 101.3 kPa). If the pressure is different, the moist air equations can be employed.

Understanding the dynamics of moisture and air will provide a solid foundation for understanding the principles of cooling and air-conditioning systems. Figure 1.32 shows several processes on the psychrometric chart. Figure 1.32a exhibits cooling and heating processes and therefore an example of an increase and decrease in dry-bulb temperature. In these processes, only a change in sensible heat is encountered. There is no latent heat involved due to the constant humidity ratio of the air. Figure 1.32b is an example of a dehumidification process at the constant dry-bulb temperature with decreasing humidity ratio. A very common example is given in Figure 1.32c which includes both cooling and dehumidification, resulting in a decrease of both the dry-bulb and wet-bulb temperatures, as well as the humidity ratio. Figure 1.32d exhibits a process of adiabatic humidification at the constant wet-bulb temperature (1-2), for instance spray type humidification. If it is done by heated water, it will result in (1-2′). Figure 1.32e displays a chemical dehumidification process as the water vapor is absorbed or adsorbed from the air by using a hydroscopic material. It is isolated because of the constant enthalpy as the humidity ratio decreases. The last one (Figure 1.32f) represents a mixing process of two streams of air (i.e., one at state 1 and other at state 2), and their mixture reaches state 3.

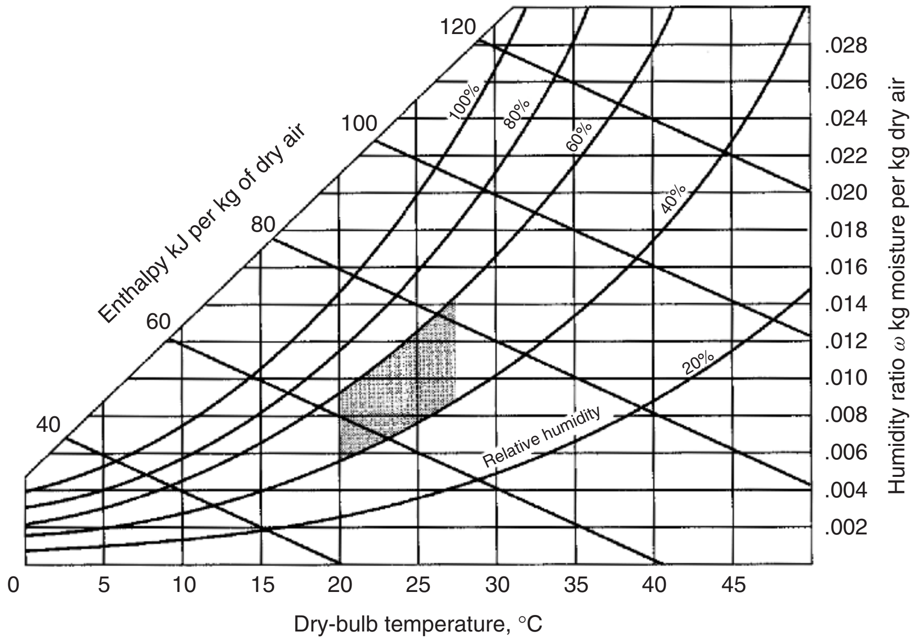

**Figure 1.31** Psychrometric chart.

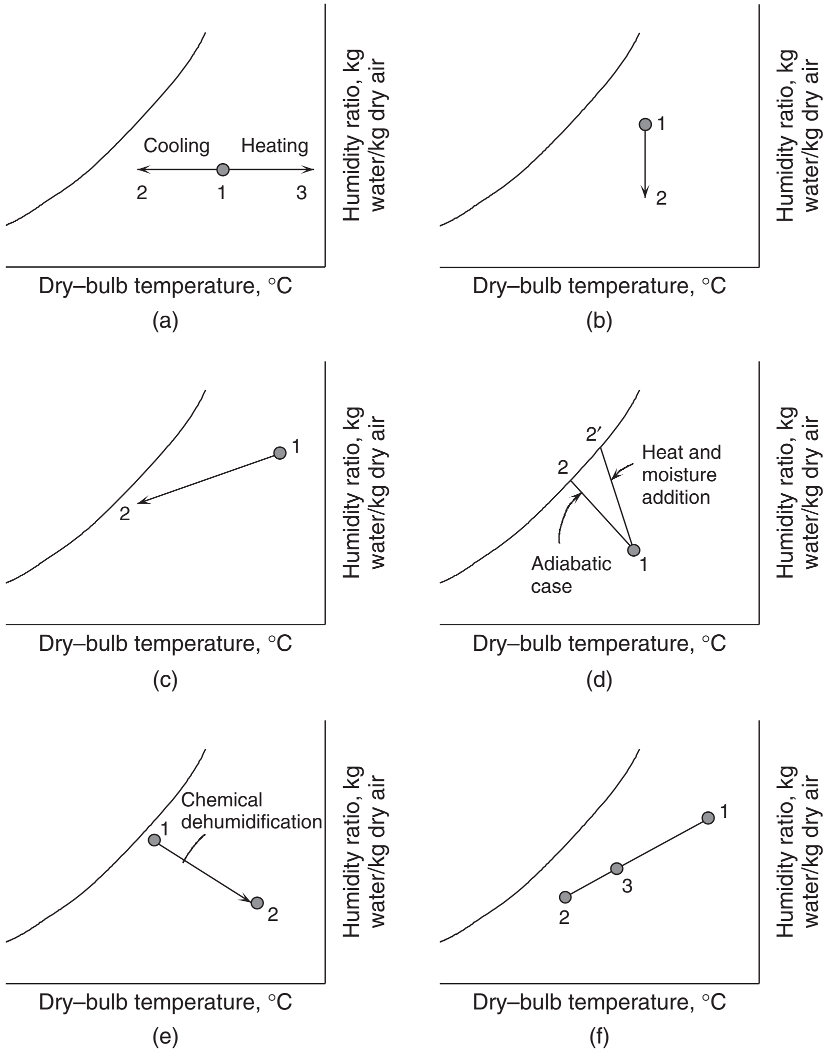

**Figure 1.32** Some processes on the psychrometric chart. (a) Cooling and heating. (b) Dehumidification. (c) Cooling and dehumidification. (d) Adiabatic humidification. (e) Chemical dehumidification. (f) Mixture of two moist air flows.

## 1.11 General Aspects of Fluid Flow

For a good understanding of the operation of refrigeration systems and their components as well as the behavior of fluid flow, an extensive background on fluid mechanics is essential. In addition to learning the principles of fluid flow, the student and/or engineer should develop an understanding of the properties of fluids, which should enable him or her to solve practical refrigeration problems.

In practice, refrigeration engineers come into contact everyday or at least on an occasional basis with a large variety of fluid flow problems such as

- subcooled liquid refrigerant, water, brine, and other liquids,
- mixtures of boiling liquid refrigerant and its vapor,
- mixtures of refrigerants and absorbents,
- mixtures of air and water vapor as in humid air, and
- low- and high-side vaporous refrigerant and other gases.

In order to deal effectively with the fluid flow systems it is necessary to identify flow categories, defined in predominantly mathematical terms, which will allow appropriate analysis to be undertaken by identifying suitable and acceptable simplifications. Example of the categories to be introduced includes variation of the flow parameters with time (steady or unsteady) or variations along the flow path (uniform or non-uniform). Similarly, compressibility effects may be important in high-speed gas flows, but may be ignored in many liquid flow situations.

## *1.11.1 Classification of Fluid Flows*

There are several criteria to classify the fluid flows into the following categories:

- uniform or nonuniform,
- steady or unsteady state,
- one-, two-, or three-dimensional,
- laminar or turbulent, and
- compressible or incompressible.

Also, the liquids flowing in channels may be classified according to their regions, for example, subcritical, critical, or supercritical, and the gas flows may be categorized as subsonic, transsonic, supersonic, or hypersonic.

### 1.11.1.1 Uniform Flow and Nonuniform Flow

If the velocity and cross-sectional area are constant in the direction of flow, the flow is uniform. Otherwise, the flow is nonuniform.

### 1.11.1.2 Steady Flow

This is defined as a flow in which the flow conditions do not change with time. However, we may have a steady-flow in which the velocity, pressure, and cross-section of the flow may vary from point to point but do not change with time. Therefore, we need to distinguish this by dividing it into the *steady uniform flow* and the *steady nonuniform flow*. In the steady uniform flow, all conditions (e.g., velocity, pressure, and cross-sectional area) are uniform and do not vary with time or position. For example, uniform flow of water in a duct is considered steady uniform flow. If the conditions (e.g., velocity, cross-sectional area) change from point to point (e.g., from cross-section to cross-section) but not with time, it is called steady nonuniform flow. For example, a liquid flows at a constant rate through a tapering pipe running completely full.

### 1.11.1.3 Unsteady Flow

If the conditions vary with time, the flow becomes unsteady. At a given time, the velocity at every point in the flow field is the same, but the velocity changes with time, referring to the *unsteady*

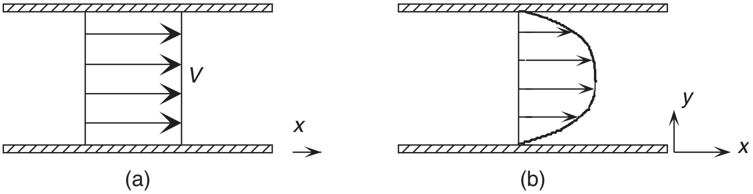

**Figure 1.33** Velocity profiles for flows. (a) One-dimensional flow. (b) Two-dimensional flow.

*uniform flow*, for example, accelerating flow, a fluid through a pipe of uniform bore running full. In the unsteady uniform flow, the conditions in cross-sectional area and velocity vary with time from one point to another, for example, a wave traveling along a channel.

### 1.11.1.4 One-, Two-, and Three-Dimensional Flow

The flow of real fluids occurs in three dimensions. However, in the analysis the conditions are simplified to either one-dimensional or two-dimensional, depending on the flow problem under consideration. If all fluid and flow parameters (e.g. velocity, pressure, elevation, temperature, density, and viscosity) are considered to be uniform throughout any cross-section and vary only along the direction of flow (Figure 1.33a), the flow becomes one-dimensional. Two-dimensional flow is the flow in which the fluid and flow parameters are assumed to have spatial gradients in two directions, that is, $x$ and $y$ axes (Figure 1.33b). In fact, in a three-dimensional flow the fluid and flow parameters vary in three directions, that is, $x$, $y$ and $z$ axes, and the gradients of the parameters occur in all three directions.

### 1.11.1.5 Laminar Flow and Turbulent Flow

This is one of the most important classifications of fluid flow and depends primarily upon the arbitrary disturbances, irregularities, or fluctuations in the flow field, based on the internal characteristics of the flow. In this regard, there are two significant parameters such as velocity and viscosity. If the flow occurs at a relatively low velocity and/or with a highly viscous fluid, resulting in a fluid flow in an orderly manner without fluctuations, the flow is referred to as laminar. As the flow velocity increases and the viscosity of fluid decreases, the fluctuations will take place gradually, referring to a *transition state* which is dependent on the fluid viscosity, the flow velocity, and the geometric details. In this regard, the Reynolds number is introduced to represent the characteristics of the flow conditions relative to the transition state. As the flow conditions deviate more from the transition state, a more chaotic flow field, that is, turbulent flow occurs. It is obvious that increasing Reynolds number increases the chaotic nature of the turbulence. Turbulent flow is therefore defined as a characteristic representative of the irregularities in the flow field.

The differences between laminar flow and turbulent flow can be distinguished by the Reynolds number, which is expressed by

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{\rho VD}{\mu} \tag{1.109}$$

In fact, the Reynolds number indicates the ratio of inertia force to viscous force. One can point out that at high Reynolds numbers the inertia forces predominate, resulting in turbulent flow, while at low Reynolds numbers the viscous forces become dominant, which makes the flow laminar. In a circular duct, the flow is laminar when Re is less than 2100 and turbulent when Re is greater than 4000. In a duct with a rough surface, the flow is turbulent at Re values as low as 2700.

### 1.11.1.6 Compressible Flow and Incompressible Flow

All actual fluids are normally compressible, leading to the fact that their density changes with pressure. However, in most cases during the analysis it is assumed that the changes in density are negligibly small. This refers to the incompressible flow.

## *1.11.2 Viscosity*

This is known as one of the most significant fluid properties and is defined as a measure of the fluid's resistance to deformation. In gases, the viscosity increases with increasing temperature, resulting in a greater molecular activity and momentum transfer. The viscosity of an ideal gas is a function of molecular dimensions and absolute temperature only, based on the kinetic theory of gases. However, in liquids, molecular cohesion between molecules considerably affects the viscosity, and the viscosity decreases with increasing temperature due to the fact that the cohesive forces are reduced by increasing the temperature of the fluid (causing a decrease in shear stress), resulting in an increase in the rate of molecular interchange; therefore, the net result is apparently a reduction in the viscosity. The coefficient of viscosity of an ideal fluid is zero, meaning that an ideal fluid is inviscid, so that no shear stresses occur in the fluid, despite the fact that shear deformations are finite. Nevertheless, all real fluids are viscous.

There are two types of viscosities, namely, the *dynamic viscosity* which is the ratio of a shear stress to a fluid strain (velocity gradient) and the *kinematic viscosity* which is defined as the ratio of dynamic viscosity to density. The dynamic viscosity is expressed based on Figure 1.34, leading to the fact that the shear stress within a fluid is proportional to the spatial rate of change of fluid strain normal to the flow:

$$\mu = \frac{\tau}{du/\mathrm{d}y} \tag{1.110}$$

where the unit of $\mu$ is Ns/m$^2$ or kg/ms in the SI system and $\mathrm{lbf} \cdot \mathrm{s/ft}^2$ in the English system.

The kinematic viscosity then becomes

$$\nu = \frac{\mu}{\rho} \tag{1.111}$$

where the units of $\nu$ is m$^2$/s in the SI system and ft$^2$/s in the English system.

There are some other units (e.g., the cgs system of units) for the dynamic and kinematic viscosities that find applications as follows:

$$1 \text{ pose} = 1 \text{ dyne} \cdot \mathrm{s/cm}^2 = 1\,\mathrm{g/cm} \cdot \mathrm{s} = 0.1\,\mathrm{kg} \cdot \mathrm{m} \cdot \mathrm{s}.$$

$$1 \text{ stoke} = 1\,\mathrm{cm}^2/\mathrm{s} = 10^{-4}\ \mathrm{m}^2/\mathrm{s}.$$

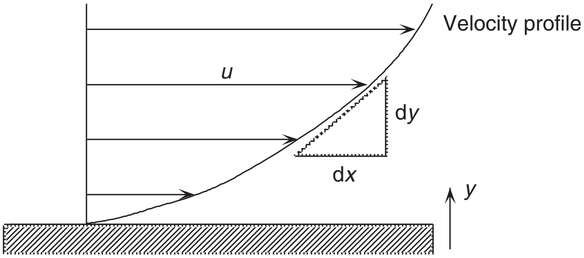

**Figure 1.34** Schematic of velocity profile.

From the viscosity point of view, the types of fluids may be classified into Newtonian and non-Newtonian fluids.

#### 1.11.2.1 Newtonian Fluids

These fluids have a dynamic viscosity dependent upon temperature and pressure and are independent of the magnitude of the velocity gradient. For such fluids, Equation 1.110 is applicable. Some examples are water and air.

#### 1.11.2.2 Non-Newtonian Fluids

The fluids which cannot be represented by Equation 1.110 are called non-Newtonian fluids. These fluids are very common in practice and have a more complex viscous behavior due to the deviation from the Newtonian behavior. There are several approximate expressions to represent their viscous behavior. Some examples are slurries, polymer solutions, oil paints, toothpaste, and sludges.

### *1.11.3 Continuity Equation*

This is based on the *conservation of mass* principle. The requirement that mass be conserved at every point in a flowing fluid imposes certain restrictions on the velocity $u$ and density $\rho$. Therefore, the rate of mass change is zero, referring to that for a steady flow; the mass of fluid in the control volume remains constant and therefore the mass of fluid entering per unit time is equal to the mass of fluid exiting per unit time. Let us apply this to a steady flow in a stream tube (Figure 1.35). The equation of continuity for the flow of a compressible fluid through a stream tube is

$$\rho_1 \delta A_1 u_1 = \rho_2 \delta A_2 u_2 = \text{constant} \tag{1.112}$$

where $\rho_1 \delta A_1 u_1$ is the mass entering per unit time and $\rho_2 \delta A_2 u_2$ is the mass exiting per unit time for the sections 1 and 2.

In practice, for the flow of a real fluid through a pipe or a conduit, the mean velocity is used since the velocity varies from wall to wall. Therefore, Equation 1.112 can be rewritten as

$$\rho_1 A_1 \overline{u}_1 = \rho_2 A_2 \overline{u}_2 = \dot{m} \tag{1.113}$$

where $\overline{u}_1$ and $\overline{u}_2$ are the mean velocities at sections 1 and 2.

For the fluids that are considered as incompressible, Equation 1.113 is simplified to the following, since $\rho_1 = \rho_2$:

$$A_1 \overline{u}_1 = A_2 \overline{u}_2 = \dot{V} \tag{1.114}$$

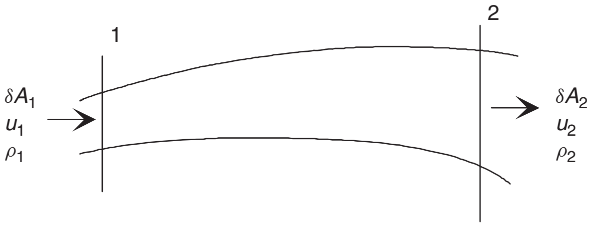

**Figure 1.35** Fluid flow in a stream tube.

## 1.12 General Aspects of Heat Transfer

Thermal processes involving the transfer of heat from one point to another are often encountered in the food industry, as in other industries. The heating and cooling of liquid or solid food products, the evaporation of water vapors, and the removal of heat liberated by a chemical reaction are common examples of processes that involve heat transfer. It is of great importance for food technologists, refrigeration engineers, researchers, and so on, to understand the physical phenomena and practical aspects of heat transfer, along with some knowledge of the basic laws, governing equations, and related boundary conditions.

In order to transfer heat, there must be a driving force, which is the temperature difference between the points where heat is taken and where the heat originates. For example, consider that when a long slab of food product is subjected to heating on the left side, the heat flows from the left-hand side to the right-hand side, which is colder. It is said that heat tends to flow from a point of high temperature to a point of low temperature, with the temperature difference being the driving force.

Many of the generalized relationships used in heat-transfer calculations have been determined by means of dimensional analysis and empirical considerations. It has been found that certain standard dimensionless groups appear repeatedly in the final equations. It is necessary for people working in the food cooling industry to recognize the more important of these groups. Some of the most commonly used dimensionless groups that appear frequently in the heat-transfer literature are given in Table 1.3.

In the utilization of these groups, care must be taken to use equivalent units so that all the dimensions cancel out. Any system of units may be used in a dimensionless group as long as the final result will permit all units to disappear by cancellation.

Basically, heat is transferred in three ways: conduction, convection, and radiation (the so-called modes of heat transfer). In many cases, heat transfer takes place by all three of these methods simultaneously. Figure 1.36 shows the different types of heat-transfer processes as modes. When a temperature gradient exists in a stationary medium, which may be a solid or a fluid, the heat transfer occurring across the medium is by conduction, the heat transfer occurring between a surface and a moving fluid at different temperatures is by convection, and the heat transfer occurring

**Table 1.3** Some of the most important heat-transfer dimensionless parameters.

| Name | Symbol | Definition | Mode |
|---|---|---|---|
| Biot number | Bi | $hY/k$ | Steady- and unsteady-state conduction |
| Fourier number | Fo | $at/Y^2$ | Unsteady-state conduction |
| Graetz number | Gz | $GY^2c_p/k$ | Laminar convection |
| Grashof number | Gr | $g\beta\Delta TY^3/\nu^2$ | Natural convection |
| Rayleigh number | Ra | $Gr \times Pr$ | Natural convection |
| Nusselt number | Nu | $hY/k_\mathrm{f}$ | Natural or forced convection, boiling, or condensation |
| Peclet number | Pe | $UY/a = Re \times Pr$ | Forced convection (for small Pr) |
| Prandtl number | Pr | $c_p\mu/k = \nu/a$ | Natural or forced convection, boiling, or condensation |
| Reynolds number | Re | $UY/\nu$ | Forced convection |
| Stanton number | St | $h/\rho Uc_p = Nu/RePr$ | Forced convection |

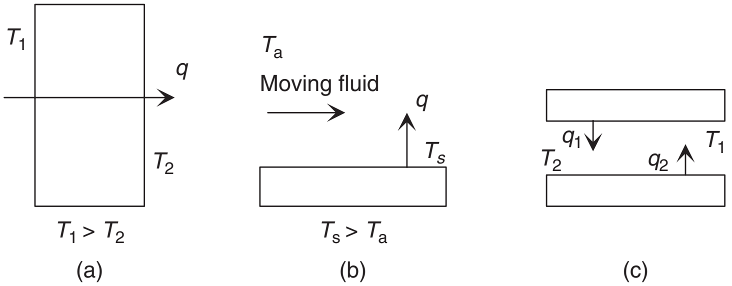

**Figure 1.36** Schematic representations of heat-transfer modes. (a) Conduction through a solid. (b) Convection from a surface to a moving fluid. (c) Radiation between two surfaces.

between two surfaces at different temperatures, in the absence of an intervening medium, is by radiation, where all surfaces of finite temperature emit energy in the form of electromagnetic waves.

## *1.12.1 Conduction Heat Transfer*

Conduction is a mode of transfer of heat from one part of a material to another part of the same material, or from one material to another in physical contact with it, without appreciable displacement of the molecules forming the substance. For example, the heat transfer in a food product subject to cooling in a medium is by conduction.

In solid objects, the conduction of heat is partly due to the impact of adjacent molecules vibrating about their mean positions and partly due to internal radiation. When the solid object is a metal, there are also large numbers of mobile electrons which can easily move through the matter, passing from one atom to another, and they contribute to the redistribution of energy in the metal object. Actually, the contribution of the mobile electrons predominates in metals, which explains the relation that is found to exist between the thermal and electrical conductivity of such materials.

### 1.12.1.1 Fourier's Law of Heat Conduction

Fourier's law states that the instantaneous rate of heat flow through an individual homogeneous solid object is directly proportional to the cross-sectional area $A$ (i.e., the area at right angles to the direction of heat flow) and to the temperature difference driving force across the object with respect to the length of the path of the heat flow, $dT/\mathrm{d}x$. This is an empirical law based on observation.

Figure 1.37 presents an illustration of Fourier's law of heat conduction. Here, a thin slab object of thickness $\mathrm{d}x$ and surface area $F$ has one face at a temperature $T$ and the other at a lower temperature $(T - dT)$ where heat flows from the high-temperature side to the low-temperature side, with a temperature change in the direction of the heat flow $dT$. Therefore, under Fourier's law the heat-transfer equation results in

$$Q = -kA\frac{dT}{\mathrm{d}x} \tag{1.115}$$

Here, we have a term *thermal conductivity*, $k$, of the object that can be defined as the heat flow per unit area per unit time when the temperature decreases by one degree in unit distance. Its units are usually written as $\mathrm{W/m\cdot{}^\circ C}$ or $\mathrm{W/m\cdot K}$.

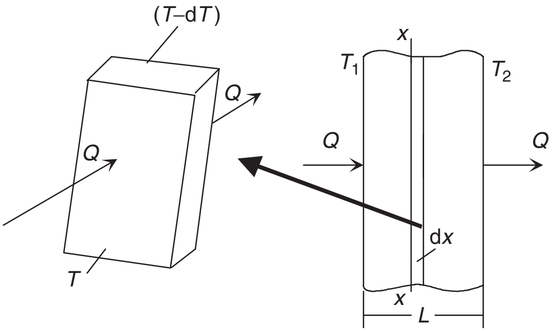

**Figure 1.37** Schematic illustration of conduction in a slab object.

Integrating Equation 1.115 from $T_1$ to $T_2$ for $dT$ and from 0 to $L$ for d$x$, the solution becomes

$$Q = -k\frac{A}{L}(T_2 - T_1) = k\frac{A}{L}(T_1 - T_2) \tag{1.116}$$

## 1.12.2 Convection Heat Transfer

Convection is the heat-transfer mode that takes place within a fluid by mixing one portion of the fluid with another. Convection heat transfer may be classified according to the nature of the flow. When the flow is caused by some mechanical or external means such as a fan, a pump, or atmospheric wind, it is called *forced convection*. On the other hand, for *natural (free) convection* the flow is induced by buoyancy forces in the fluid that arise from density variations caused by temperature variations in the fluid. For example, when a hot food product is exposed to the atmosphere, natural convection occurs, whereas in a cold store forced convection heat transfer takes place between air flow and a food product subject to this flow.

Heat transfer through solid objects is by conduction alone, whereas heat transfer from a solid surface to a liquid or gas takes place partly by conduction and partly by convection. Whenever there is an appreciable movement of the gas or liquid, heat transfer by conduction in the gas or liquid becomes negligibly small compared with the heat transfer by convection. However, there is always a thin boundary layer of liquid on a surface, and through this thin film the heat is transferred by conduction. The convection heat transfer occurring within a fluid is due to the combined effects of conduction and bulk fluid motion. Generally the heat that is transferred is the *sensible*, or internal thermal, heat of the fluid. However, there are convection processes for which there is also *latent* heat exchange, which is generally associated with a phase change between the liquid and vapor states of the fluid.

### 1.12.2.1 Newton's Law of Cooling

Newton's law of cooling states that the heat transfer from a solid surface to a fluid is proportional to the difference between the surface and fluid temperatures and the surface area. This is a particular characteristic of the convection heat-transfer mode and is defined as

$$Q = hA(T_s - T_f) \tag{1.117}$$

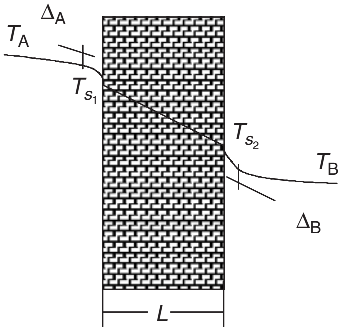

**Figure 1.38** A wall subject to convection heat transfer from both sides.

where $h$ is referred to as the *convection heat-transfer coefficient* (the *heat-transfer coefficient*, the *film coefficient*, or the *film conductance*). It encompasses all the effects that influence the convection mode and depends on conditions in the boundary layer, which is affected by factors such as surface geometry, the nature of the fluid motion, and the thermal and physical properties.

In Equation 1.117, a radiation term is not included. The calculation of radiation heat transfer will be discussed later. In many heat-transfer problems, the radiation effect on the total heat transfer is negligible compared with the heat transferred by conduction and convection from the surface to the fluid. When the surface temperature is high, or when the surface loses heat by natural convection, then the heat transfer due to radiation is of a similar magnitude as that lost by convection.

In order to better understand Newton's law of cooling, consider the heat transfer from a high-temperature fluid A to a low-temperature fluid B through a wall of thickness $x$ (Figure 1.38). In fluid A the temperature decreases rapidly from $T_A$ to $T_{s1}$ in the region of the wall, and similarly in fluid B from $T_{s2}$ to $T_B$. In most cases the fluid temperature is approximately constant throughout its bulk, apart from a thin film ($\Delta_A$ or $\Delta_B$) near the solid surface bounding the fluid. The heat transfers per unit surface area from fluid A to the wall and that from the wall to fluid B are

$$q = h_A(T_A - T_{s1}) \tag{1.118}$$

$$q = h_B(T_{s2} - T_B) \tag{1.119}$$

Also, the heat transfer in thin films is by conduction only as follows:

$$q = \frac{k_A}{\Delta_A}(T_A - T_{s1}) \tag{1.120}$$

$$q = \frac{h_B}{\Delta_B}(T_{s2} - T_B) \tag{1.121}$$

Equating Equations 1.118–1.121, the convection heat-transfer coefficients can be found to be $h_A = k_A/\Delta_A$, and $h_B = k_B/\Delta_B$. Thus, the heat transfer in the wall per unit surface area becomes

$$q = \frac{k}{L}(T_{s1} - T_{s2}) \tag{1.122}$$

For a steady-state heat-transfer case, Equation 1.118 is equal to Equation 1.119 and hence to Equation 1.122

$$q = h_A(T_A - T_{s1}) = h_B(T_{s2} - T_B) = \frac{k}{L}(T_{s1} - T_{s2}) \tag{1.123}$$

The following expression can be extracted from Equation 1.123:

$$q = \frac{(T_A - T_B)}{(1/h_A + L/k + 1/h_B)} \tag{1.124}$$

An analogy can be made with Equation 1.117, and Equation 1.124 becomes

$$Q = HA(T_A - T_B) \tag{1.125}$$

where $1/H = [(1/h_A) + (L/k) + (1/h_B)]$. $H$ is the overall heat-transfer coefficient and consists of various heat-transfer coefficients.

## *1.12.3 Radiation Heat Transfer*

An object emits radiant energy in all directions unless its temperature is absolute zero. If this energy strikes a receiver, part of it may be absorbed and part may be reflected. Heat transfer from a hot to a cold object in this manner is known as *radiation heat transfer*. It is clear that the higher the temperature, the greater is the amount of energy radiated. If, therefore, two objects at different temperatures are placed so that the radiation from each object is intercepted by the other, then the body at the lower temperature will receive more energy than it radiates, and thereby its internal energy will increase; in conjunction with this the internal energy of the object at the higher temperature will decrease. Radiation heat transfer frequently occurs between solid surfaces, although radiation from gases also takes place. Certain gases emit and absorb radiation at certain wavelengths only, whereas most solids radiate over a wide range of wavelengths. The radiative properties of some gases and solids may be found in heat-transfer-related books.

Radiation striking an object can be absorbed by the object, reflected from the object, or transmitted through the object. The fractions of the radiation absorbed, reflected, and transmitted are called the absorptivity $a$, the reflectivity $r$, and the transmittivity $t$, respectively. By definition, $a + r + t = 1$. For most solids and liquids in practical applications, the transmitted radiation is negligible and hence $a + r = 1$. A body which absorbs all radiation is called a *blackbody*. For a blackbody $a = 1$ and $r = 0$.

### 1.12.3.1 The Stefan–Boltzmann Law

This law was found experimentally by Stefan and proved theoretically by Boltzmann. The law states that the emissive power of a blackbody is directly proportional to the fourth power of its absolute temperature. The Stefan–Boltzmann law enables calculation of the amount of radiation emitted in all directions and over all wavelengths simply from knowledge of the temperature of the blackbody. This law is given as follows:

$$E_\mathrm{b} = \sigma T_s^4 \tag{1.126}$$

where $\sigma$ stands for the Stefan–Boltzmann constant, and its value is $5.669 \times 10^{-8}\ \mathrm{W/m^2 \cdot K^4}$. $T_s$ stands for the absolute temperature of the surface.

The energy emitted by a non-blackbody becomes

$$E_{nb} = \varepsilon\sigma T_s^4 \tag{1.127}$$

Then the heat transferred from an object's surface to its surroundings per unit area is

$$q = \varepsilon\sigma(T_s^4 - T_\mathrm{a}^4) \tag{1.128}$$

It is important to explain that if the emissivity of the object at $T_s$ is much different from the emissivity of the object at $T_\mathrm{a}$, then the gray object approximation may not be sufficiently accurate. In this case, it is a good approximation to take the absorptivity of the object 1 when receiving radiation from a source at $T_\mathrm{a}$ as being equal to the emissivity of object 1 when emitting radiation at $T_\mathrm{a}$. This results in

$$q = \varepsilon_{T_s}\sigma T_s^4 - \varepsilon_{T_\mathrm{a}}\sigma T_\mathrm{a}^4 \tag{1.129}$$

There are numerous applications for which it is convenient to express the net radiation heat transfer (radiation heat exchange) in the following form:

$$Q = h_\mathrm{r}A(T_s - T_\mathrm{a}) \tag{1.130}$$

After combining Equations 1.120 and 1.121, the radiation heat-transfer coefficient can be found as follows:

$$h_\mathrm{r} = \varepsilon\sigma(T_s + T_\mathrm{a})(T_s^2 + T_\mathrm{a}^2) \tag{1.131}$$

It is important to note that the radiation heat-transfer coefficient depends strongly on temperature, whereas the temperature dependence of the convection heat-transfer coefficient is generally weak.

The surface within the surroundings may also simultaneously transfer heat by convection to the surroundings. The total rate of heat transfer from the surface is the sum of the convection and radiation modes:

$$Q_\mathrm{t} = Q_\mathrm{c} + Q_\mathrm{r} = h_\mathrm{c}A(T_s - T_\mathrm{a}) + \varepsilon\sigma A(T_s^4 - T_\mathrm{a}^4) \tag{1.132}$$

## 1.13 Concluding Remarks

In this chapter, some general, but key, aspects of thermodynamics, fluid flow, and heat transfer have been presented. Understanding these topics is important as these will serve as background information for the forthcoming chapters.

## Nomenclature

| | |
|---|---|
| $a$ | acceleration, m/s$^2$; thermal diffusivity, m$^2$/s; absorptivity |
| $A$ | cross-sectional area, m$^2$; surface area, m$^2$ |
| $c$ | mass fraction |
| $c_p$ | constant-pressure specific heat, kJ/kg · K |
| $c_v$ | constant-volume specific heat, kJ/kg · K |
| COP | coefficient of performance |
| $E$ | energy, kJ |
| $\dot{E}$ | rate of energy, kW |
| $Ex$ | amount of exergy, kJ |
| $Ex_\mathrm{destroyed}$ | exergy destruction, kJ |
| $\dot{E}x$ | rate of exergy, kW |
| $F$ | force; drag force, N |
| Fo | Fourier number |

| | |
|---|---|
| $g$ | acceleration due to gravity (= 9.81 m/s$^2$) |
| Gr | Grashof number |
| Gz | Graetz number |
| $h$ | specific enthalpy, kJ/kg; heat-transfer coefficient, W/m$^2 \cdot$°C |
| $H$ | entalpy, kJ; overall heat-transfer coefficient, W/m$^2 \cdot$°C; head, m |
| $k$ | specific heat ratio; thermal conductivity, W/m $\cdot$ °C |
| KE | kinetic energy, W or kW |
| $L$ | thickness, m |
| $m$ | mass, kg |
| $\dot{m}$ | mass flow rate, kg/s |
| $M$ | molecular weight, kg/kmol |
| $n$ | mole number, kmol |
| Nu | Nusselt number |
| $P$ | pressure, kPa |
| Pe | Peclet number |
| $PE$ | potential energy, W or kW |
| Pr | Prandtl number |
| $\dot{q}$ | heat rate per unit area, W/m$^2$ |
| $Q$ | amount of heat transfer, kJ |
| $\dot{Q}$ | heat-transfer rate, kW |
| $r$ | reflectivity; radial coordinate; radial distance, m |
| $R$ | gas constant, kJ/kg $\cdot$ K; radius, m |
| $\overline{R}$ | universal gas constant, kJ/kg $\cdot$ K |
| Ra | Rayleigh number |
| Re | Reynolds number |
| $s$ | specific entropy, kJ/kg |
| $S$ | entropy, kJ/K |
| $S_{\mathrm{gen}}$ | entropy generation, kJ/K |
| St | Stanton number |
| $t$ | time, s; transmittivity |
| $T$ | temperature, °C or K |
| $T_{\mathrm{s}}$ | absolute temperature of the object surface, K |
| $u$ | specific internal energy, kJ/kg |
| $U$ | internal energy, kJ; flow velocity, m/s |
| $v$ | specific volume, m$^3$/kg |
| $\overline{v}$ | molal specific volume, kmol/kg |
| $V$ | volume, m$^3$; velocity, m/s |
| $\dot{V}$ | volumetric flow rate, m$^3$/s |
| $W$ | amount of work, kJ |
| $\dot{W}$ | power, W or kW |
| $x$ | quality, kg/kg |
| $X$ | length for plate, m |
| $y$ | mole fraction |
| $Z$ | compressibility factor |

## Greek Letters

| | |
|---|---|
| $\Delta T$ | temperature difference, K; overall temperature difference, °C or K |
| $\varepsilon$ | surface emissivity |
| $\eta$ | efficiency |
| $\eta_{\mathrm{th}}$ | thermal efficiency |
| $\eta_{\mathrm{ex}}$ | exergy (second-law) efficiency |

| | |
|---|---|
| $\mu$ | dynamic viscosity, kg/ms; root of the characteristic equation |
| $\psi$ | flow exergy, kJ/kg |
| $\nu$ | kinematic viscosity, m$^2$/s |
| $\rho$ | density, kg/m$^3$ |
| $\sigma$ | Stefan–Boltzmann constant, W/m$^2 \cdot$ K$^4$ |
| $\tau$ | shear stress, N/m$^2$ |
| $\phi$ | relative humidity, % |
| $\omega$ | humidity ratio, kg/kg |

### *Subscripts and Superscripts*

| | |
|---|---|
| a | air; medium; surroundings |
| av | average |
| db | dry-bulb |
| H | high-temperature |
| in | input |
| l | liquid |
| liq | liquid |
| L | low-temperature |
| out | output |
| tot | total |
| v | vapor |
| vap | vapor |
| wb | wet-bulb |
| 0 | surroundings; ambient; environment; reference |

## Study Problems

### *Introduction, Thermodynamic Properties*

**1.1** Why are SI units most widely used throughout the world?

**1.2** What is the difference between mass and weight?

**1.3** What is specific heat? Define two specific heats used. Is specific heat a function of temperature?

**1.4** Explain operating principle of thermocouples. What are some typical applications depending on the type of thermocouples? What is the main advantage of thermocouple over other temperature sensors?

**1.5** Consider the flow of a refrigerant vapor through a compressor, which is operating at steady-state conditions. Do mass flow rate and volume flow rate of the refrigerant across the compressor remain constant?

**1.6** Consider a refrigeration system consisting of a compressor, an evaporator, a condenser, and an expansion valve. Do you evaluate each component as a closed system or as a control volume; as a steady-flow system or unsteady-flow system? Explain.

**1.7** What is the difference between an adiabatic system and an isolated system?

**1.8** Define intensive and extensive properties. Identify the following properties as intensive or extensive: mass, volume, density, specific volume, energy, specific enthalpy, total entropy, temperature, pressure.

**1.9** Define sensible and latent heats, and latent heat of fusion. What are their units.

**1.10** What is the weight of a 10-kg substance in N, kN, $kg_f$, and lbf?

**1.11** The vacuum pressure of a tank is given to be 40 kPa. If the atmospheric pressure is 95 kPa, what is the gage pressure and absolute pressure in kPa, $kN/m^2$, $lbf/in^2$, psi, and mm Hg.

**1.12** Express −40 °C temperature in Fahrenheit (°F), Kelvin (K), and Rankine (R) units.

**1.13** The temperature of air changes by 10 °C during a process. Express this temperature change in Kelvin (K), Fahrenheit (°F), and Rankine (R) units.

**1.14** The specific heat of water at 25 °C is given to be 4.18 kJ/kg · °C. Express this value in kJ/kg · K, J/g · °C, kcal/kg · °C, and Btu/lbm · °F.

**1.15** A 0.2-kg of R134a at 700 kPa pressure initially at 4 °C is heated until 50% of mass is vaporized. Determine the temperature at which the refrigerant is vaporized, and sensible and latent heat transferred to the refrigerant.

**1.16** A 0.5-lbm of R134a at 100 psia pressure initially at 40 °F is heated until 50% of mass is vaporized. Determine the temperature at which the refrigerant is vaporized, and sensible and latent heat transferred to the refrigerant.

**1.17** A 2-kg ice initially at −18 °C is heated until 75% of mass is melted. Determine sensible and latent heat transferred to the water. The specific heat of ice at 0 °C is 2.11 kJ/kg · °C. The latent heat of fusion of water at 0 °C is 334.9 kJ/kg.

**1.18** A 2-kg ice initially at −18 °C is heated until it exists as liquid water at 20 °C. The specific heat of ice at 0 °C is 2.11 kJ/kg · °C. The latent heat of fusion of water at 0 °C is 334.9 kJ/kg. Determine sensible and latent heat transferred to the water.

**1.19** Refrigerant-134a enters the evaporator of a refrigeration system at −24 °C with a quality of 25% at a rate of 0.22 kg/s. If the refrigerant leaves evaporator as a saturated vapor, determine the rate of heat transferred to the refrigerant. If the refrigerant is heated by water in the evaporator, which experiences a temperature rise of 16 °C, determine the mass flow rate of water.

## *Ideal Gases and the First Law of Thermodynamics*

**1.20** What is compressibility factor?

**1.21** What is an isentropic process? Is a constant-entropy process necessarily reversible and adiabatic?

**1.22** What is the difference between heat and work.

**1.23** An elastic tank contains 0.8 kmol of air at 23 °C and 600 kPa. Determine the volume of the tank. The volume is now doubled at the same pressure. What is the temperature at this state?

**1.24** An elastic tank contains 1.4 lb mol of air at 79 °F and 80 psia. Determine the volume of the tank. The volume is now doubled at the same pressure. What is the temperature at this state?

**1.25** A 50-liter piston-cylinder device contains oxygen at 52 °C and 170 kPa. Now the oxygen is heated until the temperature reaches 77 °C. What is the amount of heat transfer during this process?

**1.26** A 50-liter rigid tank contains oxygen at 52 °C and 170 kPa. Now the oxygen is heated until the temperature reaches 77 °C. What is the amount of heat transfer during this process?

**1.27** A 50-liter rigid tank contains oxygen at 52 °C and 170 kPa. Now the oxygen is heated until the temperature reaches 77 °C. What is the entropy change during this process?

**1.28** A rigid tank contains 2.5 kg oxygen at 52 °C and 170 kPa. Now the oxygen is heated in an isentropic process until the temperature reaches 77 °C. What is the pressure at final state? What is the work interaction during this process?

**1.29** A piston-cylinder device contains 2.5 kg oxygen at 52 °C and 170 kPa. Now the oxygen is heated until the temperature reaches 77 °C. Determine the work done and the amount of heat transfer during this process?

## *Refrigerators, Heat Pumps, and the Carnot Refrigeration Cycle*

**1.30** Show that for given values $Q_L$ and $Q_H$ the COPs of a refrigerator and a heat pump are related to each other by $\mathrm{COP_{HP}} = \mathrm{COP_R} + 1$.

**1.31** How can the COP of a Carnot refrigerator be increased?

**1.32** A Carnot refrigerator is used to keep a space at 18 °C by rejecting heat to a reservoir at 35 °C. If the heat removal from the cooled space is 12,000 kJ/h, determine the COP of the refrigerator and the power input in kW.

**1.33** A Carnot refrigerator is used to keep a space at 65 °F by rejecting heat to a reservoir at 90 °F. If the heat removal from the cooled space is 10,500 Btu/h, determine the COP of the refrigerator and the power input in kW.

**1.34** A Carnot refrigerator is used to keep a space at −20 °C. If the COP of the refrigerator is 7.5, what is the temperature of the reservoir to which heat is rejected? For a power input of 3.7 kW, what is the rate of heat rejected to high-temperature reservoir?

## *Exergy*

**1.35** What are the two main reasons that cause irreversibility?

**1.36** What happens to the parameters mass, energy, entropy, and exergy during an irreversible process: conserved, decreases, or increases?

**1.37** How does an exergy analysis help the goal of more efficient energy resource use? What are the advantageous of using an exergy analysis?

## *Psychrometrics*

**1.38** What is the difference between humidity ratio and relative humidity?

**1.39** Why is heating usually accompanied by humidification and cooling by dehumidification?

**1.40** Consider moist air at 24 °C at sea level with a relative humidity of 70%. Using psychrometric chart, determine humidity ratio, wet-bulb temperature, and enthalpy of moist air.

**1.41** Consider moist air at 25 °C at sea level with a relative humidity of 40%. What is the partial pressure of water vapor in the air? The saturation pressure of water at 25 °C is 3.17 kPa.

**1.42** Consider moist air at 80 °F at sea level with a relative humidity of 40%. What is the partial pressure of water vapor in the air? The saturation pressure of water at 80 °F is 0.507 psia.

**1.43** Air at 1 atm and 32 °C with a relative humidity of 20% enters an evaporative cooling section whose effectiveness is 80%. What is the air temperature at the exit of the evaporative cooler?

## *General Aspects of Fluid Flow*

**1.44** What is the physical meaning of the Reynolds number? What makes the flow laminar and what makes it turbulent?

**1.45** What is viscosity? How does viscosity change with temperature for gases and for liquids?

## *General Aspects of Heat Transfer*

**1.46** What are the modes of heat transfer? Explain mechanism of each mode.

**1.47** A 20-cm thick wall of a house made of brick ($k = 0.72\text{W/m} \cdot {}^\circ\text{C}$) is subjected to inside air at 22 °C with a convection heat-transfer coefficient of $15\,\text{W/m}^2 \cdot {}^\circ\text{C}$. The inner surface temperature of the wall is 18 °C and the outside air temperature is −1 °C. Determine the outer surface temperature of the wall and the heat-transfer coefficient at the outer surface.

**1.48** A satellite is subjected to solar energy at a rate of $300\ \text{W/m}^2$. The absorptivity of the surface is 0.75 and its emissivity is 0.60. Determine the equilibrium temperature of the satellite.

**1.49** A satellite is subjected to solar energy at a rate of $95\,\text{Btu/h} \cdot \text{ft}^2$. The absorptivity of the surface is 0.80 and its emissivity is 0.65. Determine the equilibrium temperature of the satellite.

**1.50** An 80-cm-diameter spherical tank made of steel contains liquefied natural gas (LNG) at −160 °C. The tank is insulated with 4-cm-thick insulation ($k = 0.015\,\text{W/m} \cdot {}^\circ\text{C}$). The tank is subjected to ambient air at 18 °C with a convection heat-transfer coefficient of $20\,\text{W/m}^2 \cdot {}^\circ\text{C}$. How long will it take for the temperature of the LNG to drop to −150 °C. Neglect the thermal resistance of the steel tank. The density and the specific heat of LNG are $425\,\text{kg/m}^3$ and $3.475\,\text{kJ/kg} \cdot {}^\circ\text{C}$, respectively.

# References

Borgnakke, C. and Sonntag, R. (2008) *Fundamentals of Thermodynamics*, 7th edn, John Wiley & Sons, Ltd., New York.

Cengel, Y.A. and Boles, M.A. (2008) *Thermodynamics: An Engineering Approach*, 6th edn, McGraw Hill, New York..

Dincer, I. (2002) The role of exergy in energy policy making. *Energy Policy*, **30**, 137–149.

Dincer, I. (2003) *Refrigeration Systems and Applications*, 1st edn, John Wiley & Sons, Ltd., New York.

Dincer, I. and Rosen, M.A. (2005) Thermodynamic aspects of renewables and sustainable development. *Renewable and Sustainable Energy Reviews*, **9**, 169–189.

Klein, S.A. (2006) Engineering equation solver (EES), F-Chart Software, www.fChart.com.

Marquand, C. and Croft, D. (1997) *Thermofluids – An Integrated Approach to Thermodynamics and Fluid Mechanics Principles*, John Wiley & Sons, Ltd., New York.

Moran, M.J. and Shapiro, H.N. (2007) *Fundamentals of Engineering Thermodynamics*, 6th edn, John Wiley & Sons, Ltd., New York.

Raznjevic, K. (1995) *Handbook of Thermodynamic Tables*, 2nd edn, Begell House, New York.