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General Aspects of Thermodynamics

1.1 Introduction

Refrigeration has a diverse nature 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 due to 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 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, that 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 further on. While the information in some situations is limited, it is desirable that the reader comprehend 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, etc., discusses the relationships between the aspects and provides some key examples for better understanding.

This chapter primarily focuses on general aspects of thermodynamics, ranging from dimensions and units to psychrometric processes, and specifically discusses systems of units, thermodynamic systems, thermodynamic laws, pure substances, ideal and real gases, refrigerators and heat pumps, Carmot cycles, and psychrometrics and its processes. We also introduce performance assessment criteria through energy and exergy efficiencies and energetic and exergetic coefficients of performance (COPs) by the thermodynamic laws. The chapter presents lots of examples to show how to utilize thermodynamic tools, particularly balance equations, for design, analysis, and assessment.

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1.2 Dimensions and Units

In the area of refrigeration it is critical to employ dimensions and units correctly for analysis, design, and assessment. It is commonly accepted that any physical quantity can be characterized by dimensions. Their magnitudes are measured/recognized in units. There are numerous commonly accepted dimensions, namely mass (m), length (L), time (t), and temperature (T), which are treated as primary quantities. There are also several other quantities, such as force (F), pressure (P), velocity (V), energy (E), and exergy (Ex), which are treated as the derived dimensions. We discuss several of these in the following subsections.

1.2.1 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 SI units, and the *English System of Units* (the *English Engineering System*). SI units are the most widely used throughout the world, although the English System is utilized as the traditional system of North America. In this book, SI units are primarily employed. Appendix A contains some common conversions. The dimensions, such as mass, length, force, density, specific volume, mass flow rate, volumetric flow rate, temperature and pressure, are briefly described below.

1.2.1.1 Mass

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 SI and its unit in the English System is the pound mass (lb_m). The basic unit of time for both unit systems is the second (s). The following relationships exist between the two unit systems:

$$1\ \text{kg} = 2.2046\ \text{lb}_\text{m} \quad \text{or} \quad 1\ \text{lb}_\text{m} = 0.4536\ \text{kg}$$

$$1\ \text{kg/s} = 7936.6\ \text{lb}_\text{m}/\text{h} = 2.2046\ \text{lb}_\text{m}/\text{s}$$

$$1\ \text{lb}_\text{m}/\text{h} = 0.000126\ \text{kg/s}$$

$$1\ \text{lb}_\text{m}/\text{s} = 0.4536\ \text{kg/s}$$

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 defined as

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

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

1.2.1.2 Length

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

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

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

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

1.2.1.3 Force

A 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):

$$1\ \text{N} = 0.22481\ \text{lb}_\text{f} \quad \text{or} \quad 1\ \text{lb}_\text{f} = 4.448\ \text{N}$$

The four aspects, that is, 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 acceleration in the direction of the force, as given below:

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

Equation (1.2) shows the force required to accelerate a mass of one kilogram at a rate of one meter per square second as $1\ \text{N} = 1\ \text{kg m/s}^2$.

It is important to note the value of the earth's gravitational acceleration as $9.80665\ \text{m/s}^2$ (generally taken as $9.81\ \text{m/s}^2$) in the SI system and $32.174\ \text{ft/s}^2$ in the English System, which indicates that a body falling freely toward the surface of the earth is subject to the action of gravity alone. Some common conversion factors are listed in Appendix in A.

1.2.1.4 Density and Specific Volume

Specific volume is defined as 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/lb) 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:

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

Its units are kg/m^3 in the SI system and $\text{lb}_\text{m}/\text{ft}^3$ 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 affected by temperature and pressure. The related interconversions are

$$1\ \text{kg/m}^3 = 0.06243\ \text{lb}_\text{m}/\text{ft}^3 \quad \text{or} \quad 1\ \text{lb}_\text{m}/\text{ft}^3 = 16.018\ \text{kg/m}^3$$

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

1.2.1.5 Mass Flow Rate and Volumetric Flow Rate

Mass flow rate is defined as the mass flowing per unit time (kg/s in the SI system and lb/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:

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

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

1.2.1.6 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 Celsius (°C) and Fahrenheit (°F). Normally, the Celsius scale is used with the SI unit system and the Fahrenheit scale with the English System. There are also two more scales, the Kelvin scale (K) and the Rankine scale (R), which are sometimes employed in thermodynamic applications. The relations between these scales are summarized as follows:

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

$$T(\mathrm{K}) = T(^\circ\mathrm{C}) + 273.15 = \frac{T(\mathrm{R})}{1.8} = \frac{T(^\circ\mathrm{F}) + 459.67}{1.8} \tag{1.9}$$

$$T(^\circ\mathrm{F}) = 1.8T(^\circ\mathrm{C}) + 32 = 1.8(T(\mathrm{K}) - 273.15) + 32 \tag{1.10}$$

$$T(\mathrm{R}) = 1.8T(\mathrm{K}) = T(^\circ\mathrm{F}) + 459.67 \tag{1.11}$$

Furthermore, the temperature differences result in

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

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

Here, Kelvin is a unit of temperature measurement: zero Kelvin (0 K) is the absolute zero and is equal to −273.15 °C. Both 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 many devices. In general, the following devices are in common use:

- **Liquid-in-glass thermometers**. It is known that in these thermometers the volume of 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. Amongst 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 °C and +800 °C, the majority of such thermometers are made of platinum. In industry, in addition to platinum, nickel (−60 °C to +180 °C) and copper (−30 °C to +220 °C) are frequently used to manufacture resistance thermometers. Resistance thermometers can be provided with two, three, or four wire connections and for higher accuracy at least three 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 the sheath material (stainless steel for the temperature range from −50 °C to +200 °C or nylon for the temperature range from −50 °C 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 multi-element 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 °C to 100 °C, the maximum temperature may reach 200 °C. To keep high accuracy these units are normally supplied with three-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 (the 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 (the so-called *cold junction*). The thermo-electric 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 of 850 °C, is used in applications in the plastics industry. Chromel-alumel-type thermocouples, with a maximum of about 1100 °C, are suitable for combustion applications in ovens and furnaces. In addition, it is possible to reach about 1600 °C 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 semi-conductors and act as thermal resistors with a high (usually negative) temperature coefficient. Thermistors operate either by self-heating or are 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 °C 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, such as digital hand-held, traceable, and traceable infrared, is available on the market for various thermal applications. The temperature ranges change from −50 °C to 1000 °C with high accuracy, for example ±0.3%, and resolution, for example ±0.1 °C). Digital thermometers are configured for compatibility with one of the common temperature transducers and the thermistor. Each of the configurations linearizes the analog output from the transducer to produce a stable and accurate digital temperature display. The use of advanced digital design techniques gives indicators with extremely accurate readings and long battery life.

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 and also contamination effects, lead lengths, immersion, heat transfer and controller interfacing. In temperature control there are many sources of error which can be 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 in this book. The selection of temperature measurement devices is a complex task and is discussed briefly here. It is extremely important to remember to choose “the right tool for the right job”. Recently, data acquisition devices have come into common use.

1.2.1.7 Pressure

When we deal with liquids and gases, pressure becomes one of the most important components. Pressure is defined as 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 written as

$$P = \frac{F}{A} \tag{1.12}$$

Table 1.1 Some of most common thermocouples

Type	Common names	Temperature range (°C)
T	Copper–constantan (C/C)	−250–400
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	0–1600

The unit for pressure in SI is the force of one newton acting on a square meter 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, lb_f/ft^2. Here are some pressure conversions:

$$\begin{aligned}
1 \text{ Pa} &= 0.020886 \text{ lb}_f/\text{ft}^2 = 1.4504 \times 10^{-4} \text{ lb}_f/\text{in}^2 = 4.015 \times 10^{-3} \text{ in water} \\
&= 2.953 \times 10^{-4} \text{ in Hg} \\
1 \text{ lb}_f/\text{ft}^2 &= 47.88 \text{ Pa} \\
1 \text{ lb}_f/\text{in}^2 &= 1 \text{ psi} = 6894.8 \text{ Pa} \\
1 \text{ bar} &= 1 \times 10^5 \text{ Pa}
\end{aligned}$$

Here, we introduce the basic pressure definitions, and a summary of these basic pressure measurement relationships is shown in Figure 1.1. There are basically four categories of pressure, atmospheric, gauge, absolute, and vacuum, which are described in the following subsections.

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.

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

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 a reference level for other types of pressure measurements, for example gauge pressure.

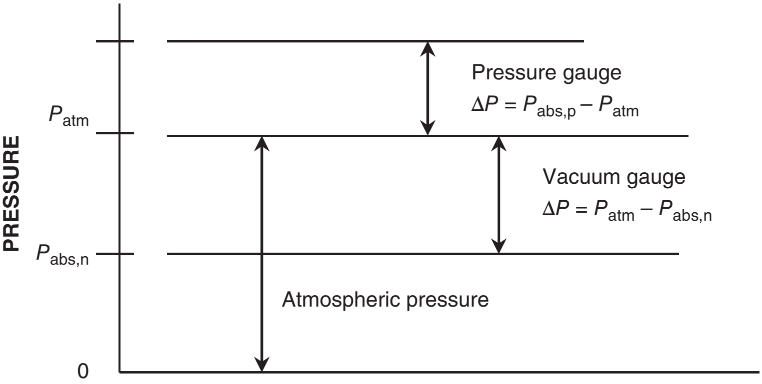


Figure 1.1 Illustration of pressure relationships.

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.

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:

$$\text{absolute pressure} = \text{gauge pressure} + \text{atmospheric pressure (in kPa)} \tag{1.13}$$

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 absolute pressure and the atmospheric pressure existing in the gauge being read by most pressure gauges and indicators.

Vacuum A vacuum is a pressure lower than atmospheric and occurs only in closed systems, except in outer space. It is also called the *negative gauge pressure*. 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 (non-mechanical 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, the two principal types of measuring devices for refrigeration applications are manometers and Bourdon gauges. However, in many cases manometers are not preferred due to the excessive length of tube needed, their inconvenience for pressures much in excess of 1 atm, and lower 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. 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 that the *saturation pressure* is the pressure of a liquid or vapor at saturation conditions.

1.3 Thermodynamics

Thermodynamics is defined as the science of energy and exergy while in most books it is defined as the science of energy and entropy. There are essentially four laws of thermodynamics, the zeroth law, the first law, the second law and the third law, and two of these essentially shape the thermodynamic principles, namely first law and the second law. We can now connect this to our original definition through two paths: (i) the energy path, which comes from the first law of thermodynamics, and (ii) the exergy path, which comes from the second law of thermodynamics. The definition provided here is more consistent with the concepts for the following reasons:

- both energy and exergy quantities are denoted in the same units
- energy efficiency comes from the concept of energy, and exergy efficiency comes from the concept of exergy.

Thermodynamics is a critical subject in designing, analyzing, assessing, and improving refrigeration systems, and appropriate teaching of thermodynamics is critical in correctly applying the concepts and first and second laws to refrigeration systems and applications.

1.3.1 Thermodynamic Systems

In thermodynamics, any device or process or combination of these devices or processes contains a certain quantity of matter being studied. It is important to carefully define the term "system" as that portion of all matter under consideration. The types of thermodynamic systems can be classified into two main categories 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 comes in or goes out. In some books it is called *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*.

Note that there is a special form of thermodynamic system called an *isolated system*, which is a closed system in which no mass, heat or work crosses the system boundary. Such systems are assumed to be not affected by the surroundings.

1.3.2 Thermodynamic Laws

The science of thermodynamics is guided and governed by certain laws, just as in any society things are guided and governed by certain laws or rules or regulations. These are essential to provide systematic order and harmony.

In thermodynamics there are four laws as follows:

- *Zeroth law of thermodynamics*: This laws states that if there are two objects (A and B) in thermal equilibrium with another object (C), all three objects (A, B, and C) are ultimately in thermal equilibrium with each other. This is recognized as a special law and a basis of thermometers for temperature measurements.
- *First law of thermodynamics*: This law is known as the energy conservation law and states that energy is neither created nor destroyed due to the conservation. However, it may change from heat to work (such as a thermal power plant, where we supply heat to produce mechanical work) and work to heat (such as a heat pump, where we provide mechanical/electrical work to achieve heating). This is the most widely recognized and utilized law. It has a big deficiency in that it does not recognize irreversibilities and inefficiencies. So, the bottom line is that this law is necessary, but not sufficient.
- *Second law of thermodynamics*: This law is recognized as the law which practically governs all thermodynamic systems and measures all irreversibilities and inefficiencies, helps to achieve only the practically possible performance, and confirm what is practically impossible. We can define it as a true measuring tool for irreversibilities and inefficiencies. It is known as “non-conservation of exergy” principle. This will be further discussed and treated under the exergy section.
- *Third law of thermodynamics*: This law is helpful in determining the absolute values of entropy and their changes during the processes. It states that there is a situation (such as freezing) where the molecules of a substance in the solid phase are considered to be not moving (stationary) at absolute zero. This condition is recognized as the state where molecular order exists with minimum energy. This brings us to a point where the entropy of a pure crystalline substance at absolute zero temperature becomes zero, which is known as the third law of thermodynamics. This law confirms the absolute reference point for the purpose of determining entropy. The entropy determined relative to this point is called absolute entropy, which is useful in the thermodynamic analysis of chemical reactions.

In the following two subsections we will only discuss both first and second laws of thermodynamics in detail as the governing laws for the thermodynamic systems while the other two laws (zeroth and third laws) are related to states and/or conditions only.

1.3.3 First Law of Thermodynamics

Thermodynamics is the science of energy and entropy, and 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 due to the fact that transfers and conversions of energy are governed by these two laws, and in this chapter we will 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:

$$E_{\mathrm{in}} - E_{\mathrm{out}} = \Delta E_{\mathrm{system}} \tag{1.14}$$

In rate form,

$$\dot{E}_{\mathrm{in}} - \dot{E}_{\mathrm{out}} = \Delta \dot{E}_{\mathrm{system}} \tag{1.15}$$

For a closed system undergoing a process between initial and final states involving heat and work interactions with the surroundings (Figure 1.2), it is written as:

$$\begin{aligned} E_{\mathrm{in}} - E_{\mathrm{out}} &= \Delta E_{\mathrm{system}} \\ (Q_{\mathrm{in}} + W_{\mathrm{in}}) - (Q_{\mathrm{out}} + W_{\mathrm{out}}) &= \Delta U + \Delta \mathrm{KE} + \Delta \mathrm{PE} \end{aligned} \tag{1.16}$$

If there are no changes in kinetic and potential energies, it becomes:

$$(Q_{\mathrm{in}} + W_{\mathrm{in}}) - (Q_{\mathrm{out}} + W_{\mathrm{out}}) = \Delta U = m(u_2 - u_1) \tag{1.17}$$

Let us consider a control volume involving a steady-flow process. Mass is entering and leaving the system and there are heat and work interactions with the surroundings (Figure 1.3). During a 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

$$\begin{aligned} \dot{E}_{\mathrm{in}} - \dot{E}_{\mathrm{out}} &= \Delta \dot{E}_{\mathrm{system}} = 0 \\ \dot{E}_{\mathrm{in}} &= \dot{E}_{\mathrm{out}} \\ \dot{Q}_{\mathrm{in}} + \dot{W}_{\mathrm{in}} + \dot{m}h_{\mathrm{in}} &= \dot{Q}_{\mathrm{out}} + \dot{W}_{\mathrm{out}} + \dot{m}h_{\mathrm{out}} \end{aligned} \tag{1.18}$$

Here, the changes in kinetic and potential energies are considered negligible.

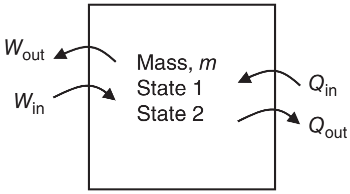


Figure 1.2 A general closed system with heat and work interactions.

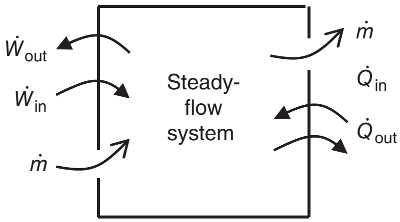


Figure 1.3 A general steady-flow control volume with mass, heat, and work interactions.

An important consequence of the FLT 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.

1.3.4 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., a 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 operating with an 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). This simply shows the impossibility of running a refrigerator or heat pump without any work input.

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 into another form of potential energy. The energy is transferred from one ball to the other. The last ball gains kinetic energy to go up again. The cycle continues but each time the ball goes 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.

In addition, the SLT introduces two key concepts, namely entropy and exergy. It 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 the 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". This 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 exerted in minimizing the entropy generation (and hence irreversibility) in thermodynamic systems and applications.

The SLT is recognized as a useful tool in determining the following:

- direction of process
- behavior of system
- irreversibility level

- actual performance
- effect of surroundings
- effect of varying operating conditions and state properties
- operational possibilities.

Consequently, the SLT appears to be a clear linkage between entropy and usefulness of energy, and SLT analysis has found applications even in a large variety of non-engineering disciplines, for example chemistry, economics, ecology, environment, sociology, etc., which is far from the conventional engineering thermodynamic applications.

1.3.4.1 Exergy and its Importance

The science of thermodynamics is built primarily on two fundamentally-driven 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 always remains constant. The SLT asserts that energy has quality as well as quantity, and that actual processes occur in the direction of decreasing quality of energy. The high-temperature thermal energy (heat) 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 introduced as a potential thermodynamic method 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 the efficiency which provide a true measure of the system performance and a critical indicator of how much deviation takes place from the ideality (reversibility). It primarily determines the true magnitudes of thermodynamic losses, inefficiencies and irreversibilities and their exact states and locations. It also potentially helps in quantifying and assessing the environmental impact and sustainability. Furthermore, an effective utilization of exergy analysis tools is enhanced by including improvement, assessment and optimization studies.

Energy and exergy efficiencies are considered by many to be useful for the performance assessments of energy systems and applications. By considering both of these efficiencies, both quantity (through energy) and quality (through exergy) of the energy used to achieve a specific task can easily determined. Improving the efficiencies of energy systems is, thus, recognized an important task for meeting energy policy and strategy objectives. Such efforts can assist in attaining better use of energy and natural resources [1].

An engineer designing a refrigeration system or plant is often expected to aim for achieving the highest possible exergy efficiency or exergetic coefficient of performance at the lowest cost under the prevailing technical, economic, environmental, and legal conditions, and fulfilling ethical, ecological, and social responsibilities. Note that exergy methods can assist in such activities and offer unique insights into possible improvements with special emphasis on the environment and sustainability. Exergy analysis, in this regard, appears to be a critically 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 or not and by how much it is possible to design more efficient energy systems by reducing inefficiencies. As a matter of fact, it is a key tool for system design, analysis, assessment, evaluation and improvement.

More specifically, the maximum energy availability or 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. It is important to note that 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 accordingly [1].

A system is said to be in the dead state when it is in thermodynamic equilibrium with its environment. In 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 $T_0 = 25\,°\text{C}$ (77 °F) and $P_0 = 1$ atm (101.325 kPa or 14.7 psia), respectively. 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 as per ASHRAE's Sustainability Roadmap [2].

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, the 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 the thermal energy of a system can be converted to work. In another words, the exergy of thermal energy is less than the magnitude of thermal energy.

1.3.4.2 Reversibility and Irreversibility

These two concepts are considered highly important components in analyzing thermodynamic systems. The *reversibility* is defined as the statement that both the system and its surroundings can be returned to their initial states, which represent the ideality for a given or considered case. The irreversibility, however, 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, electrical and mechanical effects, etc. For instance, as an actual system provides an amount of work that is less than the ideal reversible work, so 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.

1.3.4.3 Reversible Work and Exergy Destruction

The reversible work W_{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_{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

$$Ex_{\text{destroyed}} = W_{\text{rev, out}} - W_{\text{out}} \quad \text{or} \quad Ex_{\text{destroyed}} = W_{\text{in}} - W_{\text{rev, in}} \tag{1.19}$$

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.

1.3.5 Dincer's Six-step Approach

In teaching or explaining thermodynamics, a novel approach is proposed through Dincer's six-step approach. This approach consists of six critical steps, namely property, state, process, cycle, balance equations, and performance assessment, as illustrated in Figure 1.4. This provides a logical order and makes it simple to understand and comprehend the thermodynamic concepts and aspects.

Step 1: Property

Property is defined as a physical characteristic or quantity of a substance which is treated as a key attribute of any thermodynamic system. Any two properties usually

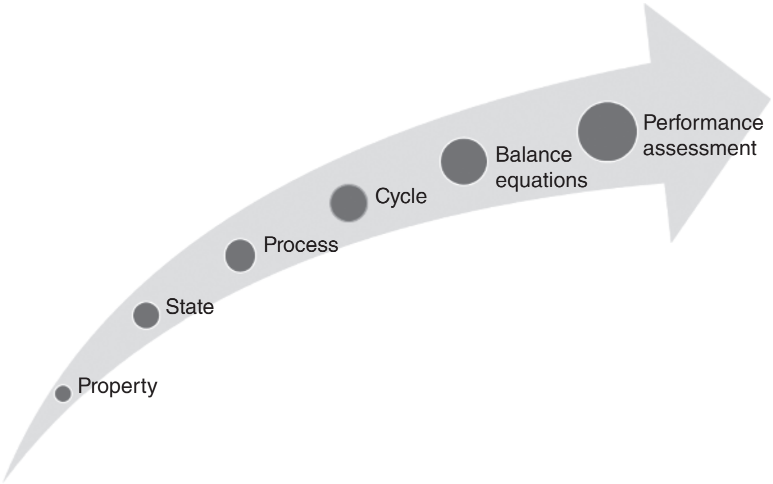


Figure 1.4 An illustration of Dincer's six-step approach.

define the state or condition of the substance, from which all other properties can be derived. There are measureable properties, such as temperature, pressure, mass, and volume, and there are also non-measurable properties, such as internal energy, enthalpy, entropy, and exergy. 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.

Step 2: State

State is defined as a true condition of any thermodynamic system defined by the thermodynamic properties (particularly the measureable properties). In order to obtain the other unknown properties for the subject matter state, there is a need at least for two known properties. There may be situations where the unknown properties of any state are obtained by using a known property and the state-related information, such saturated liquid, saturated vapor, etc. In a refrigeration system, there are four common state points: saturated vapor after the evaporator, superheated vapor after the compressor, saturated liquid after the condenser, and a mixture of liquid and vapor after the throttling valve.

Step 3: Process

Process is defined as 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 processes such as isothermal (constant temperature), isobaric (constant pressure), and isochoric (constant volume). A refrigeration process is generally expressed by the conditions or properties of the refrigerant at the beginning and end of the process.

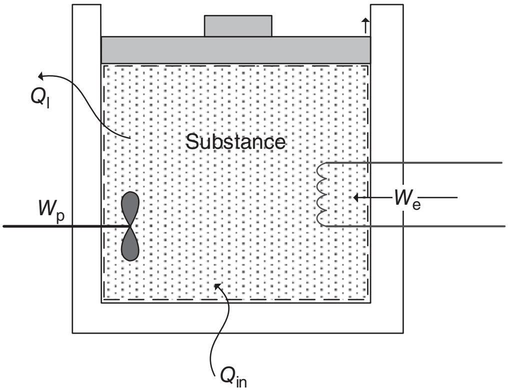


Figure 1.5 A closed system.

Step 4: Cycle

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 continuously be reused in a closed system. In thermodynamics, there are cycles, such as closed type and open type. A good example comes from the air Brayton cycle which has both closed and open versions.

Step 5: Balance Equations

Writing the thermodynamic balance equations for mass, energy, entropy, and exergy correctly is the most critical step in thermodynamics. This step is vital in correctly designing, analyzing, and evaluating thermodynamic systems and their components. In this section, we present two examples (one with a closed system and one with an open system) of how to write balance equations. In both cases all inputs must be written as equivalent to all outputs, which is the basis of this step, by keeping in mind that entropy generation hast to be included as an input term and that exergy destruction has to be included as an output term.

For a closed system: Assume that we have a closed system, as shown in Figure 1.5. This closed system consists of a piston-cylinder mechanism (which brings up a boundary movement work), heat input from a hot source (Q_{in}), electrical work from an electrical heater (W_e), a paddle work to run a fan (W_p), heat loss (Q_l), and a fixed amount of mass within the system (since there is no mass flow crossing the boundary).

Here, we need to write the balance equations between the initial and final states for mass, energy, entropy, and exergy for the closed system shown in Figure 1.5 as follows:

- Mass balance equation (MBE)

$$m_i = m_f = m = \text{constant} \tag{1.20}$$

- Energy balance equation (EBE)
  We can apply two approaches: (i) taking the boundary movement work with $W_b = P(V_f - V_i)$ as a work output (so-called: work done by the system):

$$m_i u_i + Q_{in} + W_e + W_p = m_f u_f + Q_l + W_b \tag{1.21}$$

or (ii) considering PV for both initial (as input) and final (as output) state-specific boundary work terms as follows:

$$m_i u_i + Q_{in} + W_e + W_p + PV_i = m_f u_f + Q_l + PV_f \tag{1.22}$$

- Entropy balance equation (EnBE)
  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. The 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 within a closed system. Accordingly, the EnBE is written as the input entropy to the system plus entropy generation, equal to the final entropy of the system

  $$m_i s_i + \frac{Q_{in}}{T_s} + S_{gen} = m_f s_f + \frac{Q_l}{T} \tag{1.23}$$

  where T_s is source temperature since heat is supplied from a source. In addition, T is expected to be an immediate boundary temperature T_b or surface temperature of the system T_{su}. Usually T is taken to be the reference environment temperature T_0, which may cause some issues when the exergy destructions are calculated as it eliminates the exergy losses. The suggestion is to use T_{su} or T_b. If neither is known, using the average temperature, $T_{av} = (T_i + T_f)/2$ is a less-complicated approach. It is important to note that entropy is not associated with work, and hence no work terms are included in the balance equation.
- Exergy balance equation (ExBE)
  The nature of exergy is opposite to that of entropy in that exergy can be *destroyed*, but it cannot be created. Therefore, it is known that 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. The work exergy becomes equivalent to the actual (practical or irreversible) work. That's why we include actual work directly in the exergy balance equation. For the system illustrated in Figure 1.5, the exergy balance equation is written as follows:

  $$m_i ex_i + Ex^{Q_{in}} + W_e + W_p = m_f ex_f + Ex^{Q_l} + W_b + Ex_{dest} \tag{1.24}$$

  where ex is the specific exergy (kJ/kg), with

  $$Ex^{Q_{in}} = \left(1 - \frac{T_0}{T_s}\right) Q_{in}, \quad Ex^{Q_l} = \left(1 - \frac{T_0}{T}\right) Q_l \text{ and } Ex_{dest} = T_0 S_{gen}.$$

  where T is expected to be the immediate boundary temperature T_b or the surface temperature of the system T_{su}. Usually it is taken to be the reference environment temperature T_0, which may cause some issues when the exergy destructions are calculated as it eliminates the exergy losses. The suggestion is to use T_{su} or T_b. If neither is known, using the average temperature, $T_{av} = (T_i + T_f)/2$ is a less-complicated approach.
  Furthermore, the specific non-flow exergies with respect to a reference (dead state) state are written as follows:

  $$ex_i = (u_i - u_0) - T_0(s_i - s_0) \text{ and } ex_f = (u_f - u_0) - T_0(s_f - s_0).$$

  Remember that in writing exergy balance equations, the exergy destruction term must be included as an output term, as shown above.

Example 1.1

A piston cylinder assembly, as shown in Figure 1.6, initially contains refrigerant R-134a at 100 kPa and 20 °C with a mass of 1.3 kg. The piston cylinder assembly is heated from the heat source at 200 °C until the temperature reaches 140 °C. The piston starts moving when the pressure inside the cylinder reaches 140 kPa. Determine (a) the work done, (b) the amount of heat transfer, (c) the exergy destruction, and (d) the energy and exergy efficiencies of this process. Take $T_0 = 25\,°\text{C}$.

Solution

a) For the piston cylinder assembly shown in Figure 1.6, one can write the thermodynamic balance equations as follows:

$$\text{MBE:}\quad m_1 = m_2$$

$$\text{EBE:}\quad m_1u_1 + Q_{\text{in}} = m_2u_2 + W_{\text{b}}$$

$$\text{EnBE:}\quad m_1s_1 + \frac{Q_{\text{in}}}{T_{\text{s}}} + S_{\text{gen}} = m_2s_2$$

$$\text{ExBE:}\quad m_1ex_1 + Q_{\text{in}}\left(1 - \frac{T_0}{T_{\text{s}}}\right) = m_2ex_2 + W_{\text{b}} + Ex_{\text{dest}}$$

For R-134a, from the EES package [3] one can obtain the properties at state 1 as

$$\left.\begin{aligned} T_1 &= 20\,°\text{C} \\ P_1 &= 100\,\text{kPa} \end{aligned}\right\} \begin{aligned} u_1 &= 248.8\,\text{kJ/kg} \\ s_1 &= 1.092\,\text{kJ/kg K} \\ v_1 &= 0.2337\,\text{m}^3/\text{kg} \end{aligned}$$

Similarly, for state 2:

$$\left.\begin{aligned} T_1 &= 140\,°\text{C} \\ P_1 &= 140\,\text{kPa} \end{aligned}\right\} \begin{aligned} u_2 &= 350.9\,\text{kJ/kg} \\ s_2 &= 1.383\,\text{kJ/kg K} \\ v_2 &= 0.2384\,\text{m}^3/\text{kg} \end{aligned}$$

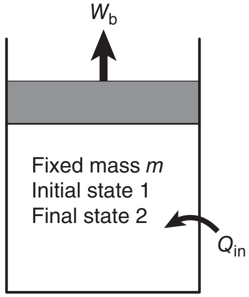
Similarly, for the reference state:

$$\left.\begin{aligned} T_0 &= 25\,°\text{C} \\ P_0 &= 101.321\,\text{kPa} \end{aligned}\right\} \begin{aligned} u_0 &= 252.6\,\text{kJ/kg} \\ s_0 &= 1.105\,\text{kJ/kg K} \end{aligned}$$

The specific exergy at state 1 can be calculated as

$$\begin{aligned} ex_1 = u_1 - u_0 - T_0(s_1 - s_0) &= 248.8 - 252.6 - (25 + 273)(0.1092 - 1.105) \\ &= 0.1458\,\text{kJ/kg} \end{aligned}$$

Figure 1.6 A closed system for Example 1.1.



Similarly, for state 2:

$$ex_2 = u_2 - u_0 - T_0(s_2 - s_0) = 350.9 - 252.6 - (25 + 273)(1.383 - 1.105)$$
$$= 15.34\,\text{kJ/kg}$$

From MBE, $m_1 = m_2 = 1.3\,\text{kg}$

The piston movement starts when the pressure inside the cylinder reaches 140 kPa, so the boundary work can be calculated as

$$W_b = m_1 P_2(v_2 - v_1) = (1.3\,\text{kg})(140\,\text{kPa})(0.2384 - 0.2337)\,\text{m}^3/\text{kg} = 0.8544\,\text{kJ}$$

b) The amount of heat transfer can be determined from the EBE as

$$Q_{in} = m_2(u_2 - u_1) + W_b = (1.3\,\text{kg})(350.9 - 248.8)\,\text{kJ/kg} + 0.8544\,\text{kJ} = 133.6\,\text{kJ}$$

c) The exergy destroyed can be determined from the ExBE as

$$Ex_{dest} = m_1(ex_1 - ex_2) - W_b + Q_{in}\left(1 - \frac{T_0}{T_s}\right)$$
$$= (1.3\,\text{kg})(0.1458 - 15.34)\,\text{kJ/kg} - 0.8544\,\text{kJ} + 133.6\left(1 - \frac{25 + 273}{200 + 273}\right)\text{kJ}$$
$$= 28.81\,\text{kJ}$$

d) The energy efficiency of the process can be determined as

$$\eta_{en} = \frac{m_2(u_2 - u_1) + W_b}{Q_{in}} = 1 = 100\%$$

The exergy efficiency of the process can be determined as

$$\eta_{ex} = \frac{m_2(ex_2 - ex_1) + W_b}{Q_{in}\left(1 - \frac{T_0}{T_s}\right)} = \frac{1.3(15.34 - 0.1458) + 0.8544}{133.6\left(1 - \frac{25 + 273}{200 + 273}\right)} = 0.4169 = 41.69\%$$

For an open system Assume that we have an open (control volume) system, as shown in Figure 1.7. This open system consists of flow energies at states 1, 2, and 3, work input

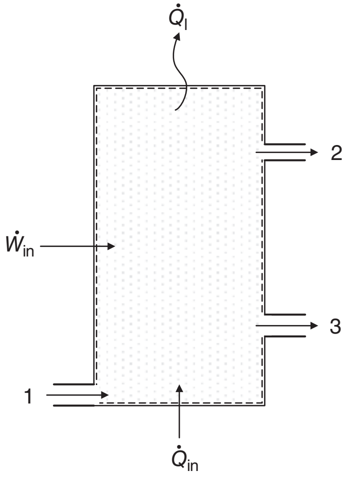


Figure 1.7 An open system.

($\dot{W}_{\text{in}}$), heat input ($\dot{Q}_{\text{in}}$) and heat loss ($\dot{Q}_{\text{l}}$). The flow energy consists of flow enthalpies, flow kinetic energies, and flow potential energies in the balance equations.

Here, we need to write the balance equations between the initial and final states for mass, energy, entropy, and exergy for the open system shown in Figure 1.7 as follows:

- MBE

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3 \tag{1.25}$$

- EBE

$$\dot{m}_1(h_1 + ke_1 + pe_1) + \dot{W}_{\text{in}} + \dot{Q}_{\text{in}} = \dot{m}_2(h_2 + ke_2 + pe_2) + \dot{m}_3(h_3 + ke_3 + pe_3) + \dot{Q}_{\text{l}} \tag{1.26}$$

where h, ke and pe are specific enthalpy, specific kinetic energy ($V^2/2$), and specific potential energy (gz), respectively, in kJ/kg.

If the changes in kinetic and potential energies are negligible or assumed to be negligible, Equation (1.26) results in

$$\dot{m}_1 h_1 + \dot{W}_{\text{in}} + \dot{Q}_{\text{in}} = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{Q}_{\text{l}} \tag{1.27}$$

- EnBE

$$\dot{m}_1 s_1 + \frac{\dot{Q}_{\text{in}}}{T_{\text{s}}} + \dot{S}_{\text{gen}} = \dot{m}_2 s_2 + \dot{m}_3 s_3 + \frac{\dot{Q}_{\text{l}}}{T} \tag{1.28}$$

where T is expected to be the immediate boundary temperature T_{b} or the surface temperature of the condenser T_{su}. Usually it is taken to be the reference environment temperature T_0, which may cause some issues when the exergy destructions are calculated as it eliminates the exergy losses. The suggestion is to use T_{su} or T_{b}. If neither is known, using the average temperature, $T_{\text{av}} = (T_1 + T_2 + T_3)/3$ is a less-complicated approach.

- ExBE

$$\dot{m}_1 ex_1 + \dot{W}_{\text{in}} + \dot{E}x^{Q_{\text{in}}} = \dot{m}_2 ex_2 + \dot{m}_3 ex_3 + \dot{E}x^{Q_{\text{l}}} + \dot{E}x_{\text{dest}} \tag{1.29}$$

where the specific exergies are defined as follows:

$$ex_1 = (h_1 - h_0) - T_0(s_1 - s_0),\ ex_2 = (h_2 - h_0) - T_0(s_2 - s_0) \text{ and}$$
$$ex_3 = (h_3 - h_0) - T_0(s_3 - s_0).$$

The total exergy destruction is written as $\dot{E}x_{\text{dest}} = T_0 \dot{S}_{\text{gen}}$. T_0 is the reference environment (dead state) temperature. Usually the surrounding temperature is taken to be the reference environment temperature.

Here, it is necessary to make three important points:

i) Kinetic exergy equals kinetic energy, and potential exergy equals potential energy. If the changes in kinetic exergy (kex) and potential exergy (pex) are not negligible, the flow exergy terms in Equation (1.29) should include both kinetic and potential exergy terms.
ii) In writing the exergy balance equation, the exergy destruction term must be included as an output term as done above.
iii) Work is associated with exergy as work exergy.

Example 1.2

R-134a at −20 °C enters the compressor, as shown in Figure 1.8, as a saturated vapor with a flow rate of 0.45 kg/s and leaves at 800 kPa and 45 °C. Assume the heat loss to the surrounding air from the compressor is 10% of the work input to the compressor. Calculate (a) the compressor work, (b) the amount of heat loss, (c) the exergy destruction, and (d) the energy and exergy efficiencies of the compressor. Take $T_0 = 25\,°\text{C}$ which can be taken as the boundary or surface temperature.

Solution

a) For the compressor shown in Figure 1.8 one can write the thermodynamic balance equations as follows:

$$\text{MBE}: \dot{m}_1 = \dot{m}_2$$

$$\text{EBE}: \dot{m}_1 h_1 + \dot{W}_c = \dot{m}_2 h_2 + \dot{Q}_l$$

$$\text{EnBE}: \dot{m}_1 s_1 + \dot{S}_{\text{gen}} = \dot{m}_2 s_2 + \frac{\dot{Q}_l}{T_l}$$

$$\text{ExBE}: \dot{m}_1 ex_1 + \dot{W}_{\text{c}} = \dot{m}_2 ex_2 + \dot{Q}_l\left(1 - \frac{T_0}{T_{\text{su}}}\right)$$

For R-134a, from the EES package [3], one can obtain the properties at state 1 as

$$\left.\begin{array}{l} T_1 = 20\,°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 238.4\,\text{kJ/kg} \\ s_1 = 0.9456\,\text{kJ/kg K} \end{array}$$

Similarly, for state 2:

$$\left.\begin{array}{l} T_2 = 45\,°\text{C} \\ P_2 = 800\,\text{kPa} \end{array}\right\} \begin{array}{l} h_2 = 281.6\,\text{kJ/kg} \\ s_2 = 0.9643\,\text{kJ/kg K} \end{array}$$

Similarly, for the reference state:

$$\left.\begin{array}{l} T_0 = 25\,°\text{C} \\ P_0 = 101.321\,\text{kPa} \end{array}\right\} \begin{array}{l} h_0 = 276.4\,\text{kJ/kg} \\ s_0 = 1.105\,\text{kJ/kg K} \end{array}$$

The specific exergy at state 1 can be calculated as

$$\begin{aligned} ex_1 &= h_1 - h_0 - T_0(s_1 - s_0) = 238.4 - 276.4 - (25 + 273)(0.9456 - 1.105) \\ &= 9.51\,\text{kJ/kg} \end{aligned}$$

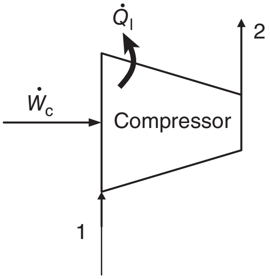


Figure 1.8 A schematic illustration of compressor.

Similarly, for state 2:

$$\begin{aligned} ex_2 &= h_2 - h_0 - T_0(s_2 - s_0) = 281.6 - 276.4 - (25 + 273)(0.9643 - 1.105) \\ &= 47.14\,\text{kJ/kg} \end{aligned}$$

It is given that $\dot{Q}_l = 0.10\dot{W}_c$

Now from the EBE, the compressor work can be determined as

$$\dot{W}_c = \frac{\dot{m}_1(h_2 - h_1)}{0.9} = \frac{(0.45\,\text{kg/s})(281.6 - 238.4)\,\text{kJ/kg}}{0.9} = 21.59\,\text{kW}$$

b) The amount of heat lost can be determined as

$$\dot{Q}_l = 0.10\dot{W}_c = 0.1 \times 21.59 = 2.159\,\text{kJ}$$

c) The exergy destruction can be determined from the ExBE as

$$\begin{aligned} \dot{E}x_{\text{dest}} &= \dot{m}_1(ex_1 - ex_2) + \dot{W}_c - \dot{Q}_l\left(1 - \frac{T_0}{T_{\text{su}}}\right) \\ &= 0.45(9.51 - 47.14) + 21.59 - 2.159\left(1 - \frac{25 + 273}{25 + 273}\right) \\ &= 4.663\,\text{kW} \end{aligned}$$

d) The energy efficiency of the compressor can be determined as

$$\eta_{\text{en,c}} = \frac{\dot{m}_1(h_2 - h_1)}{\dot{W}_c} = \frac{0.45(281.6 - 238.4)}{21.59} = 0.90 = 90\%$$

In addition, the exergy efficiency of the compressor can be determined as

$$\eta_{\text{ex,c}} = \frac{\dot{m}_1(ex_2 - ex_1)}{\dot{W}_c} = \frac{0.45(47.14 - 9.51)}{21.59} = 0.7841 = 78.41\%$$

Step 6: Performance Assessment

A typical approach to assess the performance is to use the efficiency, which ranges between 0 and 1 or 0% and 100%. A general definition of the efficiency is given as follows:

$$\text{Efficiency} = \frac{\text{useful output}}{\text{total input}} \tag{1.30}$$

If the efficiency is defined under the FLT, it is named as first law or thermal or energy efficiency. If it is defined under the SLT, it is called the second law or exergy efficiency. Defining efficiency in a conceptually correct manner is of the greatest importance in thermodynamics. Throughout this book we use energy efficiency under the FLT and exergy efficiency under the SLT, which can be defined as follows:

$$\text{Energy efficiency} = \frac{\text{useful energy output}}{\text{total energy input}} \tag{1.31}$$

$$\text{Exergy efficiency} = \frac{\text{exergy output}}{\text{total exergy input}} \tag{1.32}$$

There are situations/applications where the efficiency values may be greater than 1 or 100%, such as refrigeration and heat pump applications. In such cases we use the COPs to assess the performance of refrigeration and heat pump systems. COP values can be obtained through energy and exergy analyses, and are then called energetic COP and exergetic COP, which can be defined as follows:

$$\text{Energetic COP} = \text{COP}_{\text{en}} = \frac{\text{useful energy output}}{\text{total energy input}} \tag{1.33}$$

$$\text{Exergetic COP} = \text{COP}_{\text{ex}} = \frac{\text{exergy output}}{\text{total exergy input}} \tag{1.34}$$

Although there is comprehensive coverage and discussion about efficiencies and COPs in the following sections and chapters, we list both the energy and exergy efficiencies of a thermal power plant (so-called heat engine since it utilizes heat to produce work), and the energetic and exergetic COPs of refrigeration and heat pump systems as follows:

- For a heat engine:

$$\text{Energy efficiency: } \eta_{\text{en}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} \tag{1.35}$$

$$\text{Exergy efficiency: } \eta_{\text{ex}} = \frac{\dot{W}_{\text{net}}}{\dot{E}x^{Q_{\text{in}}}} \tag{1.36}$$

- For a refrigeration system:

$$\text{Energetic COP: } \text{COP}_{\text{en}} = \frac{\dot{Q}_{\text{L}}}{\dot{W}_{\text{in}}} \tag{1.37}$$

$$\text{Exergetic COP: } \text{COP}_{\text{ex}} = \frac{\dot{E}x^{Q_{\text{L}}}}{\dot{W}_{\text{in}}} \tag{1.38}$$

- For a heat pump system:

$$\text{Energetic COP: } \text{COP}_{\text{en}} = \frac{\dot{Q}_{\text{H}}}{\dot{W}_{\text{in}}} \tag{1.39}$$

$$\text{Exergetic COP: } \text{COP}_{\text{ex}} = \frac{\dot{E}x^{Q_{\text{H}}}}{\dot{W}_{\text{in}}} \tag{1.40}$$

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

$$\eta_{\text{th,rev}} = 1 - \frac{T_{\text{L}}}{T_{\text{H}}} \tag{1.41}$$

where T_{H} is the source temperature and T_{L} is the sink temperature where heat is rejected (i.e., lake, ambient air etc.). This is the maximum efficiency for a heat engine operating between two reservoirs at T_{H} and T_{L}.

A refrigerator or heat pump operating on a reversed Carnot cycle would supply maximum cooling (in the case of a refrigerator) and maximum heating (in the case of a heat pump), and the COPs of such reversible cycles for refrigeration and heat pump are defined as follows:

$$\mathrm{COP}_{\mathrm{R,rev}} = \frac{1}{T_\mathrm{H}/T_\mathrm{L} - 1} \tag{1.42}$$

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{1}{1 - T_\mathrm{L}/T_\mathrm{H}} \tag{1.43}$$

1.3.6 Pure Substances

A pure substance is one that has a homogeneous and invariable chemical composition. While keeping 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 phase has the same chemical composition, but a mixture of liquid air and gaseous air cannot be considered a pure substance because the composition of each phase differs. A thorough understanding of the pure substance is of significance, particularly for air-conditioning applications. The thermodynamic properties of water and steam can be taken from tables and charts, given in almost all thermodynamic books, based on 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.

1.3.6.1 State and Change of State

The state of a system or substance is defined as the condition of the system or substance characterized by the 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 the 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 1.9 shows typical examples of ice melting and water boiling.

A clearer presentation of the solid, liquid, and vapor phases of water is exhibited on the temperature–volume (T–v) diagram in Figure 1.10. 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 quality $x = 0$ and 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 1 or from 0 to 100% in percent. 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**: On 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 (one is present only), but 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 the 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*.

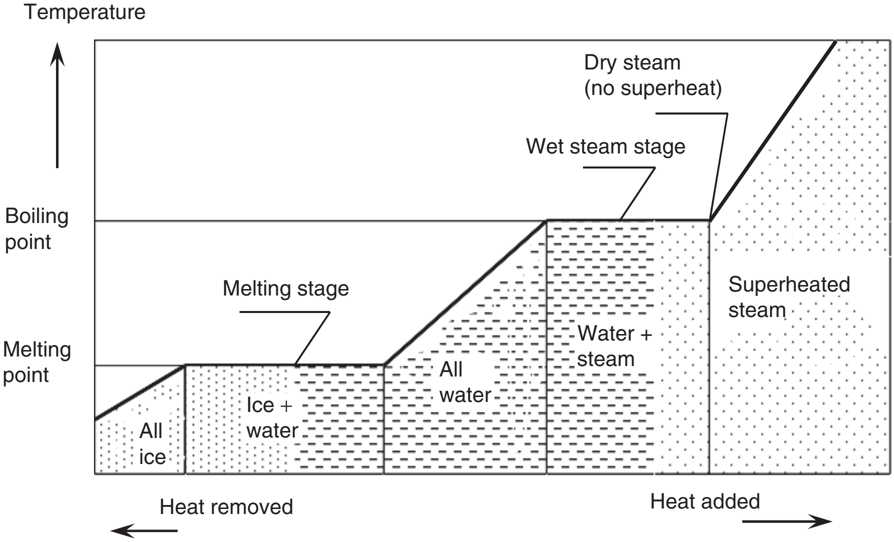


Figure 1.9 The state-change diagram of water.

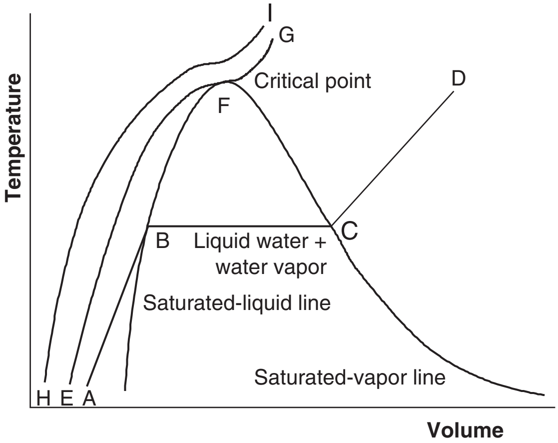


Figure 1.10 Temperature–volume diagram for the phase change of water.

1.3.6.2 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's consider a mass (m) with a quality (x). The volume is the sum of the volumes of the liquid and vapor, as defined below:

$$V = V_{\text{liq}} + V_{\text{vap}} \tag{1.44}$$

This can also be written in terms of specific volumes as

$$mv = m_{\text{liq}}v_{\text{liq}} + m_{\text{vap}}v_{\text{vap}} \tag{1.45}$$

Dividing all terms by the total mass results in

$$v = (1 - x)v_{\text{liq}} + xv_{\text{vap}} \tag{1.46}$$

and

$$v = v_{\text{liq}} + xv_{\text{liq,vap}} \tag{1.47}$$

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

1.3.6.3 Sensible Heat, Latent Heat and Latent Heat of Fusion

It is known that all substances can hold a certain amount of heat; that 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 the *latent heat*. In fact, this 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 in its dry saturated state and in 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 one kilogram of ice at 0 °C to one kilogram 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 one kilogram of water at 0 °C changes it back to ice.

1.3.6.4 Specific Heat

The energy required to change (raise or 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).

1.3.6.5 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 similarly be written in a similar form to Equations (1.46) and (1.47):

$$u = (1 - x)u_{\mathrm{liq}} + xu_{\mathrm{vap}} \tag{1.48}$$

and

$$u = u_{\mathrm{liq}} + xu_{\mathrm{liq,vap}} \tag{1.49}$$

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

1.3.6.6 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 a 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 similarly be written as

$$h = (1 - x)h_{\mathrm{liq}} + xh_{\mathrm{vap}} \tag{1.50}$$

and

$$h = h_{\mathrm{liq}} + xh_{\mathrm{liq,vap}} \tag{1.51}$$

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

1.3.6.7 Specific Entropy

Entropy is a property resulting from the 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 while 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

$$\Delta s = s_2 - s_1 \tag{1.52}$$

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

$$s = (1 - x)s_{\mathrm{liq}} + xs_{\mathrm{vap}} \tag{1.53}$$

and

$$s = s_{\mathrm{liq}} + xs_{\mathrm{liq,vap}} \tag{1.54}$$

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

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

$$s_2 - s_1 = c \ln \frac{T_2}{T_1} \tag{1.55}$$

where c is the average specific heat of the substance.

An isentropic (i.e., constant entropy) process is defined as follows:

$$s_2 = s_1 \tag{1.56}$$

1.3.6.8 Energy Change and Energy Transfer

Energy is the capacity for doing work. Although the total energy of s system may consist of many kind of energies, we generally consider internal, kinetic and potential energies in our engineering thermodynamics. 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 thus expressed as

$$\Delta E = E_2 - E_1 = \Delta U + \Delta \mathrm{KE} + \Delta \mathrm{PE} \tag{1.57}$$

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

$$\Delta E = \Delta U = m(u_2 - u_1) \tag{1.58}$$

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

$$\dot{E} = \frac{E}{\Delta t} \text{ (kW or Btu/lbm)} \tag{1.59}$$

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 units of kJ/kg or Btu/lbm (1 kJ/kg = 0.430 Btu/lbm):

$$e = \frac{E}{m} \text{ (kJ/kg or Btu/lbm)} \tag{1.60}$$

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

1.3.6.9 Flow Energy

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 flow energy entering is equal to mass times enthalpy $\dot{m}h$ (kW). Note that the total flow energy generally includes flow enthalpy, flow kinetic energy and flow potential energy. There are some common cases where the changes in kinetic and potential energies are negligible. If this happens, we cover flow enthalpy only as stated earlier.

1.3.6.10 Heat Transfer

The definitive experiments, which was carried out by the Scottish physicist James Joule, showed that heat is a form of energy convertible into other forms. Heat is commonly 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 obviously provide higher heat transfer rates.

Heat or heat transfer has the same unit as energy. The symbol for heat and heat transfer is Q (kJ). Heat or heat transfer per unit mass is denoted by q (kJ/kg). Heat or 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*.

1.3.6.11 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 or flow energy 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".

1.3.6.12 Thermodynamic Tables

Thermodynamic tables were first published in 1936 as steam tables by Keenan and Keyes, and later in 1969 and 1978 were revised and republished. The use of thermodynamic tables of many substances ranging from water to several refrigerants is very common in process design calculations [4]. Appendix B contains the thermophysical properties of water, air and various other substances and refrigerants. In literature they are also called either steam tables or vapor tables. In this book we will refer to them as thermodynamic tables. These tables are normally given in different distinct phases (parts), for example four different parts for water (saturated water, superheated vapor water, compressed liquid water, saturated solid-saturated vapor water) and two distinct parts for R-134a (saturated and superheated). Each table is arranged according to the values of temperature and pressure, and also contains the values of four other thermodynamic parameters: specific volume, internal energy, enthalpy, and entropy. When we 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 difficult. However, further practice will provide sufficient confidence to do carry out the calculations. Beside these thermodynamic tables, recently much attention has been paid to computerized tables for design calculations. Of course, although this eliminates several reading problems, students may not understand the concepts and comprehend the subject very well. This is why in thermodynamics courses it is a must for students to know how to obtain thermodynamic data from the appropriate thermodynamic tables. These days almost every thermodynamics book is accompanied by a software package (e.g., [3]) that includes thermodynamic tables in a form (such as a database) that enables the necessary data for calculations to be obtained. Of course, there are also still handbooks of thermodynamic tables available for practicing engineers, students, etc. As well as these, there are also online resources offered by various groups and organizations, some free of charge.

1.4 Ideal and Real 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:

$$Pv = RT \tag{1.61}$$

and

$$PV = mRT \tag{1.62}$$

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

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

$$R = \frac{\overline{R}}{M} \tag{1.63}$$

where $\overline{R}$= 8.314 kJ/kmolK.

Equations (1.61) and (1.62) may be written in a mole-basis form as follows:

$$P\overline{v} = RT \tag{1.64}$$

and

$$PV = n\overline{R}T \tag{1.65}$$

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 thermodynamic tables and graphs from the following expressions:

$$\Delta u = (u_2 - u_1) = c_\mathrm{v}(T_2 - T_1) \tag{1.66}$$

$$\Delta h = (h_2 - h_1) = c_\mathrm{p}(T_2 - T_1) \tag{1.67}$$

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

$$c_\mathrm{p} - c_\mathrm{v} = R \tag{1.68}$$

For the entire range of states, the ideal gas model may be 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:

$$Pv = ZRT \quad or \quad Z = \frac{Pv}{RT} \tag{1.69}$$

Figure 1.11 shows a generalized compressibility chart for simple substances. In the chart there are two important parameters: the reduced temperature ($T_\mathrm{r} = T/T_\mathrm{c}$) and the reduced pressure ($P_\mathrm{r} = P/P_\mathrm{c}$). To calculate the compressibility factor, the values of T_r and P_r should be calculated by using the critical temperature and pressure values of the respective substance, which may easily be taken from the thermodynamic tables

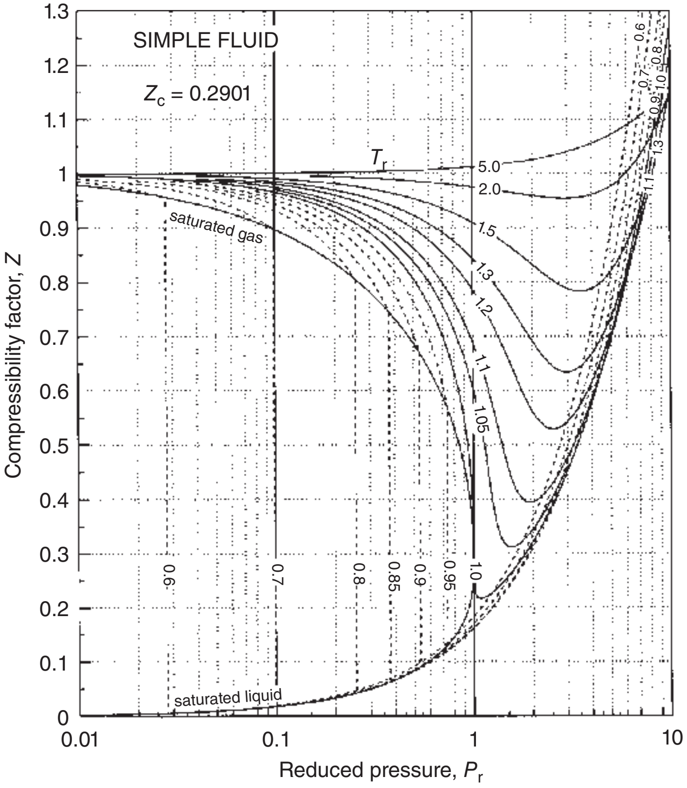


Figure 1.11 Generalized compressibility chart for simple substances [5].

(e.g., Appendix B). As can be seen in Figure 1.11, 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 (1.69) becomes Equation (1.61). 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, the van der Waals equation, and the Redlich and Kwong equation. However, some of these equations of state are complicated due to 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

$$P_1V_1 = P_2V_2 \tag{1.70}$$

where the subscripts refer to the initial and final states.

Equation (1.70) 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 due to 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

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \tag{1.71}$$

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

$$\frac{P_1}{T_1} = \frac{P_2}{T_2} \tag{1.72}$$

Equations (1.71) and (1.72) 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:

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{1.73}$$

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

$$\mathrm{d}U = mc_{\mathrm{v0}}\mathrm{d}T \tag{1.74}$$

and the specific internal energy becomes

$$\mathrm{d}u = c_{\mathrm{v0}}\mathrm{d}T \tag{1.75}$$

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

$$\mathrm{d}H = mc_{\mathrm{p0}}\mathrm{d}T \tag{1.76}$$

and the specific enthalpy then becomes

$$\mathrm{d}h = c_{\mathrm{p0}}\mathrm{d}T \tag{1.77}$$

The entropy change of an ideal gas, based on the general entropy equation in terms of $T\mathrm{d}s = \mathrm{d}u + P\mathrm{d}v$ and $T\mathrm{d}s = \mathrm{d}h - v\mathrm{d}P$ as well as the ideal gas equation $Pv = RT$, can be obtained in two ways by substituting Equations (1.75) and (1.77):

$$s_2 - s_1 = c_{\mathrm{v0}} \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1} \tag{1.78}$$

$$s_2 - s_1 = c_{\mathrm{p0}} \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \tag{1.79}$$

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

$$Pv^k = P_1 v_1^k = P_2 v_2^k \tag{1.80}$$

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

$$k = \frac{c_{\mathrm{p0}}}{c_{\mathrm{v0}}} \tag{1.81}$$

Based on Equation (1.80) and the ideal gas equation, the following expressions can be obtained:

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{v_1}{v_2}\right)^{k-1} \tag{1.82}$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \tag{1.83}$$

$$\left(\frac{P_2}{P_1}\right) = \left(\frac{v_1}{v_2}\right)^{k} \tag{1.84}$$

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

Let's consider a closed system with ideal gas, undergoing an adiabatic reversible process with a constant specific heat. The work can be derived from the FLT equation as follows:

$$W_{1-2} = \frac{mR(T_2 - T_1)}{(1 - k)} = \frac{(P_2 V_2 - P_1 V_1)}{(1 - k)} \tag{1.85}$$

which 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 (1.80), (1.82)–(1.84), and (1.85) can be rewritten with the polytropic exponent under $Pv^n = \text{constant}$ as

$$Pv^n = P_1 v_1^n = P_2 v_2^n \tag{1.86}$$

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{n/n-1} = \left(\frac{v_1}{v_2}\right)^{n} = \left(\frac{V_1}{V_2}\right)^{n} \tag{1.87}$$

$$W_{1-2} = \frac{mR(T_2 - T_1)}{(1 - n)} = \frac{(P_2 V_2 - P_1 V_1)}{(1 - n)} \tag{1.88}$$

In order to give a clear idea about the type of process and its exponent, it is important to show the values of n for four different types of polytropic processes for ideal gases, as illustrated in Figure 1.12, as follows:

- $n = 0$ for an isobaric (constant pressure) process
- $n = 1$ for an isothermal (constant temperature) process
- $n = k$ for an isentropic (constant entropy) process
- $n = \infty$ for an isochoric (constant volume) process

As is obvious in Figure 1.12, there are two quadrants where n varies from zero to infinity 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. These are comprehensively treated in a widely accepted book by Cengel and Boles [6].

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

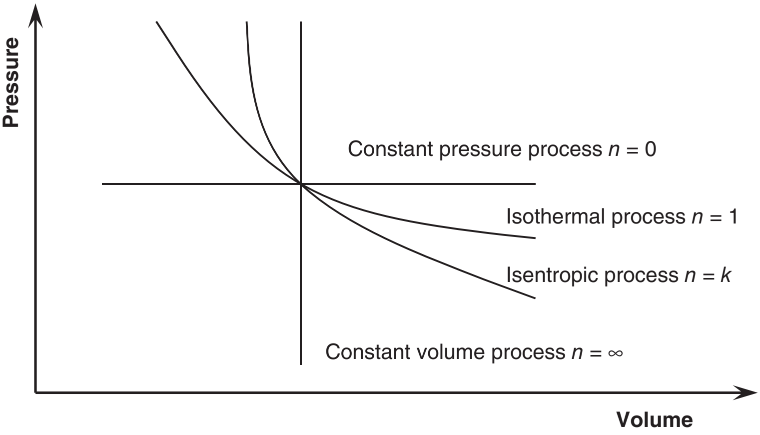


Figure 1.12 Representation of four different polytropic processes on a pressure–volume diagram.

Table 1.2 Equations for gas and gas mixtures and relevant models

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

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 chemical composition. Note that entropy increase becomes greater when more different gases are mixed in a system.

Example 1.3
Determine the specific volume of refrigerant R-134a at 1000 kPa and 40 °C considering R-134a as an ideal gas.

Solution
For R-134 a, from the EES package [3] one can obtain

$$R = 0.08149\,\text{kPam}^3/\text{kg}$$

The specific volume of refrigerant R-134a can be determined from ideal gas equation as

$$v = \frac{RT}{P} = \frac{(0.08149\,\text{kPam}^3/\text{kg K})(313\,\text{K})}{1000\,\text{kPa}} = 0.02551\,\text{m}^3/\text{kg}$$

Example 1.4
Determine the specific volume and compressibility factor for refrigerant R-134a at 1000 kPa and 40 °C by considering R-134a as a real gas.

Solution
For R-34a, from the EES package [3] one can obtain

$$T_c = 374.2\,\text{K},\ P_c = 4059\,\text{kPa}$$

Now, we first need to calculate reduced temperature and reduced pressure in order to determine the compressibility factor as follows:

$$T_r = \frac{T_1}{T_c} = \frac{40 + 273}{374.2} = 0.8365$$

$$P_r = \frac{P_1}{P_c} = \frac{1000}{4059} = 0.246$$

From the compressibility chart (such as Figure 1.11) with the values of $T_r = 0.84$ and $P_r = 0.25$, one can get the value of Z as $Z = 0.80$.

Now, the specific volume can be calculated as $v = v_{\text{ideal}}Z = 0.02551 \times 0.8 = 0.2040\,\text{m}^3/\text{kg}$.

Alternatively, the values of Z and the specific volume can also be calculated from the EES package [3], corresponding to state point as

$$\left.\begin{array}{l} T_1 = 40\,°\text{C} \\ P_1 = 1000\,\text{kPa} \end{array}\right\} \begin{array}{l} v = 0.02038\,\text{m}^3/\text{kg} \\ Z = 0.7992 \end{array}$$

1.5 Refrigerators and Heat Pumps

Refrigerators and heat pumps are devices used to transfer heat from a low-temperature medium to a high-temperature medium in order ultimately achieve cooling or heating effects. These are treated as cyclic devices. Figure 1.13a shows the schematic of a

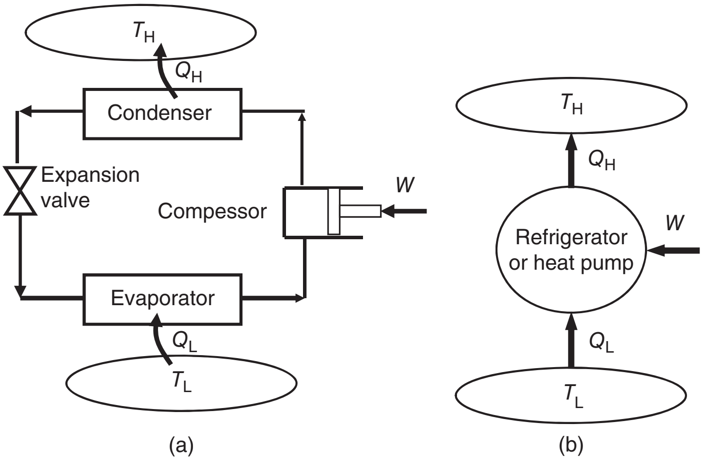


Figure 1.13 (a) The vapor-compression refrigeration cycle. (b) Simplified schematic of refrigeration cycle.

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 the expansion valve and its pressure and temperature drops. 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 for refrigerator or heat pump 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 1.13b.

An energy balance (the FLT) on the refrigeration cycle gives

$$Q_H = Q_L + W \tag{1.89}$$

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

$$\mathrm{COP_R} = \frac{Q_L}{W} \tag{1.90}$$

This can also be expressed as

$$\mathrm{COP_R} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{Q_H/Q_L - 1} \tag{1.91}$$

A heat pump is basically the same device as a refrigerator. The only difference between refrigerator and heat pumps is that we use refrigerator for cooling and heat pump for heating. Therefore, the useful output of refrigerator is evaporator cooling load while the useful output for heat pump is the condenser heating load. 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

$$\mathrm{COP_{HP}} = \frac{Q_H}{W} \tag{1.92}$$

This can also be expressed as

$$\mathrm{COP_{HP}} = \frac{Q_\mathrm{H}}{Q_\mathrm{H} - Q_\mathrm{L}} = \frac{1}{1 - Q_\mathrm{L}/Q_\mathrm{H}} \tag{1.93}$$

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

$$\mathrm{COP_{HP}} = \mathrm{COP_R} + 1 \tag{1.94}$$

This shows that the COP of a heat pump is always greater than the COP of the corresponding refrigerator. Note that the COP of a refrigerator may be less than or greater than 1.

1.5.1 The Carnot Refrigerators and Heat Pumps

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 1.14a). The maximum theoretical performance can be calculated, establishing criteria against which real refrigeration cycles can be compared. The following processes take place in the Carnot refrigeration cycle, as shown on the temperature–entropy diagram in Figure 1.14b:

- 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, corresponding to:

$$Q_\mathrm{L} = T_\mathrm{L}(s_1 - s_4) \tag{1.95}$$

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

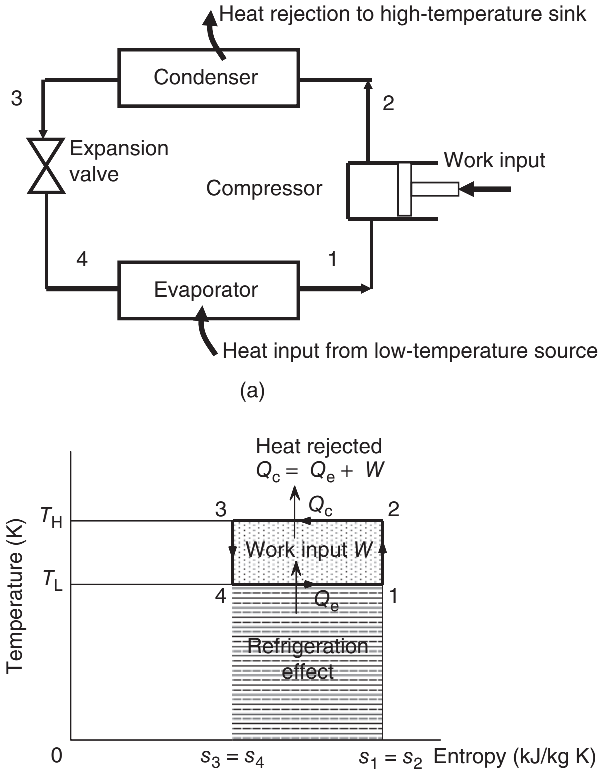
$$W = (T_\mathrm{H} - T_\mathrm{L})(s_1 - s_4) \tag{1.96}$$

After inserting Equations (1.95) and (1.96) into Equation (1.90), we find the following equation, which is dependent on the process temperatures:

$$\mathrm{COP_{R,rev}} = \frac{Q_\mathrm{L}}{W} = \frac{Q_\mathrm{L}}{Q_\mathrm{H} - Q_\mathrm{L}} = \frac{T_\mathrm{L}}{T_\mathrm{H} - T_\mathrm{L}} \tag{1.97}$$

This can also be expressed as

$$\mathrm{COP_{R,rev}} = \frac{1}{Q_\mathrm{H}/Q_\mathrm{L} - 1} = \frac{1}{T_\mathrm{H}/T_\mathrm{L} - 1} \tag{1.98}$$



(a)

(b)

Figure 1.14 (a) A schematic illustration of refrigerator and (b) *T-s* diagram of the Carnot refrigeration cycle.

For a reversible heat pump, the following relations apply:

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{Q_H}{W} = \frac{Q_H}{Q_H - Q_L} = \frac{T_H}{T_H - T_L} \tag{1.99}$$

or

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{1}{1 - Q_L/Q_H} = \frac{1}{1 - T_L/T_H} \tag{1.100}$$

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 1.15 and 1.16 show that the COP of a reversible refrigerator increases with increasing T_L and decreasing T_H.

Example 1.5

A refrigeration cycle, e.g. Figure 14.a, is used to keep a food department at −18 °C in an environment at 20 °C. The total heat gain to the food department is estimated to be

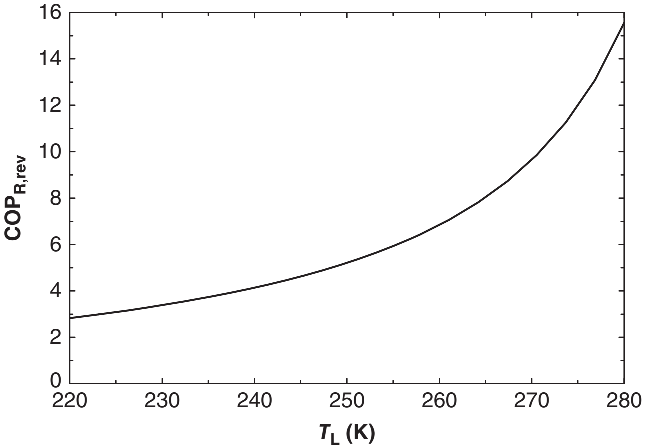


Figure 1.15 The COP of a reversible refrigerator as a function of T_L. T_H is taken as 298 K.

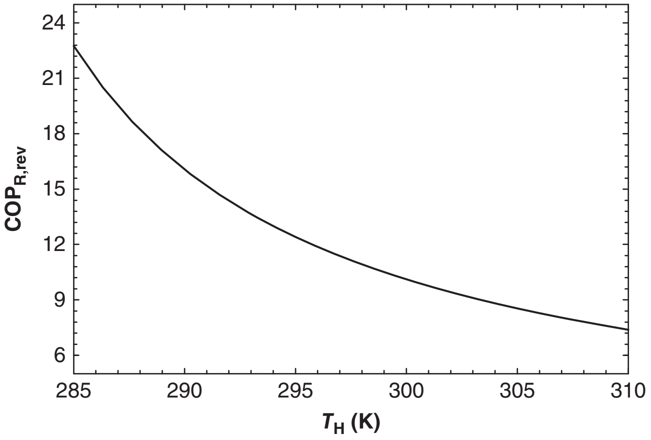


Figure 1.16 The COP of a reversible refrigerator as a function of T_H. T_L is taken as 273 K.

1800 kJ/h and the heat rejection in the condenser is 3000 kJ/h. Calculate (a) the power input to the compressor in kW, (b) the COP of the refrigerator, (c) the Carnot COP of the refrigerator, (d) the minimum power input to the compressor if a reversible refrigerator was used, and (e) the exergetic COP of the refrigerator.

Solution

a) The power input to the compressor is determined from energy balance on the refrigeration cycle:

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{H}} - \dot{Q}_{\text{L}} = 3000 - 1800 = 1200\,\text{kJ/h} = (1200\,\text{kJ/h})\left(\frac{1\,\text{kW}}{3600\,\text{kJ/h}}\right)$$
$$= \mathbf{0.333\,kW}$$

b) The COP of the refrigerator can be determined as

$$\mathrm{COP}_\mathrm{R} = \frac{\dot{Q}_\mathrm{L}}{\dot{W}_\mathrm{in}} = \frac{(1200/3600)\,\mathrm{kW}}{0.333\,\mathrm{kW}} = \mathbf{1.5}$$

c) The Carnot or maximum COP of the refrigerator can be determined as

$$\mathrm{COP}_\mathrm{R,Carnot} = \frac{T_\mathrm{L}}{T_\mathrm{H} - T_\mathrm{L}} = \frac{-18+273}{(20+273)-(-18+273)} = \frac{255}{(293)-(255)} = \mathbf{6.711}$$

d) From the definition of the maximum COP of the refrigerator, the minimum power input to the compressor is determined as

$$\dot{W}_\mathrm{min} = \frac{\dot{Q}_\mathrm{L}}{\mathrm{COP}_\mathrm{R,Carnot}} = \frac{(1200/3600)\,\mathrm{kW}}{6.711} = \mathbf{0.0745\,kW}$$

e) The exergetic COP of the refrigerator can be determined as

$$\mathrm{COP}_\mathrm{R,ex} = \frac{\mathrm{COP}_\mathrm{R}}{\mathrm{COP}_\mathrm{R,Carnot}} = \frac{1.5}{6.711} = \mathbf{0.2235}$$

Example 1.6

A heat pump, e.g. Figure 14.a (with a difference that condenser heat is used for heating purposes), is used to keep a house at 20 °C by extracting heat from the outside environment at 5 °C. The total heat gain by the house is estimated to be 200 kW by using 45 kW of power. Calculate (a) the COP of the heat pump, (b) the Carnot COP of the heat pump, and (c) the exergetic COP of the heat pump.

Solution

a) The COP of the heat pump can be determined as

$$\mathrm{COP}_\mathrm{hp} = \frac{\dot{Q}_\mathrm{H}}{\dot{W}_\mathrm{in}} = \frac{200\,\mathrm{kW}}{45\,\mathrm{kW}} = \mathbf{4.444}$$

b) The maximum COP or Carnot COP of the heat pump can be determined as

$$\mathrm{COP}_\mathrm{hp,Carnot} = \frac{T_\mathrm{H}}{T_\mathrm{H} - T_\mathrm{L}} = \frac{20+273}{(20+273)-(5+273)} = \frac{293}{(293)-(278)} = \mathbf{19.53}$$

c) The exergetic COP of the heat pump can be determined as

$$\mathrm{COP}_\mathrm{hp,ex} = \frac{\mathrm{COP}_\mathrm{hp}}{\mathrm{COP}_\mathrm{hp,Carnot}} = \frac{4.444}{19.53} = \mathbf{0.2275}$$

Example 1.7

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*) receives heat from a source at 1000 K. After this, both engines reject heat to a medium at 300 K. At 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

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

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

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

It is obvious from this example that 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:

$$\eta_{\text{ex}} = \frac{\eta_{\text{th}}}{\eta_{\text{th,rev}}} \tag{1.101}$$

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

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

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

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:

$$\eta_{\text{ex}} = \frac{W_{\text{out}}}{W_{\text{rev,out}}} \tag{1.102}$$

This definition is more general since it can be applied to processes (in turbines, piston–cylinder devices, etc.) as well as to 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:

$$\eta_{\text{ex}} = \frac{W_{\text{rev,in}}}{W_{\text{in}}} \tag{1.103}$$

For cyclic devices such as refrigerators and heat pumps, this can also be expressed in terms of the COPs as

$$\eta_{\text{ex}} = \frac{\text{COP}}{\text{COP}_{\text{rev}}} \tag{1.104}$$

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

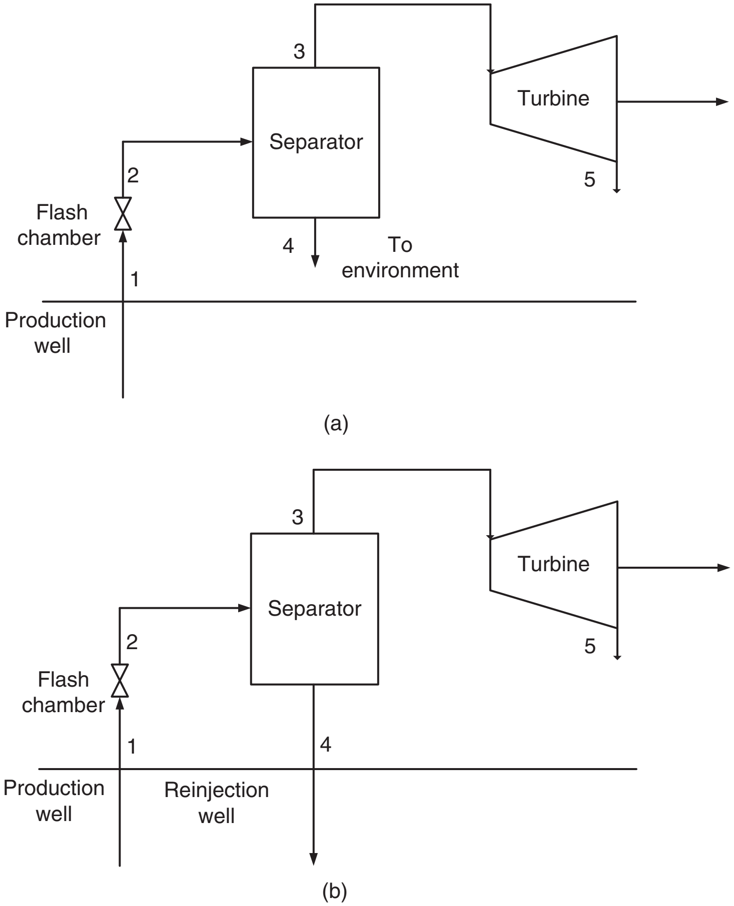


Figure 1.17 A single-flash geothermal power plant for two cases: (a) without reinjection and (b) with reinjection.

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

$$\eta_{\mathrm{ex}} = \frac{\text{exergy output}}{\text{exergy input}} = 1 - \frac{\text{exergy destroyed}}{\text{exergy supplied}} \tag{1.105}$$

Example 1.8

In this example we consider a single-flash geothermal power plant for two cases: (i) without reinjection, as shown in Figure 1.17a, and (ii) with reinjection at a temperature of 90 °C back into the reinjection well, as shown in Figure 1.17b, using geothermal liquid water at 170 °C at a rate of 120 kg/s as the heat source and producing 4500 kW of net power at an environment temperature of 25 °C. (a) Write all mass, energy, entropy, and exergy balance equations for the flash chamber and turbine, (b) calculate the plant exergy destructions for both cases, and (c) calculate both the energy and the exergy efficiencies of the plant. Take $T_0 = 25\,°\mathrm{C}$ for both cases.

Solution

We conduct a thermodynamic analysis of this power plant by considering both energy and exergy approaches.

(i) Without reinjection

For the geothermal power plant shown in Figure 1.17a, one can write the thermodynamic balance equations for each component as follows:

For the flash chamber:
MBE: $\dot{m}_1 = \dot{m}_2$
EBE: $\dot{m}_1 h_1 = \dot{m}_2 h_2$
EnBE: $\dot{m}_1 s_1 + \dot{S}_{\text{gen}} = \dot{m}_2 s_2$
ExBE: $\dot{m}_1 ex_1 = \dot{m}_2 ex_2 + \dot{E}x_{\text{dest}}$
For the separator:
MBE: $\dot{m}_2 = \dot{m}_3 + \dot{m}_4$
EBE: $\dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_4 h_4$
EnBE: $\dot{m}_2 s_2 + \dot{S}_{\text{gen}} = \dot{m}_4 s_4 + \dot{m}_3 h_3$
ExBE: $\dot{m}_2 ex_2 = \dot{m}_3 ex_3 + \dot{m}_4 ex_4 + \dot{E}x_{\text{dest}}$
For the turbine:
MBE: $\dot{m}_3 = \dot{m}_5$
EBE: $\dot{m}_3 h_3 = \dot{m}_5 h_5 + \dot{W}_{\text{t}}$
EnBE: $\dot{m}_3 s_3 + \dot{S}_{\text{gen}} = \dot{m}_5 s_5$
ExBE: $\dot{m}_3 ex_3 = \dot{m}_5 ex_5 + \dot{W}_t + \dot{E}x_{\text{dest}}$

The properties of geothermal water at the inlet of the plant and at the dead state are obtained from the EES package [3]:

$$T_1 = 170°\text{C, liquid} \longrightarrow h_1 = 719.3 \text{ kJ/kg}, \; s_1 = 2.042 \;\; \text{kJ/kg K}$$
$$T_0 = 25°\text{C}, \; P_0 = 1 \text{ atm} \longrightarrow h_0 = 104.8 \text{ kJ/kg}, \; s_0 = 0.3669 \;\; \text{kJ/kg K}$$

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

$$\dot{E}_{\text{in}} = \dot{m}(h_1 - h_0) = (120 \;\text{kg/s})[(719.3 - 104.8)\text{ kJ/kg}] = 73733 \;\text{kW}$$

The exergy of geothermal water is found to be

$$\begin{aligned}\dot{E}x_{\text{in}} &= \dot{m}[(h_1 - h_0) - T_0(s_1 - s_0)] \\ &= (120 \text{ kg/s}) \\ &\quad [(719.3 - 104.8)\text{ kJ/kg} - (25 + 273 \text{ K})(2.042 - 0.3669)\text{kJ/kg K}] \\ &= 13830 \text{ kW}\end{aligned}$$

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

$$\dot{E}x_{\text{in}} - \dot{W}_{\text{net,out}} - \dot{E}x_{\text{dest}} = 0$$
$$13830 - 4500 - \dot{E}x_{\text{dest}} = 0 \longrightarrow \dot{E}x_{\text{dest}} = 9330 \text{ kW}$$

b) The energy efficiency of the power plant is

$$\eta_{\text{en}} = \frac{\dot{W}_{\text{net,out}}}{\dot{E}_{\text{in}}} = \frac{4500 \;\text{kW}}{73733 \;\text{kW}} = 0.061 = 6.1\%$$

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

$$\eta_{ex} = \frac{\dot{W}_{net,out}}{\dot{E}x_{in}} = \frac{4500\ \text{kW}}{13830\ \text{kW}} = 0.325 = 32.5\%$$

(i) With reinjection

For the geothermal power plant with the reinjection option, shown in Figure 1.17b, one can write the thermodynamic balance equations for each component as given above for the geothermal power plant without reinjection.

The energy input of the plant can be determined as

$$\dot{E}_{in} = \dot{m}_1(h_1 - h_0) - \dot{m}_4(h_4 - h_0) = 41082\ \ \text{kW}$$

The exergy of the geothermal water is

$$\dot{E}x_{in} = \dot{m}_1(ex_1 - ex_0) - \dot{m}_4(ex_4 - ex_0) = 10702\ \ \text{kW}$$

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

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

$$10702 - 4500 - \dot{E}x_{dest} = 0 \longrightarrow \dot{E}x_{dest} = 6202\ \text{kW}$$

The energy efficiency of the power plant is

$$\eta_{en} = \frac{\dot{W}_{net,out}}{\dot{E}_{in}} = \frac{4500\ \ \text{kW}}{41082\ \ \text{kW}} = 0.1095 = 10.95\%$$

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

$$\eta_{ex} = \frac{\dot{W}_{net,out}}{\dot{E}x_{in}} = \frac{4500\ \ \text{kW}}{10702\ \ \text{kW}} = 0.4205 = 42.05\%$$

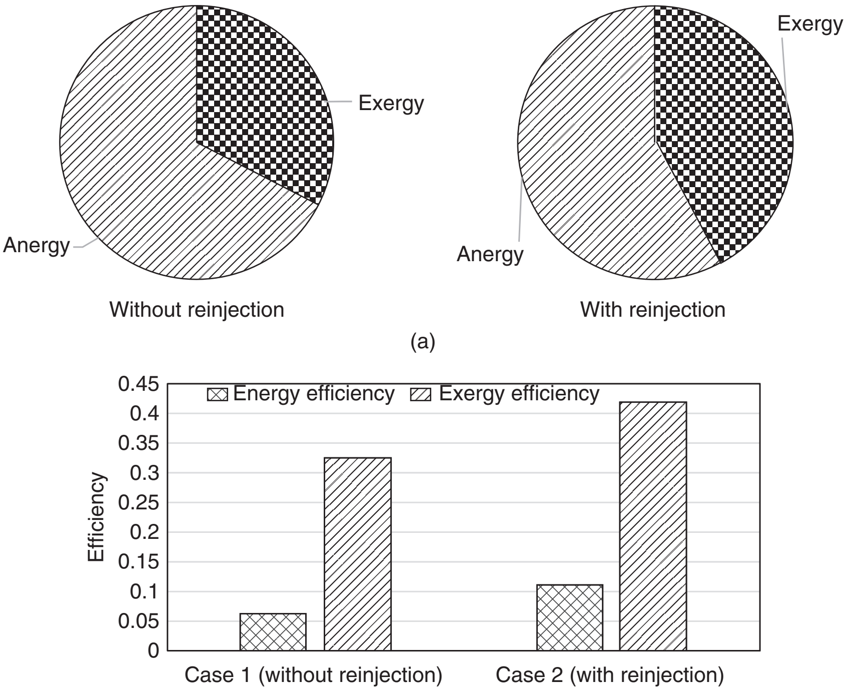
The results of these two cases are presented in Figures 1.18a and 1.18b for comparison purposes. Figure 1.18a shows the exergy and anergy contents of both cases. It is important to note that energy consists of two parts, namely exergy (useful part) and anergy (non-useful part). Figure 1.18b shows both energy and exergy efficiencies for case 1 without reinjection and case 2 with reinjection. It is clearly evident that case 2 is more efficient both energetically and exergetically. It is clear from the figure that in case of reinjection, both the energy and exergy efficiencies are higher compared to without reinjection.

Example 1.9

In this example, a house is designed to be heated by one of two options: Case 1 with an electric heater, as shown in Figure 1.19a, and Case 2 with a heat pump, as shown in Figure 1.19b.

Case 1: With an electric heater

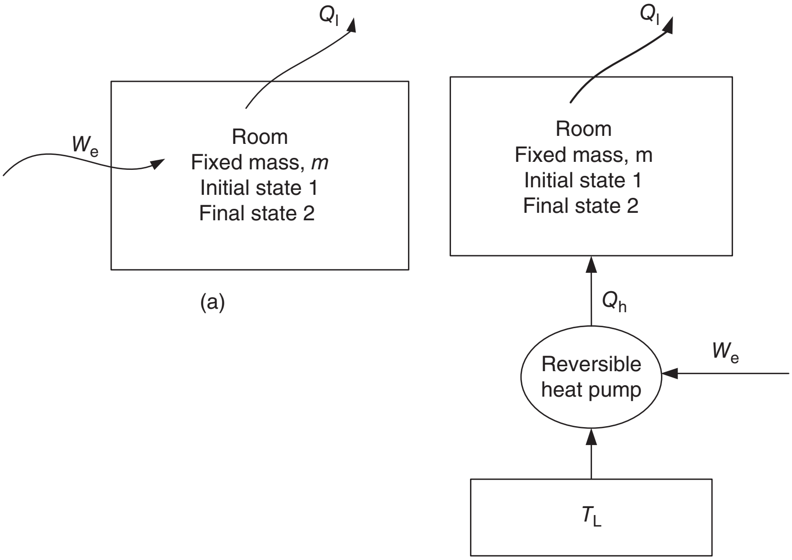
An electric resistance heater is used to heat the air inside a room of 30 m^3 volume to 23 °C when the outdoor temperature is 10 °C. Assume that the heat loss from the room



(a)

(b)

Figure 1.18 Comparisons of (a) exergy and anergy contents, and (b) energy and exergy efficiencies for Cases 1 and 2.



(a)

(b)

Figure 1.19 House heating options: (a) an electric heater and (b) a heat pump.

is 10% of the electrical heater work. Calculate (a) the electric work required by the heater, (b) the energy efficiency of the process, and (c) the exergy efficiency of the process. Take $T_0 = 10\,°\text{C}$ and $C_v = 0.718\,\text{kJ/kg K}$.

Solution

For the process shown below, one can write the thermodynamic balance equations as follows:

$$\text{MBE: } m_1 = m_2$$

$$\text{EBE: } m_1u_1 + W_e = m_2u_2 + Q_l$$

$$\text{EnBE: } m_1s_1 + S_{gen} = m_2s_2 + \frac{Q_l}{T_s}$$

$$\text{ExBE: } m_1ex_1 + W_e = m_2ex_2 + Q_l\left(1 - \frac{T_0}{T_s}\right) + Ex_{dest}$$

For air, from the EES package [3]:

$$\text{at } T_1 = 10\,°\text{C},\ \rho = 1.231\,\text{kg/m}^3$$

Now, the mass of air can be calculated as

$$m_1 = \rho V = 1.23(30) = 36.92\,\text{kg}$$

The change in specific internal energy can be calculated as

$$u_2 - u_1 = C_v(T_2 - T_1) = 0.718(23 - 13) = 9.327\,\text{kJ/kg}$$

Similarly, the change in specific entropy can be calculated as

$$(s_2 - s_1) = C_v\ln\frac{T_2}{T_1} = 0.718\ln\frac{296}{283} = 0.0322\,\text{kJ/kg K}$$

The change in specific exergy can be determined as

$$(ex_2 - ex_1) = u_2 - u_1 - T_0(s_2 - s_1) = (9.327 - (283)(0.0322)) = 0.2078\,\text{kJ/kg}$$

From the MBE, $m_1 = m_2 = 36.92\,\text{kg}$ and $Q_l = 0.1W_e$ (given).

a) So, the electric work required by the heater can be determined from the EBE as

$$W_e = \frac{m_2(u_2 - u_1)}{0.9} = \frac{36.92(9.327)}{0.9} = 382.6\,\text{kJ}$$

b) The energy efficiency of the process can be determined as

$$\eta_{en,process} = \frac{m_2(u_2 - u_1)}{W_e} = 0.9$$

c) The exergy efficiency of the process can be determined as

$$\eta_{ex,process} = \frac{m_2(ex_2 - ex_1)}{W_e} = 0.02006$$

Case 2: With a heat pump

Instead of the electric heater a heat pump is used to heat the air inside the room to 23 °C when the outdoor temperature is 10 °C. Assume that the heat loss from the room is 10%

of the heat given to the room. Determine (a) the electric work required by the heat pump, (b) the actual COP of the heat pump, and (c) the exergy efficiency of the heat pump.

Solution

a) For the process shown below, one can write the thermodynamic balance equations as follows:

$$\text{MBE: } m_1 = m_2$$

$$\text{EBE: } m_1 u_1 + Q_\text{h} = m_2 u_2 + Q_\text{l}$$

$$\text{EnBE:} m_1 s_1 + \frac{Q_\text{h}}{T_\text{h}} + S_\text{gen} = m_2 s_2 + \frac{Q_\text{l}}{T_\text{s}}$$

$$\text{ExBE:} m_1 ex_1 + Q_\text{h}\left(1 - \frac{T_0}{T_\text{s}}\right) = m_2 ex_2 + Q_\text{l}\left(1 - \frac{T_0}{T_\text{s}}\right) + Ex_\text{dest}$$

From the MBE, $m_1 = m_2 = 36.92\,\text{kg}$.

So, the heat given to the room by the heat pump can be determined from the EBE as

$$Q_\text{h} = \frac{m_2(u_2 - u_1)}{0.9} = 382.6\,\text{kJ}$$

In order to determine the electric work done required by the heat pump, first the COP of the heat pump needs to be calculated:

$$\text{COP}_\text{hp,rev} = \frac{T_\text{h}}{T_\text{h} - T_l} = \frac{23 + 273}{(23 + 273) - (10 + 273)} = 22.77$$

The minimum work requirement for the heat pump is determined from the COP definition for a heat pump as

$$W_\text{e} = \frac{Q_\text{h}}{(\text{COP}_\text{hp,rev})} = \frac{382.6}{(22.77)} = 16.8\,\text{kJ}$$

b) The actual COP of the heat pump can be determined as

$$\text{COP}_\text{hp} = \frac{m_2(u_2 - u_1)}{W_\text{e}} = 20.49$$

c) The exergetic efficiency of the heat pump can be determined as

$$\eta_\text{ex,process} = \frac{m_2(ex_2 - ex_1)}{W_\text{e}} = 0.4568$$

The results of these two cases are presented in Figures 1.20a and 1.20b for comparison purposes. Figure 1.20a shows electrical work requirements in kilojoules for both cases. It is important to note that Case 1 consumes over 22 times more than what the heat pump requires in Case 2. Figure 1.20b shows both energy and exergy efficiencies for Case 1 with an electrical heater and Case 2 with a heat pump. It is clearly evident that Case 2 is over 22 times more efficient than the corresponding electrical heater application in Case 1.

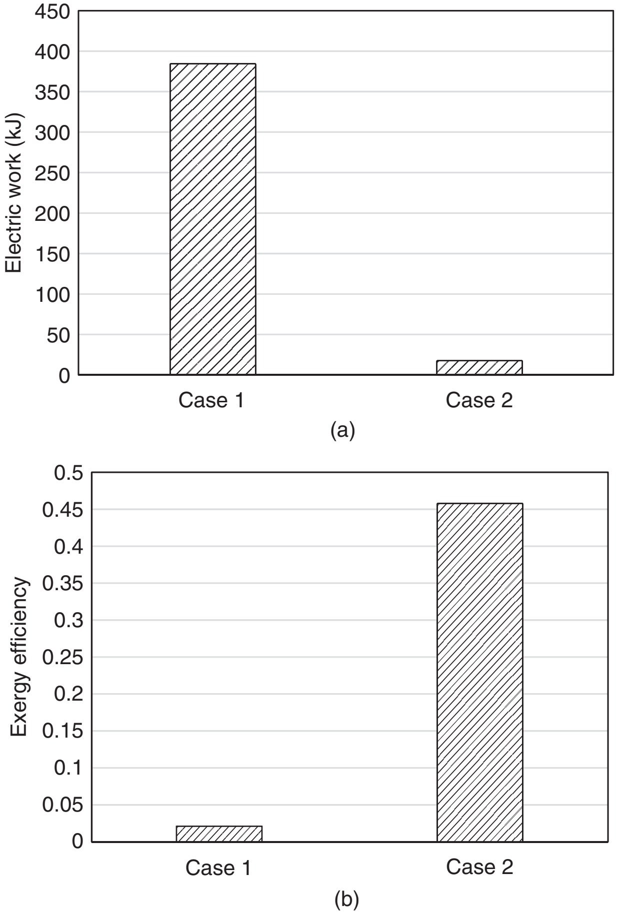


Figure 1.20 Comparisons of (a) work requirements and (b) exergy efficiencies for Cases 1 and 2.

1.6 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 heating, ventilating and air conditioning (HVAC) community. It plays a key role, not only in the heating and cooling processes, and the resulting comfort of the occupants, but 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. Furthermore, the storage conditions are different for each food commodity and should be implemented accordingly.

1.6.1 Common Definitions in Psychrometrics

There are many definitions which need to be introduced. Here, we cover some of the most common terms used 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, gaseous pollutants, etc.). Dry air does not contain 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 a specific 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 1.21). It occurs when 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:

$$\phi = \frac{P_\mathrm{v}}{P_\mathrm{s}} = \frac{\rho_\mathrm{v}}{\rho_\mathrm{s}} = \frac{v_\mathrm{s}}{v_\mathrm{v}} \tag{1.106}$$

where P_v is the partial pressure of vapor (in Pa or kPa) and P_s is the saturation pressure of vapor at the same temperature (in Pa or kPa), which can be taken directly from the saturated water table. The total pressure is $P = P_\mathrm{a} + P_\mathrm{v}$. According to Figure 1.21, $\phi = P_1/P_3$.

Humidity ratio. The humidity ratio of moist air (the *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

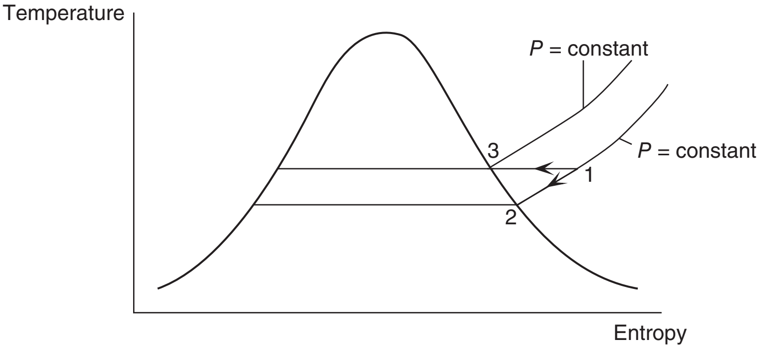


Figure 1.21 Representation of dew point temperature on a *T–s* diagram.

same temperature and pressure:

$$\omega = \frac{m_\mathrm{v}}{m_\mathrm{a}} = 0.622\ \frac{P_\mathrm{v}}{P_\mathrm{a}} \tag{1.107}$$

where $m_\mathrm{v} = P_\mathrm{v}V/R_\mathrm{v}T$ and $m_\mathrm{a} = P_\mathrm{a}V/R_\mathrm{a}T$ since both water vapor and air, as well as their mixtures, are treated as ideal gases.

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:

$$\phi = \frac{\omega P_\mathrm{a}}{0.622 P_\mathrm{s}} \tag{1.108}$$

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 1.22), some of the water in the wick is evaporated into the surrounding air, 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

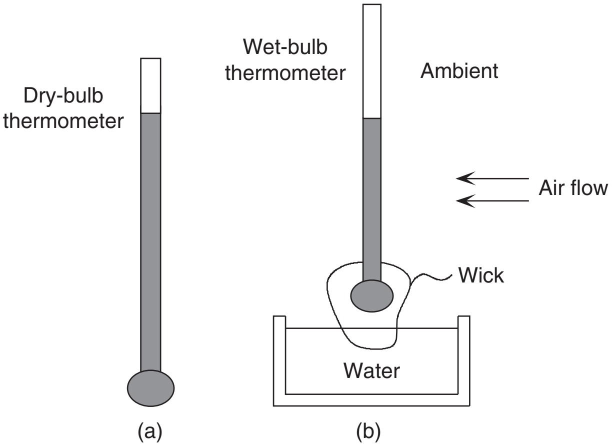


Figure 1.22 Illustration of (a) a dry-bulb thermometer and (b) a wet-bulb thermometer.

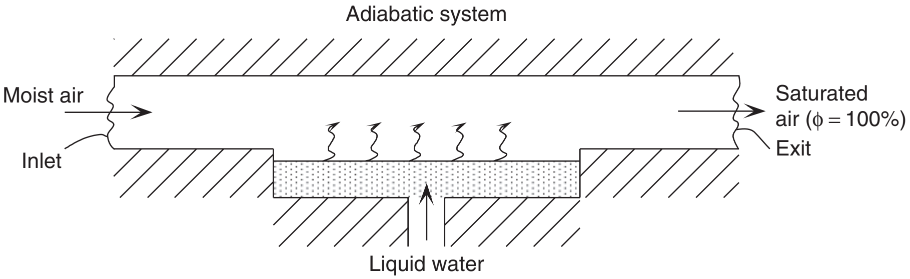


Figure 1.23 An adiabatic saturation process.

system is identified as the *adiabatic saturation temperature* and the process is called the *adiabatic saturation process* (Figure 1.23).

1.6.2 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's consider a cooling process, with negligible kinetic and potential energies and no work involved, that has two inputs and one output, as illustrated in Figure 1.24. Before going into a detailed analysis of this process, the general mass and energy balance equations may be written as follows:

- The MBEs are:

$$\Sigma\dot{m}_{a,i} = \Sigma\dot{m}_{a,e} \tag{1.109}$$

$$\Sigma\dot{m}_{v,i} + \Sigma\dot{m}_{l,i} = \Sigma\dot{m}_{a,e} + \Sigma\dot{m}_{l,e} \tag{1.110}$$

- The EBE is written as

$$\dot{Q}_i + \Sigma\dot{m}_i h_i = \Sigma\dot{m}_e h_e \tag{1.111}$$

Let's now write the respective balance equations for the subject matter system in Figure 1.24 as follows:

$$\dot{m}_{a,1} = \dot{m}_{a,3} = \dot{m}_a \tag{1.112}$$

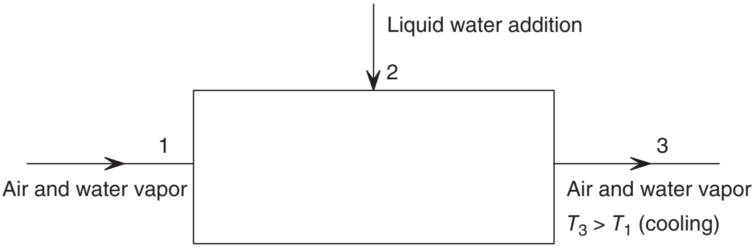


Figure 1.24 Schematic of the system.

$$\dot{m}_{\mathrm{v},1} + \dot{m}_{l,2} = \dot{m}_{\mathrm{v},3} \tag{1.113}$$

$$\dot{Q}_{\mathrm{i}} + \dot{m}_{\mathrm{a}} h_{\mathrm{a},1} + \dot{m}_{\mathrm{v},1} h_{\mathrm{v},1} + \dot{m}_{\mathrm{l},2} h_{\mathrm{l},2} = \dot{m}_{\mathrm{a}} h_{\mathrm{a},3} + \dot{m}_{\mathrm{v},3} h_{\mathrm{v},3} \tag{1.114}$$

Equation (1.114) can be arranged in terms of the humidity ratio under $\omega = m_{\mathrm{v}}/m_{\mathrm{a}}$ (Equation (1.107)):

$$\frac{\dot{Q}_{\mathrm{i}}}{\dot{m}_{\mathrm{a}}} + h_{\mathrm{a},1} + \omega_1 h_{\mathrm{v},1} + (\omega_1 - \omega_2) h_{\mathrm{l},2} = h_{\mathrm{a},3} + \omega_3 h_{\mathrm{v},3} \tag{1.115}$$

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

1.6.3 The Psychrometric Chart

This chart was developed in the early 1900s by a German engineer named Richard Mollier. It is the graph, as shown in Figure 1.25, which 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 moist air. It is important to note that the chart can only be used for atmospheric pressure (i.e., 1 bar or 100 kPa). If the pressure is different, the moist air equations should be used.

Understanding the dynamics of moisture and air will provide a solid foundation for understanding the principles of cooling and air-conditioning systems. Figure 1.26 shows several processes on the psychrometric chart. Figure 1.26a exhibits cooling and heating processes, and therefore is 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 1.26b 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 1.26c, which includes both

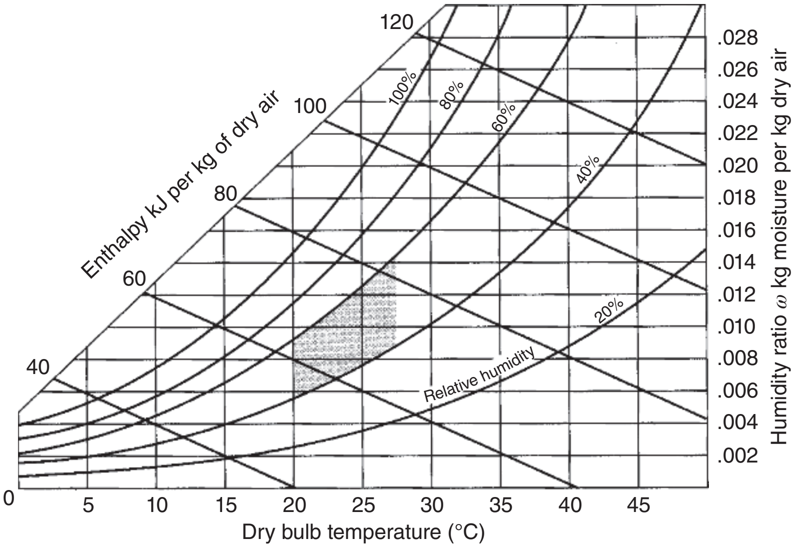


Figure 1.25 A psychrometric chart.

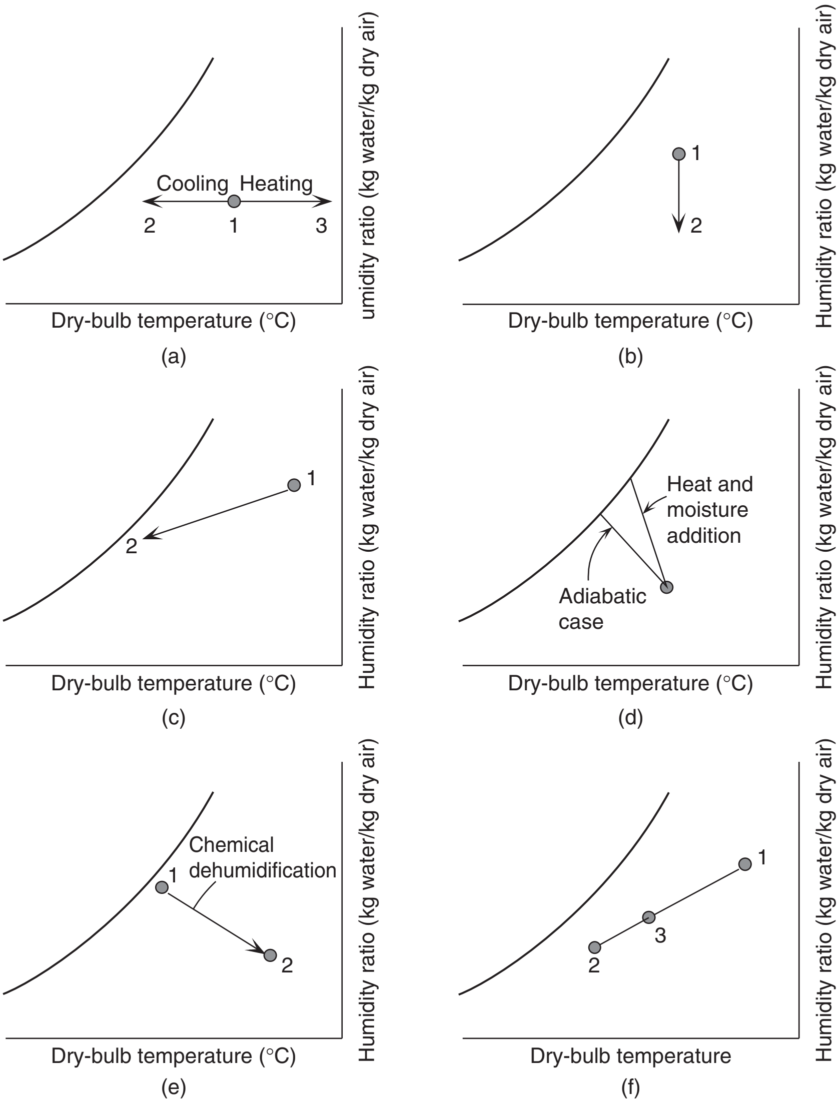


Figure 1.26 Some processes on the psychrometric chart: (a) cooling and heating, (b) dehumidification, (c) cooling and dehumidification, (d) adiabatic humidification, (e) chemical dehumidification, and (f) mixture of two moist air flows.

cooling and dehumidification, resulting in a decrease of both the dry-bulb and wet-bulb temperatures, as well as the humidity ratio. Figure 1.26d 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 1.26e 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. Figure 1.26f 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.

Example 1.10

In this example we study the above psychrometric processes (as given in Figure 1.26) thermodynamically by writing all balance equations for mass, energy, entropy, and exergy, and defining both energy and exergy efficiencies.

Case 1: Simple Heating

Air is heated by a heating coil in which hot water is flowing at an average temperature of 90 °C. Using the values given in the figure below, determine (a) the exergy destruction, (b) the energy efficiency for this process, and (c) the exergy efficiency for this process.

Solution

a) For the simple heating process shown in Figure 1.27, one can write the thermodynamic balance equations as follows:

$$\text{MBE:}\quad \dot{m}_{a1} = \dot{m}_{a2}\quad \text{(for dry air)}$$

$$\dot{m}_{w1} = \dot{m}_{w2}\quad \text{(for water)}$$

$$\text{EBE:}\quad \dot{m}_{a1}h_1 + \dot{Q}_{in} = \dot{m}_{a2}h_2$$

$$\text{EnBE:}\quad \dot{m}_{a1}s_1 + \frac{\dot{Q}_{in}}{T_s} + \dot{S}_{gen} = \dot{m}_{a2}s_2$$

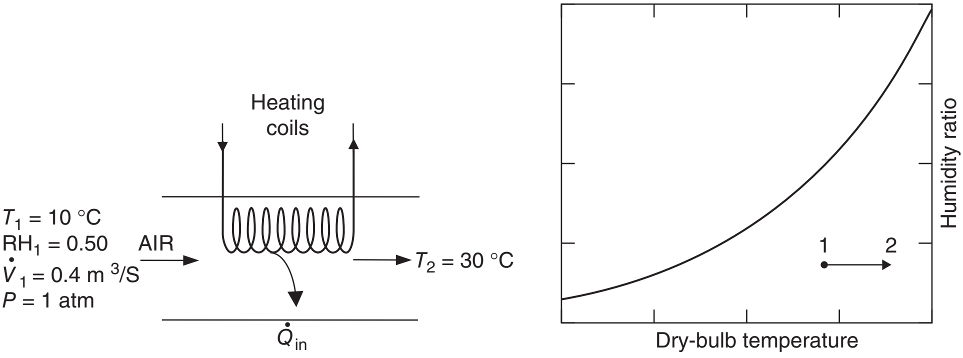
$$\text{ExBE:}\quad \dot{m}_{a1}ex_1 + \dot{Q}_{in}\left(1 - \frac{T_0}{T_s}\right) = \dot{m}_{a2}ex_2 + Ex_{dest}$$

For moist air, from the EES packages [3] one can obtain the properties at state 1 as

$$\left.\begin{aligned} T_1 &= 10\,^\circ\text{C} \\ P_1 &= 101.321\,\text{kPa} \\ \text{RH}_1 &= 0.50 \end{aligned}\right\} \left\{\begin{aligned} v_1 &= 0.8033\,\text{m}^3/\text{kg} \\ h_1 &= 19.59\,\text{kJ/kg} \\ s_1 &= 5.68\,\text{kJ/kg K} \\ \omega_1 &= 0.003792\,\text{kg water/kg air} \end{aligned}\right.$$

The dead state temperature is taken to be the same as the inlet temperature of air. Since it is a simple heating process, the specific humidity does not change, so $\omega_1 = \omega_2$. The properties at state 2 can be obtain as

$$\left.\begin{aligned} T_2 &= 30\,^\circ\text{C} \\ P_2 &= 101.321\,\text{kPa} \\ \omega_2 &= 0.003792\,\text{kg water/kg air} \end{aligned}\right\} \begin{aligned} h_2 &= 39.82\,\text{kJ/kg} \\ s_2 &= 5.749\,\text{kJ/kg K} \end{aligned}$$



(a) (b)

Figure 1.27 (a) A simple heating process and (b) illustration of the process in the chart.

The specific exergy at state 1 can be calculated as

$$ex_1 = h_1 - h_0 - T_0(s_1 - s_0) = 19.59 - 19.59 - (10 + 273)(5.68 - 5.68) = 0\,\text{kJ/kg}$$

Similarly, for state 2:

$$\begin{aligned} ex_2 &= h_2 - h_0 - T_0(s_2 - s_0) = 39.82 - 19.59 - (10 + 273)(5.749 - 5.68) \\ &= 0.687\,\text{kJ/kg} \end{aligned}$$

The mass flow rate of the air can be calculated as

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{0.4}{0.8033} = 0.498\,\text{kg/s}$$

From the MBE, $\dot{m}_{a1} = \dot{m}_{a2}$
By using the EBE one can find the rate of heat input as

$$\dot{Q}_{in} = \dot{m}_{a1}(h_2 - h_1) = 0.498(39.82 - 19.59) = 10.07\,\text{kW}$$

The exergy destruction can be determined from the ExBE as

$$\begin{aligned} \dot{E}x_{dest} &= \dot{m}_{a1}(ex_1 - ex_2) + \dot{Q}_{in}\left(1 - \frac{T_0}{T_s}\right) \\ &= 0.498(0 - 0.687) + 10.07\left(1 - \frac{10 + 273}{90 + 273}\right) = 1.878\,\text{kW} \end{aligned}$$

b) For the simple heating process, the energy efficiency can be determined as follows:

$$\eta_{en} = \left(\frac{\dot{m}_{a2}h_2 - \dot{m}_{a1}h_1}{\dot{Q}_{in}}\right) = \frac{10.07}{10.07} = 1 = 100\%$$

c) Similarly, for the simple heating process, the exergy efficiency can be determined as follows:

$$\eta_{ex} = \left(\frac{\dot{m}_{a2}ex_2 - \dot{m}_{a1}ex_1}{\dot{Q}_{in}\left(1 - \frac{T_0}{T_s}\right)}\right) = 0.1541 = 15.4\%$$

Case 2: Simple Cooling

Air is cooled by a cooling coil. Using the values given in Figure 1.28, determine (a) the exergy destruction, (b) the energy efficiency for this process, and (c) the exergy efficiency for this process. Take $T_0 = 25\,°\text{C}$ and $\text{RH}_0 = 0.80$.

Solution

a) For the simple cooling process shown below, one can write the thermodynamic balance equations as follows:

MBE: $\dot{m}_{a1} = \dot{m}_{a2}$ (for dry air)
$\dot{m}_{w1} = \dot{m}_{w2}$ (for water)
EBE: $\dot{m}_{a1}h_1 = \dot{m}_{a2}h_2 + \dot{Q}_{out}$

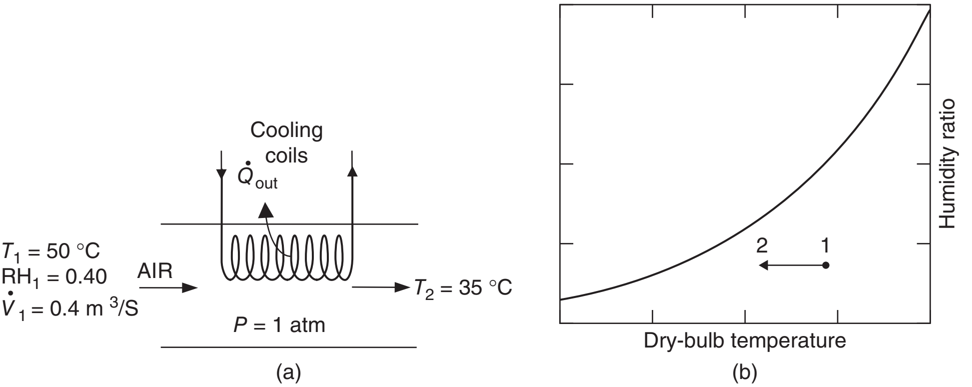


Figure 1.28 (a) A simple cooling process and (b) illustration of the process in the chart.

$$\text{ExBE:}\quad \dot{m}_{a1}s_1 + \dot{S}_{gen} = \dot{m}_{a2}s_2 + \frac{\dot{Q}_{out}}{T_s}$$

$$\text{ExBE:}\quad \dot{m}_{a1}ex_1 = \dot{m}_{a2}ex_2 + \dot{Q}_{out}\left(1 - \frac{T_0}{T_s}\right) + \dot{E}x_{dest}$$

For moist air, from the EES package [3] one can obtain the properties at state 1 as

$$\left.\begin{array}{l} T_1 = 50\,^\circ\mathrm{C} \\ P_1 = 101.321\,\mathrm{kPa} \\ \mathrm{RH}_1 = 0.40 \end{array}\right\}\begin{array}{ll} v_1 & = 0.9263\,\mathrm{m^3/kg} \\ h_1 & = 132.9\,\mathrm{kJ/kg} \\ s_1 & = 6.062\,\mathrm{kJ/kg\,K} \\ \omega_1 & = 0.03188\,\mathrm{kg\,water/kg\,air} \end{array}$$

The properties at reference state 0 can be obtain as

$$\left.\begin{array}{l} T_0 = 25\,^\circ\mathrm{C} \\ P_0 = 101.321\,\mathrm{kPa} \\ \mathrm{RH}_0 = 0.80 \end{array}\right\}\begin{array}{ll} h_0 & = 65.77\,\mathrm{kJ/kg} \\ s_0 & = 5.841\,\mathrm{kJ/kg\,K} \end{array}$$

Since it is simple cooling process, the specific humidity does not change, so $\omega_1 = \omega_2$. The properties at state 2 can be obtain as

$$\left.\begin{array}{l} T_2 = 35\,^\circ\mathrm{C} \\ P_2 = 101.321\,\mathrm{kPa} \\ \omega_2 = 0.03188\,\mathrm{kg\,water/kg\,air} \end{array}\right\}\begin{array}{ll} h_2 & = 117\,\mathrm{kJ/kg} \\ s_2 & = 6.011\,\mathrm{kJ/kg\,K} \end{array}$$

The specific exergy at state 1 can be calculated as

$$ex_1 = h_1 - h_0 - T_0(s_1 - s_0) = 1.332\,\mathrm{kJ/kg}$$

Similarly, for state 2:

$$ex_2 = h_2 - h_0 - T_0(s_2 - s_0) = 0.4696\,\mathrm{kJ/kg}$$

The mass flow rate of the air can be calculated as

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{0.4\,\mathrm{m^3/s}}{0.9263\,\mathrm{m^3/kg}} = 0.4318\,\mathrm{kg/s}$$

From the MBE, $\dot{m}_{a1} = \dot{m}_{a2}$

By using the EBE one can find the rate of heat input as

$$\dot{Q}_{\mathrm{out}} = \dot{m}_{\mathrm{a1}}(h_1 - h_2) = 0.4318(132.9 - 117) = 6.9\,\mathrm{kW}$$

The exergy destruction can be determined from the ExBE as

$$\begin{aligned}\dot{E}x_{\mathrm{dest}} &= \dot{m}_{\mathrm{a1}}(ex_1 - ex_2) - \dot{Q}_{\mathrm{out}}\left(1 - \frac{T_0}{T_{\mathrm{s}}}\right)\\ &= 0.4318(1.332 - 0.4696) + 6.9\left(1 - \frac{25+273}{35+273}\right)\\ &= 0.1485\,\mathrm{kW}\end{aligned}$$

b) For the simple heating process the energy efficiency can be defined as follows:

$$\eta_{\mathrm{en}} = \left(\frac{\dot{Q}_{\mathrm{out}}}{\dot{m}_{\mathrm{a1}}h_1 - \dot{m}_{\mathrm{a2}}h_2}\right) = 1 = 100\%$$

c) Similarly, for the simple heating process the energy efficiency can be defined as follows:

$$\eta_{\mathrm{ex}} = \left(\frac{\dot{Q}_{\mathrm{out}}\left(1 - \frac{T_0}{T_{\mathrm{s}}}\right)}{\dot{m}_{\mathrm{a1}}ex_1 - \dot{m}_{\mathrm{a2}}ex_2}\right) = 0.601 = 60.1\%$$

Case 3: Heating with Humidification

A heating process with humidification is considered. Using the values given in Figure 1.29, determine (a) the exergy destruction, (b) the energy efficiency for this process, and (c) the exergy efficiency for this process. Take the reference state as the inlet state.

Solution

a) For the simple heating process shown below, one can write the thermodynamic balance equations as follows:

$$\begin{aligned}\text{MBE:}\quad \dot{m}_{\mathrm{a1}} &= \dot{m}_{\mathrm{a2}} = \dot{m}_{\mathrm{a3}} \quad \text{(for dry air)}\\ \dot{m}_{\mathrm{w1}} &= \dot{m}_{\mathrm{w2}} \quad \text{(for water)}\\ \dot{m}_{\mathrm{w1}} + \dot{m}_{\mathrm{w}} &= \dot{m}_{\mathrm{w3}} \quad \text{(for water)}\end{aligned}$$

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

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

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

$$\text{EnBE:}\quad \dot{m}_{\mathrm{a1}}s_1 + \frac{\dot{Q}_{\mathrm{in}}}{T_{\mathrm{s}}} + \dot{S}_{\mathrm{gen}} = \dot{m}_{\mathrm{a2}}s_2 \quad \text{(process 1 – 2)}$$

$$\dot{m}_{\mathrm{a2}}s_2 + \dot{m}_{\mathrm{w}}s_{\mathrm{w}} + \dot{S}_{\mathrm{gen}} = \dot{m}_{\mathrm{a3}}s_3 \quad \text{(state 2 – 3)}$$

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

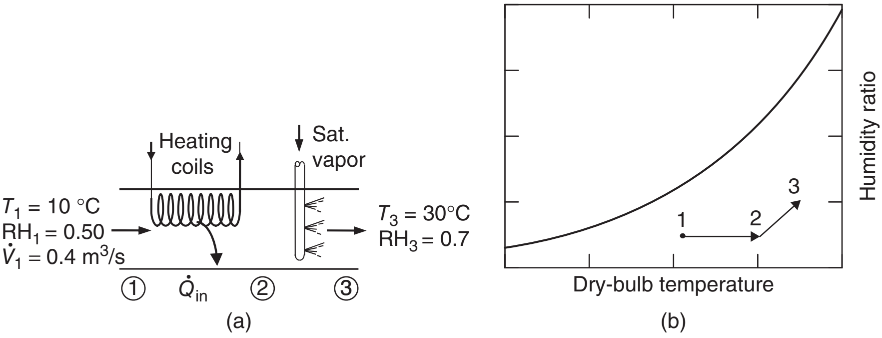


Figure 1.29 (a) A heating with humidification process and (b) illustration of the process in the chart.

$$\text{ExBE:}\quad \dot{m}_{a1}ex_{a1} + \dot{Q}_{in}\left(1 - \frac{T_0}{T_s}\right) = \dot{m}_{a2}ex_2 + \dot{E}x_{dest} \quad \text{(process 1 − 2)}$$

$$\dot{m}_{a2}ex_{a2} + \dot{m}_w ex_w = \dot{m}_{a3}ex_{a3} + \dot{E}x_{dest} \quad \text{(process 2 − 3)}$$

$$\dot{m}_{a1}ex_{a1} + \dot{Q}_{in}(1 - \tfrac{T_0}{T_s}) + \dot{m}_w ex_w = \dot{m}_{a3}ex_3 + \dot{E}x_{dest} \quad \text{(process 1 − 3)}$$

For moist air, from the EES package [3] one can obtain the properties at state 1 as

$$\left.\begin{array}{l} T_1 = 10\,^\circ\text{C} \\ P_1 = 101.321\,\text{kPa} \\ \text{RH}_1 = 0.50 \end{array}\right\} \begin{array}{ll} v_1 & = 0.8033\,\text{m}^3/\text{kg} \\ h_1 & = 19.59\,\text{kJ/kg} \\ s_1 & = 5.68\,\text{kJ/kg K} \\ \omega_1 & = 0.003792\,\text{kg water/kg air} \end{array}$$

Since 1–2 is a simple heating process, the specific humidity does not changes, so $\omega_1 = \omega_2$
The properties at state 2 can be obtain as

$$\left.\begin{array}{l} T_2 = 20\,^\circ\text{C} \\ P_2 = 101.321\,\text{kPa} \\ \omega_2 = 0.003792\,\text{kg water/kg air} \end{array}\right\} \begin{array}{ll} h_2 & = 29.7\,\text{kJ/kg} \\ s_2 & = 5.715\,\text{kJ/kg K} \end{array}$$

The properties at state 3 can be obtain as

$$\left.\begin{array}{l} T_3 = 30\,^\circ\text{C} \\ P_3 = 101.321\,\text{kPa} \\ \text{RH}_3 = 0.70 \end{array}\right\} \begin{array}{ll} h_3 & = 78.19\,\text{kJ/kg} \\ s_3 & = 5.883\,\text{kJ/kg K} \end{array}$$

The specific exergy at state 1 can be calculated as

$$ex_1 = h_1 - h_0 - T_0(s_1 - s_0) = 0\,\text{kJ/kg}$$

Similarly for state 2 and 3, respectively

$$ex_2 = h_2 - h_0 - T_0(s_2 - s_0) = 0.1796\,\text{kJ/kg}$$
$$ex_3 = h_3 - h_0 - T_0(s_3 - s_0) = 1.319\,\text{kJ/kg}$$

For water, using the EES package [3] one can obtain the properties as

$$\left.\begin{aligned} T_w &= 100\,^\circ\text{C} \\ x_w &= 1 \end{aligned}\right\} \begin{aligned} h_w &= 2676\,\text{kJ/kg} \\ s_w &= 7.354\,\text{kJ/kg K} \end{aligned}$$

The specific exergy of water can be calculated as

$$ex_w = h_w - h_{w0} - T_0(s_w - s_{w0}) = 595\,\text{kJ/kg}$$

By using the energy balance for process 1–2, the rate at which heat is added can be found as

$$\dot{Q}_{in} = 5.036\,\text{kW}$$

Using the ExBE, the exergy destruction is determined as

$$\dot{Ex}_{dest} = 6.076\,\text{kW}$$

b) For the heating with humidification process, the energy efficiency can be determined as follows:

$$\eta_{en} = \left(\frac{\dot{m}_{a3}h_3 - \dot{m}_{a1}h_1}{\dot{Q}_{in} + \dot{m}_w h_w}\right) = 1 = 100\%$$

c) Similarly, for the heating with humidification process, the exergy efficiency can be determined as

$$\eta_{ex} = \left(\frac{\dot{m}_{a3}ex_3 - \dot{m}_{a1}ex_1}{\dot{Q}_{in}\left(1 - \frac{T_0}{T_s}\right) + \dot{m}_w ex_w}\right) = 0.0991 = 9.9\%$$

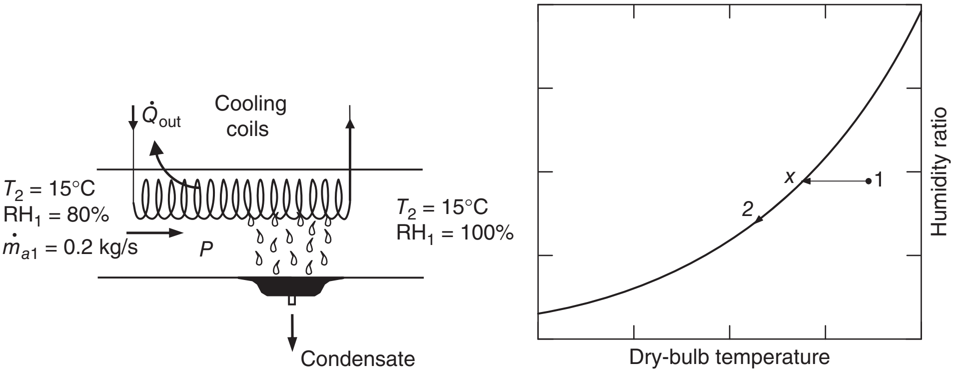
Case 4: Cooling with Dehumidification

For the cooling with dehumidification process, use the given data shown in Figure 1.30. Determine (a) the exergy destruction, (b) the energy efficiency for this process, and (c) the exergy efficiency for this process. Take $T_0 = 25\,^\circ\text{C}$ and $\text{RH}_0 = 0.90$.

Solution

a) For the cooling with dehumidification process shown below, one can write the thermodynamic balance equations as follows:

$$\begin{aligned} &\text{MBE:} \quad \dot{m}_{a1} = \dot{m}_{a2} \quad \text{(for dry air)} \\ &\qquad\quad \dot{m}_{w1} = \dot{m}_{w2} + \dot{m}_w \quad \text{(for water)} \\ &\text{EBE:} \quad \dot{m}_{a1}h_1 = \dot{m}_{a2}h_2 + \dot{Q}_{out} + \dot{m}_w h_w \\ &\text{EnBE:} \quad \dot{m}_{a1}s_1 + \dot{S}_{gen} = \dot{m}_{a2}s_2 + \frac{\dot{Q}_{out}}{T_s} + \dot{m}_w s_w \\ &\text{ExBE:} \quad \dot{m}_{a1}ex_1 = \dot{m}_{a2}ex_2 + \left|\dot{Q}_{out}\left(1 - \frac{T_0}{T_s}\right)\right| + \dot{m}_w ex_w + Ex_{dest} \end{aligned}$$



(a)

(b)

Figure 1.30 (a) A cooling with dehumidification process and (b) illustration of the process in the chart.

By using the EES package [3] one can obtain the properties at state 1

$$\left.\begin{aligned} T_1 &= 35\,°\mathrm{C} \\ P_1 &= 101.321\,\mathrm{kPa} \\ \mathrm{RH}_1 &= 0.80 \end{aligned}\right\} \begin{aligned} \omega_1 &= 0.02516\,\mathrm{kg\,water/kg\,air} \\ h_1 &= 99.72\,\mathrm{kJ/kg} \\ s_1 &= 5.954\,\mathrm{kJ/kg\,K} \end{aligned}$$

Similarly, for state 2:

$$\left.\begin{aligned} T_2 &= 15\,°\mathrm{C} \\ P_2 &= 101.321\,\mathrm{kPa} \\ \mathrm{RH}_1 &= 1.0 \end{aligned}\right\} \begin{aligned} \omega_2 &= 0.01064\,\mathrm{kg\,water/kg\,air} \\ h_2 &= 41.98\,\mathrm{kJ/kg} \\ s_2 &= 5.759\,\mathrm{kJ/kg\,K} \end{aligned}$$

The specific exergy at state 1 can be calculated as

$$ex_1 = h_1 - h_0 - T_0(s_1 - s_0) = 0.7727\,\mathrm{kJ/kg}$$

Similarly, for state 2:

$$ex_2 = h_2 - h_0 - T_0(s_2 - s_0) = 0.142\,\mathrm{kJ/kg}$$

For water using the EES, one can obtain the properties as

$$\left.\begin{aligned} T_w &= 15\,°\mathrm{C} \\ x_w &= 0 \end{aligned}\right\} \begin{aligned} h_w &= 62.92\,\mathrm{kJ/kg} \\ s_w &= 0.2242\,\mathrm{kJ/kg\,K} \end{aligned}$$

The specific exergy of water can be calculated as

$$ex_w = h_w - h_{w0} - T_0(s_w - s_{w0}) = 0.169\,\mathrm{kJ/kg}$$

The mass flow rate of condense water can be found from the MBE as

$$\dot{m}_w = \dot{m}_{a1}(\omega_1 - \omega_2) = 0.002905\,\mathrm{kg/s}$$

By using the EBE, the heat removal rate can be found as

$$\dot{Q}_{\text{out}} = 11.36\,\text{kW}$$

Using the ExBE, the exergy destruction is determined as

$$\dot{E}x_{\text{dest}} = 0.1083\,\text{kW}$$

b) For the cooling with dehumidification process, the energy efficiency can be determined as follows:

$$\eta_{\text{en}} = \left(\frac{\dot{m}_{\text{w}}h_{\text{w}} + \dot{Q}_{\text{out}}}{\dot{m}_{\text{a1}}h_1 - \dot{m}_{\text{a2}}h_2}\right) = 1 = 100\%$$

c) Similarly, for cooling with dehumidification, the exergy efficiency can be determined as

$$\eta_{\text{ex}} = \left(\frac{\left|\dot{Q}_{\text{out}}\left(1 - \frac{T_0}{T_{\text{s}}}\right)\right| + \dot{m}_{\text{w}}ex_{\text{w}}}{\dot{m}_{\text{a1}}ex_1 - \dot{m}_{\text{a2}}ex_2}\right) = 0.1415 = 14.1\%$$

Case 5: Simple Adiabatic Mixing

For the simple adiabatic mixing process and using the data shown in Figure 1.31, determine (a) the exergy destruction, (b) the energy efficiency for this process, and (c) the exergy efficiency. Take $T_0 = 10\,°\text{C}$ and $\text{RH}_0 = 0.90$.

State 1 $\dot{m}_{\text{a3}} = 1\,\text{kg/s}$

$T_1 = 15\,°\text{C}$

$\text{RH}_1 = 1$

State 2 $\dot{m}_{\text{a3}} = 0.5\,\text{kg/s}$

$T_2 = 35\,°\text{C}$

$\text{RH}_2 = 0.7$

Solution

a) For the adiabatic mixing process shown in Figure 1.31, one can write the thermodynamic balance equations as follows:

$$\begin{aligned} \text{MBE}: \quad & \dot{m}_{\text{a1}} + \dot{m}_{\text{a2}} = \dot{m}_{\text{a3}} \ \text{(for air)} \\ & \dot{m}_{\text{w1}} + \dot{m}_{\text{w2}} = \dot{m}_{\text{w3}} \ \text{(for water)} \\ \text{EBE}: \quad & \dot{m}_{\text{a1}}h_1 + \dot{m}_{\text{a2}}h_2 = \dot{m}_{\text{a3}}h_3 \\ \text{EnBE}: \quad & \dot{m}_{\text{a1}}s_1 + \dot{S}_{\text{gen}} + \dot{m}_{\text{a2}}s_2 = \dot{m}_{\text{a3}}s_3 \\ \text{EBE}: \quad & m_{\text{a1}}ex_1 + \dot{m}_{\text{a2}}ex_2 = \dot{m}_{\text{a3}}ex_3 + \dot{E}x_{\text{dest}} \end{aligned}$$

By using the EES package [3] one can obtain the properties at state 1

$$\left.\begin{array}{l} T_1 = 15\,°\text{C} \\ P_1 = 101.321\,\text{kPa} \\ \text{RH}_1 = 1 \end{array}\right\} \begin{array}{l} \omega_1 = 0.01064\,\text{kg water/kg air} \\ h_1 = 41.98\,\text{kJ/kg} \\ s_1 = 5.759\,\text{kJ/kg K} \end{array}$$

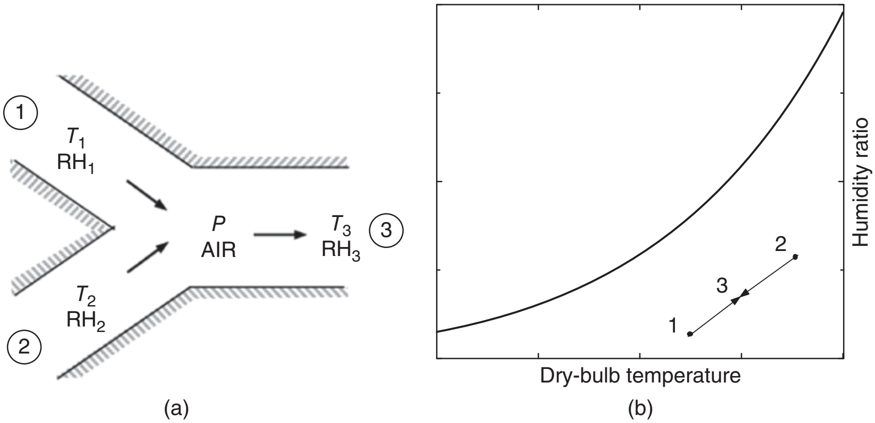


Figure 1.31 (a) A simple adiabatic mixing process and (b) illustration of the process in the chart.

Similarly, at state 2:

$$\left.\begin{aligned} T_2 &= 35\,^\circ\mathrm{C} \\ P_2 &= 101.321\,\mathrm{kPa} \\ \mathrm{RH}_2 &= 0.7 \end{aligned}\right\}\begin{aligned} \omega_2 &= 0.01064\,\mathrm{kg\,water/kg\,air} \\ h_2 &= 99.72\,\mathrm{kJ/kg} \\ s_2 &= 5.954\,\mathrm{kJ/kg\,K} \end{aligned}$$

The specific exergy at state 1 can be calculated as

$$ex_1 = h_1 - h_0 - T_0(s_1 - s_0) = 0.1434\,\mathrm{kJ/kg}$$

Similarly, for state 2:

$$ex_2 = h_2 - h_0 - T_0(s_2 - s_0) = 2.723\,\mathrm{kJ/kg}$$

By using the EBE, the enthalpy at state 3 can be found as

$$h_3 = 61.23\,\mathrm{kJ/kg}$$

By using the MBE one can find the specific humidity at state 3 as

$$\omega_1 \dot{m}_{a1} + \omega_2 \dot{m}_{a2} = \omega_3 \dot{m}_{a3}$$
$$\omega_3 = 0.01548\,\mathrm{kg\,water/kg\,air}$$

The properties at state 3 can be found as

$$\left.\begin{aligned} w_3 &= 0.01548\,\mathrm{kg\,water/kg\,air} \\ P_3 &= 101.321\,\mathrm{kPa} \\ h_3 &= 61.23\,\mathrm{kJ/kg} \end{aligned}\right\} s_3 = 5.826\,\mathrm{kJ/kg\,K}$$

The specific exergy at state 3 can be calculated as

$$ex_3 = h_3 - h_0 - T_0(s_3 - s_0) = 0.6614\,\mathrm{kJ/kg}$$

The exergy destruction in the process can be determined from the ExBE as

$$\dot{E}x_{\mathrm{dest}} = exd = 0.5128\,\mathrm{kW}$$

b) For the adiabatic process, the energy efficiency can be determined as follows:

$$\eta_{\text{en}} = \left(\frac{\dot{m}_{\text{a3}} h_3}{\dot{m}_{\text{a1}} h_1 + \dot{m}_{\text{a2}} h_2}\right) = 1 = 100\%$$

c) Similarly, for the adiabatic mixing process, the exergy efficiency can be defined as follows:

$$\eta_{\text{ex}} = \left(\frac{\dot{m}_3 ex_3}{\dot{m}_{\text{a2}} ex_2 + \dot{m}_{\text{a1}} ex_1}\right) = 0.659 = 65.9\%$$

1.7 Concluding Remarks

In this chapter, we present major topics under general aspects of thermodynamics, covering dimensions and units, thermodynamic laws (particularly the first and second laws of thermodynamics), the six-step approach in thermodynamics (consisting of property, state, process, cycle, balance equations, and performance assessment), ideal and real gases, Carnot refrigerators and heat pumps, and psychrometrics. There are specific examples selected and given for every section to make it clear and allow the reader to understand the concepts and principles better and comprehend sufficient knowledge for the next chapters. It has also some parametric studies and their results to highlight the critical aspects in the analyses and performance assessments.

Nomenclature

a	acceleration, m/s^2
A	cross-sectional area, m^2; surface area, m^2
c	specific heat, kJ/kg K
c_{p}	constant-pressure specific heat, kJ/kg K
c_{v}	constant-volume specific heat, kJ/kg K
e	specific energy, kJ/kg
E	energy, kJ
$\dot{E}$	energy rate, W or kW
ex	specific exergy, kJ/kg
Ex	exergy, kJ
$\dot{E}x$	exergy rate, W or kW
F	force; drag force, N
g	acceleration due to gravity (=9.81 m/s^2)
h	specific enthalpy, kJ/kg
H	enthalpy, kJ
k	adiabatic exponent, specific heat ratio
ke	specific kinetic energy, kJ/kg

L	length, m
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
M	molecular weight, kg/kmol
n	mole number, kmol
P	pressure, kPa
P_r	reduced pressure, kPa
pe	specific potential energy, kJ/kg
q	specific heat transfer, kJ/kg
Q	heat transfer, kJ
$\dot{Q}$	heat transfer rate, W or kW
R	gas constant, kJ/kg K
$\overline{\mathrm{R}}$	universal gas constant, kJ/kg K
RH	relative humidity (%)
s	specific entropy, kJ/kg K
S	entropy, kJ/K
S_{gen}	entropy generation, kJ/K
$\dot{S}_{gen}$	entropy generation rate, W or kW
t	time, s
T	temperature, °C or K
T_r	reduced temperature, K
u	specific internal energy, kJ/kg
U	internal energy, kJ
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	specific work, kJ/kg
W	work, kJ
$\dot{W}$	work rate or power, W or kW
x	quality, kg/kg
y	mole fraction
Z	compressibility factor

Greek Letters

Δ	difference
η	efficiency
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3
ϕ	relative humidity, %
ω	humidity ratio, kg/kg

Subscripts and Superscripts

a	air; medium; surroundings
abs	absolute
av	average
atm	atmosphere
b	boundary
c	compressor
C	Carnot
db	dry-bulb
dest	destruction
e	electrical; exit
en	energy
ex	exergy
f	final
g	gauge
gen	generation
H	high-temperature
hp	heat pump
HP	heat pump
i	initial; inlet
in	in; input
irr	irreversible
l	loss
liq	liquid
L	low-temperature
min	minimum
net	net
out	out; output
p	paddle
r	refrigerant
R	reversible; reversed; refrigerator
rev	reversible
s	source
th	thermal
tot	total
v	vapor
vap	vapor
wb	wet-bulb
0	surroundings; ambient; environment; reference
1, 2, 3…	state points

Acronyms

COP	coefficient of performance
EBE	energy balance equation
EES	engineering equation solver
EnBE	entropy balance equation
ExBE	exergy balance equation
FLT	first law of thermodynamics
MBE	mass balance equation
RH	relative humidity
SLT	second law of thermodynamics

Study Problems

1.1 Explain why SI units are internationally accepted and the most widely used throughout the world.

1.2 Explain mathematically the difference between mass and weight.

1.3 What is specific heat? Define two common types of specific heats used in thermodynamics. Is specific heat a function of temperature?

1.4 Explain the basic operating principle of thermocouples. What are some typical applications depending on the type of thermocouples? Explain why people prefer digital thermometers over other old analog devices.

1.5 Consider the flow of a refrigerant vapor through a compressor, which is operating at steady state conditions. Do both mass flow rate and volumetric 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. Explain how one can evaluate each component as a closed system or as a control volume; as a steady-flow system or unsteady-flow system? Explain.

1.7 Explain the difference between an adiabatic system and an isolated system, and illustrate both schematically and compare them accordingly.

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 with right units.

1.10 What is the weight of a 80-kg substance in N, kN, kgf, and lbf?

1.11 The vacuum pressure of a tank is given as 40 kPa. If the atmospheric pressure is 95 kPa, what are the gage pressures and absolute pressures in kPa, kN/m^2, lbf/in^2, psi, and mm Hg?

1.12 Express −38 °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 as 4.18 kJ/kg·°C. Express this value in kJ/kg·K, J/gr·°C, kCal/kg·°C, and Btu/lbm·°F.

1.15 0.2 kg of R-134a at 700 kPa pressure initially at 4 °C is heated until 50% of the mass is vaporized. Calculate the temperature at which the refrigerant is vaporized, and sensible and latent heats transferred to the refrigerant.

1.16 A 0.5 kg of R134a at 1 MPa pressure initially at 20 °C is heated until 60% 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 the mass has melted. Calculate both sensible and latent heats 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. Calculate sensible and latent heat transferred to the water.

1.19 R-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 the 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.

1.20 Describe the compressibility factor and illustrate it mathematically.

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

1.22 Explain the difference between heat and work.

1.23 An non-rigid tank contains 0.8 kmol of air at 23 °C and 600 kPa. Calculate the volume of the tank. The volume is then doubled at the same pressure. Find the temperature at this state.

1.24 An non-rigid tank contains 2.5 kmol of air at 40 °C and 180 kPa. Find the volume of the tank. The volume is then doubled at the same pressure. Calculate the temperature at this state.

1.25 A 10-liter piston-cylinder device contains air at 25 °C and 180 kPa. The oxygen is heated until the temperature reaches 80 °C. Find the amount of heat transfer during this process.

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

1.27 A 30-liter rigid tank contains nitrogen at 40 °C and 250 kPa. The oxygen is heated until the temperature reaches 90 °C. Find the entropy change during this process.

1.28 A rigid tank contains 3 kg oxygen at 50 °C and 150 kPa. The oxygen is heated in an isentropic process until the temperature reaches 70 °C. Calculate (a) the final pressure and (b) the work interaction during this process.

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

1.30 Derive both energetic and exergetic COPs for a refrigerator and a heat pump and relate them to each other mathematically.

1.31 Illustrate mathematically how to increase the COP of a Carnot refrigerator?

1.32 A Carnot refrigerator is used to keep a space at 20 °C by rejecting heat to a reservoir at 35 °C. If the heat removal from the cooled space is 15,000 kJ/h, find the COP of the refrigerator and the power input in kW.

1.33 A Carnot refrigerator is used to keep a space at 3 °C by rejecting heat to a reservoir at 45 °C. If the heat removal from the cooled space is 3.8 times more than the power input, find 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, calculate the rate of heat rejected to a high-temperature reservoir.

1.35 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. Calculate (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 is used.

1.36 Explain what causes irreversibilities in a thermodynamic system.

1.37 Explain how the properties of mass, energy, entropy, and exergy behave during an irreversible process: conserved, decrease, or increase?

1.38 Explain the importance of exergy and how it helps achieve more efficient energy-resource use.

1.39 What is the difference between humidity ratio and relative humidity? Show this mathematically.

1.40 Explain why heating usually is accompanied by humidification and cooling by dehumidification?

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

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

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

1.44 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 Example 1.9, determine the exergy destruction and the exergy efficiency for this process.

1.45 Design an evaporative cooler for a house in the desert for the hottest summer day where the maximum ambient temperature is 50 °C and the maximum relative humidity is 100%, and find the cooling capacity. Make reasonable and conceptually correct assumptions for calculations.

References

1 Dincer, I., Rosen, M.A. (2013). *Exergy*, 2nd edn, Elsevier, New York.
2 ASHRAE. (2006). *ASHRAE's Sustainability Roadmap*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.
3 EES (2016). Engineering Equation Solver developed by S.A. Klein for Microsoft windows operating systems. Professional V9.902 (2015), Madison, via http://www.fchart.com.
4 Dincer, I. (2003). *Refrigeration Systems and Applications*, Wiley, Chichester.
5 Van Wylen, G., Sonntag, R., Borgnakke, C. (1998). *Fundamentals of Classical Thermodynamics*, 5th edn, Wiley, New York.
6 Cengel, Y.A., Boles, M.A. (2015). Thermodynamics: An Engineering Approach, 8th edn, McGraw-Hill, New York.