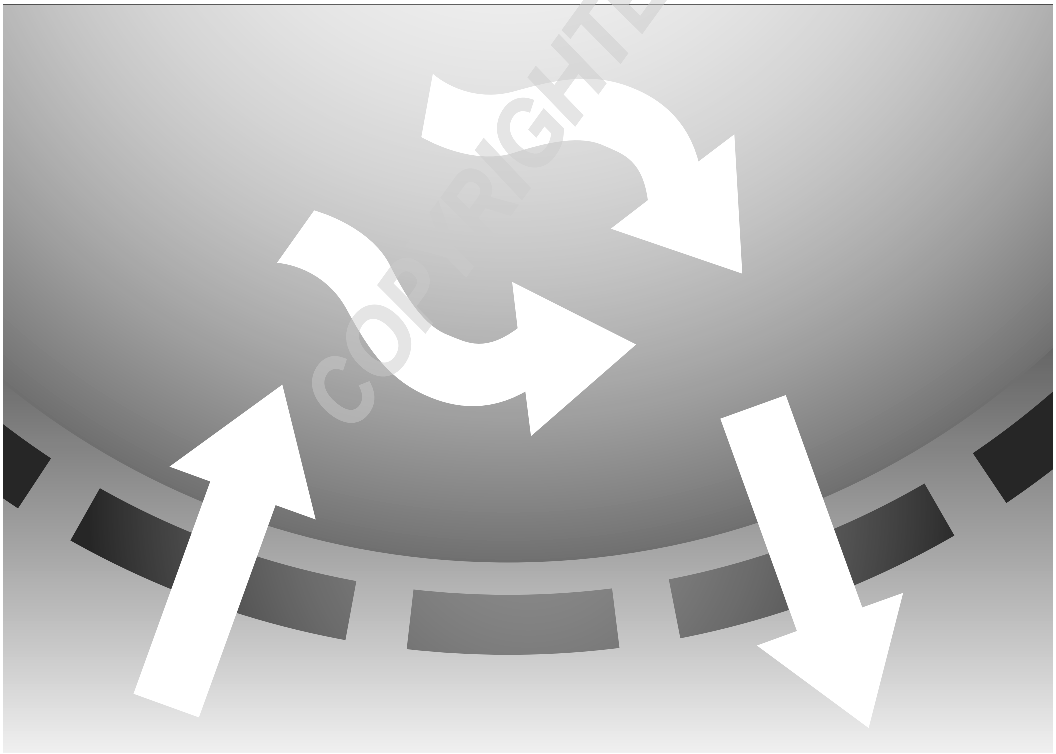
CHAPTER 1

Introduction



From the study of thermodynamics, you have learned that energy can be transferred by interactions of a system with its surroundings. These interactions are called work and heat. However, thermodynamics deals with the end states of the process during which an interaction occurs and provides no information concerning the nature of the interaction or the time rate at which it occurs. The objective of this text is to extend thermodynamic analysis through the study of the *modes* of heat transfer and through the development of relations to calculate heat transfer *rates.*

In this chapter we lay the foundation for much of the material treated in the text. We do so by raising several questions: *What is heat transfer? How is heat transferred? Why is it important?* One objective is to develop an appreciation for the fundamental concepts and principles that underlie heat transfer processes. A second objective is to illustrate the manner in which a knowledge of heat transfer may be used with the first law of thermodynamics (*conservation of energy*) to solve problems relevant to technology and society.

1.1 What and How?

A simple, yet general, definition provides sufficient response to the question: What is heat transfer?

> *Heat transfer (or heat) is thermal energy in transit due to a spatial temperature difference.*

Whenever a temperature difference exists in a medium or between media, heat transfer must occur.

As shown in Figure 1.1, we refer to different types of heat transfer processes as *modes.* When a temperature gradient exists in a stationary medium, which may be a solid or a fluid, we use the term *conduction* to refer to the heat transfer that will occur across the medium. In contrast, the term *convection* refers to heat transfer that will occur between a surface and a moving fluid when they are at different temperatures. The third mode of heat transfer is termed *thermal radiation.* All surfaces of finite temperature emit energy in the form of electromagnetic waves. Hence, in the absence of an intervening medium, there is net heat transfer by radiation between two surfaces at different temperatures.

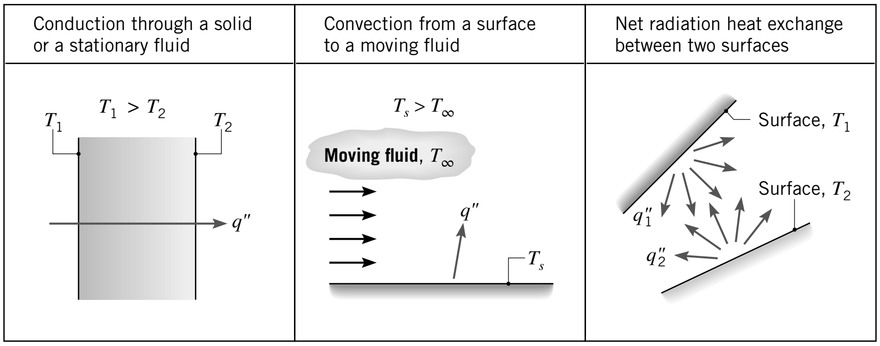


FIGURE 1.1 Conduction, convection, and radiation heat transfer modes.

1.2 *Physical Origins and Rate Equations*

As engineers, it is important that we understand the *physical mechanisms* which underlie the heat transfer modes and that we be able to use the rate equations that quantify the amount of energy being transferred per unit time.

1.2.1 Conduction

At mention of the word *conduction*, we should immediately conjure up concepts of *atomic* and *molecular activity* because processes at these levels sustain this mode of heat transfer. Conduction may be viewed as the transfer of energy from the more energetic to the less energetic particles of a substance due to interactions between the particles.

The physical mechanism of conduction is most easily explained by considering a gas and using ideas familiar from your thermodynamics background. Consider a gas in which a temperature gradient exists, and assume that there is *no bulk, or macroscopic, motion.* The gas may occupy the space between two surfaces that are maintained at different temperatures, as shown in Figure 1.2. We associate the temperature at any point with the energy of gas molecules in proximity to the point. This energy is related to the random translational motion, as well as to the internal rotational and vibrational motions, of the molecules.

Higher temperatures are associated with higher molecular energies. When neighboring molecules collide, as they are constantly doing, a transfer of energy from the more energetic to the less energetic molecules must occur. In the presence of a temperature gradient, energy transfer by conduction must then occur in the direction of decreasing temperature. This would be true even in the absence of collisions, as is evident from Figure 1.2. The hypothetical plane at x_o is constantly being crossed by molecules from above and below due to their *random* motion. However, molecules from above are associated with a higher temperature than those from below, in which case there must be a *net* transfer of energy in the positive x-direction. Collisions between molecules enhance this energy transfer. We may speak of the net transfer of energy by random molecular motion as a *diffusion* of energy.

The situation is much the same in liquids, although the molecules are more closely spaced and the molecular interactions are stronger and more frequent. Similarly, in a solid, conduction may be attributed to atomic activity in the form of lattice vibrations. The modern

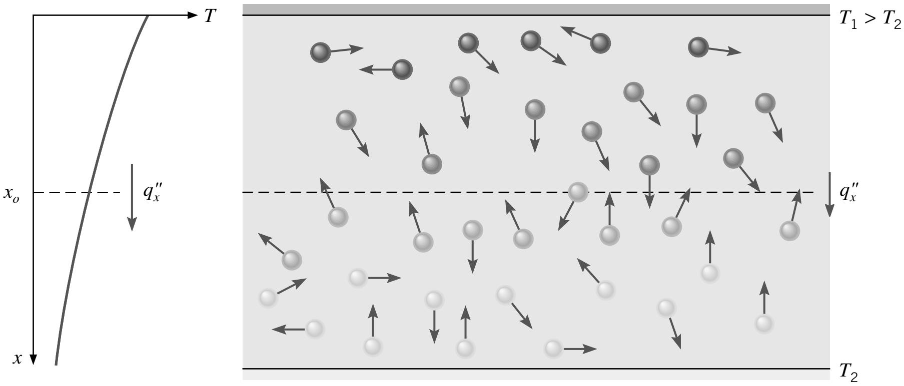


FIGURE 1.2 Association of conduction heat transfer with diffusion of energy due to molecular activity.

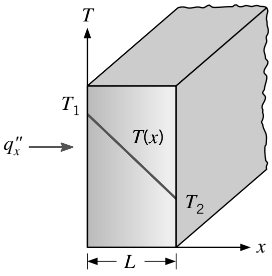


FIGURE 1.3 One-dimensional heat transfer by conduction (diffusion of energy).

view is to ascribe the energy transfer to *lattice waves* induced by atomic motion. In an electrical nonconductor, the energy transfer is exclusively via these lattice waves; in a conductor, it is also due to the translational motion of the free electrons. We treat the important properties associated with conduction phenomena in Chapter 2 and in Appendix A.

Examples of conduction heat transfer are legion. The exposed end of a metal spoon suddenly immersed in a cup of hot coffee is eventually warmed due to the conduction of energy through the spoon. On a winter day, there is significant energy loss from a heated room to the outside air. This loss is principally due to conduction heat transfer through the wall that separates the room air from the outside air.

Heat transfer processes can be quantified in terms of appropriate *rate equations.* These equations may be used to compute the amount of energy being transferred per unit time. For heat conduction, the rate equation is known as *Fourier's law.* For the one-dimensional plane wall shown in Figure 1.3, having a temperature distribution $T(x)$, the rate equation is expressed as

$$q_x'' = -k\frac{dT}{dx} \tag{1.1}$$

The *heat flux* q_x''(W/m^2) is the heat transfer rate in the x-direction *per* unit area *perpendicular* to the direction of transfer, and it is proportional to the *temperature gradient*, dT/dx, in this direction. The parameter k is a *transport* property known as the *thermal conductivity* (W/m · K) and is a characteristic of the wall material. The minus sign is a consequence of the fact that heat is transferred in the direction of decreasing temperature. Under the *steady-state conditions* shown in Figure 1.3, where the temperature distribution is *linear*, the temperature gradient may be expressed as

$$\frac{dT}{dx} = \frac{T_2 - T_1}{L}$$

and the heat flux is then

$$q_x'' = -k\frac{T_2 - T_1}{L}$$

or

$$q_x'' = k\frac{T_1 - T_2}{L} = k\frac{\Delta T}{L} \tag{1.2}$$

Note that this equation provides a *heat flux*, that is, the rate of heat transfer per *unit area.* The *heat rate* by conduction, q_x(W), through a plane wall of area A is then the product of the flux and the area, $q_x = q_x'' \cdot A$.

IHT* EXAMPLE 1.1

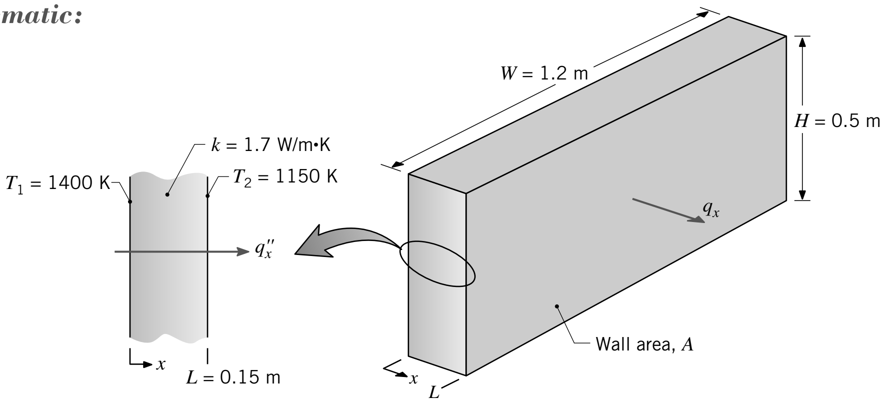
The wall of an industrial furnace is constructed from 0.15-m-thick fireclay brick having a thermal conductivity of 1.7 W/m · K. Measurements made during steady-state operation reveal temperatures of 1400 and 1150 K at the inner and outer surfaces, respectively. What is the rate of heat loss through a wall that is 0.5 m × 1.2 m on a side?

SOLUTION

Known: Steady-state conditions with prescribed wall thickness, area, thermal conductivity, and surface temperatures.

Find: Rate of heat loss through wall.

Schematic:



Assumptions:

1. Steady-state conditions.
2. One-dimensional conduction through the wall.
3. Constant thermal conductivity.

Analysis: Since heat transfer through the wall is by conduction, the heat flux may be determined from Fourier's law. Using Equation 1.2, we have

$$q_x'' = k\frac{\Delta T}{L} = 1.7\ \text{W/m}\cdot\text{K} \times \frac{250\ \text{K}}{0.15\ \text{m}} = 2833\ \text{W/m}^2$$

The heat flux represents the rate of heat transfer through a section of unit area, and it is uniform (invariant) across the surface of the wall. The rate of heat loss through the wall of area $A = H \times W$ is then

$$q_x = (HW)q_x'' = (0.5\ \text{m} \times 1.2\ \text{m})2833\ \text{W/m}^2 = 1700\ \text{W} \qquad \triangleleft$$

Comments: Note the direction of heat flow and the distinction between heat flux and heat rate.

*This icon identifies examples that are available in tutorial form in the *Interactive Heat Transfer (IHT)* software that accompanies the text. Each tutorial is brief and illustrates a basic function of the software. *IHT* can be used to solve simultaneous equations, perform parameter sensitivity studies, and graph the results. Use of *IHT* will reduce the time spent solving more complex end-of-chapter problems.

1.2.2 Convection

The convection heat transfer *mode* is comprised of *two mechanisms.* In addition to energy transfer due to *random molecular motion* (*diffusion*), energy is also transferred by the *bulk*, or *macroscopic, motion* of the fluid. This fluid motion is associated with the fact that, at any instant, large numbers of molecules are moving collectively or as aggregates. Such motion, in the presence of a temperature gradient, contributes to heat transfer. Because the molecules in the aggregate retain their random motion, the total heat transfer is then due to a superposition of energy transport by the random motion of the molecules and by the bulk motion of the fluid. The term *convection* is customarily used when referring to this cumulative transport, and the term *advection* refers to transport due to bulk fluid motion alone.

We are especially interested in convection heat transfer between a fluid in motion and a bounding surface when the two are at different temperatures. Consider fluid flow over the hot surface of Figure 1.4. A consequence of the fluid–surface interaction is the development of a region in the fluid through which the velocity varies from zero at the surface to a finite value u_∞ associated with the flow. This region of the fluid is known as the *hydrodynamic*, or *velocity, boundary layer.* Moreover, if the surface and flow temperatures differ, there will be a region of the fluid through which the temperature varies from T_s at $y = 0$ to T_∞ in the outer flow. This region, called the *thermal boundary layer*, may be smaller, larger, or the same size as that through which the velocity varies. In any case, if $T_s > T_\infty$, convection heat transfer will occur from the surface to the outer flow.

The convection heat transfer mode is sustained both by random molecular motion and by the bulk motion of the fluid within the boundary layer. The contribution due to random molecular motion (diffusion) dominates near the surface where the fluid velocity is low. In fact, at the interface between the surface and the fluid ($y = 0$), the fluid velocity is zero, and heat is transferred by this mechanism only. The contribution due to bulk fluid motion originates from the fact that the boundary layer *grows* as the flow progresses in the x-direction. In effect, the heat that is conducted into this layer is swept downstream and is eventually transferred to the fluid outside the boundary layer. Appreciation of boundary layer phenomena is essential to understanding convection heat transfer. For this reason, the discipline of fluid mechanics will play a vital role in our later analysis of convection.

Convection heat transfer may be classified according to the nature of the flow. We speak of *forced convection* when the flow is caused by external means, such as by a fan, a pump, or atmospheric winds. As an example, consider the use of a fan to provide forced convection air cooling of hot electrical components on a stack of printed circuit boards (Figure 1.5*a*). In contrast, for *free* (or *natural*) *convection*, the flow is induced by buoyancy forces, which are due to density differences caused by temperature variations in the fluid. An example is the free convection heat transfer that occurs from hot components on

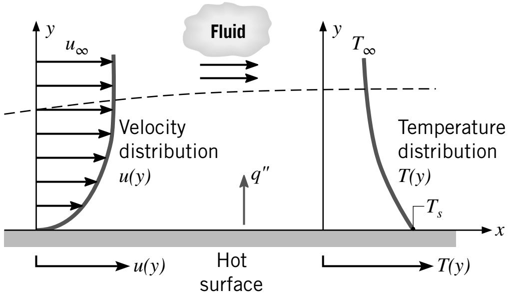


FIGURE 1.4 Boundary layer development in convection heat transfer.

a vertical array of circuit boards in air (Figure 1.5*b*). Air that makes contact with the components experiences an increase in temperature and hence a reduction in density. Since it is now lighter than the surrounding air, buoyancy forces induce a vertical motion for which warm air ascending from the boards is replaced by an inflow of cooler ambient air.

While we have presumed *pure* forced convection in Figure 1.5*a* and *pure* natural convection in Figure 1.5*b*, conditions corresponding to *mixed* (*combined*) *forced* and *natural convection* may exist. For example, if velocities associated with the flow of Figure 1.5*a* are small and/or buoyancy forces are large, a secondary flow that is comparable to the imposed forced flow could be induced. In this case, the buoyancy-induced flow would be normal to the forced flow and could have a significant effect on convection heat transfer from the components. In Figure 1.5*b*, mixed convection would result if a fan were used to force air upward between the circuit boards, thereby assisting the buoyancy flow, or downward, thereby opposing the buoyancy flow.

We have described the convection heat transfer mode as energy transfer occurring within a fluid due to the combined effects of conduction and bulk fluid motion. Typically, the energy that is being transferred is the *sensible*, or internal thermal, energy of the fluid. However, for some convection processes, there is, in addition, *latent* heat exchange. This latent heat exchange is generally associated with a phase change between the liquid and vapor states of the fluid. Two special cases of interest in this text are *boiling* and *condensation.* For example, convection heat transfer results from fluid motion induced by vapor bubbles generated at the bottom of a pan of boiling water (Figure 1.5*c*) or by the condensation of water vapor on the outer surface of a cold water pipe (Figure 1.5*d*).

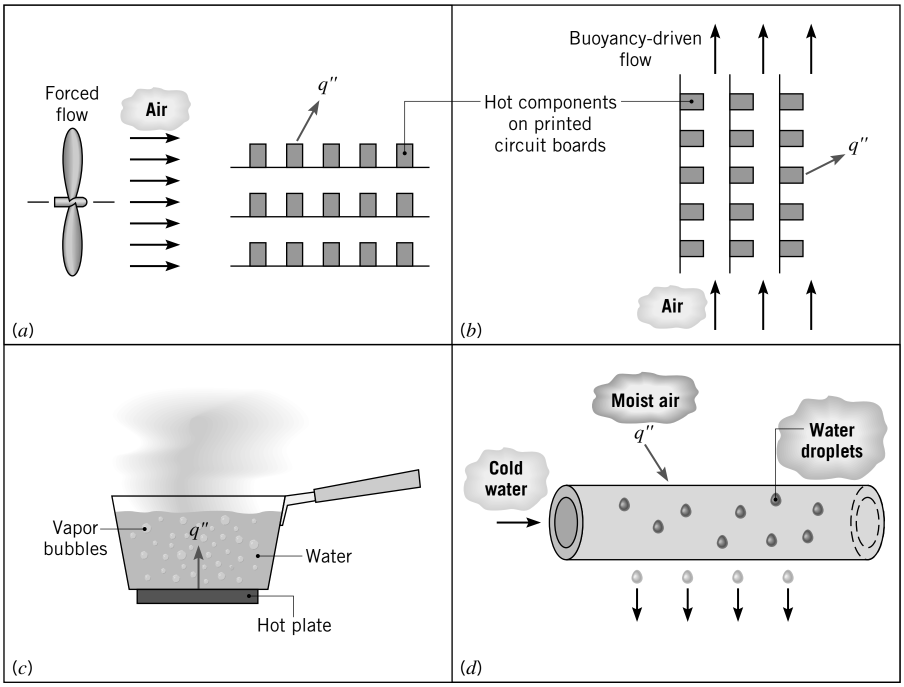


FIGURE 1.5 Convection heat transfer processes. (a) Forced convection. (b) Natural convection. (c) Boiling. (d) Condensation.

TABLE 1.1 Typical values of the convection heat transfer coefficient

Process	h ($W/m^2 \cdot K$)
Free convection	
Gases	2–25
Liquids	50–1000
Forced convection	
Gases	25–250
Liquids	100–20,000
Convection with phase change	
Boiling or condensation	2500–100,000

Regardless of the nature of the convection heat transfer process, the appropriate rate equation is of the form

$$q'' = h(T_s - T_\infty) \tag{1.3a}$$

where q'', the convective *heat flux* (W/m^2), is proportional to the difference between the surface and fluid temperatures, T_s and T_∞, respectively. This expression is known as *Newton's law of cooling*, and the parameter h ($W/m^2 \cdot K$) is termed the *convection heat transfer coefficient.* This coefficient depends on conditions in the boundary layer, which are influenced by surface geometry, the nature of the fluid motion, and an assortment of fluid thermodynamic and transport properties.

Any study of convection ultimately reduces to a study of the means by which h may be determined. Although consideration of these means is deferred to Chapter 6, convection heat transfer will frequently appear as a boundary condition in the solution of conduction problems (Chapters 2 through 5). In the solution of such problems we presume h to be known, using typical values given in Table 1.1.

When Equation 1.3a is used, the convection heat flux is presumed to be *positive* if heat is transferred *from* the surface ($T_s > T_\infty$) and *negative* if heat is transferred *to* the surface ($T_\infty > T_s$). However, nothing precludes us from expressing Newton's law of cooling as

$$q'' = h(T_\infty - T_s) \tag{1.3b}$$

in which case heat transfer is positive if it is to the surface.

1.2.3 Radiation

Thermal radiation is energy *emitted* by matter that is at a nonzero temperature. Although we will focus on radiation from solid surfaces, emission may also occur from liquids and gases. Regardless of the form of matter, the emission may be attributed to changes in the electron configurations of the constituent atoms or molecules. The energy of the radiation field is transported by electromagnetic waves (or alternatively, photons). While the transfer of energy by conduction or convection requires the presence of a material medium, radiation does not. In fact, radiation transfer occurs most efficiently in a vacuum.

Consider radiation transfer processes for the surface of Figure 1.6*a*. Radiation that is *emitted* by the surface originates from the thermal energy of matter bounded by the surface, and the rate at which energy is released per unit area (W/m^2) is termed the surface *emissive power, E.* There is an upper limit to the emissive power, which is prescribed by the *Stefan–Boltzmann law*

$$E_b = \sigma T_s^4 \tag{1.4}$$

where T_s is the *absolute temperature* (K) of the surface and σ is the *Stefan–Boltzmann constant* ($\sigma = 5.67 \times 10^{-8}\ W/m^2 \cdot K^4$). Such a surface is called an ideal radiator or *blackbody.*

The heat flux emitted by a real surface is less than that of a blackbody at the same temperature and is given by

$$E = \varepsilon \sigma T_s^4 \tag{1.5}$$

where ε is a radiative property of the surface termed the *emissivity.* With values in the range $0 \le \varepsilon \le 1$, this property provides a measure of how efficiently a surface emits energy relative to a blackbody. It depends strongly on the surface material and finish, and representative values are provided in Appendix A.

Radiation may also be *incident* on a surface from its surroundings. The radiation may originate from a special source, such as the sun, or from other surfaces to which the surface of interest is exposed. Irrespective of the source(s), we designate the rate at which all such radiation is incident on a unit area of the surface as the *irradiation G* (Figure 1.6*a*).

A portion, or all, of the irradiation may be *absorbed* by the surface, thereby increasing the thermal energy of the material. The rate at which radiant energy is absorbed per unit surface area may be evaluated from knowledge of a surface radiative property termed the *absorptivity* α. That is,

$$G_{\text{abs}} = \alpha G \tag{1.6}$$

where $0 \le \alpha \le 1$. If $\alpha < 1$ and the surface is *opaque*, portions of the irradiation are *reflected.* If the surface is *semitransparent*, portions of the irradiation may also be *transmitted.* However, whereas absorbed and emitted radiation increase and reduce, respectively, the thermal energy of matter, reflected and transmitted radiation have no effect on this

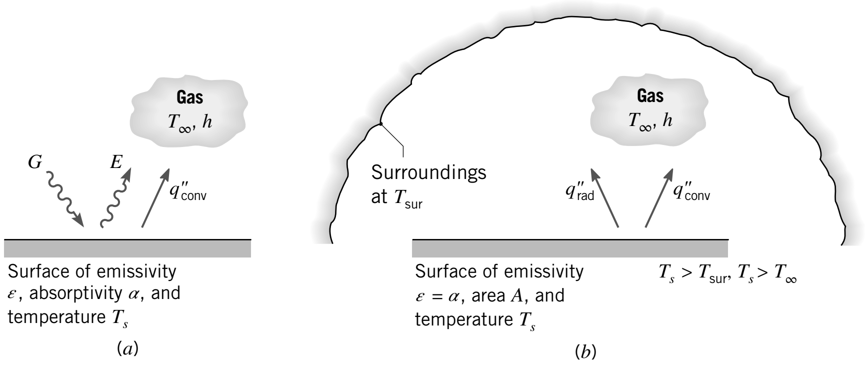


Figure 1.6 Radiation exchange: (a) at a surface and (b) between a surface and large surroundings.

energy. Note that the value of α depends on the nature of the irradiation, as well as on the surface itself. For example, the absorptivity of a surface to solar radiation may differ from its absorptivity to radiation emitted by the walls of a furnace.

In many engineering problems (a notable exception being problems involving solar radiation or radiation from other very high temperature sources), liquids can be considered opaque to radiation heat transfer, and gases can be considered transparent to it. Solids can be opaque (as is the case for metals) or *semitransparent* (as is the case for thin sheets of some polymers and some semiconducting materials).

A special case that occurs frequently involves radiation exchange between a small surface at T_s and a much larger, isothermal surface that completely surrounds the smaller one (Figure 1.6*b*). The *surroundings* could, for example, be the walls of a room or a furnace whose temperature T_{sur} differs from that of an enclosed surface ($T_{\text{sur}} \neq T_s$). We will show in Chapter 12 that, for such a condition, the irradiation may be approximated by emission from a blackbody at T_{sur}, in which case $G = \sigma T_{\text{sur}}^4$. If the surface is assumed to be one for which $\alpha = \varepsilon$ (a *gray surface*), the *net* rate of radiation heat transfer *from* the surface, expressed per unit area of the surface, is

$$q''_{\text{rad}} = \frac{q}{A} = \varepsilon E_b(T_s) - \alpha G = \varepsilon\sigma(T_s^4 - T_{\text{sur}}^4) \tag{1.7}$$

This expression provides the difference between thermal energy that is released due to radiation emission and that gained due to radiation absorption.

For many applications, it is convenient to express the net radiation heat exchange in the form

$$q_{\text{rad}} = h_r A(T_s - T_{\text{sur}}) \tag{1.8}$$

where, from Equation 1.7, the *radiation heat transfer coefficient* h_r is

$$h_r \equiv \varepsilon\sigma(T_s + T_{\text{sur}})(T_s^2 + T_{\text{sur}}^2) \tag{1.9}$$

Here we have modeled the radiation mode in a manner similar to convection. In this sense we have *linearized* the radiation rate equation, making the heat rate proportional to a temperature difference rather than to the difference between two temperatures to the fourth power. Note, however, that h_r depends strongly on temperature, whereas the temperature dependence of the convection heat transfer coefficient h is generally weak.

The surfaces of Figure 1.6 may also simultaneously transfer heat by convection to an adjoining gas. For the conditions of Figure 1.6*b*, the total rate of heat transfer *from* the surface is then

$$q = q_{\text{conv}} + q_{\text{rad}} = hA(T_s - T_\infty) + \varepsilon A\sigma(T_s^4 - T_{\text{sur}}^4) \tag{1.10}$$

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EXAMPLE 1.2

An uninsulated steam pipe passes through a room in which the air and walls are at 25°C. The outside diameter of the pipe is 70 mm, and its surface temperature and emissivity are 200°C and 0.8, respectively. What are the surface emissive power and irradiation? If the coefficient associated with free convection heat transfer from the surface to the air is 15 W/m$^2 \cdot$ K, what is the rate of heat loss from the surface per unit length of pipe?

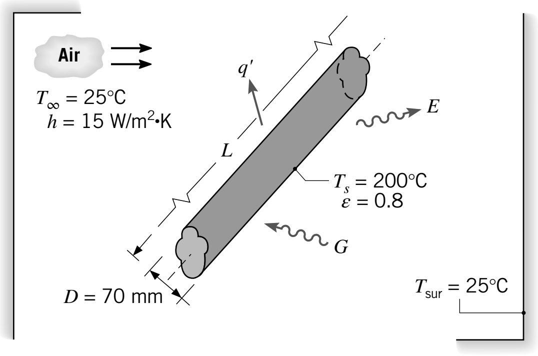
SOLUTION

Known: Uninsulated pipe of prescribed diameter, emissivity, and surface temperature in a room with fixed wall and air temperatures.

Find:

1. Surface emissive power and irradiation.
2. Rate of heat loss from pipe per unit length, q'.

Schematic:



Assumptions:

1. Steady-state conditions.
2. Radiation exchange between the pipe and the room is between a small surface and a much larger enclosure.
3. The surface emissivity and absorptivity are equal.

Analysis:

1. The surface emissive power may be evaluated from Equation 1.5, while the irradiation corresponds to $G = \sigma T_{sur}^4$. Hence

$$E = \varepsilon \sigma T_s^4 = 0.8(5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4)(473\ \text{K})^4 = 2270\ \text{W/m}^2 \qquad \triangleleft$$

$$G = \sigma T_{sur}^4 = 5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4 (298\ \text{K})^4 = 447\ \text{W/m}^2 \qquad \triangleleft$$

2. Heat loss from the pipe is by convection to the room air and by radiation exchange with the walls. Hence, $q = q_{conv} + q_{rad}$ and from Equation 1.10, with $A = \pi DL$,

$$q = h(\pi DL)(T_s - T_\infty) + \varepsilon(\pi DL)\sigma(T_s^4 - T_{sur}^4)$$

The rate of heat loss per unit length of pipe is then

$$q' = \frac{q}{L} = 15\ \text{W/m}^2 \cdot \text{K}(\pi \times 0.07\,\text{m})(200 - 25)°\text{C}$$
$$+\, 0.8(\pi \times 0.07\ \text{m})\ 5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4 (473^4 - 298^4)\text{K}^4$$

$$q' = 577\ \text{W/m} + 421\ \text{W/m} = 998\ \text{W/m} \qquad \triangleleft$$

Comments:

1. Note that temperature may be expressed in units of °C or K when evaluating the temperature difference for a convection (or conduction) heat transfer rate. However, temperature must be expressed in kelvins (K) when evaluating a radiation transfer rate.
2. The net rate of radiation heat transfer from the pipe may be expressed as

$$q'_{\text{rad}} = \pi D(E - \alpha G)$$

$$q'_{\text{rad}} = \pi \times 0.07 \text{ m } (2270 - 0.8 \times 447) \text{ W/m}^2 = 421 \text{ W/m}$$

3. In this situation, the radiation and convection heat transfer rates are comparable because T_s is large compared to T_{sur} and the coefficient associated with free convection is small. For more moderate values of T_s and the larger values of h associated with forced convection, the effect of radiation may often be neglected. The radiation heat transfer coefficient may be computed from Equation 1.9. For the conditions of this problem, its value is $h_r = 11 \text{ W/m}^2 \cdot \text{K}$.

1.2.4 The Thermal Resistance Concept

The three modes of heat transfer were introduced in the preceding sections. As is evident from Equations 1.2, 1.3, and 1.8, the heat transfer rate can be expressed in the form

$$q = q''A = \frac{\Delta T}{R_t} \tag{1.11}$$

where ΔT is a relevant temperature difference and A is the area normal to the direction of heat transfer. The quantity R_t is called a *thermal resistance* and takes different forms for the three different modes of heat transfer. For example, Equation 1.2 may be multiplied by the area A and rewritten as $q_x = \Delta T/R_{t,\text{cond}}$, where $R_{t,\text{cond}} = L/kA$ is a thermal resistance associated with conduction, having the units K/W. The thermal resistance concept will be considered in detail in Chapter 3 and will be seen to have great utility in solving complex heat transfer problems.

1.3 *Relationship to Thermodynamics*

If you have taken a thermodynamics course, you are aware that heat exchange plays a vital role in the first and second laws of thermodynamics because it is a primary mechanism for energy transfer between a system and its surroundings. While thermodynamics may be used to determine the *amount* of energy required in the form of heat for a system to pass from one state to another, it considers neither the mechanisms that provide for heat exchange nor the methods that exist for computing the *rate* of heat exchange. The discipline of heat transfer specifically seeks to quantify the rate at which heat is exchanged through the rate equations expressed, for example, by Equations 1.2, 1.3, and 1.7. Indeed, heat transfer principles enable the engineer to implement the concepts of thermodynamics. For example, the size of a power plant to be constructed cannot be determined from thermodynamics alone; the principles of heat transfer must also be invoked at the design stage.

This section considers the relationship of heat transfer to thermodynamics. Since the *first law* of thermodynamics (the *law of conservation of energy*) provides a useful, often essential, starting point for the solution of heat transfer problems, Section 1.3.1 will provide a development of the general formulations of the first law. The ideal (Carnot) efficiency of a *heat engine*, as determined by the *second law* of thermodynamics, will be reviewed in Section 1.3.2. It will be shown that a realistic description of the heat transfer between a heat engine and its surroundings limits the actual efficiency of a heat engine.

1.3.1 Relationship to the First Law of Thermodynamics (Conservation of Energy)

The first law of thermodynamics states that the total energy of a system is conserved, where *total energy* consists of mechanical energy (which is composed of kinetic and potential energy) and internal energy, as shown schematically in Figure 1.7. Internal energy can be subdivided into thermal energy (which will be defined more carefully later) and other forms of internal energy, such as chemical and nuclear energy. Since total energy is conserved, the only way that the amount of energy in a system can change is if energy crosses its boundaries. For a *closed system* (a region of fixed mass), there are only two ways energy can cross the system boundaries: heat transfer through the boundaries and work done on or by the system. This leads to the following statement of the first law for a closed system, which is familiar if you have taken a course in thermodynamics:

$$\Delta E_{\mathrm{st}}^{\mathrm{tot}} = Q - W \tag{1.12a}$$

where $\Delta E_{\mathrm{st}}^{\mathrm{tot}}$ is the change in the total energy stored in the system, Q is the *net* heat transferred to the system, and W is the *net* work done by the system. This is schematically illustrated in Figure 1.8*a*.

The first law can also be applied to a *control volume* (or *open system*), a region of space bounded by a *control surface* through which mass may flow. Mass entering and leaving the control volume carries energy with it; this process, termed *energy advection*, adds a third way in which energy can cross the boundaries of a control volume. To summarize, the

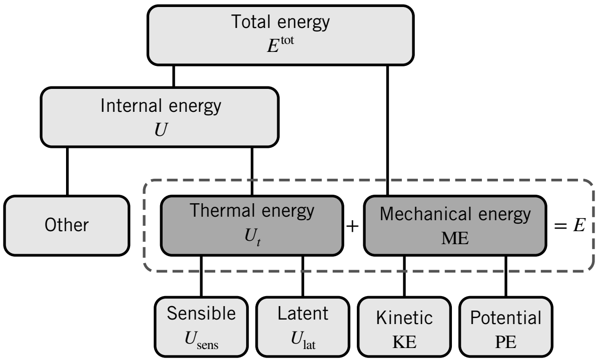


FIGURE 1.7 The components of total energy. The sum of thermal and mechanical energy, E, is of interest in the field of heat transfer.

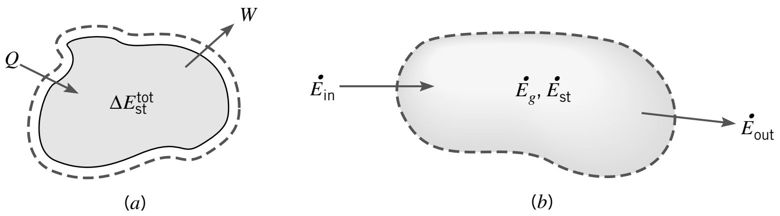


Figure 1.8 Conservation of: (a) total energy for a closed system over a time interval and (b) thermal and mechanical energy for a control volume at an instant.

first law of thermodynamics can be simply stated as follows for both a control volume and a closed system:

1. Conservation of Total Energy: First Law of Thermodynamics over a Time Interval (Δt)

The increase in the amount of energy stored in a control volume must equal the amount of energy that enters the control volume, minus the amount of energy that leaves the control volume.

As engineers, we often focus our attention on the thermal and mechanical forms of energy. We must recognize that the sum of thermal and mechanical energy is *not* conserved, because conversion can occur between other forms of energy and thermal or mechanical energy. For example, during combustion the amount of chemical energy in the system will decrease and the amount of thermal energy in the system will increase. If an electric motor operates within the system, it will cause conversion from electrical to mechanical energy. We can think of such energy conversions as resulting in *thermal or mechanical energy generation* (which can be either positive or negative). So a statement of the first law that is well suited for heat transfer analysis is:

2. Conservation of Thermal and Mechanical Energy over a Time Interval (Δt)

The increase in the amount of thermal and mechanical energy stored in a control volume must equal the amount of thermal and mechanical energy that enters the control volume, minus the amount of thermal and mechanical energy that leaves the control volume, plus the amount of thermal and mechanical energy that is generated within the control volume.

This expression applies over a time interval Δt, and all the energy terms are measured in joules. Since the first law must be satisfied at every *instant* of time t, we can also formulate the law on a *rate basis*. That is, at any instant, there must be a balance between all *energy rates*, as measured in joules per second (W). In words, this is expressed as:

3. Conservation of Thermal and Mechanical Energy at an Instant (t)

The rate of increase of thermal and mechanical energy stored in a control volume must equal the rate at which thermal and mechanical energy enters the control volume, minus the rate at which thermal and mechanical energy leaves the control volume, plus the rate at which thermal and mechanical energy is generated within the control volume.

If the inflow and generation of thermal and mechanical energy exceed the outflow, the amount of thermal and mechanical energy stored (accumulated) in the control volume must increase. If the outflow of thermal and mechanical energy exceeds the generation and inflow, the amount of stored thermal and mechanical energy in the control volume must decrease. If the inflow and generation equal the outflow, a *steady-state* condition must prevail such that there will be no change in the amount of thermal and mechanical energy stored in the control volume.

We will now rewrite the two boxed conservation of thermal and mechanical energy statements as equations. To do so, we let E stand for the sum of thermal and mechanical energy (in contrast to the symbol E^{tot} for total energy and as shown in Figure 1.7). The change in thermal and mechanical energy stored over the time interval Δt is then ΔE_{st}. The subscripts *in* and *out* refer to energy entering and leaving the control volume. Finally, thermal and mechanical energy generation is given the symbol E_g. Thus, boxed conservation statement 2 can be written as:

$$\Delta E_{\text{st}} = E_{\text{in}} - E_{\text{out}} + E_g \tag{1.12b}$$

Next, using a dot over a term to indicate a rate, boxed conservation statement 3 becomes:

$$\dot{E}_{\text{st}} \equiv \frac{dE_{\text{st}}}{dt} = \dot{E}_{\text{in}} - \dot{E}_{\text{out}} + \dot{E}_g \tag{1.12c}$$

This expression is illustrated schematically in Figure 1.8*b*.

Every application of the first law must begin with the identification of an appropriate control volume and its control surface, to which an analysis is subsequently applied. The first step is to indicate the control surface by drawing a dashed line. The second step is to decide whether to perform the analysis for a time interval Δt (Equation 1.12b) or on a rate basis (Equation 1.12c). This choice depends on the objective of the solution and on how information is given in the problem. The next step is to identify the energy terms that are relevant in the problem you are solving. To develop your confidence in taking this last step, the remainder of this section is devoted to clarifying the terms in the thermal and mechanical energy equations, Equations 1.12b and 1.12c.

Thermal and Mechanical Energy, E As noted earlier, *mechanical energy* is the sum of kinetic energy (KE = $\frac{1}{2}mV^2$, where m and V are mass and velocity, respectively) and potential energy (PE = mgz, where g is the gravitational acceleration and z is the vertical coordinate).

As shown in Figure 1.7, *thermal energy* consists of a *sensible component* U_{sens}, which accounts for translational, rotational, and/or vibrational motion of the atoms/molecules comprising the matter and a *latent component* U_{lat}, which relates to intermolecular forces influencing phase change between solid, liquid, and vapor states. The sensible energy is the portion associated mainly with changes in temperature (although it can also depend on pressure). The latent energy is the component associated with changes in phase. For example, if the material in the control volume increases in temperature, its sensible energy increases. If the material in the control volume changes from solid to liquid (*melting*) or from liquid to vapor (*vaporization, evaporation, boiling*), the latent energy increases. Conversely, if the phase change is from vapor to liquid (*condensation*) or from liquid to solid (*solidification, freezing*), the latent energy decreases.

Based on this discussion, the *change in stored thermal and mechanical energy* is given by $\Delta E_{\text{st}} = \Delta(\text{KE} + \text{PE} + U_t)$, where $U_t = U_{\text{sens}} + U_{\text{lat}}$. In many heat transfer problems, the changes in kinetic and potential energy are small relative to changes in the thermal energy, and can be neglected. In addition, if there is no phase change, then the only relevant term will be the change in sensible energy, that is, $\Delta E_{\text{st}} = \Delta U_{\text{sens}}$. For either an ideal gas or an *incompressible* substance (one whose density can be treated as constant), the change in sensible energy can be expressed as $\Delta U_{\text{sens}} = mc_v \Delta T$ or in terms of rates, $dU_{\text{sens}}/dt = mc_v dT/dt$.

Energy Generation, E_g To understand energy generation, it is helpful to remember that thermal energy is only a portion of the *internal energy*, the energy associated with the thermodynamic state of matter. In addition to the sensible and latent components, internal energy includes other components such as a *chemical component*, which accounts for energy stored in the chemical bonds between atoms; a *nuclear component*, which accounts for the binding forces in the nucleus; and components such as electrical or magnetic energy. *Energy generation*, E_g, is associated with conversion from some other form of internal energy (chemical, nuclear, electrical, or magnetic) to thermal or mechanical energy. It is a *volumetric phenomenon*. That is, it occurs within the control volume and is generally proportional to the size of this volume. We have already noted that energy generation occurs when chemical energy is converted to sensible energy by combustion. Another example is the conversion from electrical energy that occurs due to resistance heating when an electric current is passed through a conductor. That is, if an electric current I passes through a resistance R in the control volume, electrical energy is dissipated at a rate I^2R, which corresponds to the rate at which thermal energy is generated (released) within the volume.

Inflow and Outflow, E_{in} and E_{out} Energy inflow and outflow are surface phenomena, that is, they are associated exclusively with processes occurring at the control surface. As discussed previously, the energy inflow and outflow terms include heat transfer and work interactions occurring at the system boundaries (e.g., due to displacement of a boundary, a rotating shaft, and/or electromagnetic effects). For cases in which mass crosses the control volume boundary (e.g., for situations involving fluid flow), the inflow and outflow terms also include energy (thermal and mechanical) that is advected (carried) by mass entering and leaving the control volume. For instance, if the mass flow rate entering through the boundary is $\dot{m}$, then the rate at which thermal and mechanical energy enters with the flow is $\dot{m}(u_t + \frac{1}{2}V^2 + gz)$, where u_t is the thermal energy per unit mass. The *mass flow rate* may be expressed as $\dot{m} = \rho V A_c$ where ρ is the fluid density and A_c is the cross-sectional area of the channel through which the fluid flows. The *volumetric flow rate* is simply $\dot{\forall} = VA_c = \dot{m}/\rho$.

When the first law is applied to a control volume with fluid crossing its boundary, it is customary to divide the work term into two contributions. The first contribution, termed *flow work*, is associated with the work done by pressure forces moving fluid through the boundary. For a *unit mass*, the amount of work is the product of the pressure and the specific volume of the fluid (pv). The symbol $\dot{W}$ is traditionally used for the rate at which the remaining work (not including flow work) is performed. If operation is under steady-state conditions ($dE_{\text{st}}/dt = 0$) and there is no thermal or mechanical energy generation, Equation 1.12c reduces to the following form of the steady-flow energy equation (see Figure 1.9), which will be familiar if you have taken a thermodynamics course:

$$\dot{m}(u_t + pv + \tfrac{1}{2}V^2 + gz)_{\text{in}} - \dot{m}(u_t + pv + \tfrac{1}{2}V^2 + gz)_{\text{out}} + q - \dot{W} = 0 \qquad (1.12\text{d})$$

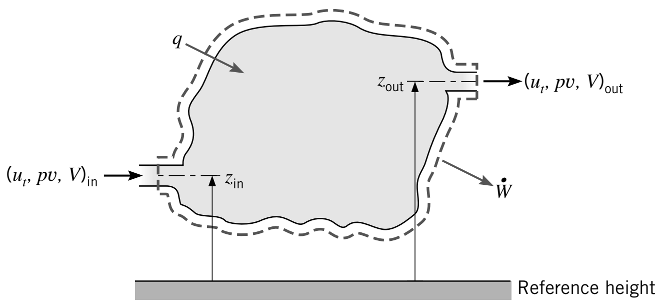


FIGURE 1.9 Conservation of energy for a steady-flow, open system.

Terms within the parentheses are expressed for a unit mass of fluid at the inflow and outflow locations. When multiplied by the mass flow rate $\dot{m}$, they yield the rate at which the corresponding form of energy (thermal, flow work, kinetic, and potential) enters or leaves the control volume. The sum of thermal energy and flow work per unit mass may be replaced by the enthalpy per unit mass, $i = u_t + pv$.

In most open system applications of interest in this text, changes in latent energy between the inflow and outflow conditions of Equation 1.12d may be neglected, so the thermal energy reduces to only the sensible component. If the fluid is approximated as an *ideal gas* with *constant specific heats*, the difference in enthalpies (per unit mass) between the inlet and outlet flows may then be expressed as $(i_{\text{in}} - i_{\text{out}}) = c_p(T_{\text{in}} - T_{\text{out}})$, where c_p is the specific heat at constant pressure and T_{in} and T_{out} are the inlet and outlet temperatures, respectively. If the fluid is an *incompressible liquid*, its specific heats at constant pressure and volume are equal, $c_p = c_v \equiv c$, and for Equation 1.12d the change in sensible energy (per unit mass) reduces to $(u_{t,\text{in}} - u_{t,\text{out}}) = c(T_{\text{in}} - T_{\text{out}})$. Unless the pressure drop is extremely large, the difference in flow work terms, $(pv)_{\text{in}} - (pv)_{\text{out}}$, is negligible for a liquid. Thus for *either* ideal gases *or* incompressible liquids with no latent energy change, it is often appropriate to use the approximation $i_{\text{in}} - i_{\text{out}} = c_p(T_{\text{in}} - T_{\text{out}})$.

Simplified Steady-Flow Thermal Energy Equation Having already assumed steady-state conditions and no thermal or mechanical energy generation, we also adopt the approximation $i_{\text{in}} - i_{\text{out}} = c_p(T_{\text{in}} - T_{\text{out}})$, as discussed in the preceding paragraph. Further assuming negligible changes in kinetic and potential energy and negligible work, Equation 1.12d reduces to the *simplified steady-flow thermal energy equation*:

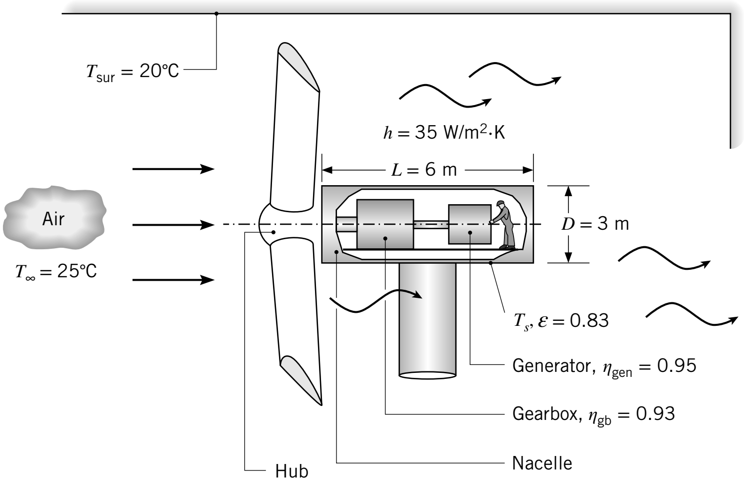
$$q = \dot{m}c_p(T_{\text{out}} - T_{\text{in}}) \tag{1.12e}$$

The right-hand side of Equation 1.12e represents the net rate of outflow of enthalpy (thermal energy plus flow work) for an ideal gas or of thermal energy for an incompressible liquid.

Since many engineering applications satisfy the assumptions of Equation 1.12e, it is commonly used for the analysis of heat transfer in moving fluids. It will be used in this text in the study of convection heat transfer in internal flow.

EXAMPLE 1.3

The blades of a wind turbine turn a large shaft at a relatively slow speed. The rotational speed is increased by a gearbox that has an efficiency of $\eta_{gb} = 0.93$. In turn, the gearbox output shaft drives an electric generator with an efficiency of $\eta_{gen} = 0.95$. The cylindrical *nacelle*, which houses the gearbox, generator, and associated equipment, is of length $L = 6$ m and diameter $D = 3$ m. If the turbine produces $P = 2.5$ MW of electrical power, and the air and surroundings temperatures are $T_\infty = 25°C$ and $T_{sur} = 20°C$, respectively, determine the exterior nacelle temperature. The emissivity of the nacelle is $\varepsilon = 0.83$, and the convective heat transfer coefficient is $h = 35\ \text{W/m}^2 \cdot \text{K}$. The surface of the nacelle that is adjacent to the blade hub can be considered to be adiabatic, and solar irradiation may be neglected.

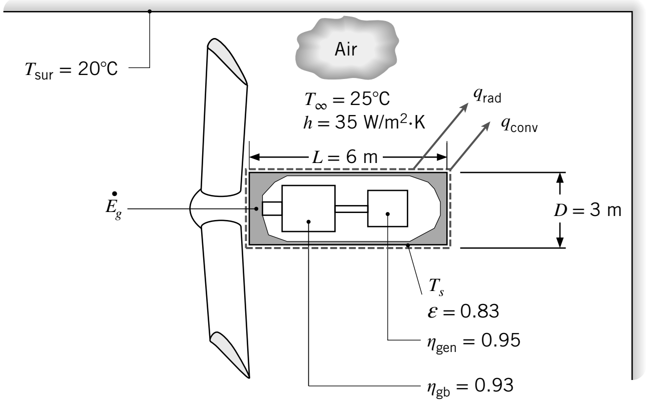


SOLUTION

Known: Electrical power produced by a wind turbine. Gearbox and generator efficiencies, dimensions and emissivity of the nacelle, ambient and surrounding temperatures, and heat transfer coefficient.

Find: Exterior nacelle temperature.

Schematic:



Assumptions:

1. Steady-state conditions.
2. Large surroundings, $\alpha = \varepsilon$.
3. Surface of the nacelle that is adjacent to the hub is adiabatic.

Analysis: The first step is to perform an energy balance on the nacelle to determine the rate of heat transfer from the nacelle to the air and surroundings under steady-state conditions. This step can be accomplished using either conservation of *total* energy or conservation of *thermal and mechanical* energy; we will compare these two approaches.

Conservation of Total Energy The first of the three boxed statements of the first law in Section 1.3 can be converted to a rate basis and expressed in equation form as follows:

$$\frac{dE_{\text{st}}^{\text{tot}}}{dt} = \dot{E}_{\text{in}}^{\text{tot}} - \dot{E}_{\text{out}}^{\text{tot}} \tag{1}$$

Under steady-state conditions, this reduces to $\dot{E}_{\text{in}}^{\text{tot}} - \dot{E}_{\text{out}}^{\text{tot}} = 0$. The $\dot{E}_{\text{in}}^{\text{tot}}$ term corresponds to the mechanical work entering the nacelle $\dot{W}$, and the $\dot{E}_{\text{out}}^{\text{tot}}$ term includes the electrical power output P and the rate of heat transfer leaving the nacelle q. Thus

$$\dot{W} - P - q = 0 \tag{2}$$

Conservation of Thermal and Mechanical Energy Alternatively, we can express conservation of thermal and mechanical energy, starting with Equation 1.12c. Under steady-state conditions, this reduces to

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} + \dot{E}_g = 0 \tag{3}$$

Here, $\dot{E}_{\text{in}}$ once again corresponds to the mechanical work $\dot{W}$. However, $\dot{E}_{\text{out}}$ now includes *only* the rate of heat transfer leaving the nacelle q. It does *not* include the electrical power, since E represents only the thermal and mechanical forms of energy. The electrical power appears in the generation term, because mechanical energy is converted to electrical energy in the generator, giving rise to a negative source of mechanical energy. That is, $\dot{E}_g = -P$. Thus, Equation (3) becomes

$$\dot{W} - q - P = 0 \tag{4}$$

which is equivalent to Equation (2), as it must be. Regardless of the manner in which the first law of thermodynamics is applied, the following expression for the rate of heat transfer evolves:

$$q = \dot{W} - P \tag{5}$$

The mechanical work and electrical power are related by the efficiencies of the gearbox and generator,

$$P = \dot{W}\eta_{\text{gb}}\eta_{\text{gen}} \tag{6}$$

Equation (5) can therefore be written as

$$q = P\left(\frac{1}{\eta_{\text{gb}}\eta_{\text{gen}}} - 1\right) = 2.5 \times 10^6 \text{ W} \times \left(\frac{1}{0.93 \times 0.95} - 1\right) = 0.33 \times 10^6 \text{ W} \tag{7}$$

Application of the Rate Equations Heat transfer is due to convection and radiation from the exterior surface of the nacelle, governed by Equations 1.3a and 1.7, respectively. Thus

$$q = q_{\text{rad}} + q_{\text{conv}} = A[q''_{\text{rad}} + q''_{\text{conv}}]$$

$$= \left[\pi DL + \frac{\pi D^2}{4}\right][\varepsilon\sigma(T_s^4 - T_{\text{sur}}^4) + h(T_s - T_\infty)] = 0.33 \times 10^6 \text{ W}$$

or

$$\left[\pi \times 3 \text{ m} \times 6 \text{ m} + \frac{\pi \times (3 \text{ m})^2}{4}\right]$$

$$\times [0.83 \times 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 (T_s^4 - (273 + 20)^4)\text{K}^4$$

$$+ 35 \text{ W/m}^2 \cdot \text{K}(T_s - (273 + 25)\text{K})] = 0.33 \times 10^6 \text{ W}$$

The preceding equation does not have a closed-form solution, but the nacelle surface temperature can be easily determined by trial and error or by using a software package such as the *Interactive Heat Transfer* (*IHT*) software accompanying your text. Doing so yields

$$T_s = 416 \text{ K} = 143°\text{C}$$

Comments:

1. The temperature inside the nacelle must be greater than the exterior surface temperature of the nacelle T_s, because the heat generated within the nacelle must be transferred from the interior of the nacelle to its surface, and from the surface to the air and surroundings.
2. The high temperature would preclude, for example, performance of routine maintenance by a worker, as illustrated in the problem statement. Thermal management approaches involving fans or blowers must be employed to reduce the temperature to an acceptable level.
3. Improvements in the efficiencies of either the gearbox or the generator would provide more electrical power and reduce the size and cost of the thermal management hardware. As such, improved efficiencies would increase revenue generated by the wind turbine and decrease both its capital and operating costs.
4. The heat transfer coefficient would not be a steady value but would vary periodically as the blades sweep past the nacelle. Therefore, the value of the heat transfer coefficient represents a *time-averaged* quantity.

IHT

EXAMPLE 1.4

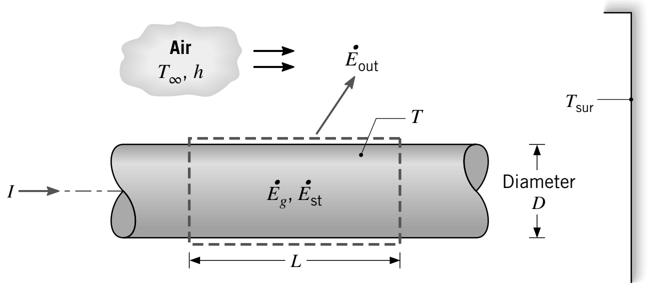
A long conducting rod of diameter D and electrical resistance per unit length R'_e is initially in thermal equilibrium with the ambient air and its surroundings. This equilibrium is disturbed when an electrical current I is passed through the rod. Develop an equation that could be used to compute the variation of the rod temperature with time during the passage of the current.

SOLUTION

Known: Temperature of a rod of prescribed diameter and electrical resistance changes with time due to passage of an electrical current.

Find: Equation that governs temperature change with time for the rod.

Schematic:



Assumptions:

1. At any time t, the temperature of the rod is uniform.
2. Constant properties (ρ, c_v, $\varepsilon = \alpha$).
3. Radiation exchange between the outer surface of the rod and the surroundings is between a small surface and a large enclosure.

Analysis: In this problem, there is no mechanical energy component. So relevant energy terms include heat transfer by convection and radiation from the surface, thermal energy generation due to ohmic heating within the conductor, and a change in thermal energy storage. Since we wish to determine the rate of change of the temperature, the first law should be applied at an instant of time. Hence, applying Equation 1.12c to a control volume of length L about the rod, it follows that

$$\dot{E}_g - \dot{E}_{\text{out}} = \dot{E}_{\text{st}}$$

where thermal energy generation is due to the electric resistance heating,

$$\dot{E}_g = I^2 R'_e L$$

Heating occurs uniformly within the control volume and could also be expressed in terms of a volumetric heat generation rate $\dot{q}$(W/m^3). The generation rate for the entire control volume is then $\dot{E}_g = \dot{q}V$, where $\dot{q} = I^2 R'_e/(\pi D^2/4)$. Energy outflow is due to convection and net radiation from the surface, Equations 1.3a and 1.7, respectively,

$$\dot{E}_{\text{out}} = h(\pi DL)(T - T_\infty) + \varepsilon\sigma(\pi DL)(T^4 - T_{\text{sur}}^4)$$

and the change in energy storage is due only to sensible energy change,

$$\dot{E}_{\text{st}} = \frac{dU_{\text{sens}}}{dt} = mc_v \frac{dT}{dt} = \rho V c_v \frac{dT}{dt}$$

where ρ and c_v are the mass density and the specific heat, respectively, of the rod material, and V is the volume of the rod, $V = (\pi D^2/4)L$. Substituting the rate equations into the energy balance, it follows that

$$I^2 R'_e L - h(\pi DL)(T - T_\infty) - \varepsilon\sigma(\pi DL)(T^4 - T_{\text{sur}}^4) = \rho c_v \left(\frac{\pi D^2}{4}\right) L \frac{dT}{dt}$$

Hence

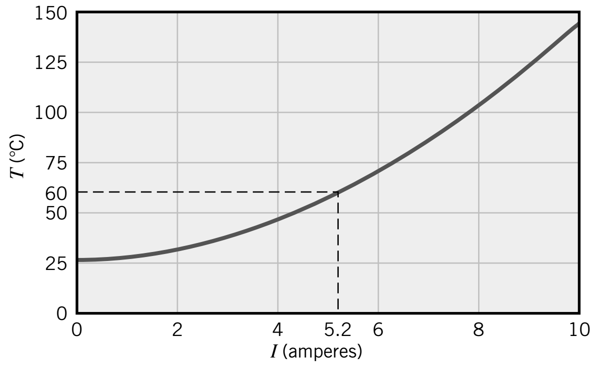
$$\frac{dT}{dt} = \frac{I^2 R'_e - \pi Dh(T - T_\infty) - \pi D\varepsilon\sigma(T^4 - T_{\text{sur}}^4)}{\rho c_v (\pi D^2/4)} \qquad \triangleleft$$

Comments:

1. Solids and liquids can usually be treated as incompressible, in which case $c_v = c_p = c$. This assumption will be made in the remainder of this text.
2. The preceding equation could be solved for the time dependence of the rod temperature by integrating numerically. A steady-state condition would eventually be reached for which $dT/dt = 0$. The rod temperature is then determined by an algebraic equation of the form

$$\pi Dh(T - T_\infty) + \pi D\varepsilon\sigma(T^4 - T_{\text{sur}}^4) = I^2 R'_e$$

3. For fixed environmental conditions (h, T_∞, T_{sur}), as well as a rod of fixed geometry (D) and properties (ε, R'_e), the steady-state temperature depends on the rate of thermal energy generation and hence on the value of the electric current. Consider an uninsulated copper wire ($D = 1$ mm, $\varepsilon = 0.8$, $R'_e = 0.4\ \Omega/\text{m}$) in a large enclosure ($T_{\text{sur}} = 300$ K) through which cooling air is circulated ($h = 100\ \text{W/m}^2 \cdot \text{K}$, $T_\infty = 300$ K). Substituting these values into the foregoing equation, the rod temperature has been computed for operating currents in the range $0 \le I \le 10$ A, and the following results were obtained:



4. If a maximum operating temperature of $T = 60$°C is prescribed for safety reasons, the current should not exceed 5.2 A. At this temperature, heat transfer by radiation (0.6 W/m) is much less than heat transfer by convection (10.4 W/m). Hence, if one wished to operate at a larger current while maintaining the rod temperature within the safety limit, the convection coefficient would have to be increased by increasing the velocity of the circulating air. For $h = 250\ \text{W/m}^2 \cdot \text{K}$, the maximum allowable current could be increased to 8.1 A.
5. The *IHT* software is especially useful for solving equations, such as the energy balance in Comment 1, and generating the graphical results of Comment 2.

IHT

EXAMPLE 1.5

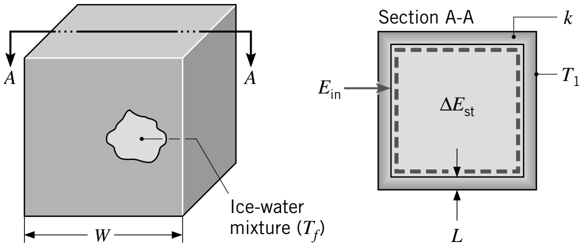
Consider a mass m of ice at the fusion temperature ($T_f = 0°C$) that is enclosed in a cubical container of width W on a side. The container wall is of thickness L and thermal conductivity k. If the outer surface of the wall is heated to a temperature $T_1 > T_f$ to melt the ice, obtain an expression for the time needed to melt the entire mass of ice.

SOLUTION

Known: Mass and temperature of ice. Dimensions, thermal conductivity, and outer surface temperature of containing wall.

Find: Expression for time needed to melt the ice.

Schematic:



Assumptions:

1. Inner surface of wall is at T_f throughout the process.
2. Constant properties.
3. Steady-state, one-dimensional conduction through each wall.
4. Conduction area of one wall may be approximated as W^2 ($L \ll W$).

Analysis: Since we must determine the melting time t_m, the first law should be applied over the time interval $\Delta t = t_m$. Hence, applying Equation 1.12b to a control volume about the ice–water mixture, it follows that

$$E_{in} = \Delta E_{st} = \Delta U_{lat}$$

where the increase in energy stored within the control volume is due exclusively to the change in latent energy associated with conversion from the solid to liquid state. Heat is transferred to the ice by means of conduction through the container wall. Since the temperature difference across the wall is assumed to remain at $(T_1 - T_f)$ throughout the melting process, the wall conduction rate is constant

$$q_{cond} = k(6W^2)\frac{T_1 - T_f}{L}$$

and the amount of energy inflow is

$$E_{in} = \left[k(6W^2)\frac{T_1 - T_f}{L} \right] t_m$$

The amount of energy required to effect such a phase change per unit mass of solid is termed the *latent heat of fusion* h_{sf}. Hence the increase in energy storage is

$$\Delta E_{\text{st}} = mh_{sf}$$

By substituting into the first law expression, it follows that

$$t_m = \frac{mh_{sf}L}{6W^2k(T_1 - T_f)} \quad \triangleleft$$

Comments:

1. Several complications would arise if the ice were initially subcooled. The storage term would have to include the change in sensible (internal thermal) energy required to take the ice from the subcooled to the fusion temperature. During this process, temperature gradients would develop in the ice.
2. Consider a cavity of width $W = 100$ mm on a side, wall thickness $L = 5$ mm, and thermal conductivity $k = 0.05$ W/m · K. The mass of the ice in the cavity is

   $$m = \rho_s(W - 2L)^3 = 920 \text{ kg/m}^3 \times (0.100 - 0.01)^3 \text{ m}^3 = 0.67 \text{ kg}$$

   If the outer surface temperature is $T_1 = 30°\text{C}$, the time required to melt the ice is

   $$t_m = \frac{0.67 \text{ kg} \times 334{,}000 \text{ J/kg} \times 0.005 \text{ m}}{6(0.100 \text{ m})^2 \times 0.05 \text{ W/m} \cdot \text{K}(30 - 0)°\text{C}} = 12{,}430 \text{ s} = 207 \text{ min}$$

   The density and latent heat of fusion of the ice are $\rho_s = 920 \text{ kg/m}^3$ and $h_{sf} = 334$ kJ/kg, respectively.
3. Note that the units of K and °C cancel each other in the foregoing expression for t_m. Such cancellation occurs frequently in heat transfer analysis and is due to both units appearing in the context of a *temperature difference*.

The Surface Energy Balance We will frequently have occasion to apply the conservation of energy requirement at the surface of a medium. In this special case, the control surfaces are located on either side of the physical boundary and enclose no mass or volume (see Figure 1.10).

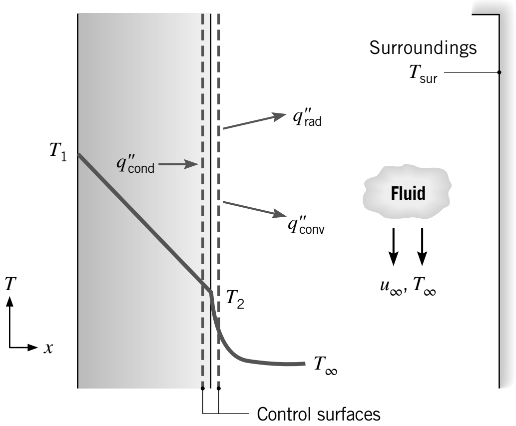


FIGURE 1.10 The energy balance for conservation of energy at the surface of a medium.

Accordingly, the generation and storage terms of the conservation expression, Equation 1.12c, are no longer relevant, and it is necessary to deal only with surface phenomena. For this case, the conservation requirement becomes

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = 0 \tag{1.13}$$

Even though energy generation may be occurring in the medium, the process would not affect the energy balance at the control surface. Moreover, this conservation requirement holds for both *steady-state* and *transient* conditions.

In Figure 1.10, three heat transfer terms are shown for the control surface. On a unit area basis, they are conduction from the medium *to* the control surface (q''_{cond}), convection *from* the surface to a fluid (q''_{conv}), and net radiation exchange from the surface to the surroundings (q''_{rad}). The energy balance then takes the form

$$q''_{\text{cond}} - q''_{\text{conv}} - q''_{\text{rad}} = 0 \tag{1.14}$$

and we can express each of the terms using the appropriate rate equations, Equations 1.2, 1.3a, and 1.7.

EXAMPLE 1.6

Humans are able to control their rates of heat production and heat loss to maintain a nearly constant core temperature of $T_c = 37°\text{C}$ under a wide range of environmental conditions. This process is called *thermoregulation*. From the perspective of calculating heat transfer between a human body and its surroundings, we focus on a layer of skin and fat, with its outer surface exposed to the environment and its inner surface at a temperature slightly less than the core temperature, $T_i = 35°\text{C} = 308\text{ K}$. Consider a person with a skin/fat layer of thickness $L = 3$ mm and effective thermal conductivity $k = 0.3\text{ W/m}\cdot\text{K}$. The person has a surface area $A = 1.8\text{ m}^2$ and is dressed in a bathing suit. The emissivity of the skin is $\varepsilon = 0.95$.

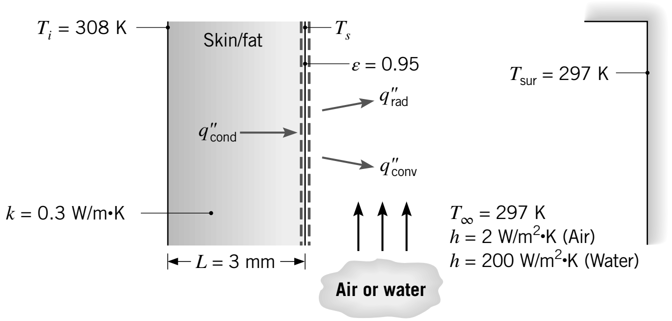
1. When the person is in still air at $T_\infty = 297$ K, what is the skin surface temperature and rate of heat loss to the environment? Convection heat transfer to the air is characterized by a free convection coefficient of $h = 2\text{ W/m}^2\cdot\text{K}$.
2. When the person is in water at $T_\infty = 297$ K, what is the skin surface temperature and rate of heat loss? Heat transfer to the water is characterized by a convection coefficient of $h = 200\text{ W/m}^2\cdot\text{K}$.

SOLUTION

Known: Inner surface temperature of a skin/fat layer of known thickness, thermal conductivity, emissivity, and surface area. Ambient conditions.

Find: Skin surface temperature and rate of heat loss for the person in air and the person in water.

Schematic:



Assumptions:

1. Steady-state conditions.
2. One-dimensional heat transfer by conduction through the skin/fat layer.
3. Thermal conductivity is uniform.
4. In part 1, radiation exchange between the skin surface and the surroundings is between a small surface and a large enclosure at the air temperature.
5. Liquid water is opaque to thermal radiation.
6. Bathing suit has no effect on heat loss from body.
7. Solar radiation is negligible.
8. Body is completely immersed in water in part 2.

Analysis:

1. The skin surface temperature may be obtained by performing an energy balance at the skin surface. From Equation 1.13,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = 0$$

It follows that, on a unit area basis,

$$q''_{\text{cond}} - q''_{\text{conv}} - q''_{\text{rad}} = 0$$

or, rearranging and substituting from Equations 1.2, 1.3a, and 1.7,

$$k\frac{T_i - T_s}{L} = h(T_s - T_\infty) + \varepsilon\sigma(T_s^4 - T_{\text{sur}}^4)$$

The only unknown is T_s, but we cannot solve for it explicitly because of the fourth-power dependence of the radiation term. Therefore, we must solve the equation iteratively, which can be done by hand or by using *IHT* or some other equation solver. To expedite a hand solution, we write the radiation heat flux in terms of the radiation heat transfer coefficient, using Equations 1.8 and 1.9:

$$k\frac{T_i - T_s}{L} = h(T_s - T_\infty) + h_r(T_s - T_{\text{sur}})$$

Solving for T_s, with $T_{\text{sur}} = T_\infty$, we have

$$T_s = \frac{\dfrac{kT_i}{L} + (h + h_r)T_\infty}{\dfrac{k}{L} + (h + h_r)}$$

We estimate h_r using Equation 1.9 with a guessed value of $T_s = 305$ K and $T_\infty = 297$ K, to yield $h_r = 5.9\ \text{W/m}^2 \cdot \text{K}$. Then, substituting numerical values into the preceding equation, we find

$$T_s = \frac{\dfrac{0.3\ \text{W/m}\cdot\text{K} \times 308\ \text{K}}{3 \times 10^{-3}\ \text{m}} + (2 + 5.9)\ \text{W/m}^2\cdot\text{K} \times 297\ \text{K}}{\dfrac{0.3\ \text{W/m}\cdot\text{K}}{3 \times 10^{-3}\ \text{m}} + (2 + 5.9)\ \text{W/m}^2\cdot\text{K}} = 307.2\ \text{K}$$

With this new value of T_s, we can recalculate h_r and T_s, which are unchanged. Thus the skin temperature is 307.2K ≅ 34°C. ◁

The rate of heat loss can be found by evaluating the conduction through the skin/fat layer:

$$q_s = kA\frac{T_i - T_s}{L} = 0.3\ \text{W/m}\cdot\text{K} \times 1.8\ \text{m}^2 \times \frac{(308 - 307.2)\ \text{K}}{3 \times 10^{-3}\ \text{m}} = 146\ \text{W} \quad \triangleleft$$

2. Since liquid water is opaque to thermal radiation, heat loss from the skin surface is by convection only. Using the previous expression with $h_r = 0$, we find

$$T_s = \frac{\dfrac{0.3\ \text{W/m}\cdot\text{K} \times 308\ \text{K}}{3 \times 10^{-3}\ \text{m}} + 200\ \text{W/m}^2\cdot\text{K} \times 297\ \text{K}}{\dfrac{0.3\ \text{W/m}\cdot\text{K}}{3 \times 10^{-3}\ \text{m}} + 200\ \text{W/m}^2\cdot\text{K}} = 300.7\ \text{K} \quad \triangleleft$$

and

$$q_s = kA\frac{T_i - T_s}{L} = 0.3\ \text{W/m}\cdot\text{K} \times 1.8\ \text{m}^2 \times \frac{(308 - 300.7)\ \text{K}}{3 \times 10^{-3}\ \text{m}} = 1320\ \text{W} \quad \triangleleft$$

Comments:

1. When using energy balances involving radiation exchange, the temperatures appearing in the radiation terms must be expressed in kelvins, and it is good practice to use kelvins in all terms to avoid confusion.
2. In part 1, the rates of heat loss due to convection and radiation are 37 W and 109 W, respectively. Thus, it would not have been reasonable to neglect radiation. Care must be taken to include radiation when the heat transfer coefficient is small (as it often is for natural convection to a gas), even if the problem statement does not give any indication of its importance.

3. A typical rate of metabolic heat generation is 100 W. If the person stayed in the water too long, the core body temperature would begin to fall. The large rate of heat loss in water is due to the higher heat transfer coefficient, which in turn is due to the much larger thermal conductivity of water compared to air.
4. The skin temperature of 34°C in part 1 is comfortable, but the skin temperature of 28°C in part 2 is uncomfortably cold.

Application of the Conservation Laws: Methodology In addition to being familiar with the transport rate equations described in Section 1.2, the heat transfer analyst must be able to work with the energy conservation requirements of Equations 1.12 and 1.13. The application of these balances is simplified if a few basic rules are followed.

1. The appropriate control volume must be defined, with the control surfaces represented by a dashed line or lines.
2. The appropriate time basis must be identified.
3. The relevant energy processes must be identified, and each process should be shown on the control volume by an appropriately labeled arrow.
4. The conservation equation must then be written, and appropriate rate expressions must be substituted for the relevant terms in the equation.

Note that the energy conservation requirement may be applied to a *finite* control volume or a *differential* (infinitesimal) control volume. In the first case, the resulting expression governs overall system behavior. In the second case, a differential equation is obtained that can be solved for conditions at each point in the system. Differential control volumes are introduced in Chapter 2, and both types of control volumes are used extensively throughout the text.

1.3.2 Relationship to the Second Law of Thermodynamics and the Efficiency of Heat Engines

In this section, we show how heat transfer plays a crucial role in managing and promoting the efficiency of a broad range of energy conversion devices. Recall that a heat engine is any device that operates continuously or cyclically and that converts heat to work. Examples include internal combustion engines, power plants, and thermoelectric devices (to be discussed in Section 3.7.2). Improving the efficiency of heat engines is a subject of great importance; for example, more efficient combustion engines consume less fuel to produce a given amount of work and reduce the corresponding emissions of pollutants and carbon dioxide. More efficient thermoelectric devices can generate more electricity from waste heat. Regardless of the energy conversion device, its size, weight, and cost can all be reduced through improvements in its energy conversion efficiency.

The second law of thermodynamics is often invoked when efficiency is of concern and can be expressed in a variety of different but equivalent ways. The *Kelvin–Planck statement* is particularly relevant to the operation of heat engines [1]. It states:

> *It is impossible for any system to operate in a thermodynamic cycle and deliver a net amount of work to its surroundings while receiving energy by heat transfer from a single thermal reservoir.*

Recall that a thermodynamic cycle is a process for which the initial and final states of the system are identical. Consequently, the energy stored in the system does not change between the initial and final states, and the first law of thermodynamics (Equation 1.12a) reduces to $W = Q$.

A consequence of the Kelvin–Planck statement is that a heat engine must exchange heat with two (or more) reservoirs, receiving thermal energy from the higher-temperature reservoir and rejecting thermal energy to the lower-temperature reservoir. Thus, converting all of the input heat to work is impossible, and $W = Q_{\text{in}} - Q_{\text{out}}$, where Q_{in} and Q_{out} are both defined to be positive. That is, Q_{in} is the heat transferred from the high temperature source to the heat engine, and Q_{out} is the heat transferred from the heat engine to the low temperature sink.

The efficiency of a heat engine is defined as the fraction of heat transferred into the system that is converted to work, namely

$$\eta \equiv \frac{W}{Q_{\text{in}}} = \frac{Q_{\text{in}} - Q_{\text{out}}}{Q_{\text{in}}} = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} \tag{1.15}$$

The second law also tells us that, for a *reversible* process, the ratio $Q_{\text{out}}/Q_{\text{in}}$ is equal to the ratio of the absolute temperatures of the respective reservoirs [1]. Thus, the efficiency of a heat engine undergoing a reversible process, called the *Carnot efficiency* η_C, is given by

$$\eta_C = 1 - \frac{T_c}{T_h} \tag{1.16}$$

where T_c and T_h are the absolute temperatures of the low- and high-temperature reservoirs, respectively. The Carnot efficiency is the maximum possible efficiency that any heat engine can achieve operating between those two temperatures. Any *real* heat engine, which will necessarily undergo an irreversible process, will have an efficiency lower than η_C.

From our knowledge of thermodynamics, we know that, for heat transfer to take place reversibly, it must occur through an infinitesimal temperature difference between the reservoir and heat engine. However, from our newly acquired knowledge of heat transfer mechanisms, as embodied, for example, in Equations 1.2, 1.3, and 1.7, we now realize that, for heat transfer to occur, there *must* be a nonzero temperature difference between the reservoir and the heat engine. This reality introduces irreversibility and reduces the efficiency.

With the concepts of the preceding paragraph in mind, we now consider a more realistic model of a heat engine [2–5] in which heat is transferred into the engine through a thermal resistance $R_{t,h}$, while heat is extracted from the engine through a second thermal resistance $R_{t,c}$ (Figure 1.11). The subscripts h and c refer to the hot and cold sides of the heat engine, respectively. As discussed in Section 1.2.4, these thermal resistances are associated with heat transfer between the heat engine and the reservoirs across a nonzero temperature difference, by way of the mechanisms of conduction, convection, and/or radiation. For example, the resistances could represent conduction through the walls separating the heat engine from the two reservoirs. Note that the reservoir temperatures are still T_h and T_c but that the temperatures seen by the heat engine are $T_{h,i} < T_h$ and $T_{c,i} > T_c$, as shown in the diagram. The heat engine is still assumed to be *internally* reversible, and its efficiency is still the Carnot efficiency. However, the Carnot efficiency is *now based on the internal*

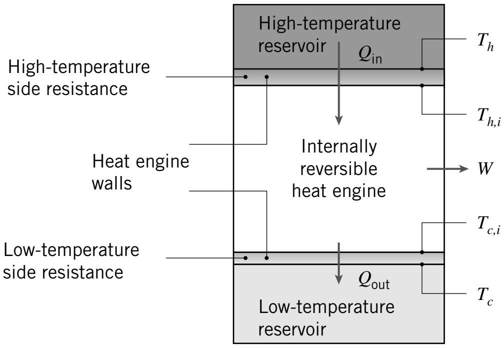


FIGURE 1.11 Internally reversible heat engine exchanging heat with high- and low-temperature reservoirs through thermal resistances.

temperatures $T_{h,i}$ and $T_{c,i}$. Therefore, a modified efficiency that accounts for realistic (irreversible) heat transfer processes η_m is

$$\eta_m = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{T_{c,i}}{T_{h,i}} \tag{1.17}$$

where the ratio of heat *flows* over a time interval, $Q_{\text{out}}/Q_{\text{in}}$, has been replaced by the corresponding ratio of heat *rates*, $q_{\text{out}}/q_{\text{in}}$. This replacement is based on applying energy conservation at an instant in time,[1] as discussed in Section 1.3.1. Utilizing the definition of a thermal resistance, the heat transfer rates into and out of the heat engine are given by

$$q_{\text{in}} = (T_h - T_{h,i})/R_{t,h} \tag{1.18a}$$

$$q_{\text{out}} = (T_{c,i} - T_c)/R_{t,c} \tag{1.18b}$$

Equations 1.18 can be solved for the internal temperatures, to yield

$$T_{h,i} = T_h - q_{\text{in}}R_{t,h} \tag{1.19a}$$

$$T_{c,i} = T_c + q_{\text{out}}R_{t,c} = T_c + q_{\text{in}}(1 - \eta_m)R_{t,c} \tag{1.19b}$$

In Equation 1.19b, q_{out} has been related to q_{in} and η_m, using Equation 1.17. The more realistic, modified efficiency can then be expressed as

$$\eta_m = 1 - \frac{T_{c,i}}{T_{h,i}} = 1 - \frac{T_c + q_{\text{in}}(1 - \eta_m)R_{t,c}}{T_h - q_{\text{in}}R_{t,h}} \tag{1.20}$$

Solving for η_m results in

$$\eta_m = 1 - \frac{T_c}{T_h - q_{\text{in}}R_{\text{tot}}} \tag{1.21}$$

---

[1] The heat engine is assumed to undergo a continuous, steady-flow process, so that all heat and work processes are occurring simultaneously, and the corresponding terms would be expressed in watts (W). For a heat engine undergoing a cyclic process with sequential heat and work processes occurring over different time intervals, we would need to introduce the time intervals for each process, and each term would be expressed in joules (J).

where $R_{\text{tot}} = R_{t,h} + R_{t,c}$. It is readily evident that $\eta_m = \eta_C$ only if the thermal resistances $R_{t,h}$ and $R_{t,c}$ could somehow be made infinitesimally small (or if $q_{\text{in}} = 0$). For realistic (nonzero) values of R_{tot}, $\eta_m < \eta_C$, and η_m further deteriorates as either R_{tot} or q_{in} increases. As an extreme case, note that $\eta_m = 0$ when $T_h = T_c + q_{\text{in}}R_{\text{tot}}$, meaning that no power could be produced even though the Carnot efficiency, as expressed in Equation 1.16, is nonzero.

In addition to the efficiency, another important parameter to consider is the power output of the heat engine, given by

$$\dot{W} = q_{\text{in}}\eta_m = q_{\text{in}}\left[1 - \frac{T_c}{T_h - q_{\text{in}}R_{\text{tot}}}\right] \tag{1.22}$$

It has already been noted in our discussion of Equation 1.21 that the efficiency is equal to the maximum Carnot efficiency ($\eta_m = \eta_C$) if $q_{\text{in}} = 0$. However, under these circumstances the power output $\dot{W}$ is zero according to Equation 1.22. To increase $\dot{W}$, q_{in} must be increased at the expense of decreased efficiency. In any real application, a balance must be struck between maximizing the efficiency and maximizing the power output. If provision of the heat input is inexpensive (for example, if waste heat is converted to power), a case could be made for sacrificing efficiency to maximize power output. In contrast, if fuel is expensive or emissions are detrimental (such as for a conventional fossil fuel power plant), the efficiency of the energy conversion may be as or more important than the power output. In any case, heat transfer and thermodynamic principles should be used to determine the actual efficiency and power output of a heat engine.

Although we have limited our discussion of the second law to heat engines, the preceding analysis shows how the principles of thermodynamics and heat transfer can be combined to address significant problems of contemporary interest.

EXAMPLE 1.7

In a large steam power plant, the combustion of coal provides a heat rate of $q_{in} = 2500$ MW at a flame temperature of $T_h = 1000$ K. Heat is rejected from the plant to a river flowing at $T_c = 300$ K. Heat is transferred from the combustion products to the exterior of large tubes in the boiler by way of radiation and convection, through the boiler tubes by conduction, and then from the interior tube surface to the working fluid (water) by convection. On the cold side, heat is extracted from the power plant by condensation of steam on the exterior condenser tube surfaces, through the condenser tube walls by conduction, and from the interior of the condenser tubes to the river water by convection. Hot and cold side thermal resistances account for the combined effects of conduction, convection, and radiation and, under *design conditions*, they are $R_{t,h} = 8 \times 10^{-8}$ K/W and $R_{t,c} = 2 \times 10^{-8}$ K/W, respectively.

1. Determine the efficiency and power output of the power plant, accounting for heat transfer effects to and from the cold and hot reservoirs. Treat the power plant as an internally reversible heat engine.
2. Over time, coal slag will accumulate on the combustion side of the boiler tubes. This *fouling process* increases the hot side resistance to $R_{t,h} = 9 \times 10^{-8}$ K/W. Concurrently, biological matter can accumulate on the river water side of the condenser tubes, increasing the cold side resistance to $R_{t,c} = 2.2 \times 10^{-8}$ K/W. Find the efficiency and power output of the plant under fouled conditions.

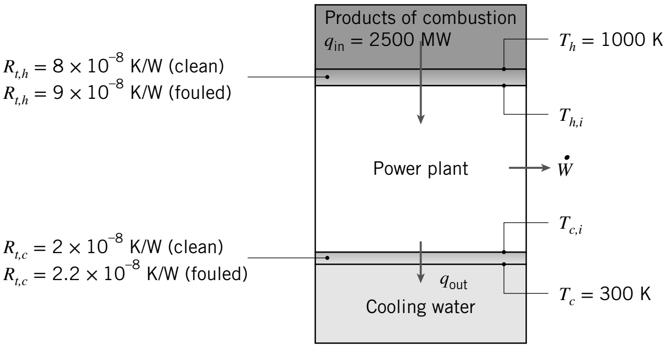
SOLUTION

Known: Source and sink temperatures and heat input rate for an internally reversible heat engine. Thermal resistances separating heat engine from source and sink under clean and fouled conditions.

Find:

1. Efficiency and power output for clean conditions.
2. Efficiency and power output under fouled conditions.

Schematic:



Assumptions:

1. Steady-state conditions.
2. Power plant behaves as an internally reversible heat engine, so its efficiency is the modified efficiency.

Analysis:

1. The modified efficiency of the internally reversible power plant, considering realistic heat transfer effects on the hot and cold sides of the power plant, is given by Equation 1.21:

$$\eta_m = 1 - \frac{T_c}{T_h - q_{\text{in}} R_{\text{tot}}}$$

where, for clean conditions

$$R_{\text{tot}} = R_{t,h} + R_{t,c} = 8 \times 10^{-8} \text{ K/W} + 2 \times 10^{-8} \text{ K/W} = 1.0 \times 10^{-7} \text{ K/W}$$

Thus

$$\eta_m = 1 - \frac{T_c}{T_h - q_{\text{in}} R_{\text{tot}}} = 1 - \frac{300 \text{ K}}{1000 \text{ K} - 2500 \times 10^6 \text{ W} \times 1.0 \times 10^{-7} \text{ K/W}} = 0.60 = 60\% \quad \triangleleft$$

The power output is given by

$$\dot{W} = q_{\text{in}} \eta_m = 2500 \text{ MW} \times 0.60 = 1500 \text{ MW} \quad \triangleleft$$

2. Under fouled conditions, the preceding calculations are repeated to find

$$\eta_m = 0.583 = 58.3\% \text{ and } \dot{W} = 1460 \text{ MW} \qquad \triangleleft$$

Comments:

1. The actual efficiency and power output of a power plant operating between these temperatures would be much less than the foregoing values, since there would be other irreversibilities internal to the power plant. Even if these irreversibilities were considered in a more comprehensive analysis, fouling effects would still reduce the plant efficiency and power output.

2. The Carnot efficiency is $\eta_C = 1 - T_c/T_h = 1 - 300 \text{ K}/1000 \text{ K} = 70\%$. The corresponding power output would be $\dot{W} = q_{\text{in}}\eta_C = 2500 \text{ MW} \times 0.70 = 1750 \text{ MW}$. Thus, if the effect of irreversible heat transfer from and to the hot and cold reservoirs, respectively, were neglected, the power output of the plant would be significantly overpredicted.

3. Fouling reduces the power output of the plant by $\Delta P = 40$ MW. If the plant owner sells the electricity at a price of \$0.08/kW · h, the daily lost revenue associated with operating the fouled plant would be $C = 40{,}000 \text{ kW} \times \$0.08/\text{kW} \cdot \text{h} \times 24 \text{ h/day} = \$76{,}800/\text{day}$.

1.4 *Units and Dimensions*

The physical quantities of heat transfer are specified in terms of *dimensions*, which are measured in terms of units. Four *basic* dimensions are required for the development of heat transfer: length (L), mass (m), time (t), and temperature (T). All other physical quantities of interest may be related to these four basic dimensions.

In the United States, dimensions have been customarily measured in terms of the *English system of units*, for which the *base units* are:

Dimension		Unit
Length (L)	→	foot (ft)
Mass (m)	→	pound mass (lb_m)
Time (t)	→	second (s)
Temperature (T)	→	degree Fahrenheit (°F)

The units required to specify other physical quantities may then be inferred from this group.

In 1960, the SI (Système International d'Unités) system of units was defined by the Eleventh General Conference on Weights and Measures and recommended as a worldwide standard. In response to this trend, the American Society of Mechanical Engineers (ASME) has required the use of SI units in all of its publications since 1974. For this reason and because SI units are operationally more convenient than the English system, the SI system is used for calculations of this text. However, because for some time to come, engineers might also have to work with results expressed in the English system, you should be able to convert from one system to the other. For your convenience, conversion factors are provided after the Appendices.

TABLE 1.2 **SI base and supplementary units**

Quantity and Symbol	Unit and Symbol
Length (L)	meter (m)
Mass (m)	kilogram (kg)
Amount of substance	mole (mol)
Time (t)	second (s)
Electric current (I)	ampere (A)
Thermodynamic temperature (T)	kelvin (K)
Plane angle[a] (θ)	radian (rad)
Solid angle[a] (ω)	steradian (sr)

[a] Supplementary unit.

The SI *base* units required for this text are summarized in Table 1.2. With regard to these units, note that 1 mol is the amount of substance that has as many atoms or molecules as there are atoms in 12 g of carbon-12 (^{12}C); this is the gram-mole (mol). Although the mole has been recommended as the unit quantity of matter for the SI system, it is more consistent to work with the kilogram-mol (kmol, kg-mol). One kmol is simply the amount of substance that has as many atoms or molecules as there are atoms in 12 kg of ^{12}C. As long as the use is consistent within a given problem, no difficulties arise in using either mol or kmol. The molecular weight of a substance is the mass associated with a mole or kilogram-mole. For oxygen, as an example, the molecular weight $\mathcal{M}$ is 16 g/mol or 16 kg/kmol.

Although the SI unit of temperature is the kelvin, use of the Celsius temperature scale remains widespread. Zero on the Celsius scale (0°C) is equivalent to 273.15 K on the thermodynamic scale,[2] in which case

$$T(\mathrm{K}) = T(^\circ\mathrm{C}) + 273.15$$

However, temperature *differences* are equivalent for the two scales and may be denoted as °C or K. Also, although the SI unit of time is the second, other units of time (minute, hour, and day) are so common that their use with the SI system is generally accepted.

The SI units comprise a coherent form of the metric system. That is, all remaining units may be derived from the base units using formulas that do not involve any numerical factors. *Derived* units for selected quantities are listed in Table 1.3. Note that force is measured in

TABLE 1.3 **SI derived units for selected quantities**

Quantity	Name and Symbol	Formula	Expression in SI Base Units
Force	newton (N)	$\mathrm{m \cdot kg/s^2}$	$\mathrm{m \cdot kg/s^2}$
Pressure and stress	pascal (Pa)	$\mathrm{N/m^2}$	$\mathrm{kg/m \cdot s^2}$
Energy	joule (J)	$\mathrm{N \cdot m}$	$\mathrm{m^2 \cdot kg/s^2}$
Power	watt (W)	J/s	$\mathrm{m^2 \cdot kg/s^3}$

---

[2] The degree symbol is retained for designating the Celsius temperature (°C) to avoid confusion with the use of C for the unit of electrical charge (coulomb).

TABLE 1.4 Multiplying prefixes

Prefix	Abbreviation	Multiplier
femto	f	10^{-15}
pico	p	10^{-12}
nano	n	10^{-9}
micro	μ	10^{-6}
milli	m	10^{-3}
centi	c	10^{-2}
hecto	h	10^{2}
kilo	k	10^{3}
mega	M	10^{6}
giga	G	10^{9}
tera	T	10^{12}
peta	P	10^{15}
exa	E	10^{18}

newtons, where a 1-N force will accelerate a 1-kg mass at 1 m/s^2. Hence $1\ \text{N} = 1\ \text{kg} \cdot \text{m/s}^2$. The unit of pressure ($\text{N/m}^2$) is often referred to as the pascal. In the SI system, there is one unit of energy (thermal, mechanical, or electrical) called the joule (J); $1\ \text{J} = 1\ \text{N} \cdot \text{m}$. The unit for energy rate, or power, is then J/s, where one joule per second is equivalent to one watt (1 J/s = 1 W). Since working with extremely large or small numbers is frequently necessary, a set of standard prefixes has been introduced to simplify matters (Table 1.4). For example, 1 megawatt (MW) = 10^6 W, and 1 micrometer (μm) = 10^{-6} m.

1.5 *Analysis of Heat Transfer Problems: Methodology*

A major objective of this text is to prepare you to solve engineering problems that involve heat transfer processes. To this end, numerous problems are provided at the end of each chapter. In working these problems you will gain a deeper appreciation for the fundamentals of the subject, and you will gain confidence in your ability to apply these fundamentals to the solution of engineering problems.

In solving problems, we advocate the use of a systematic procedure characterized by a prescribed format. We consistently employ this procedure in our examples, and we require our students to use it in their problem solutions. It consists of the following steps:

1. *Known:* After carefully reading the problem, state briefly and concisely what is known about the problem. Do not repeat the problem statement.
2. *Find:* State briefly and concisely what must be found.
3. *Schematic:* Draw a schematic of the physical system. If application of the conservation laws is anticipated, represent the required control surface or surfaces by dashed lines on the schematic. Identify relevant heat transfer processes by appropriately labeled arrows on the schematic.

4. *Assumptions:* List all pertinent simplifying assumptions.
5. *Properties:* Compile property values needed for subsequent calculations and identify the source from which they are obtained.
6. *Analysis:* Begin your analysis by applying appropriate conservation laws, and introduce rate equations as needed. Develop the analysis as completely as possible before substituting numerical values. Perform the calculations needed to obtain the desired results.
7. *Comments:* Discuss your results. Such a discussion may include a summary of key conclusions, a critique of the original assumptions, and an inference of trends obtained by performing additional *what-if* and *parameter sensitivity* calculations.

The importance of following steps 1 through 4 should not be underestimated. They provide a useful guide to thinking about a problem before effecting its solution. In step 7, we hope you will take the initiative to gain additional insights by performing calculations that may be computer based. The software accompanying this text provides a suitable tool for effecting such calculations.

IHT

EXAMPLE 1.8

The coating on a plate is cured by exposure to an infrared lamp providing a uniform irradiation of 2000 W/m^2. It absorbs 80% of the infrared irradiation and has an emissivity of 0.50. It is also exposed to an airflow and large surroundings for which temperatures are 20°C and 30°C, respectively.

1. If the convection coefficient between the plate and the ambient air is 15 W/m$^2 \cdot$ K, what is the cure temperature of the plate?
2. The final characteristics of the coating, including wear and durability, are known to depend on the temperature at which curing occurs. An airflow system is able to control the air velocity, and hence the convection coefficient, on the cured surface, but the process engineer needs to know how the temperature depends on the convection coefficient. Provide the desired information by computing and plotting the surface temperature as a function of h for $2 \leq h \leq 100$ W/m$^2 \cdot$ K. What value of h would provide a cure temperature of 50°C?

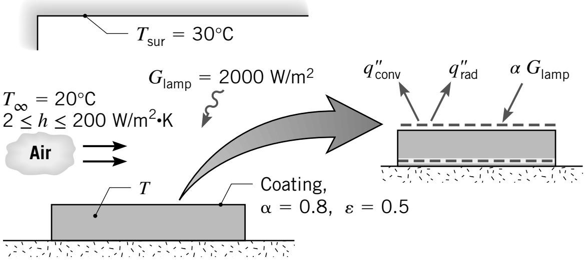
SOLUTION

Known: Coating with prescribed radiation properties is cured by irradiation from an infrared lamp. Heat transfer from the coating is by convection to ambient air and radiation exchange with the surroundings.

Find:

1. Cure temperature for $h = 15$ W/m$^2 \cdot$ K.
2. Effect of airflow on the cure temperature for $2 \leq h \leq 100$ W/m$^2 \cdot$ K. Value of h for which the cure temperature is 50°C.

Schematic:



Assumptions:

1. Steady-state conditions.
2. Negligible heat loss from back surface of plate.
3. Plate is small object in large surroundings, and coating has an absorptivity of $\alpha_{sur} = \varepsilon = 0.5$ with respect to irradiation from the surroundings.

Analysis:

1. Since the process corresponds to steady-state conditions and there is no heat transfer at the back surface, the plate must be isothermal ($T_s = T$). Hence the desired temperature may be determined by placing a control surface about the exposed surface and applying Equation 1.13 or by placing the control surface about the entire plate and applying Equation 1.12c. Adopting the latter approach and recognizing that there is no energy generation ($\dot{E}_g = 0$), Equation 1.12c reduces to

$$\dot{E}_{in} - \dot{E}_{out} = 0$$

where $\dot{E}_{st} = 0$ for steady-state conditions. With energy inflow due to absorption of the lamp irradiation by the coating and outflow due to convection and net radiation transfer to the surroundings, it follows that

$$(\alpha G)_{lamp} - q''_{conv} - q''_{rad} = 0$$

Substituting from Equations 1.3a and 1.7, we obtain

$$(\alpha G)_{lamp} - h(T - T_\infty) - \varepsilon\sigma(T^4 - T_{sur}^4) = 0$$

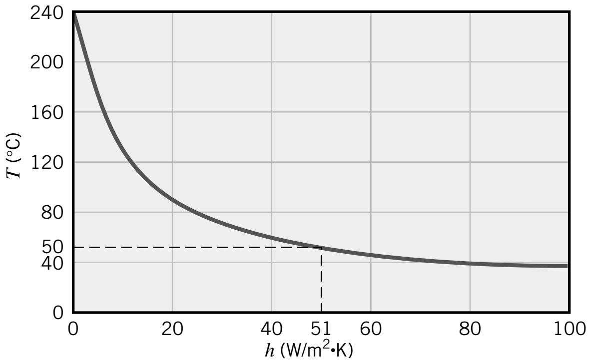
Substituting numerical values

$$\begin{aligned} &0.8 \times 2000\ \mathrm{W/m^2} - 15\ \mathrm{W/m^2 \cdot K}(T - 293)\ \mathrm{K} \\ &\qquad - 0.5 \times 5.67 \times 10^{-8}\ \mathrm{W/m^2 \cdot K^4}\,(T^4 - 303^4)\ \mathrm{K^4} = 0 \end{aligned}$$

and solving by trial-and-error, we obtain

$$T = 377\ \mathrm{K} = 104°\mathrm{C} \qquad \triangleleft$$

2. Solving the foregoing energy balance for selected values of h in the prescribed range and plotting the results, we obtain



If a cure temperature of 50°C is desired, the airflow must provide a convection coefficient of

$$h(T = 50°\text{C}) = 51.0 \text{ W/m}^2 \cdot \text{K} \qquad \triangleleft$$

Comments:

1. The coating (plate) temperature may be reduced by decreasing T_∞ and T_{sur}, as well as by increasing the air velocity and hence the convection coefficient.
2. The relative contributions of convection and radiation to heat transfer from the plate vary greatly with h. For $h = 2$ W/m$^2 \cdot$ K, $T = 204$°C and radiation dominates ($q''_{\text{rad}} \approx 1232$ W/m^2, $q''_{\text{conv}} \approx 368$ W/m^2). Conversely, for $h = 200$ W/m$^2 \cdot$ K, $T = 28$°C and convection dominates ($q''_{\text{conv}} \approx 1606$ W/m^2, $q''_{\text{rad}} \approx -6$ W/m^2). In fact, for this condition the plate temperature is slightly less than that of the surroundings and net radiation exchange is *to* the plate.

1.6 *Relevance of Heat Transfer*

We will devote much time to acquiring an understanding of heat transfer effects and to developing the skills needed to predict heat transfer rates and temperatures that evolve in certain situations. What is the value of this knowledge? To what problems may it be applied? A few examples will serve to illustrate the rich breadth of applications in which heat transfer plays a critical role.

The challenge of providing sufficient amounts of energy for humankind is well known. Adequate supplies of energy are needed not only to fuel industrial productivity, but also to supply safe drinking water and food for much of the world's population and to provide the sanitation necessary to control life-threatening diseases.

To appreciate the role heat transfer plays in the energy challenge, consider a flow chart that represents energy use in the United States, as shown in Figure 1.12*a*. Currently, about 61% of the 100 EJ of energy that is consumed annually in the United States is wasted in the form of heat. Nearly 70% of the energy used to generate electricity is lost in the form of heat. The transportation sector, which relies almost exclusively on petroleum-based fuels, utilizes only 21% of the energy it consumes; the remaining 79% is released in the form of

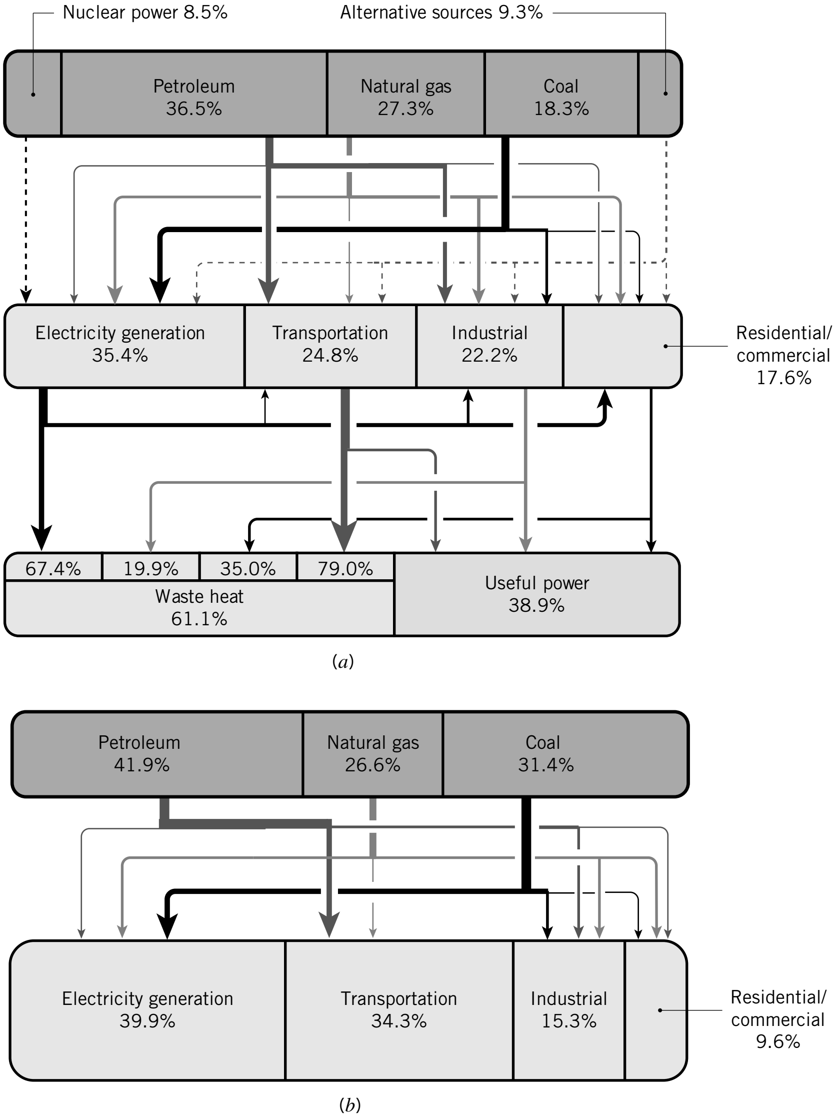


FIGURE 1.12 Flow charts for energy consumption and associated CO_2 emissions in the United States in 2012. (*a*) Energy production and consumption. (Based on data from U.S. Department of Energy and the Lawrence Livermore National Laboratory.) (*b*) Carbon dioxide by source of fossil fuel and end-use application [6]. Arrow widths represent relative magnitudes of the flow streams.

heat. Although the industrial and residential/commercial use of energy is relatively more efficient, opportunities for *energy conservation* abound. Creative thermal engineering, utilizing the tools of thermodynamics and heat transfer, can lead to new ways to (1) increase the efficiency by which energy is generated and converted, (2) reduce energy losses, and (3) harvest a large portion of the waste heat.

As evident in Figure 1.12*a*, *fossil fuels* (petroleum, natural gas, and coal) dominate the energy portfolio in many countries, such as the United States. The combustion of fossil fuels produces massive amounts of carbon dioxide; the amount of CO_2 released in the United States on an annual basis due to combustion is currently 5.07 Eg (5.07×10^{15} kg). As more CO_2 is pumped into the atmosphere, mechanisms of radiation heat transfer within the atmosphere are modified, resulting in potential changes in global temperatures. In a country like the United States, electricity generation and transportation are responsible for nearly 75% of the total CO_2 released into the atmosphere due to energy use (Figure 1.12*b*).

What are some of the ways engineers are applying the principles of heat transfer to address issues of energy and environmental *sustainability*?

The efficiency of a *gas turbine engine* can be significantly increased by increasing its operating temperature. Today, the temperatures of the combustion gases inside these engines far exceed the melting point of the exotic alloys used to manufacture the turbine blades and vanes. Safe operation is typically achieved by three means. First, relatively cool gases are injected through small holes at the leading edge of a turbine blade (Figure 1.13). These gases hug the blade as they are carried downstream and help insulate the blade from the hot combustion gases. Second, thin layers of a very low thermal conductivity, ceramic *thermal barrier coating* are applied to the blades and vanes to provide an extra layer of insulation. These coatings are produced by spraying molten ceramic powders onto the engine components using extremely high temperature sources such as plasma spray guns that can operate in excess of 10,000 kelvins. Third, the blades and vanes are designed with intricate, internal cooling passages, all carefully configured by the heat transfer engineer to allow the gas turbine engine to operate under such extreme conditions.

Alternative sources constitute a small fraction of the energy portfolio of many nations, as illustrated in the flow chart of Figure 1.12*a* for the United States. The intermittent nature of the power generated by sources such as the wind and solar irradiation limits their widespread utilization, and creative ways to store excess energy for use during low-power generation periods are urgently needed. For solar irradiation, the sun's rays can be concentrated and used to heat fluids to very high temperatures. The hot fluid can, in turn, be pumped to a power plant to drive a conventional steam turbine or other heat engine. If, however, some of the collected solar energy is stored during the day and retrieved at night, solar-generated electric power can be produced around the clock. Moreover, spreading the electric power generation over the entire day allows the same amount of electric energy to be produced each day, but at half the electric

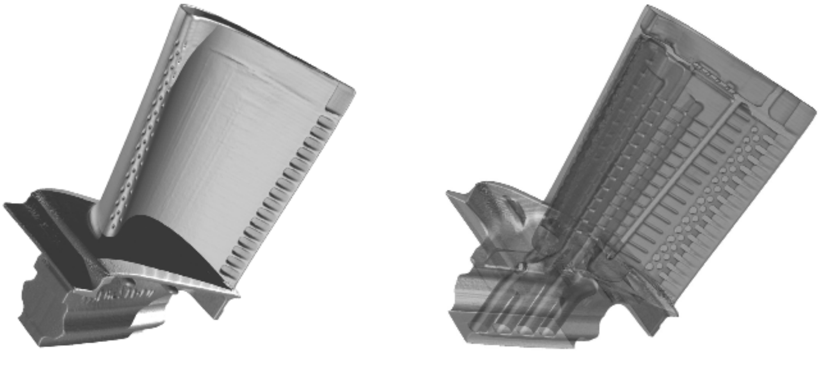

(*a*) (*b*)

FIGURE 1.13 Gas turbine blade. (*a*) External view showing holes for injection of cooling gases. (*b*) X ray view showing internal cooling passages. (Credit: Images courtesy of FarField Technology, Ltd., Christchurch, New Zealand.)

power rating. Hence, the size and cost of the power plant are reduced by a factor of approximately two with thermal energy storage. Transferring massive amounts of heat into and out of thermal energy storage devices requires novel heat transfer designs characterized by extremely low thermal resistances. Creation of such devices for thermal energy storage is a key factor to make solar electric power generation more efficient and affordable [7].

Due to the *information technology* revolution of the last several decades, strong industrial productivity growth has brought an improved quality of life worldwide. Many information technology breakthroughs have been enabled by advances in heat transfer engineering that have ensured the precise control of temperatures of systems ranging in size from nanoscale integrated circuits, to large data centers filled with heat-generating equipment. As electronic devices become faster and incorporate greater functionality, they generate more thermal energy. Simultaneously, the devices have become smaller. Inevitably, heat fluxes (W/m^2) and volumetric energy generation rates (W/m^3) keep increasing, but the operating temperatures of the devices must be held to reasonably low values to ensure their reliability. Further improvements in microprocessor technology are currently limited by our ability to cool these tiny devices [8]. Policy makers have voiced concern about limits to continually reduce the cost of computing and, in turn as a society, continue the growth in productivity that has marked the last 30 years, specifically citing the need to enhance heat transfer in electronics cooling [9]. How might your knowledge of heat transfer help ensure continued industrial productivity into the future?

Heat transfer is important not only in engineered systems but also in nature. Temperature regulates and triggers biological responses in all living systems and ultimately marks the boundary between sickness and health. Two common examples include *hypothermia*, which results from excessive cooling of the human body, and *heat stroke*, which is triggered in warm, humid environments. Both are deadly, and both are associated with core temperatures of the body exceeding physiological limits. Both are directly linked to the convection, radiation, and evaporation processes occurring at the surface of the body, the transport of heat within the body, and the metabolic energy generated volumetrically within the body.

Recent advances in *biomedical engineering*, such as laser surgery, have been enabled by successfully applying fundamental heat transfer principles [10, 11]. While increased temperatures resulting from contact with hot objects may cause thermal *burns*, beneficial *hyperthermal treatments* are used to purposely destroy, for example, cancerous lesions. In a similar manner, very low temperatures might induce *frostbite*, but purposeful localized freezing can selectively destroy diseased tissue during *cryosurgery*. Many medical therapies and devices therefore operate by destructively heating or cooling diseased tissue, while leaving the surrounding healthy tissue unaffected.

The ability to design many medical devices and to develop the appropriate protocol for their use hinges on the engineer's ability to predict and control the distribution of temperatures during thermal treatment and the distribution of chemical species in chemotherapies. The treatment of mammalian tissue is made complicated by its morphology, as shown in Figure 1.14. The flow of blood within the venular and capillary structure of a thermally treated area affects heat transfer through advection processes. Larger veins and arteries, which commonly exist in pairs throughout the body, carry blood at different temperatures and advect thermal energy at different rates. Therefore, the veins and arteries exist in a *counterflow heat exchange* arrangement with warm, arteriolar blood exchanging thermal energy with the cooler, venular blood through the intervening solid tissue. Networks of smaller capillaries can also affect local temperatures by *perfusing* blood through the treated area.

In subsequent chapters, example and homework problems will deal with the analysis of these and many other *thermal systems*.

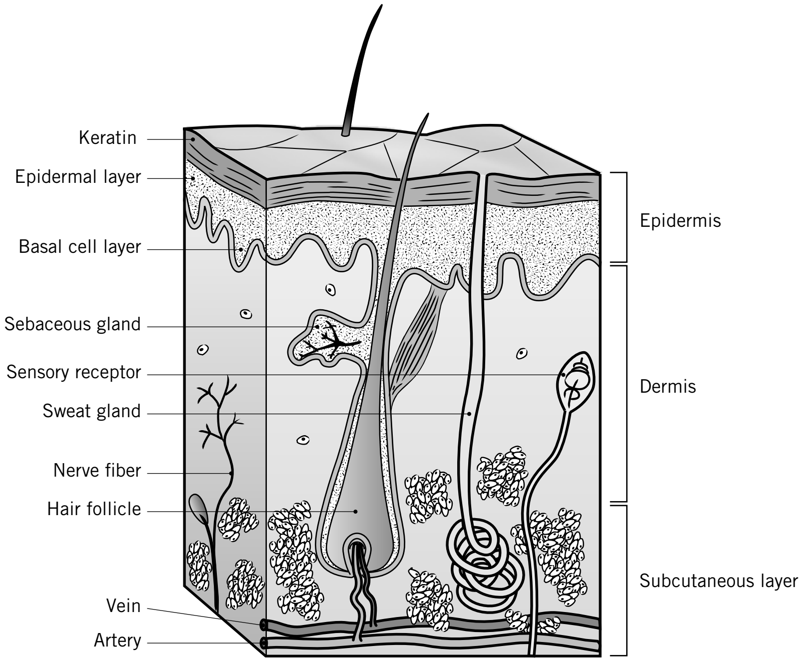


FIGURE 1.14 Morphology of human skin.

1.7 Summary

Although much of the material of this chapter will be discussed in greater detail, you should now have a reasonable overview of heat transfer. You should be aware of the several modes of transfer and their physical origins. You will be devoting much time to acquiring the tools needed to calculate heat transfer phenomena. However, before you can use these tools effectively, you must have the intuition to determine what is happening physically. Specifically, given a physical situation, you must be able to identify the relevant transport phenomena; the importance of developing this facility must not be underestimated. The example and problems at the end of this chapter will launch you on the road to developing this intuition.

You should also appreciate the significance of the rate equations and feel comfortable in using them to compute transport rates. These equations, summarized in Table 1.5, *should be committed to memory.* You must also recognize the importance of the conservation laws and the need to carefully identify control volumes. With the rate equations, the conservation laws may be used to solve numerous heat transfer problems.

Lastly, you should have begun to acquire an appreciation for the terminology and physical concepts that underpin the subject of heat transfer. Test your understanding of the important terms and concepts introduced in this chapter by addressing the following questions:

- What are the *physical mechanisms* associated with heat transfer by *conduction*, *convection*, and *radiation*?
- What is the driving potential for heat transfer? What are analogs to this potential and to heat transfer itself for the transport of electric charge?
- What is the difference between a heat *flux* and a heat *rate*? What are their units?

Table 1.5 Summary of heat transfer processes

Mode	Mechanism(s)	Rate Equation	Equation Number	Transport Property or Coefficient
Conduction	Diffusion of energy due to random molecular motion	$q''_x(\text{W/m}^2) = -k\dfrac{dT}{dx}$	(1.1)	k (W/m · K)
Convection	Diffusion of energy due to random molecular motion plus energy transfer due to bulk motion (advection)	$q''(\text{W/m}^2) = h(T_s - T_\infty)$	(1.3a)	h (W/m^2 · K)
Radiation	Energy transfer by electromagnetic waves	$q''(\text{W/m}^2) = \varepsilon\sigma(T_s^4 - T_{\text{sur}}^4)$ or $q(\text{W}) = h_r A(T_s - T_{\text{sur}})$	(1.7) (1.8)	ε h_r (W/m^2 · K)

- What is a *temperature gradient*? What are its units? What is the relationship of heat flow to a temperature gradient?
- What is the *thermal conductivity*? What are its units? What role does it play in heat transfer?
- What is *Fourier's law*? Can you write the equation from memory?
- If heat transfer by conduction through a medium occurs under *steady-state* conditions, will the temperature at a particular instant vary with location in the medium? Will the temperature at a particular location vary with time?
- What is the difference between *natural convection* and *forced convection*?
- What conditions are necessary for the development of a *hydrodynamic boundary layer*? A *thermal boundary layer*? What varies across a hydrodynamic boundary layer? Across a thermal boundary layer?
- If convection heat transfer for flow of a liquid or a vapor is not characterized by liquid/vapor phase change, what is the nature of the energy being transferred? What is it if there is such a phase change?
- What is *Newton's law of cooling*? Can you write the equation from memory?
- What role is played by the *convection heat transfer coefficient* in Newton's law of cooling? What are its units?
- What effect does convection heat transfer from or to a surface have on the solid bounded by the surface?
- What is predicted by the Stefan–Boltzmann law, and what unit of temperature must be used with the law? Can you write the equation from memory?
- What is the *emissivity*, and what role does it play in characterizing radiation transfer at a surface?
- What is *irradiation*? What are its units?
- What two outcomes characterize the response of an *opaque* surface to incident radiation? Which outcome affects the thermal energy of the medium bounded by the surface and how? What property characterizes this outcome?
- What conditions are associated with use of the *radiation heat transfer coefficient*?
- Can you write the equation used to express net radiation exchange between a small isothermal surface and a large isothermal enclosure?

- Consider the surface of a solid that is at an elevated temperature and exposed to cooler surroundings. By what mode(s) is heat transferred from the surface if (1) it is in intimate (perfect) contact with another solid, (2) it is exposed to the flow of a liquid, (3) it is exposed to the flow of a gas, and (4) it is in an evacuated chamber?
- What is the inherent difference between the application of conservation of energy over a *time interval* and at an *instant of time*?
- What is *thermal energy storage*? How does it differ from *thermal energy generation*? What role do the terms play in a surface energy balance?

EXAMPLE 1.9

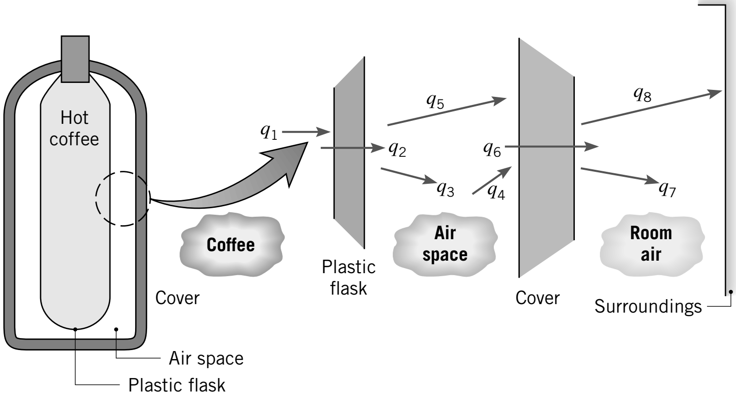
A closed plastic thermos filled with hot coffee is in a room whose air and walls are at a fixed temperature. Identify all heat transfer processes that contribute to the cooling of the coffee. Comment on features that would contribute to a superior container design.

SOLUTION

Known: Hot coffee is separated from its cooler surroundings by a plastic flask, an air space, and a plastic cover.

Find: Relevant heat transfer processes.

Schematic:



Pathways for energy transfer from the coffee are as follows:

q_1: free convection from the coffee to the flask.

q_2: conduction through the flask.

q_3: free convection from the flask to the air.

q_4: free convection from the air to the cover.

q_5: net radiation exchange between the outer surface of the flask and the inner surface of the cover.

q_6: conduction through the cover.

q_7: free convection from the cover to the room air.

q_8: net radiation exchange between the outer surface of the cover and the surroundings.

Comments: Design improvements are associated with (1) use of aluminized (low-emissivity) surfaces for the flask and cover to reduce net radiation, and (2) evacuating the air space or using a filler material to retard free convection.

References

1. Moran, M. J., and H. N. Shapiro, *Fundamentals of Engineering Thermodynamics*, Wiley, Hoboken, NJ, 2004.
2. Curzon, F. L., and B. Ahlborn, *American J. Physics*, **43**, 22, 1975.
3. Novikov, I. I., *J. Nuclear Energy II*, **7**, 125, 1958.
4. Callen, H. B., *Thermodynamics and an Introduction to Thermostatistics*, Wiley, Hoboken, NJ, 1985.
5. Bejan, A., *American J. Physics*, **64**, 1054, 1996.
6. U.S. Environmental Protection Agency, *Inventory of U.S. Greenhouse Emissions and Sinks:* 1990–2012, Publication EPA 430-R-14-003, 2014.
7. Shabgard, H., T. L. Bergman, and A. Faghri, *Energy*, **60**, 474, 2013.
8. Bar-Cohen, A., and I. Madhusudan, *IEEE Trans. Components and Packaging Tech.*, **25**, 584, 2002.
9. Miller, R., *Business Week*, November 11, 2004.
10. Diller, K. R., and T. P. Ryan, *J. Heat Transfer*, **120**, 810, 1998.
11. Datta, A.K., *Biological and Bioenvironmental Heat and Mass Transfer*, Marcel Dekker, New York, 2002.