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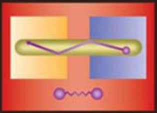
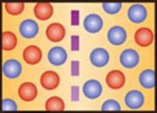
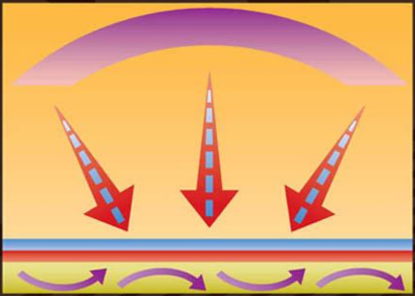




FRANK P. INCROPERA | DAVID P. DEWITT

FUNDAMENTALS OF HEAT and MASS TRANSFER

SEVENTH EDITION



Contents

Symbols

A	area, m^2
A_b	area of prime (unfinned) surface, m^2
A_c	cross-sectional area, m^2
A_p	fin profile area, m^2
A_r	nozzle area ratio
a	acceleration, m/s^2; speed of sound, m/s
Bi	Biot number
Bo	Bond number
C	molar concentration, $kmol/m^3$; heat capacity rate, W/K
C_D	drag coefficient
C_f	friction coefficient
C_t	thermal capacitance, J/K
Co	Confinement number
c	specific heat, J/kg · K; speed of light, m/s
c_p	specific heat at constant pressure, J/kg · K
c_v	specific heat at constant volume, J/kg · K
D	diameter, m
D_{AB}	binary mass diffusivity, m^2/s
D_b	bubble diameter, m
D_h	hydraulic diameter, m
d	diameter of gas molecule, nm
E	thermal plus mechanical energy, J; electric potential, V; emissive power, W/m^2
E^{tot}	total energy, J
Ec	Eckert number
$\dot{E}_g$	rate of energy generation, W
$\dot{E}_{in}$	rate of energy transfer into a control volume, W
$\dot{E}_{out}$	rate of energy transfer out of control volume, W
$\dot{E}_{st}$	rate of increase of energy stored within a control volume, W
e	thermal internal energy per unit mass, J/kg; surface roughness, m
F	force, N; fraction of blackbody radiation in a wavelength band; view factor
Fo	Fourier number
Fr	Froude number
f	friction factor; similarity variable
G	irradiation, W/m^2; mass velocity, $kg/s \cdot m^2$
Gr	Grashof number
Gz	Graetz number
g	gravitational acceleration, m/s^2
H	nozzle height, m; Henry's constant, bars
h	convection heat transfer coefficient, $W/m^2 \cdot K$; Planck's constant, J · s
h_{fg}	latent heat of vaporization, J/kg
h'_{fg}	modified heat of vaporization, J/kg
h_{sf}	latent heat of fusion, J/kg
h_m	convection mass transfer coefficient, m/s
h_{rad}	radiation heat transfer coefficient, $W/m^2 \cdot K$
I	electric current, A; radiation intensity, $W/m^2 \cdot sr$
i	electric current density, A/m^2; enthalpy per unit mass, J/kg
J	radiosity, W/m^2
Ja	Jakob number
J_i^*	diffusive molar flux of species i relative to the mixture molar average velocity, $kmol/s \cdot m^2$
j_i	diffusive mass flux of species i relative to the mixture mass average velocity, $kg/s \cdot m^2$
j_H	Colburn j factor for heat transfer
j_m	Colburn j factor for mass transfer
k	thermal conductivity, W/m · K
k_B	Boltzmann's constant, J/K
k_0	zero-order, homogeneous reaction rate constant, $kmol/s \cdot m^3$
k_1	first-order, homogeneous reaction rate constant, s^{-1}
k''_1	first-order, surface reaction rate constant, m/s
L	length, m
Le	Lewis number

M	mass, kg
$\dot{M}_i$	rate of transfer of mass for species, i, kg/s
$\dot{M}_{i,g}$	rate of increase of mass of species i due to chemical reactions, kg/s
$\dot{M}_{\text{in}}$	rate at which mass enters a control volume, kg/s
$\dot{M}_{\text{out}}$	rate at which mass leaves a control volume, kg/s
$\dot{M}_{\text{st}}$	rate of increase of mass stored within a control volume, kg/s
$\mathcal{M}_i$	molecular weight of species i, kg/kmol
Ma	Mach number
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
m_i	mass fraction of species i, ρ_i/ρ
N	integer number
N_L, N_T	number of tubes in longitudinal and transverse directions
Nu	Nusselt number
NTU	number of transfer units
N_i	molar transfer rate of species i relative to fixed coordinates, kmol/s
N_i''	molar flux of species i relative to fixed coordinates, kmol/s · m^2
$\dot{N}_i$	molar rate of increase of species i per unit volume due to chemical reactions, kmol/s · m^3
$\dot{N}_i''$	surface reaction rate of species i, kmol/s · m^2
$\mathcal{N}$	Avogadro's number
n_i''	mass flux of species i relative to fixed coordinates, kg/s · m^2
$\dot{n}_i$	mass rate of increase of species i per unit volume due to chemical reactions, kg/s · m^3
P	power, W; perimeter, m
P_L, P_T	dimensionless longitudinal and transverse pitch of a tube bank
Pe	Peclet number
Pr	Prandtl number
p	pressure, N/m^2
Q	energy transfer, J
q	heat transfer rate, W
$\dot{q}$	rate of energy generation per unit volume, W/m^3
q'	heat transfer rate per unit length, W/m
q''	heat flux, W/m^2
q^*	dimensionless conduction heat rate
R	cylinder radius, m; gas constant, J/kg · K
$\mathcal{R}$	universal gas constant, J/kmol · K
Ra	Rayleigh number
Re	Reynolds number
R_e	electric resistance, Ω
R_f	fouling factor, m^2 · K/W
R_m	mass transfer resistance, s/m^3
$R_{m,n}$	residual for the m, n nodal point
R_t	thermal resistance, K/W
$R_{t,c}$	thermal contact resistance, K/W
$R_{t,f}$	fin thermal resistance, K/W
$R_{t,o}$	thermal resistance of fin array, K/W
r_o	cylinder or sphere radius, m
r, ϕ, z	cylindrical coordinates
r, θ, ϕ	spherical coordinates
S	solubility, kmol/m^3 · atm; shape factor for two-dimensional conduction, m; nozzle pitch, m; plate spacing, m; Seebeck coefficient, V/K
S_c	solar constant, W/m^2
S_D, S_L, S_T	diagonal, longitudinal, and transverse pitch of a tube bank, m
Sc	Schmidt number
Sh	Sherwood number
St	Stanton number
T	temperature, K
t	time, s
U	overall heat transfer coefficient, W/m^2 · K; internal energy, J
u, v, w	mass average fluid velocity components, m/s
u^*, v^*, w^*	molar average velocity components, m/s
V	volume, m^3; fluid velocity, m/s
v	specific volume, m^3/kg
W	width of a slot nozzle, m
$\dot{W}$	rate at which work is performed, W
We	Weber number
X	vapor quality
X_{tt}	Martinelli parameter
X, Y, Z	components of the body force per unit volume, N/m^3
x, y, z	rectangular coordinates, m
x_c	critical location for transition to turbulence, m
$x_{\text{fd},c}$	concentration entry length, m
$x_{\text{fd},h}$	hydrodynamic entry length, m
$x_{\text{fd},t}$	thermal entry length, m
x_i	mole fraction of species i, C_i/C
Z	thermoelectric material property, K^{-1}

Greek Letters

α	thermal diffusivity, m^2/s; accommodation coefficient; absorptivity
β	volumetric thermal expansion coefficient, K^{-1}
Γ	mass flow rate per unit width in film condensation, kg/s · m
γ	ratio of specific heats
δ	hydrodynamic boundary layer thickness, m
δ_c	concentration boundary layer thickness, m
δ_p	thermal penetration depth, m
δ_t	thermal boundary layer thickness, m
ε	emissivity; porosity; heat exchanger effectiveness
ε_f	fin effectiveness
η	thermodynamic efficiency; similarity variable
η_f	fin efficiency
η_o	overall efficiency of fin array
θ	zenith angle, rad; temperature difference, K
κ	absorption coefficient, m^{-1}
λ	wavelength, μm
λ_{mfp}	mean free path length, nm

μ	viscosity, kg/s · m
ν	kinematic viscosity, m^2/s; frequency of radiation, s^{-1}
ρ	mass density, kg/m^3; reflectivity
ρ_e	electric resistivity, Ω/m
σ	Stefan–Boltzmann constant, $W/m^2 \cdot K^4$; electrical conductivity, $1/\Omega \cdot m$; normal viscous stress, N/m^2; surface tension, N/m
Φ	viscous dissipation function, s^{-2}
φ	volume fraction
ϕ	azimuthal angle, rad
ψ	stream function, m^2/s
τ	shear stress, N/m^2; transmissivity
ω	solid angle, sr; perfusion rate, s^{-1}

Subscripts

A, B	species in a binary mixture
abs	absorbed
am	arithmetic mean
atm	atmospheric
b	base of an extended surface; blackbody
C	carnot
c	cross-sectional; concentration; cold fluid; critical
cr	critical insulation thickness
cond	conduction
conv	convection
CF	counterflow
D	diameter; drag
dif	diffusion
e	excess; emission; electron
evap	evaporation
f	fluid properties; fin conditions; saturated liquid conditions
fc	forced convection
fd	fully developed conditions
g	saturated vapor conditions
H	heat transfer conditions
h	hydrodynamic; hot fluid; helical
i	general species designation; inner surface of an annulus; initial condition; tube inlet condition; incident radiation
L	based on characteristic length
l	saturated liquid conditions
lat	latent energy
lm	log mean condition
m	mean value over a tube cross section
max	maximum
o	center or midplane condition; tube outlet condition; outer
p	momentum
ph	phonon
R	reradiating surface
r, ref	reflected radiation
rad	radiation
S	solar conditions
s	surface conditions; solid properties; saturated solid conditions
sat	saturated conditions
sens	sensible energy
sky	sky conditions
ss	steady state
sur	surroundings
t	thermal
tr	transmitted
v	saturated vapor conditions
x	local conditions on a surface
λ	spectral
∞	free stream conditions

Superscripts

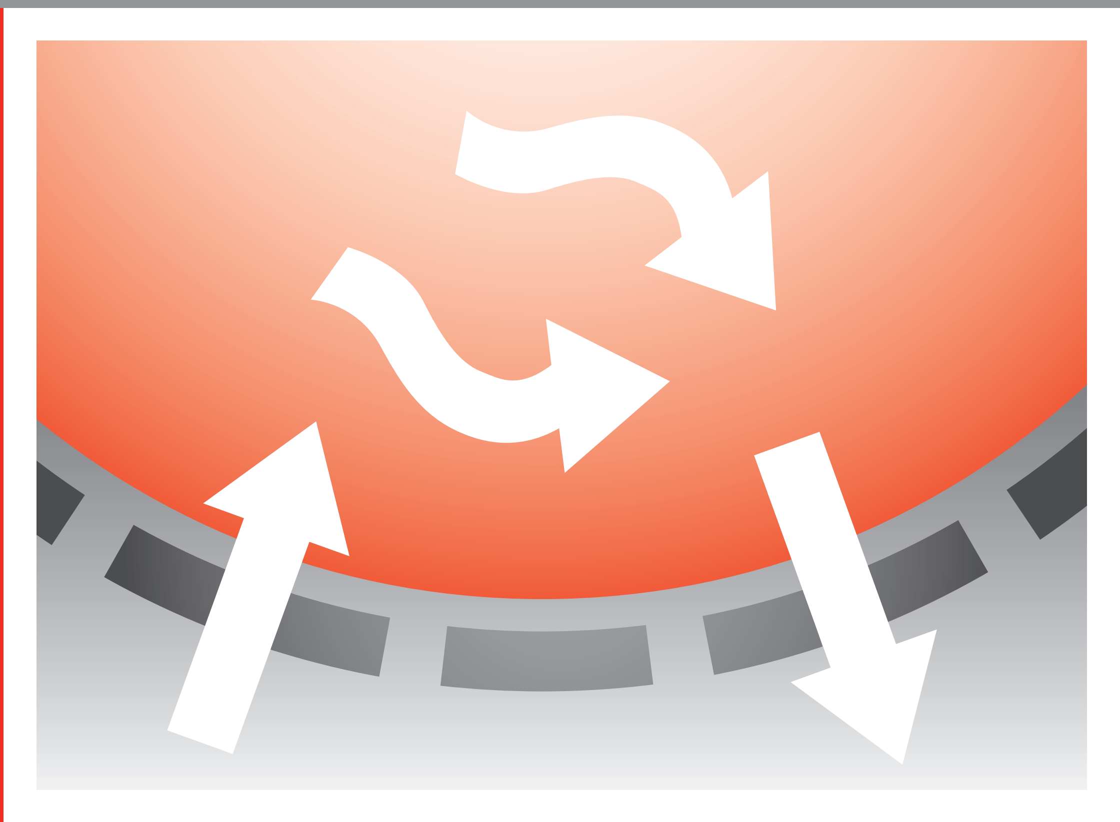
*	molar average; dimensionless quantity

Overbar

$\bar{\ }$	surface average conditions; time mean

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.

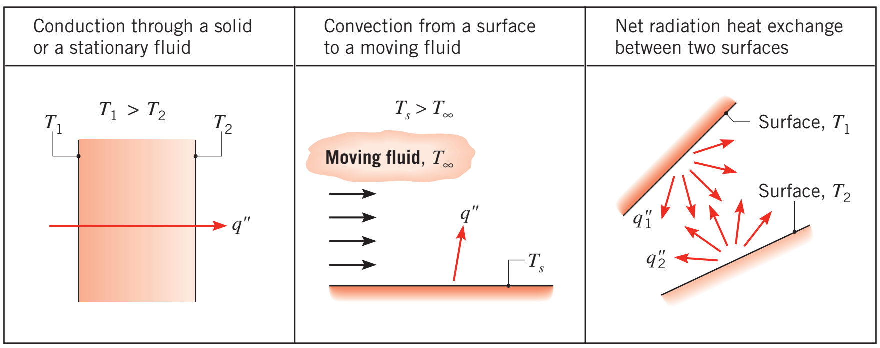


FIGURE 1.1 Conduction, convection, and radiation heat transfer modes.

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

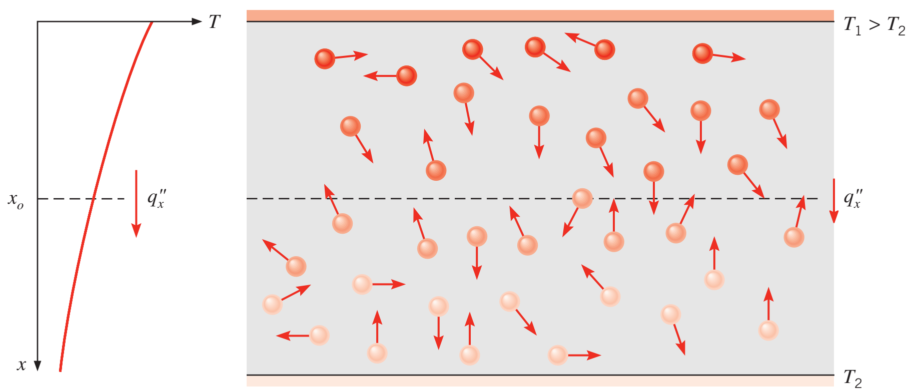


FIGURE 1.2 Association of conduction heat transfer with diffusion of energy due to molecular activity.

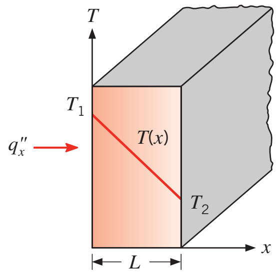


Figure 1.3 One-dimensional heat transfer by conduction (diffusion of energy).

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 *Fouriers 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 ux* 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 ux* , 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

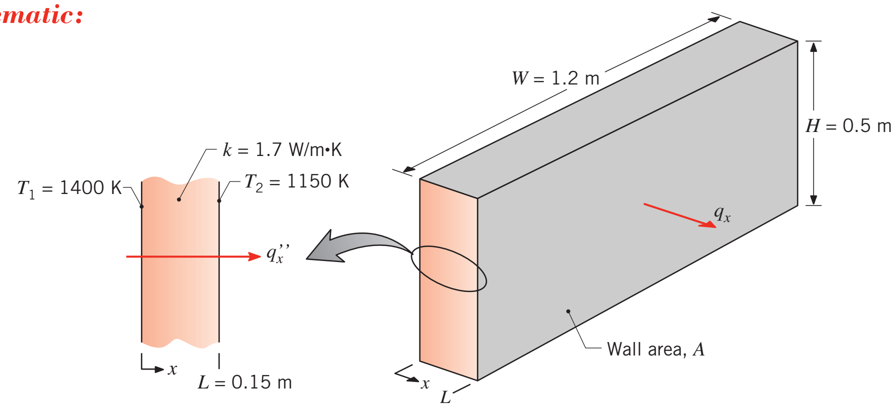
The wall of an industrial furnace is constructed from 0.15-m-thick fireclay brick having a thermal conductivity of 1.7 W/m·K. Measurements made during steady-state operation reveal temperatures of 1400 and 1150 K at the inner and outer surfaces, respectively. What is the rate of heat loss through a wall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

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 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.

We are especially interested in convection heat transfer, which occurs between a fluid in motion and a bounding surface when the two are at different temperatures. Consider fluid flow over the heated 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 a vertical array of circuit

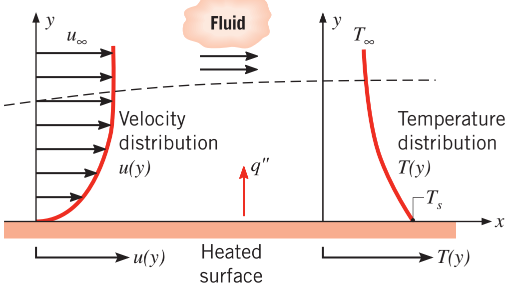


FIGURE 1.4 Boundary layer development in convection heat transfer.

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*).

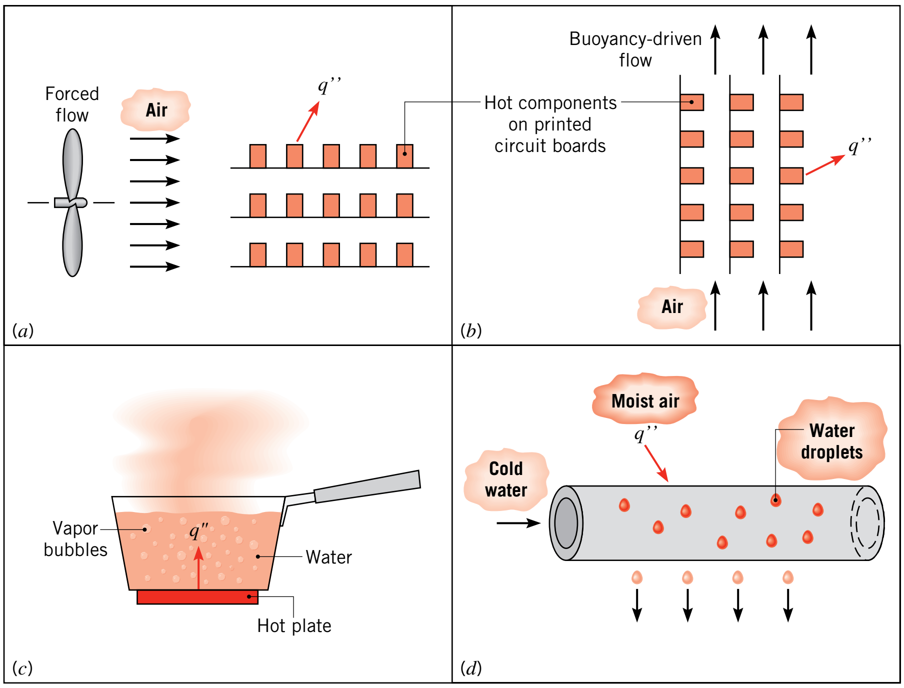


FIGURE 1.5 Convection heat transfer processes. (*a*) Forced convection. (*b*) Natural convection. (*c*) Boiling. (*d*) Condensation.

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 ux* (W/m^2), is proportional to the difference between the surface and fluid temperatures, T_s and T_∞, respectively. This expression is known as *Newtons law of cooling*, and the parameter h $(W/m^2 \cdot K)$ is termed the *convection heat transfer coef- cient.* 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 *StefanBoltzmann 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 $0 \leq \varepsilon \leq 1$, 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,

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

where $0 \leq \alpha \leq 1$. If $\alpha < 1$ and the surface is *opaque*, portions of the irradiation are *reected.* 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 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.

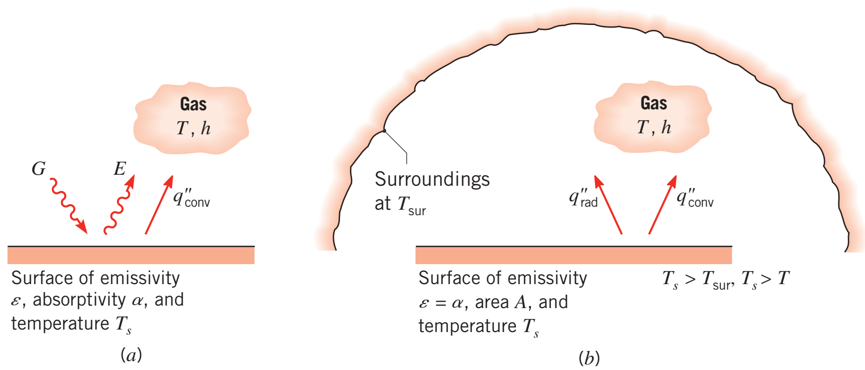


FIGURE 1.6 Radiation exchange: (*a*) at a surface and (*b*) between a surface and large surroundings.

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 ($T_{sur} \neq T_s$). 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 $G = \sigma T_{sur}^4$. 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

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

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

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

where, from Equation 1.7, the *radiation heat transfer coefcient h $_r$ is*

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

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

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

IHT

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?

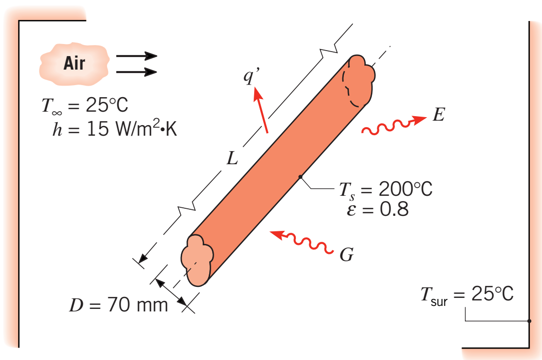
Solution

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. Pipe heat loss 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 $G = \sigma T_{\text{sur}}^4$. Hence

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

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

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

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

The heat loss per unit length of pipe is then

$$q' = \frac{q}{L} = 15\ \text{W/m}^2 \cdot \text{K}(\pi \times 0.07\ \text{m})(200 - 25)^\circ\text{C}$$

$$+\ 0.8(\pi \times 0.07\ \text{m})\ 5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4\,(473^4 - 298^4)\ \text{K}^4$$

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

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 $q_x = \Delta T/R_{t,c}$, where $R_{t,c} = L/kA$ 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*

The subjects of heat transfer and thermodynamics are highly complementary and interrelated, but they also have fundamental differences. 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 one of the primary mechanisms 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 often enable the engineer to implement the concepts of thermodynamics. For example, the actual 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.

The remainder of this section considers the relationship of heat transfer to thermodynamics. Since the *rst 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 *further limits* the actual efficiency of a heat engine.

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

At its heart, the first law of thermodynamics is simply a statement that the total energy of a system is conserved, and therefore the only way that the amount of energy in a system can change is if energy crosses its boundaries. The first law also addresses the ways in which energy can cross the boundaries of a system. For a closed system (a region of fixed mass), there are only two ways: 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:

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

where ΔE_{st}^{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 1.7*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 pass. 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 first law of thermodynamics can be very simply stated as follows for both a control volume and a closed system.

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.

In applying this principle, it is recognized that energy can enter and leave the control volume due to heat transfer through the boundaries, work done on or by the control volume, and energy advection.

The first law of thermodynamics addresses *total* energy, which consists of kinetic and potential energies (together known as mechanical energy) and internal energy. Internal energy can be further subdivided into thermal energy (which will be defined more carefully later)

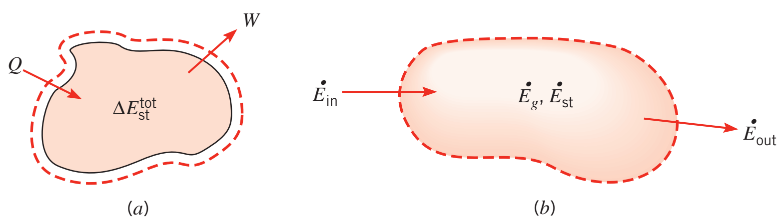


FIGURE 1.7 Conservation of energy: (*a*) for a closed system over a time interval and (*b*) for a control volume at an instant.

and other forms of internal energy, such as chemical and nuclear energy. For the study of heat transfer, we wish to focus 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, if a chemical reaction occurs that decreases the amount of chemical energy in the system, it will result in an increase in the thermal energy of the system. 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:

Thermal and Mechanical Energy Equation over a Time Interval (Δt)

The increase in the amount of thermal and mechanical energy stored in the 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 each and 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 follows:

Thermal and Mechanical Energy Equation at an Instant (t)

The rate of increase of thermal and mechanical energy stored in the 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 converse is true, thermal and mechanical energy storage 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 define symbols for each of the energy terms so that the boxed statements can be rewritten as equations. We let E stand for the sum of thermal and mechanical energy (in contrast to the symbol E^{tot} for total energy). Using the subscript *st* to denote energy stored in the control volume, 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, the first boxed statement 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, the second boxed statement becomes:

$$E_{\text{st}} \equiv \frac{dE_{\text{st}}}{dt} = E_{\text{in}} - E_{\text{out}} + E_g \tag{1.12c}$$

This expression is illustrated schematically in Figure 1.7*b*.

Equations 1.12b,c provide important and, in some cases, essential tools for solving heat transfer problems. 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 following energy terms:

- Stored thermal and mechanical energy, E_{st}.
- Thermal and mechanical energy generation, E_g.
- Thermal and mechanical energy transport across the control surfaces, that is, the inflow and outflow terms, E_{in} and E_{out}.

In the statement of the first law (Equation 1.12a), the total energy, E^{tot}, consists of kinetic energy (KE $= \frac{1}{2}mV^2$, where m and V are mass and velocity, respectively), potential energy (PE $= mgz$, where g is the gravitational acceleration and z is the vertical coordinate), and *internal energy* (U). Mechanical energy is defined as the sum of kinetic and potential energy. Most often in heat transfer problems, the changes in kinetic and potential energy are small and can be neglected. The internal energy consists of a *sensible component*, which accounts for the translational, rotational, and/or vibrational motion of the atoms/molecules comprising the matter; a *latent component*, which relates to intermolecular forces influencing phase change between solid, liquid, and vapor states; a *chemical component*, which accounts for energy stored in the chemical bonds between atoms; and a *nuclear component*, which accounts for the binding forces in the nucleus.

For the study of heat transfer, we focus attention on the sensible and latent components of the internal energy (U_{sens} and U_{lat}, respectively), which are together referred to as *thermal energy*, U_t. The sensible energy is the portion that we associate mainly with changes in temperature (although it can also depend on pressure). The latent energy is the component we associate with changes in phase. For example, 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 (*solidication, freezing*), the latent energy decreases. Obviously, if no phase change is occurring, there is no change in latent energy, and this term can be neglected.

Based on this discussion, the *stored thermal and mechanical energy* is given by $E_{\text{st}} =$ KE + PE + U_t, where $U_t = U_{\text{sens}} + U_{\text{lat}}$. In many problems, the only relevant energy term will be the sensible energy, that is, $E_{\text{st}} = U_{\text{sens}}$.

The *energy generation term* is associated with conversion from some other form of internal energy (chemical, electrical, electromagnetic, or nuclear) to thermal or mechanical energy. It is a *volumetric phenomenon*. That is, it occurs within the control volume and is generally proportional to the magnitude of this volume. For example, an exothermic chemical reaction may be occurring, converting chemical energy to thermal energy. The net effect is an increase in the thermal energy of the matter within the control volume. Another source of thermal energy 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. In all applications of interest in this text, if chemical, electrical, or nuclear effects exist, they are treated as sources (or *sinks*, which correspond to negative sources) of thermal or mechanical energy and hence are included in the generation terms of Equations 1.12b,c.

The inflow and outflow terms are *surface phenomena*. That is, they are associated exclusively with processes occurring at the control surface and are generally proportional to the surface area. As discussed previously, the energy inflow and outflow terms include heat transfer (which can be by conduction, convection, and/or radiation) 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.

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 *ow work* , is associated with work done by pressure forces moving fluid through the boundary. For a *unit mass*, the amount of work is equivalent to the product of the pressure and the specific volume of the fluid (pv). The symbol W is traditionally used for the rate at which the remaining work (not including flow work) is perfomed. If operation is under steady-state conditions ($dE_{st}/dt = 0$) and if there is no thermal or mechanical energy generation, Equation 1.12c reduces to the following form of the steady-flow energy equation (see Figure 1.8), which will be familiar if you have taken a thermodynamics course:

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

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 the 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 specic heats* , the difference in enthalpies (per unit mass) between the inlet and outlet flows may then be expressed as $(i_{in} - i_{out}) = c_p(T_{in} - T_{out})$, where c_p is

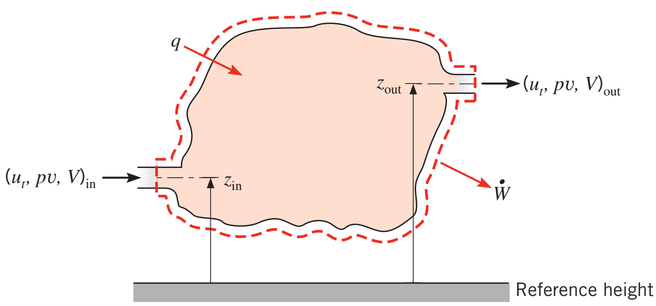


FIGURE 1.8 Conservation of energy for a steady-flow, open system.

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.

Having already assumed steady-state conditions, no changes in latent energy, and no thermal or mechanical energy generation, there are at least four cases in which further assumptions can be made to reduce Equation 1.12d to the *simplied steady-ow thermal energy equation*:

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

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.

The first two cases for which Equation 1.12e holds can readily be verified by examining Equation 1.12d. They are:

1. An ideal gas with negligible kinetic and potential energy changes and negligible work (other than flow work).
2. An incompressible liquid with negligible kinetic and potential energy changes and negligible work, *including* flow work. As noted in the preceding discussion, flow work is negligible for an incompressible liquid provided the pressure variation is not too great.

The second pair of cases cannot be directly derived from Equation 1.12d but require further knowledge of how mechanical energy is converted into thermal energy. These cases are:

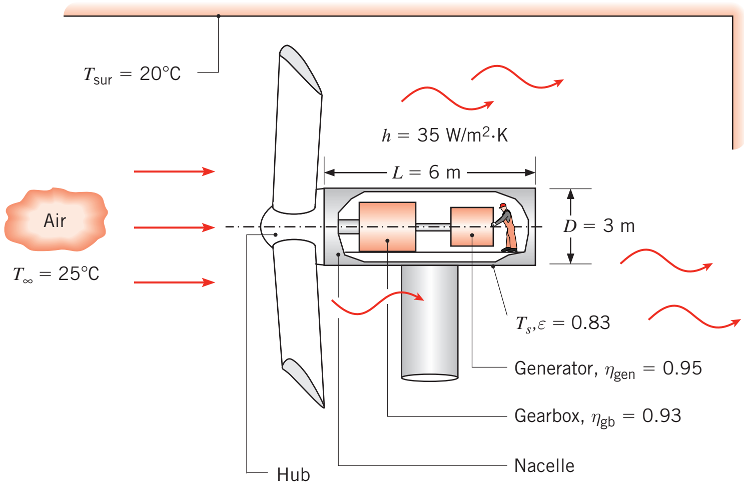
3. An ideal gas with negligible viscous dissipation and negligible pressure variation.
4. An incompressible liquid with negligible viscous dissipation.

Viscous dissipation is the conversion from mechanical energy to thermal energy associated with viscous forces acting in a fluid. It is important only in cases involving high-speed flow and/or highly viscous fluid. Since so many engineering applications satisfy one or more of the preceding four conditions, Equation 1.12e is commonly used for the analysis of heat transfer in moving fluids. It will be used in Chapter 8 in the study of convection heat transfer in internal flow.

The *mass ow rate* m of the fluid may be expressed as $m = \rho V A_c$, where ρ is the fluid density and A_c is the cross-sectional area of the channel through which the fluid flows. The *volumetric ow rate* is simply $\forall = VA_c = m/\rho$.

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 $\eta_{\text{gb}} = 0.93$. In turn, the gearbox output shaft drives an electric generator with an efficiency of $\eta_{\text{gen}} = 0.95$. 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 $T_{\text{sur}} = 20°C$, respectively, determine the minimum possible operating temperature inside the nacelle. The emissivity of the nacelle is $\varepsilon = 0.83$, and the convective heat transfer coefficient is $h = 35\ \mathrm{W/m^2 \cdot K}$. The surface of the nacelle that is adjacent to the blade hub can be considered to be adiabatic, and solar irradiation may be neglected.

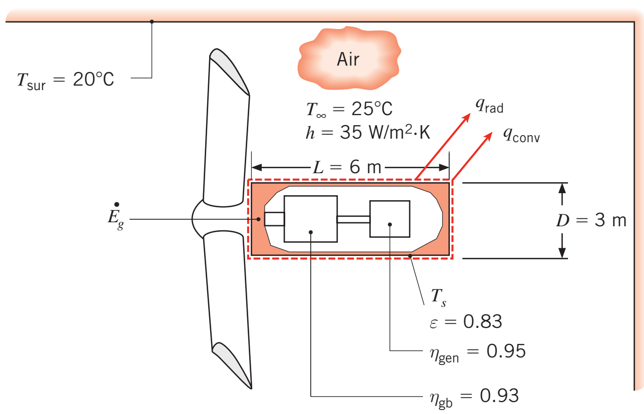


Solution

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: Minimum possible temperature inside the enclosed nacelle.

Schematic:



Assumptions:

1. Steady-state conditions.
2. Large surroundings.
3. Surface of the nacelle that is adjacent to the hub is adiabatic.

Analysis: The nacelle temperature represents the minimum possible temperature inside the nacelle, and the first law of thermodynamics may be used to determine this temperature. 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:

$$\frac{dE_{st}^{tot}}{dt} = E_{in}^{tot} - E_{out}^{tot} \tag{1}$$

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

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

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

$$E_{in} - E_{out} + E_g = 0 \tag{3}$$

Here, E_{in} once again corresponds to the mechanical work W. However, E_{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, $E_g = -P$. Thus, Equation (3) becomes

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

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:

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

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

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

Equation (5) can therefore be written as

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

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 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

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

We know that 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. Therefore, T_s represents the minimum possible temperature inside the enclosed nacelle. ◁

Comments:

1. The temperature inside the nacelle is very high. This 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.
2. Improvements in the efficiencies of either the gearbox or the generator would not only provide more electrical power, but would also 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.
3. 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

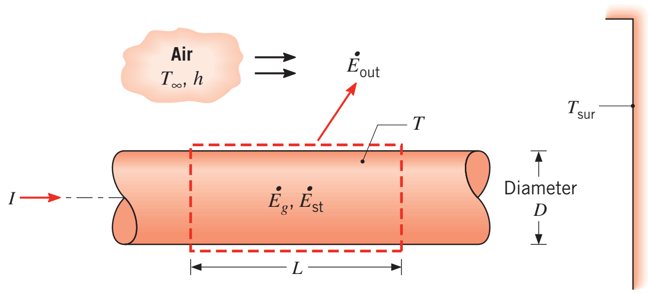
A long conducting rod of diameter D and electrical resistance per unit length R'_e is initially in thermal equilibrium with the ambient air and its surroundings. This equilibrium is disturbed when an electrical current I is passed through the rod. Develop an equation that could be used to compute the 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, $\varepsilon = \alpha$).
3. Radiation exchange between the outer surface of the rod and the surroundings is between a small surface and a large enclosure.

Analysis: The first law of thermodynamics may often be used to determine an unknown temperature. In this case, there is no mechanical energy component. So relevant 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

$$E_g - E_{out} = E_{st}$$

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

$$E_g = I^2 R'_e L$$

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

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

and the change in energy storage is due to the temperature change,

$$E_{st} = \frac{dU_t}{dt} = \frac{d}{dt}(\rho V c T)$$

The term E_{st} is associated with the rate of change in the internal thermal energy of the rod, where ρ and c 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

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

Hence

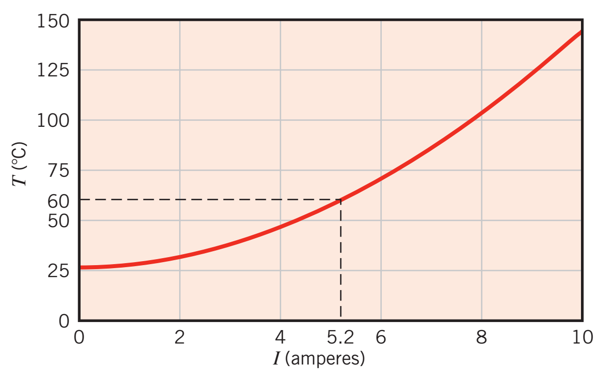
$$\frac{dT}{dt} = \frac{I^2R'_e - \pi Dh(T - T_\infty) - \pi D\varepsilon\sigma(T^4 - T^4_{\text{sur}})}{\rho c(\pi D^2/4)} \qquad \triangleleft$$

Comments:

1. 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

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

2. 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, ε = 0.8, R'_e = 0.4 Ω/m) in a relatively large enclosure (T_{sur} = 300 K) through which cooling air is circulated (h = 100 W/m$^2 \cdot$K, T_∞ = 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:

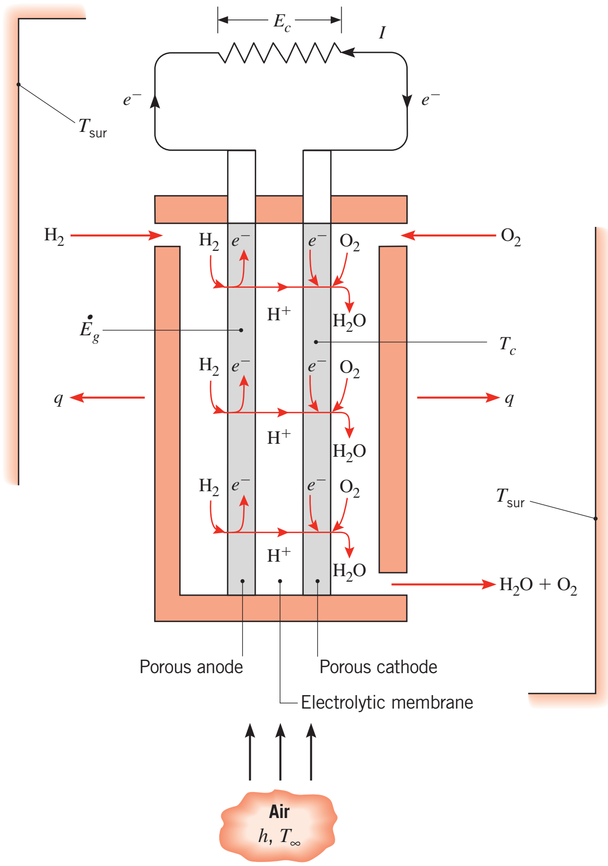


3. If a maximum operating temperature of T = 60°C is prescribed for safety reasons, the current should not exceed 5.2 A. At this temperature, heat transfer by radiation (0.6 W/m) is much less than heat transfer by convection (10.4 W/m). Hence, if one wished to operate at a larger current while maintaining 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 W/m$^2 \cdot$K, the maximum allowable current could be increased to 8.1 A.

4. 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

A hydrogen-air Proton Exchange Membrane (PEM) fuel cell is illustrated below. It consists of an *electrolytic membrane* sandwiched between porous *cathode* and *anode* materials, forming a very thin, three-layer *membrane electrode assembly* (MEA). At the anode, protons and electrons are generated ($2H_2 \rightarrow 4H^+ + 4e^-$); at the cathode, the protons and electrons recombine to form water ($O_2 + 4e^- + 4H^+ \rightarrow 2H_2O$). The overall reaction is then $2H_2 + O_2 \rightarrow 2H_2O$. The dual role of the electrolytic membrane is to transfer hydrogen ions and serve as a barrier to electron transfer, forcing the electrons to the electrical load that is external to the fuel cell.



The membrane must operate in a moist state in order to conduct ions. However, the presence of liquid water in the cathode material may block the oxygen from reaching the cathode reaction sites, resulting in the failure of the fuel cell. Therefore, it is critical to control the temperature of the fuel cell, T_c, so that the cathode side contains saturated water vapor.

For a given set of H_2 and air inlet flow rates and use of a 50 mm × 50 mm MEA, the fuel cell generates $P = I \cdot E_c = 9$ W of electrical power. Saturated vapor conditions exist in the fuel cell, corresponding to $T_c = T_{sat} = 56.4°C$. The overall electrochemical reaction is exothermic, and the corresponding thermal generation rate of $\dot{E}_g = 11.25$ W must be removed from the fuel cell by convection and radiation. The ambient and surrounding temperatures are $T_\infty = T_{sur} = 25°C$, and the relationship between the cooling air velocity and the convection heat transfer coefficient h is

$$h = 10.9 \text{ W} \cdot \text{s}^{0.8}/\text{m}^{2.8} \cdot \text{K} \times V^{0.8}$$

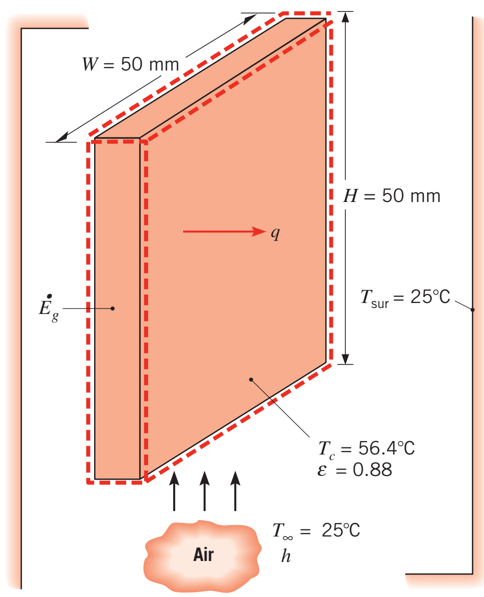
where V has units of m/s. The exterior surface of the fuel cell has an emissivity of $\varepsilon = 0.88$. Determine the value of the cooling air velocity needed to maintain steady-state operating conditions. Assume the edges of the fuel cell are well insulated.

Solution

Known: Ambient and surrounding temperatures, fuel cell output voltage and electrical current, heat generated by the overall electrochemical reaction, and the desired fuel cell operating temperature.

Find: The required cooling air velocity V needed to maintain steady-state operation at $T_c \approx 56.4°C$.

Schematic:



Assumptions:

1. Steady-state conditions.
2. Negligible temperature variations within the fuel cell.
3. Fuel cell is placed in large surroundings.
4. Edges of the fuel cell are well insulated.
5. Negligible energy entering or leaving the control volume due to gas or liquid flows.

Analysis: To determine the required cooling air velocity, we must first perform an energy balance on the fuel cell. Noting that there is no mechanical energy component, we see that $E_{\text{in}} = 0$ and $E_{\text{out}} = E_g$. This yields

$$q_{\text{conv}} + q_{\text{rad}} = E_g = 11.25 \text{ W}$$

where

$$\begin{aligned} q_{\text{rad}} &= \varepsilon A \sigma (T_c^4 - T_{\text{sur}}^4) \\ &= 0.88 \times (2 \times 0.05 \text{ m} \times 0.05 \text{ m}) \times 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 \times (329.4^4 - 298^4) \text{ K}^4 \\ &= 0.97 \text{ W} \end{aligned}$$

Therefore, we may find

$$\begin{aligned} q_{\text{conv}} &= 11.25 \text{ W} - 0.97 \text{ W} = 10.28 \text{ W} \\ &= hA(T_c - T_\infty) \\ &= 10.9 \text{ W} \cdot \text{s}^{0.8}/\text{m}^{2.8} \cdot \text{K} \times V^{0.8} A(T_c - T_\infty) \end{aligned}$$

which may be rearranged to yield

$$V = \left[\frac{10.28 \text{ W}}{10.9 \text{ W} \cdot \text{s}^{0.8}/\text{m}^{2.8} \cdot \text{K} \times (2 \times 0.05 \text{ m} \times 0.05 \text{ m}) \times (56.4 - 25^\circ\text{C})} \right]^{1.25}$$

$$V = 9.4 \text{ m/s} \qquad \triangleleft$$

Comments:

1. Temperature and humidity of the MEA will vary from location to location within the fuel cell. Prediction of the *local* conditions within the fuel cell would require a more detailed analysis.
2. The required cooling air velocity is quite high. Decreased cooling velocities could be used if heat transfer enhancement devices were added to the exterior of the fuel cell.
3. The convective heat rate is significantly greater than the radiation heat rate.
4. The chemical energy (20.25 W) of the hydrogen and oxygen is converted to electrical (9 W) and thermal (11.25 W) energy. This fuel cell operates at a conversion efficiency of (9 W)/(20.25 W) × 100 = 44%.

IHT

EXAMPLE 1.6

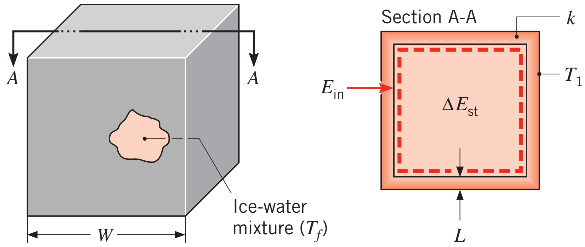
Large PEM fuel cells, such as those used in automotive applications, often require internal cooling using pure liquid water to maintain their temperature at a desired level (see Example 1.5). In cold climates, the cooling water must be drained from the fuel cell to an adjoining container when the automobile is turned off so that harmful freezing does not occur within the fuel cell. Consider a mass M of ice that was frozen while the automobile was not being operated. The ice is at the fusion temperature ($T_f = 0°\text{C}$) and 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 and, in turn, deliver cooling water to, and energize, the fuel cell.

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

$$\Delta E_{st} = Mh_{sf}$$

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 $\rho_s = 920$ 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 1.9). Accordingly, the generation and storage terms of the conservation

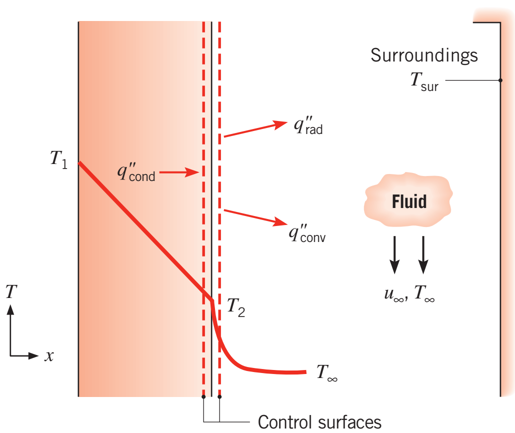


FIGURE 1.9 The energy balance for conservation of energy at the surface of a medium.

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

$$E_{in} - E_{out} = 0 \tag{1.13}$$

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 1.9, 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.

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

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

IHT

Example 1.7

Humans are able to control their heat production rate and heat loss rate to maintain a nearly constant core temperature of $T_c = 37°C$ 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, $T_i = 35°C = 308$ K. Consider a person with a skin/fat layer of thickness $L = 3$ mm and effective thermal conductivity $k = 0.3$ W/m·K. The person has a surface area $A = 1.8$ m^2 and is dressed in a bathing suit. The emissivity of the skin is $\varepsilon = 0.95$.

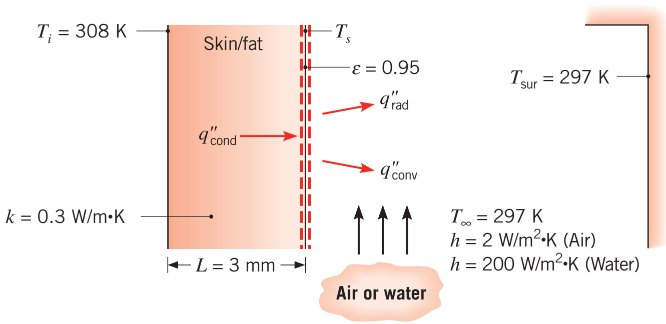
1. When the person is in still air at $T_\infty = 297$ K, what is the skin surface temperature and rate of heat loss to the environment? Convection heat transfer to the air is characterized by a free convection coefficient of $h = 2$ W/m^2·K.
2. When the person is in water at $T_\infty = 297$ K, what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

$$E_{in} - E_{out} = 0$$

It follows that, on a unit area basis,

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

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

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

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:

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

Solving for T_s, with $T_{sur} = T_\infty$, 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 $h_r = 5.9\ \mathrm{W/m^2 \cdot K}$. Then, substituting numerical values into the preceding equation, we find

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

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

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

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

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

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

and

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

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, heat losses 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 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 *nite* 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 are interested in the efficiency of heat engines. The discussion builds on your knowledge of thermodynamics and shows 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 3.8). Improving the efficiency of heat engines is a subject of extreme 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 *KelvinPlanck 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, gaining 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 $W = Q_{in} - Q_{out}$, 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

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

The second law also tells us that, for a *reversible* process, the ratio Q_{out}/Q_{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 efciency* 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 a lower efficiency.

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 1.10). 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,

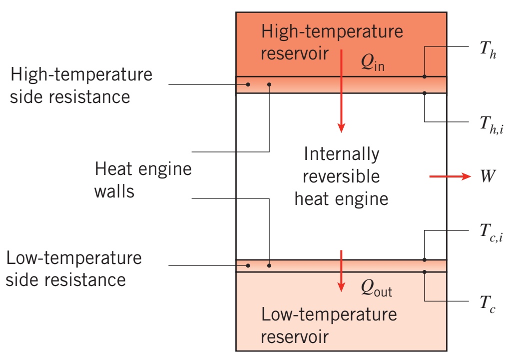


FIGURE 1.10 Internally reversible heat engine exchanging heat with high- and low-temperature reservoirs through thermal resistances.

the Carnot efficiency is *now based on the internal 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 *ows* 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

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

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

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

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

[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).

the power output W is zero according to Equation 1.22. To increase 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 thermodyamic 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 1.8

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.

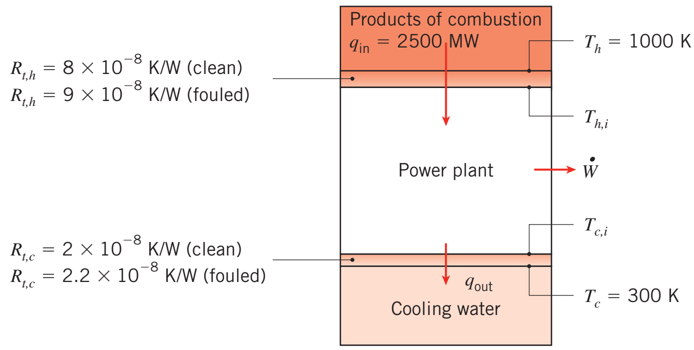
Solution

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 side 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

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

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

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

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 $W = q_{\text{in}}\eta_C = 2500\ \text{MW} \times 0.70 = 1750\ \text{MW}$. 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/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 recent years, there has been a strong trend toward the global usage of a standard set of units. In 1960, the SI (Systme International dUnits) system of units was dened 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 on the inside back cover of the text.

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 newtons, where a 1-N force will accelerate a 1-kg mass at 1 m/s^2. Hence 1 N = 1 kg · m/s^2. The unit of pressure (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 J = 1 N · 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.

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.

TABLE 1.3 SI derived units for selected quantities

Quantity	Name and Symbol	Formula	Expression in SI Base Units
Force	newton (N)	m · kg/s^2	m · kg/s^2
Pressure and stress	pascal (Pa)	N/m^2	kg/m · s^2
Energy	joule (J)	N · m	m^2 · kg/s^2
Power	watt (W)	J/s	m^2 · 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

Prex	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}

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 1.9

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 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 $2 \leq h \leq 200\ W/m^2 \cdot K$. What value of h would provide a cure temperature of 50°C?

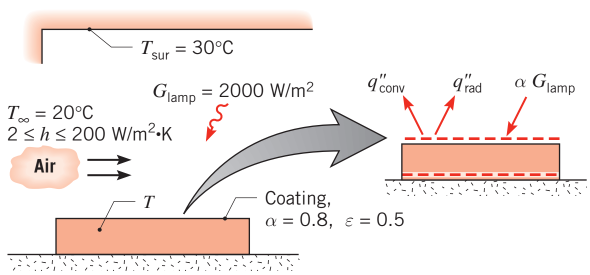
SOLUTION

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 $2 \leq h \leq 200\ W/m^2 \cdot K$. 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

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

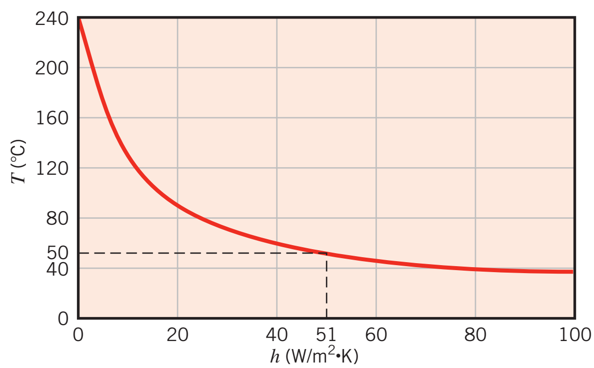
Substituting numerical values

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

and solving by trial-and-error, we obtain

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

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

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

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 $h = 2\ \mathrm{W/m^2 \cdot K}$, $T = 204°\mathrm{C}$ and radiation dominates ($q''_{rad} \approx 1232\ \mathrm{W/m^2}$, $q''_{conv} \approx 368\ \mathrm{W/m^2}$). Conversely, for $h = 200\ \mathrm{W/m^2 \cdot K}$, $T = 28°\mathrm{C}$ and convection dominates ($q''_{conv} \approx 1606\ \mathrm{W/m^2}$, $q''_{rad} \approx -6\ \mathrm{W/m^2}$). 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 1.11*a*. Currently, about 58% of the nearly 110 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 21.5% of the energy it consumes; the remaining 78.5% is released in the form of 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 1.11*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 5.99 Eg (5.99×10^{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 1.11*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

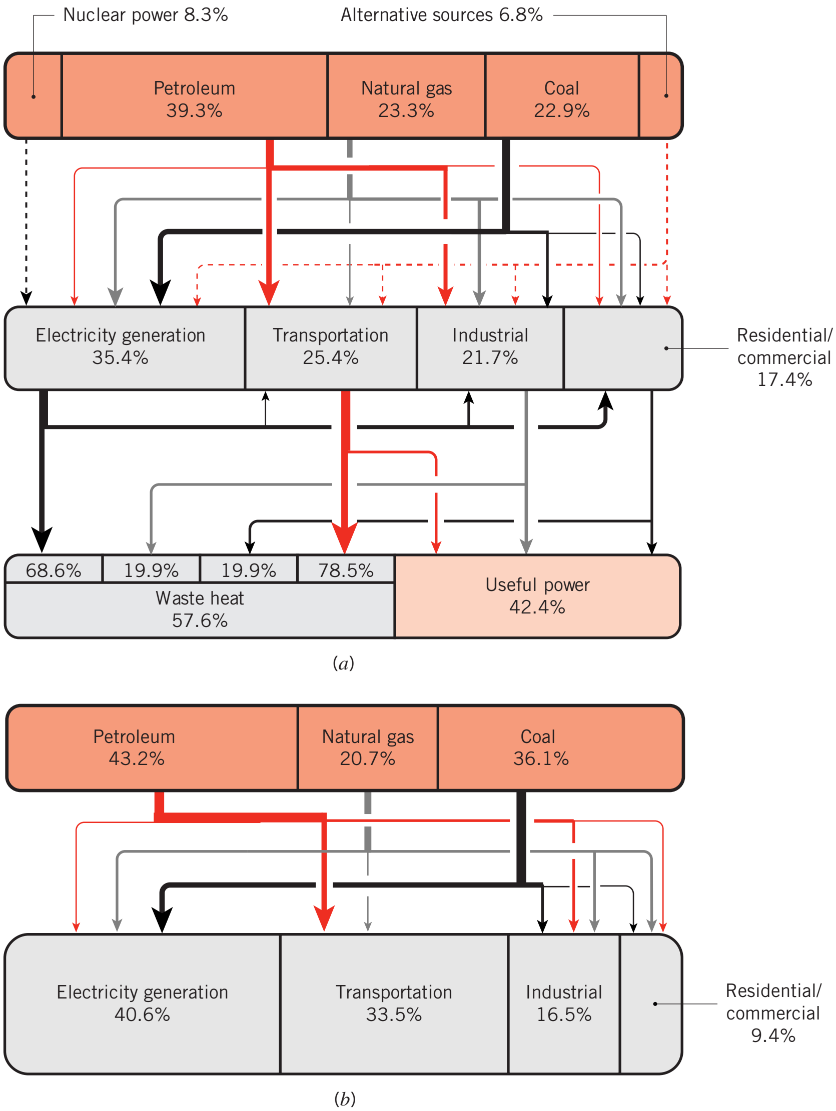


FIGURE 1.11 Flow charts for energy consumption and associated CO_2 emissions in the United States in 2007. (*a*) Energy production and consumption. (*b*) Carbon dioxide by source of fossil fuel and end-use application. Arrow widths represent relative magnitudes of the flow streams. (Credit: U.S. Department of Energy and the Lawrence Livermore National Laboratory.)

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 1.12). 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

(*a*) (*b*)

Figure 1.12 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.)

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 1.11*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. Emerging energy conversion devices such as *fuel cells* could be used to (1) combine excess electricity that is generated during the day (in a solar power station, for example) with liquid water to produce hydrogen, and (2) subsequently convert the stored hydrogen at night by recombining it with oxygen to produce electricity and water. Roadblocks hindering the widespread use of hydrogen fuel cells are their size, weight, and limited durability. As with the gas turbine engine, the efficiency of a fuel cell increases with temperature, but high operating temperatures and large temperature gradients can cause the delicate polymeric materials within a hydrogen fuel cell to fail.

More challenging is the fact that water exists inside any hydrogen fuel cell. If this water should freeze, the polymeric materials within the fuel cell would be destroyed, and the fuel cell would cease operation. Because of the necessity to utilize very pure water in a hydrogen fuel cell, common remedies such as antifreeze cannot be used. What heat transfer mechanisms must be controlled to avoid freezing of pure water within a fuel cell located at a wind farm or solar energy station in a cold climate? How might your developing knowledge of internal forced convection, evaporation, or condensation be applied to control the operating temperatures and enhance the durability of a fuel cell, in turn promoting more widespread use of solar and wind power?

Due to the *information technology* revolution of the last two 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 microscale storage media including compact discs, 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.

For *personal computers*, cooling fins (also known as *heat sinks*) are fabricated of a high thermal conductivity material (usually aluminum) and attached to the microprocessors to reduce their operating temperatures, as shown in Figure 1.13. Small fans are used to induce forced convection over the fins. The cumulative energy that is consumed worldwide, just to (1) power the small fans that provide the airflow over the fins and (2) manufacture the heat sinks for personal computers, is estimated to be over 10^9 kW·h per year [6]. How might your knowledge of conduction, convection, and radiation be used to, for example, eliminate the fan and minimize the size of the heat sink?

Further improvements in microprocessor technology are currently limited by our ability to cool these tiny devices. Policy makers have voiced concern about our ability 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 [7]. 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 [8, 9]. 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

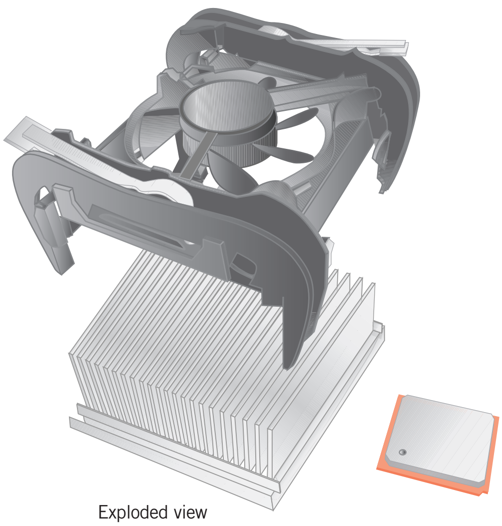


FIGURE 1.13 A finned heat sink and fan assembly (left) and microprocessor (right).

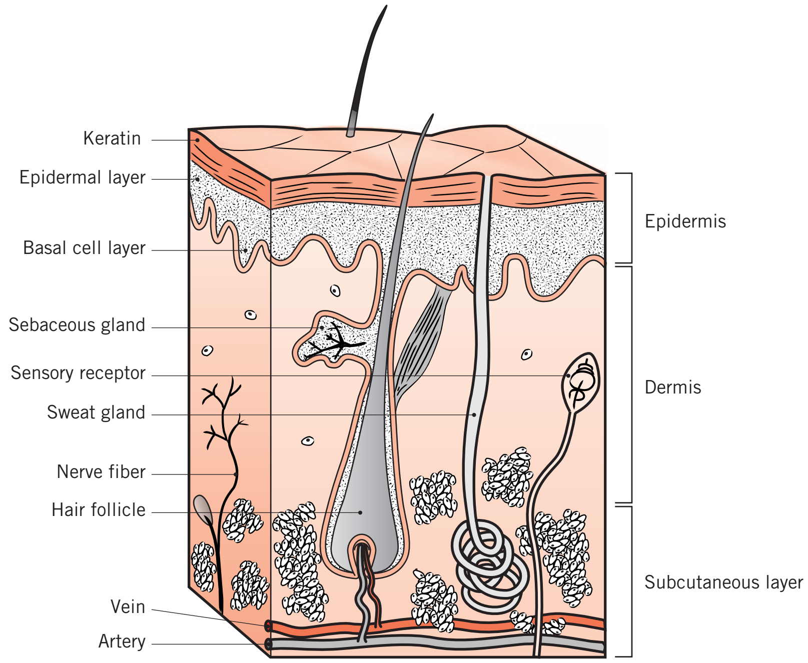


FIGURE 1.14 Morphology of human skin.

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 *counterow 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*.

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

TABLE 1.5 Summary of heat transfer processes

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

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 *ux* and a heat *rate*? What are their units?
- 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 *Fouriers 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 *Newtons law of cooling* ? Can you write the equation from memory?
- What role is played by the *convection heat transfer coefcient* 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 coefcient* ?
- 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 1.10

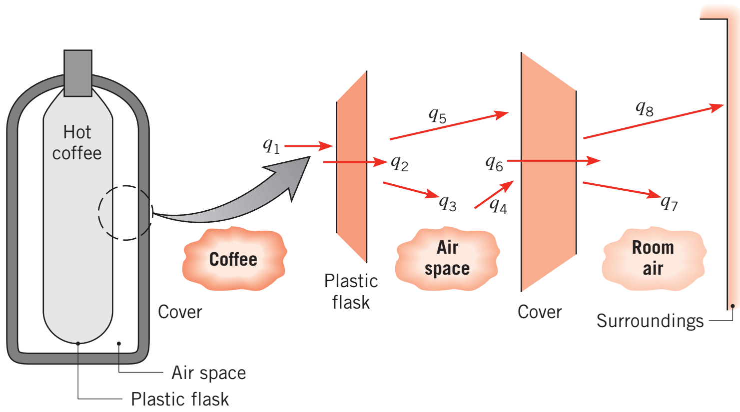
A closed container filled with hot coffee is in a room whose air and walls are at a fixed temperature. Identify all heat transfer processes that contribute to the cooling of the coffee. Comment on features that would contribute to a superior container design.

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.

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Problems

Conduction

1.1 The thermal conductivity of a sheet of rigid, extruded insulation is reported to be $k = 0.029$ W/m·K. The measured temperature difference across a 20-mm-thick sheet of the material is $T_1 - T_2 = 10°C$.

(a) What is the heat flux through a 2 m × 2 m sheet of the insulation?

(b) What is the rate of heat transfer through the sheet of insulation?

1.2 The heat flux that is applied to the left face of a plane wall is $q'' = 20$ W/m^2. The wall is of thickness $L = 10$ mm and of thermal conductivity $k = 12$ W/m·K. If the surface temperatures of the wall are measured to be 50°C on the left side and 30°C on the right side, do steady-state conditions exist?

1.3 A concrete wall, which has a surface area of 20 m^2 and is 0.30 m thick, separates conditioned room air from ambient air. The temperature of the inner surface of the wall is maintained at 25°C, and the thermal conductivity of the concrete is 1 W/m·K.

(a) Determine the heat loss through the wall for outer surface temperatures ranging from −15°C to 38°C, which correspond to winter and summer extremes, respectively. Display your results graphically.

(b) On your graph, also plot the heat loss as a function of the outer surface temperature for wall materials having thermal conductivities of 0.75 and 1.25 W/m·K. Explain the family of curves you have obtained.

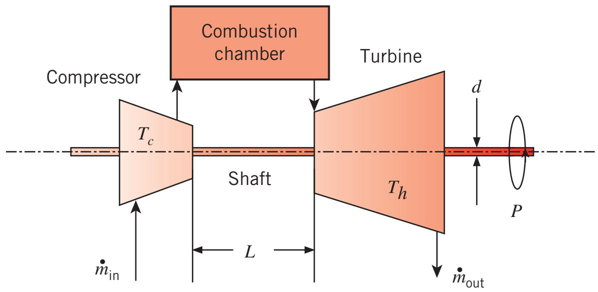
1.4 The concrete slab of a basement is 11 m long, 8 m wide, and 0.20 m thick. During the winter, temperatures are nominally 17°C and 10°C at the top and bottom surfaces, respectively. If the concrete has a thermal conductivity of 1.4 W/m·K, what is the rate of heat loss through the slab? If the basement is heated by a gas furnace operating at an efficiency of $\eta_f = 0.90$ and natural gas is priced at $C_g = \$0.02$/MJ, what is the daily cost of the heat loss?

1.5 Consider Figure 1.3. The heat flux in the x-direction is $q''_x = 10$ W/m^2, the thermal conductivity and wall thickness are $k = 2.3$ W/m·K and $L = 20$ mm, respectively, and steady-state conditions exist. Determine the value of the temperature gradient in units of K/m. What is the value of the temperature gradient in units of °C/m?

1.6 The heat flux through a wood slab 50 mm thick, whose inner and outer surface temperatures are 40 and 20°C, respectively, has been determined to be 40 W/m^2. What is the thermal conductivity of the wood?

1.7 The inner and outer surface temperatures of a glass window 5 mm thick are 15 and 5°C. What is the heat loss through a 1 m × 3 m window? The thermal conductivity of glass is 1.4 W/m·K.

1.8 A thermodynamic analysis of a proposed Brayton cycle gas turbine yields $P = 5$ MW of net power production. The compressor, at an average temperature of $T_c = 400°C$, is driven by the turbine at an average temperature of $T_h = 1000°C$ by way of an $L = 1$-m-long, $d = 70$-mm-diameter shaft of thermal conductivity $k = 40$ W/m·K.



(a) Compare the steady-state conduction rate through the shaft connecting the hot turbine to the warm compressor to the net power predicted by the thermodynamics-based analysis.

(b) A research team proposes to scale down the gas turbine of part (a), keeping all dimensions in the same proportions. The team assumes that the same hot and cold temperatures exist as in part (a) and that the net power output of the gas turbine is proportional to the overall volume of the device. Plot the ratio of the conduction through the shaft to the net power output of the turbine over the range $0.005 \text{ m} \le L \le 1$ m. Is a scaled-down device with $L = 0.005$ m feasible?

1.9 A glass window of width $W = 1$ m and height $H = 2$ m is 5 mm thick and has a thermal conductivity of $k_g = 1.4$ W/m·K. If the inner and outer surface temperatures of the glass are 15°C and −20°C, respectively, on a cold winter day, what is the rate of heat loss through the glass? To reduce heat loss through windows, it is customary to use a double pane construction in which adjoining panes are separated by an air space. If the spacing is 10 mm and the glass surfaces in contact with the air have temperatures of 10°C and −15°C, what is the rate of heat loss from a 1 m × 2 m window? The thermal conductivity of air is $k_a = 0.024$ W/m·K.

1.10 A freezer compartment consists of a cubical cavity that is 2 m on a side. Assume the bottom to be perfectly insulated. What is the minimum thickness of styrofoam insulation ($k = 0.030$ W/m · K) that must be applied to the top and side walls to ensure a heat load of less than 500 W, when the inner and outer surfaces are −10 and 35°C?

1.11 The heat flux that is applied to one face of a plane wall is $q'' = 20$ W/m^2. The opposite face is exposed to air at temperature 30°C, with a convection heat transfer coefficient of 20 W/m^2 · K. The surface temperature of the wall exposed to air is measured and found to be 50°C. Do steady-state conditions exist? If not, is the temperature of the wall increasing or decreasing with time?

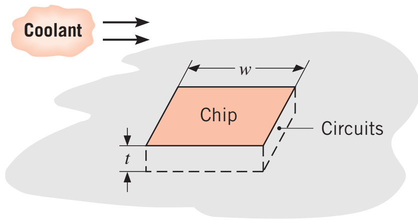
1.12 An inexpensive food and beverage container is fabricated from 25-mm-thick polystyrene ($k = 0.023$ W/m · K) and has interior dimensions of 0.8 m × 0.6 m × 0.6 m. Under conditions for which an inner surface temperature of approximately 2°C is maintained by an ice-water mixture and an outer surface temperature of 20°C is maintained by the ambient, what is the heat flux through the container wall? Assuming negligible heat gain through the 0.8 m × 0.6 m base of the cooler, what is the total heat load for the prescribed conditions?

1.13 What is the thickness required of a masonry wall having thermal conductivity 0.75 W/m · K if the heat rate is to be 80% of the heat rate through a composite structural wall having a thermal conductivity of 0.25 W/m · K and a thickness of 100 mm? Both walls are subjected to the same surface temperature difference.

1.14 A wall is made from an inhomogeneous (nonuniform) material for which the thermal conductivity varies through the thickness according to $k = ax + b$, where a and b are constants. The heat flux is known to be constant. Determine expressions for the temperature gradient and the temperature distribution when the surface at $x = 0$ is at temperature T_1.

1.15 The 5-mm-thick bottom of a 200-mm-diameter pan may be made from aluminum ($k = 240$ W/m · K) or copper ($k = 390$ W/m · K). When used to boil water, the surface of the bottom exposed to the water is nominally at 110°C. If heat is transferred from the stove to the pan at a rate of 600 W, what is the temperature of the surface in contact with the stove for each of the two materials?

1.16 A square silicon chip ($k = 150$ W/m · K) is of width $w = 5$ mm on a side and of thickness $t = 1$ mm. The chip is mounted in a substrate such that its side and back surfaces are insulated, while the front surface is exposed to a coolant. If 4 W are being dissipated in circuits mounted to the back surface of the chip, what is the steady-state temperature difference between back and front surfaces?



Convection

1.17 For a boiling process such as shown in Figure 1.5*c*, the ambient temperature T_∞ in Newton's law of cooling is replaced by the saturation temperature of the fluid T_{sat}. Consider a situation where the heat flux from the hot plate is $q'' = 20 \times 10^5$ W/m^2. If the fluid is water at atmospheric pressure and the convection heat transfer coefficient is $h_w = 20 \times 10^3$ W/m^2 · K, determine the upper surface temperature of the plate, $T_{s,w}$. In an effort to minimize the surface temperature, a technician proposes replacing the water with a dielectric fluid whose saturation temperature is $T_{sat,d} = 52$°C. If the heat transfer coefficient associated with the dielectric fluid is $h_d = 3 \times 10^3$ W/m^2 · K, will the technician's plan work?

1.18 You've experienced convection cooling if you've ever extended your hand out the window of a moving vehicle or into a flowing water stream. With the surface of your hand at a temperature of 30°C, determine the convection heat flux for (a) a vehicle speed of 35 km/h in air at −5°C with a convection coefficient of 40 W/m^2 · K and (b) a velocity of 0.2 m/s in a water stream at 10°C with a convection coefficient of 900 W/m^2 · K. Which condition would *feel* colder? Contrast these results with a heat loss of approximately 30 W/m^2 under normal room conditions.

1.19 Air at 40°C flows over a long, 25-mm-diameter cylinder with an embedded electrical heater. In a series of tests, measurements were made of the power per unit length, P', required to maintain the cylinder surface temperature at 300°C for different free stream velocities V of the air. The results are as follows:

Air velocity, V (m/s)	1	2	4	8	12
Power, P' (W/m)	450	658	983	1507	1963

(a) Determine the convection coefficient for each velocity, and display your results graphically.

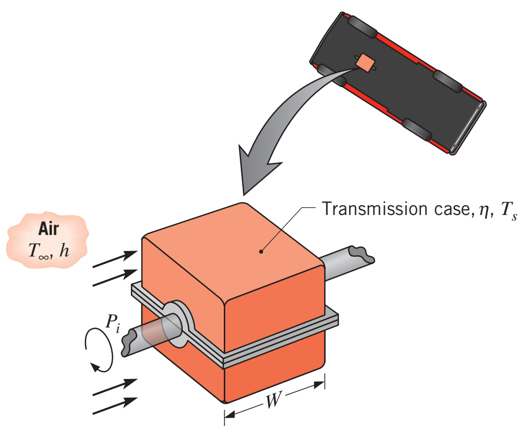
(b) Assuming the dependence of the convection coefficient on the velocity to be of the form $h = CV^n$, determine the parameters C and n from the results of part (a).

1.20 A wall has inner and outer surface temperatures of 16 and 6°C, respectively. The interior and exterior air temperatures are 20 and 5°C, respectively. The inner and outer convection heat transfer coefficients are 5 and 20 $W/m^2 \cdot K$, respectively. Calculate the heat flux from the interior air to the wall, from the wall to the exterior air, and from the wall to the interior air. Is the wall under steady-state conditions?

1.21 An electric resistance heater is embedded in a long cylinder of diameter 30 mm. When water with a temperature of 25°C and velocity of 1 m/s flows crosswise over the cylinder, the power per unit length required to maintain the surface at a uniform temperature of 90°C is 28 kW/m. When air, also at 25°C, but with a velocity of 10 m/s is flowing, the power per unit length required to maintain the same surface temperature is 400 W/m. Calculate and compare the convection coefficients for the flows of water and air.

1.22 The free convection heat transfer coefficient on a thin hot vertical plate suspended in still air can be determined from observations of the change in plate temperature with time as it cools. Assuming the plate is isothermal and radiation exchange with its surroundings is negligible, evaluate the convection coefficient at the instant of time when the plate temperature is 225°C and the change in plate temperature with time (dT/dt) is -0.022 K/s. The ambient air temperature is 25°C and the plate measures 0.3×0.3 m with a mass of 3.75 kg and a specific heat of 2770 $J/kg \cdot K$.

1.23 A transmission case measures $W = 0.30$ m on a side and receives a power input of $P_i = 150$ hp from the engine.

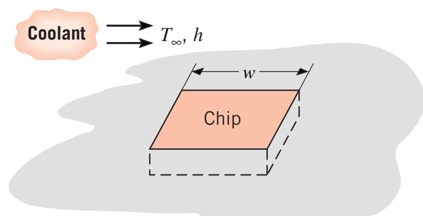


If the transmission efficiency is $\eta = 0.93$ and airflow over the case corresponds to $T_\infty = 30$°C and $h = 200$ $W/m^2 \cdot K$, what is the surface temperature of the transmission?

1.24 A cartridge electrical heater is shaped as a cylinder of length $L = 200$ mm and outer diameter $D = 20$ mm. Under normal operating conditions, the heater dissipates 2 kW while submerged in a water flow that is at 20°C and provides a convection heat transfer coefficient of $h = 5000$ $W/m^2 \cdot K$. Neglecting heat transfer from the ends of the heater, determine its surface temperature T_s. If the water flow is inadvertently terminated while the heater continues to operate, the heater surface is exposed to air that is also at 20°C but for which $h = 50$ $W/m^2 \cdot K$. What is the corresponding surface temperature? What are the consequences of such an event?

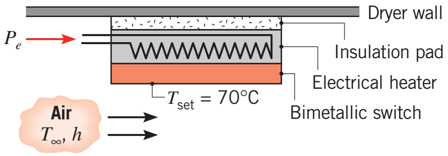
1.25 A common procedure for measuring the velocity of an airstream involves the insertion of an electrically heated wire (called a *hot-wire anemometer*) into the airflow, with the axis of the wire oriented perpendicular to the flow direction. The electrical energy dissipated in the wire is assumed to be transferred to the air by forced convection. Hence, for a prescribed electrical power, the temperature of the wire depends on the convection coefficient, which, in turn, depends on the velocity of the air. Consider a wire of length $L = 20$ mm and diameter $D = 0.5$ mm, for which a calibration of the form $V = 6.25 \times 10^{-5} h^2$ has been determined. The velocity V and the convection coefficient h have units of m/s and $W/m^2 \cdot K$, respectively. In an application involving air at a temperature of $T_\infty = 25$°C, the surface temperature of the anemometer is maintained at $T_s = 75$°C with a voltage drop of 5 V and an electric current of 0.1 A. What is the velocity of the air?

1.26 A square isothermal chip is of width $w = 5$ mm on a side and is mounted in a substrate such that its side and back surfaces are well insulated; the front surface is exposed to the flow of a coolant at $T_\infty = 15$°C. From reliability considerations, the chip temperature must not exceed $T = 85$°C.



If the coolant is air and the corresponding convection coefficient is $h = 200$ $W/m^2 \cdot K$, what is the maximum allowable chip power? If the coolant is a dielectric liquid for which $h = 3000$ $W/m^2 \cdot K$, what is the maximum allowable power?

1.27 The temperature controller for a clothes dryer consists of a bimetallic switch mounted on an electrical heater attached to a wall-mounted insulation pad.



The switch is set to open at 70°C, the maximum dryer air temperature. To operate the dryer at a lower air temperature, sufficient power is supplied to the heater such that the switch reaches 70°C (T_{set}) when the air temperature T is less than T_{set}. If the convection heat transfer coefficient between the air and the exposed switch surface of 30 mm^2 is 25 W/m$^2 \cdot$K, how much heater power P_e is required when the desired dryer air temperature is $T_\infty = 50°C$?

Radiation

1.28 An overhead 25-m-long, uninsulated industrial steam pipe of 100-mm diameter is routed through a building whose walls and air are at 25°C. Pressurized steam maintains a pipe surface temperature of 150°C, and the coefficient associated with natural convection is $h = 10$ W/m$^2 \cdot$K. The surface emissivity is $\varepsilon = 0.8$.

(a) What is the rate of heat loss from the steam line?

(b) If the steam is generated in a gas-fired boiler operating at an efficiency of $\eta_f = 0.90$ and natural gas is priced at $C_g = \$0.02$ per MJ, what is the annual cost of heat loss from the line?

1.29 Under conditions for which the same room temperature is maintained by a heating or cooling system, it is not uncommon for a person to feel chilled in the winter but comfortable in the summer. Provide a plausible explanation for this situation (with supporting calculations) by considering a room whose air temperature is maintained at 20°C throughout the year, while the walls of the room are nominally at 27°C and 14°C in the summer and winter, respectively. The exposed surface of a person in the room may be assumed to be at a temperature of 32°C throughout the year and to have an emissivity of 0.90. The coefficient associated with heat transfer by natural convection between the person and the room air is approximately 2 W/m$^2 \cdot$K.

1.30 A spherical interplanetary probe of 0.5-m diameter contains electronics that dissipate 150 W. If the probe surface has an emissivity of 0.8 and the probe does not receive radiation from other surfaces, as, for example, from the sun, what is its surface temperature?

1.31 An instrumentation package has a spherical outer surface of diameter $D = 100$ mm and emissivity $\varepsilon = 0.25$. The package is placed in a large space simulation chamber whose walls are maintained at 77 K. If operation of the electronic components is restricted to the temperature range $40 \leq T \leq 85°C$, what is the range of acceptable power dissipation for the package? Display your results graphically, showing also the effect of variations in the emissivity by considering values of 0.20 and 0.30.

1.32 Consider the conditions of Problem 1.22. However, now the plate is in a vacuum with a surrounding temperature of 25°C. What is the emissivity of the plate? What is the rate at which radiation is emitted by the surface?

1.33 If $T_s \approx T_{sur}$ in Equation 1.9, the radiation heat transfer coefficient may be approximated as

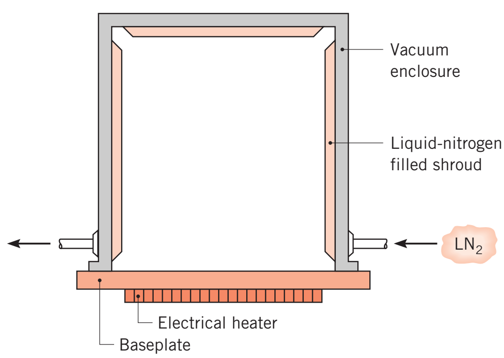
$$h_{r,a} = 4\varepsilon\sigma\overline{T}^3$$

where $\overline{T} \equiv (T_s + T_{sur})/2$. We wish to assess the validity of this approximation by comparing values of h_r and $h_{r,a}$ for the following conditions. In each case, represent your results graphically and comment on the validity of the approximation.

(a) Consider a surface of either polished aluminum ($\varepsilon = 0.05$) or black paint ($\varepsilon = 0.9$), whose temperature may exceed that of the surroundings ($T_{sur} = 25°C$) by 10 to 100 C. Also compare your results with values of the coefficient associated with free convection in air ($T_\infty = T_{sur}$), where $h\,(\text{W/m}^2 \cdot \text{K}) = 0.98\,\Delta T^{1/3}$.

(b) Consider initial conditions associated with placing a workpiece at $T_s = 25°C$ in a large furnace whose wall temperature may be varied over the range $100 \leq T_{sur} \leq 1000°C$. According to the surface finish or coating, its emissivity may assume values of 0.05, 0.2, and 0.9. For each emissivity, plot the relative error, $(h_r - h_{r,a})/h_r$, as a function of the furnace temperature.

1.34 A vacuum system, as used in sputtering electrically conducting thin films on microcircuits, is comprised of a baseplate maintained by an electrical heater at 300 K and a shroud within the enclosure maintained at 77 K by a liquid-nitrogen coolant loop. The circular baseplate, insulated on the lower side, is 0.3 m in diameter and has an emissivity of 0.25.



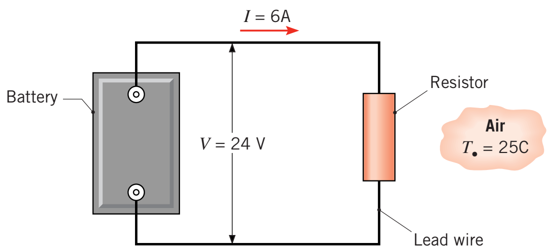
(a) How much electrical power must be provided to the baseplate heater?

(b) At what rate must liquid nitrogen be supplied to the shroud if its heat of vaporization is 125 kJ/kg?

(c) To reduce the liquid nitrogen consumption, it is proposed to bond a thin sheet of aluminum foil ($\varepsilon = 0.09$) to the baseplate. Will this have the desired effect?

Relationship to Thermodynamics

1.35 An electrical resistor is connected to a battery, as shown schematically. After a brief transient, the resistor assumes a nearly uniform, steady-state temperature of 95°C, while the battery and lead wires remain at the ambient temperature of 25°C. Neglect the electrical resistance of the lead wires.



(a) Consider the resistor as a system about which a control surface is placed and Equation 1.12c is applied. Determine the corresponding values of E_{in}(W), E_g(W), E_{out}(W), and E_{st}(W). If a control surface is placed about the entire system, what are the values of E_{in}, E_g, E_{out}, and E_{st}?

(b) If electrical energy is dissipated uniformly within the resistor, which is a cylinder of diameter $D = 60$ mm and length $L = 250$ mm, what is the volumetric heat generation rate, q (W/m^3)?

(c) Neglecting radiation from the resistor, what is the convection coefficient?

1.36 Pressurized water ($p_{in} = 10$ bar, $T_{in} = 110°C$) enters the bottom of an $L = 10$-m-long vertical tube of diameter $D = 100$ mm at a mass flow rate of $m = 1.5$ kg/s. The tube is located inside a combustion chamber, resulting in heat transfer to the tube. Superheated steam exits the top of the tube at $p_{out} = 7$ bar, $T_{out} = 600°C$. Determine the change in the rate at which the following quantities enter and exit the tube: (a) the combined thermal and flow work, (b) the mechanical energy, and (c) the total energy of the water. Also, (d) determine the heat transfer rate, q. *Hint*: Relevant properties may be obtained from a thermodynamics text.

1.37 Consider the tube and inlet conditions of Problem 1.36. Heat transfer at a rate of $q = 3.89$ MW is delivered to the tube. For an exit pressure of $p = 8$ bar, determine (a) the temperature of the water at the outlet as well as the change in (b) combined thermal and flow work, (c) mechanical energy, and (d) total energy of the water from the inlet to the outlet of the tube. *Hint:* As a first estimate, neglect the change in mechanical energy in solving part (a). Relevant properties may be obtained from a thermodynamics text.

1.38 An internally reversible refrigerator has a modified coefficient of performance accounting for realistic heat transfer processes of

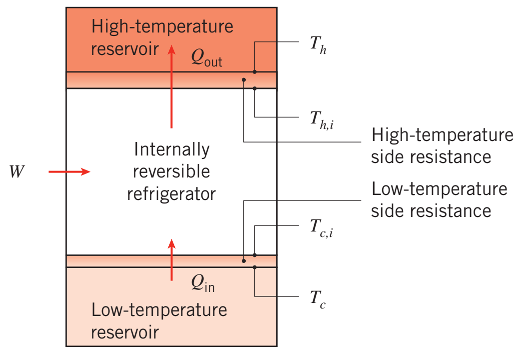
$$\mathrm{COP}_m = \frac{q_{in}}{W} = \frac{q_{in}}{q_{out} - q_{in}} = \frac{T_{c,i}}{T_{h,i} - T_{c,i}}$$

where q_{in} is the refrigerator cooling rate, q_{out} is the heat rejection rate, and W is the power input. Show that COP_m can be expressed in terms of the reservoir temperatures T_c and T_h, the cold and hot thermal resistances $R_{t,c}$ and $R_{t,h}$, and q_{in}, as

$$\mathrm{COP}_m = \frac{T_c - q_{in}R_{tot}}{T_h - T_c + q_{in}R_{tot}}$$

where $R_{tot} = R_{t,c} + R_{t,h}$. Also, show that the power input may be expressed as

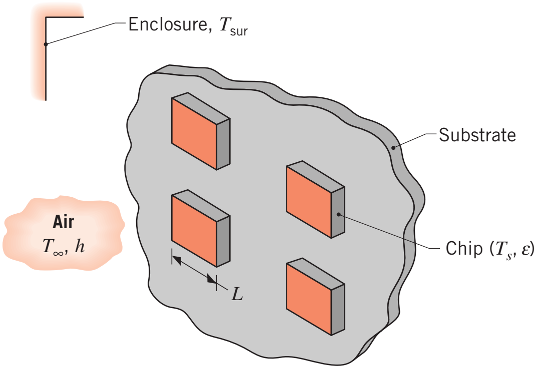
$$W = q_{in}\frac{T_h - T_c + q_{in}R_{tot}}{T_c - q_{in}R_{tot}}$$



1.39 A household refrigerator operates with cold- and hot-temperature reservoirs of $T_c = 5°C$ and $T_h = 25°C$, respectively. When new, the cold and hot side resistances are $R_{c,n} = 0.05$ K/W and $R_{h,n} = 0.04$ K/W, respectively. Over time, dust accumulates on the refrigerator's condenser coil, which is located behind the refrigerator, increasing the hot side resistance to $R_{h,d} = 0.1$ K/W. It is desired to have a refrigerator cooling rate of $q_{in} = 750$ W. Using the results of Problem 1.38, determine the modified coefficient of performance and the required power input W under (a) clean and (b) dusty coil conditions.

Energy Balance and Multimode Effects

1.40 Chips of width $L = 15$ mm on a side are mounted to a substrate that is installed in an enclosure whose walls and air are maintained at a temperature of $T_{sur} = 25°C$. The chips have an emissivity of $\varepsilon = 0.60$ and a maximum allowable temperature of $T_s = 85°C$.

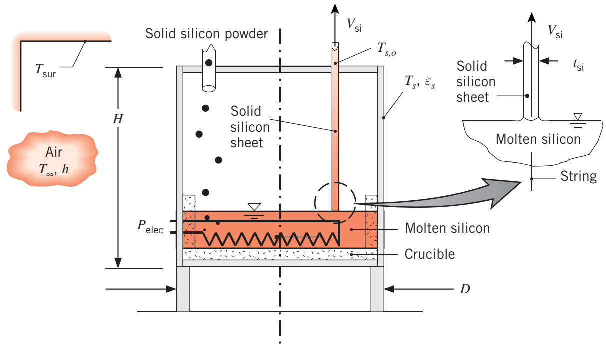


(a) If heat is rejected from the chips by radiation and natural convection, what is the maximum operating power of each chip? The convection coefficient depends on the chip-to-air temperature difference and may be approximated as $h = C(T_s - T_\infty)^{1/4}$, where $C = 4.2\ \text{W/m}^2 \cdot \text{K}^{5/4}$.

(b) If a fan is used to maintain airflow through the enclosure and heat transfer is by forced convection, with $h = 250\ \text{W/m}^2 \cdot \text{K}$, what is the maximum operating power?

1.41 Consider the transmission case of Problem 1.23, but now allow for radiation exchange with the ground/chassis, which may be approximated as large surroundings at $T_{sur} = 30°C$. If the emissivity of the case is $\varepsilon = 0.80$, what is the surface temperature?

1.42 One method for growing thin silicon sheets for photovoltaic solar panels is to pass two thin strings of high melting temperature material upward through a bath of molten silicon. The silicon solidifies on the strings near the surface of the molten pool, and the solid silicon sheet is pulled slowly upward out of the pool. The silicon is replenished by supplying the molten pool with solid silicon powder. Consider a silicon sheet that is $W_{si} = 85$ mm wide and $t_{si} = 150\ \mu$m thick that is pulled at a velocity of $V_{si} = 20$ mm/min. The silicon is melted by supplying electric power to the cylindrical growth chamber of height $H = 350$ mm and diameter $D = 300$ mm. The exposed surfaces of the growth chamber are at $T_s = 320$ K, the corresponding convection coefficient at the exposed surface is $h = 8\ \text{W/m}^2 \cdot \text{K}$, and the surface is characterized by an emissivity of $\varepsilon_s = 0.9$. The solid silicon powder is at $T_{si,i} = 298$ K, and the solid silicon sheet exits the chamber at $T_{si,o} = 420$ K. Both the surroundings and ambient temperatures are $T_\infty = T_{sur} = 298$ K.

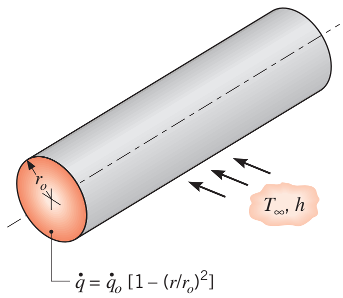


(a) Determine the electric power, P_{elec}, needed to operate the system at steady state.

(b) If the photovoltaic panel absorbs a time-averaged solar flux of $q''_{sol} = 180\ \text{W/m}^2$ and the panel has a conversion efficiency (the ratio of solar power absorbed to electric power produced) of $\eta = 0.20$, how long must the solar panel be operated to produce enough electric energy to offset the electric energy that was consumed in its manufacture?

1.43 Heat is transferred by radiation and convection between the inner surface of the nacelle of the wind turbine of Example 1.3 and the outer surfaces of the gearbox and generator. The convection heat flux associated with the gearbox and the generator may be described by $q''_{conv,gb} = h(T_{gb} - T_\infty)$ and $q''_{conv,gen} = h(T_{gen} - T_\infty)$, respectively, where the ambient temperature $T_\infty \approx T_s$ (which is the nacelle temperature) and $h = 40\ \text{W/m}^2 \cdot \text{K}$. The outer surfaces of both the gearbox and the generator are characterized by an emissivity of $\varepsilon = 0.9$. If the surface areas of the gearbox and generator are $A_{gb} = 6\ \text{m}^2$ and $A_{gen} = 4\ \text{m}^2$, respectively, determine their surface temperatures.

1.44 Radioactive wastes are packed in a long, thin-walled cylindrical container. The wastes generate thermal energy nonuniformly according to the relation $q = q_o[1 - (r/r_o)^2]$, where q is the local rate of energy generation per unit volume, q_o is a constant, and r_o is the radius of the container. Steady-state conditions are maintained by submerging the container in a liquid that is at T_∞ and provides a uniform convection coefficient h.



Obtain an expression for the total rate at which energy is generated in a unit length of the container. Use this result to obtain an expression for the temperature T_s of the container wall.

1.45 An aluminum plate 4 mm thick is mounted in a horizontal position, and its bottom surface is well insulated. A special, thin coating is applied to the top surface such that it absorbs 80% of any incident solar radiation, while having an emissivity of 0.25. The density ρ and specific heat c of aluminum are known to be 2700 kg/m^3 and 900 J/kg · K, respectively.

(a) Consider conditions for which the plate is at a temperature of 25°C and its top surface is suddenly exposed to ambient air at $T_\infty = 20$°C and to solar radiation that provides an incident flux of 900 W/m^2. The convection heat transfer coefficient between the surface and the air is $h = 20$ W/m^2 · K. What is the initial rate of change of the plate temperature?

(b) What will be the equilibrium temperature of the plate when steady-state conditions are reached?

(c) The surface radiative properties depend on the specific nature of the applied coating. Compute and plot the steady-state temperature as a function of the emissivity for $0.05 \le \varepsilon \le 1$, with all other conditions remaining as prescribed. Repeat your calculations for values of $\alpha_S = 0.5$ and 1.0, and plot the results with those obtained for $\alpha_S = 0.8$. If the intent is to maximize the plate temperature, what is the most desirable combination of the plate emissivity and its absorptivity to solar radiation?

1.46 A blood warmer is to be used during the transfusion of blood to a patient. This device is to heat blood taken from the blood bank at 10°C to 37°C at a flow rate of 200 ml/min. The blood passes through tubing of length 2 m, with a rectangular cross section 6.4 mm × 1.6 mm At what rate must heat be added to the blood to accomplish the required temperature increase? If the fluid originates from a large tank with nearly zero velocity and flows vertically downward for its 2-m length, estimate the magnitudes of kinetic and potential energy changes. Assume the blood's properties are similar to those of water.

1.47 Consider a carton of milk that is refrigerated at a temperature of $T_m = 5$°C. The kitchen temperature on a hot summer day is $T_\infty = 30$°C. If the four sides of the carton are of height and width $L = 200$ mm and $w = 100$ mm, respectively, determine the heat transferred to the milk carton as it sits on the kitchen counter for durations of $t = 10$ s, 60 s, and 300 s before it is returned to the refrigerator. The convection coefficient associated with natural convection on the sides of the carton is $h = 10$ W/m^2 · K. The surface emissivity is 0.90. Assume the milk carton temperature remains at 5°C during the process. Your parents have taught you the importance of refrigerating certain foods from the food safety perspective. Comment on the importance of quickly returning the milk carton to the refrigerator from an energy conservation point of view.

1.48 The energy consumption associated with a home water heater has two components: (i) the energy that must be supplied to bring the temperature of groundwater to the heater storage temperature, as it is introduced to replace hot water that has been used; (ii) the energy needed to compensate for heat losses incurred while the water is stored at the prescribed temperature. In this problem, we will evaluate the first of these components for a family of four, whose daily hot water consumption is approximately 100 gal. If groundwater is available at 15°C, what is the annual energy consumption associated with heating the water to a storage temperature of 55°C? For a unit electrical power cost of \$0.18/kW · h, what is the annual cost associated with supplying hot water by means of (a) electric resistance heating or (b) a heat pump having a COP of 3.

1.49 Liquid oxygen, which has a boiling point of 90 K and a latent heat of vaporization of 214 kJ/kg, is stored in a spherical container whose outer surface is of 500-mm diameter and at a temperature of −10°C. The container is housed in a laboratory whose air and walls are at 25°C.

(a) If the surface emissivity is 0.20 and the heat transfer coefficient associated with free convection at the outer surface of the container is 10 W/m^2 · K, what is the rate, in kg/s, at which oxygen vapor must be vented from the system?

(b) Moisture in the ambient air will result in frost formation on the container, causing the surface emissivity to increase. Assuming the surface temperature and convection coefficient to remain at −10°C and

10 $\text{W/m}^2 \cdot \text{K}$, respectively, compute the oxygen evaporation rate (kg/s) as a function of surface emissivity over the range $0.2 \le \varepsilon \le 0.94$.

1.50 The emissivity of galvanized steel sheet, a common roofing material, is $\varepsilon = 0.13$ at temperatures around 300 K, while its absorptivity for solar irradiation is $\alpha_S = 0.65$. Would the neighborhood cat be comfortable walking on a roof constructed of the material on a day when $G_S = 750 \text{ W/m}^2$, $T_\infty = 16°\text{C}$, and $h = 7 \text{ W/m}^2 \cdot \text{K}$? Assume the bottom surface of the steel is insulated.

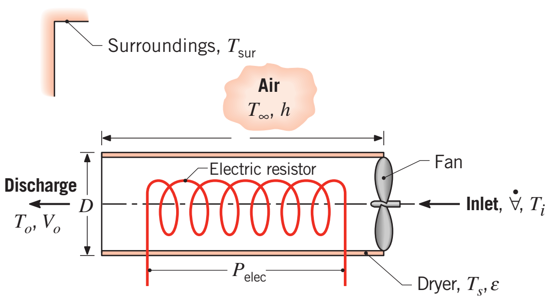
1.51 Three electric resistance heaters of length $L = 250$ mm and diameter $D = 25$ mm are submerged in a 10-gal tank of water, which is initially at 295 K. The water may be assumed to have a density and specific heat of $\rho = 990 \text{ kg/m}^3$ and $c = 4180 \text{ J/kg} \cdot \text{K}$.

(a) If the heaters are activated, each dissipating $q_1 = 500$ W, estimate the time required to bring the water to a temperature of 335 K.

(b) If the natural convection coefficient is given by an expression of the form $h = 370\,(T_s - T)^{1/3}$, where T_s and T are temperatures of the heater surface and water, respectively, what is the temperature of each heater shortly after activation and just before deactivation? Units of h and $(T_s - T)$ are $\text{W/m}^2 \cdot \text{K}$ and K, respectively.

(c) If the heaters are inadvertently activated when the tank is empty, the natural convection coefficient associated with heat transfer to the ambient air at $T_\infty = 300$ K may be approximated as $h = 0.70\,(T_s - T_\infty)^{1/3}$. If the temperature of the tank walls is also 300 K and the emissivity of the heater surface is $\varepsilon = 0.85$, what is the surface temperature of each heater under steady-state conditions?

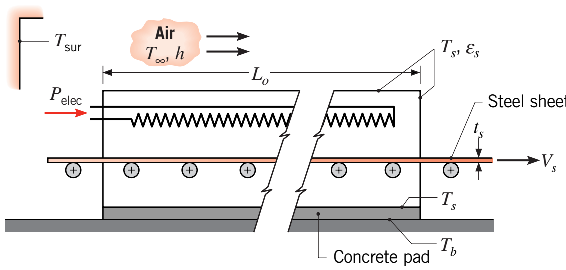
1.52 A hair dryer may be idealized as a circular duct through which a small fan draws ambient air and within which the air is heated as it flows over a coiled electric resistance wire.



(a) If a dryer is designed to operate with an electric power consumption of $P_{elec} = 500$ W and to heat air from an ambient temperature of $T_i = 20°\text{C}$ to a discharge temperature of $T_o = 45°\text{C}$, at what volumetric flow rate $\dot{\forall}$ should the fan operate? Heat loss from the casing to the ambient air and the surroundings may be neglected. If the duct has a diameter of $D = 70$ mm, what is the discharge velocity V_o of the air? The density and specific heat of the air may be approximated as $\rho = 1.10 \text{ kg/m}^3$ and $c_p = 1007 \text{ J/kg} \cdot \text{K}$, respectively.

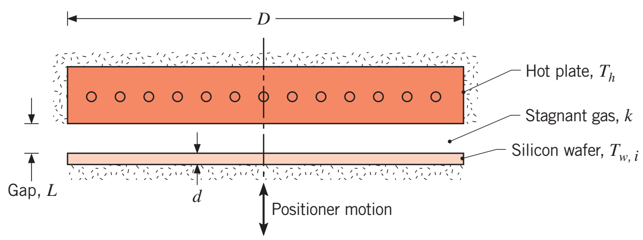
(b) Consider a dryer duct length of $L = 150$ mm and a surface emissivity of $\varepsilon = 0.8$. If the coefficient associated with heat transfer by natural convection from the casing to the ambient air is $h = 4 \text{ W/m}^2 \cdot \text{K}$ and the temperature of the air and the surroundings is $T_\infty = T_{sur} = 20°\text{C}$, confirm that the heat loss from the casing is, in fact, negligible. The casing may be assumed to have an average surface temperature of $T_s = 40°\text{C}$.

1.53 In one stage of an annealing process, 304 stainless steel sheet is taken from 300 K to 1250 K as it passes through an electrically heated oven at a speed of $V_s = 10$ mm/s. The sheet thickness and width are $t_s = 8$ mm and $W_s = 2$ m, respectively, while the height, width, and length of the oven are $H_o = 2$ m, $W_o = 2.4$ m, and $L_o = 25$ m, respectively. The top and four sides of the oven are exposed to ambient air and large surroundings, each at 300 K, and the corresponding surface temperature, convection coefficient, and emissivity are $T_s = 350$ K, $h = 10 \text{ W/m}^2 \cdot \text{K}$, and $\varepsilon_s = 0.8$. The bottom surface of the oven is also at 350 K and rests on a 0.5-m-thick concrete pad whose base is at 300 K. Estimate the required electric power input, P_{elec}, to the oven.



1.54 Convection ovens operate on the principle of inducing forced convection inside the oven chamber with a fan. A *small* cake is to be baked in an oven when the convection feature is disabled. For this situation, the free convection coefficient associated with the cake and its pan is $h_{fr} = 3\ \mathrm{W/m^2 \cdot K}$. The oven air and wall are at temperatures $T_\infty = T_{sur} = 180°\mathrm{C}$. Determine the heat flux delivered to the cake pan and cake batter when they are initially inserted into the oven and are at a temperature of $T_i = 24°\mathrm{C}$. If the convection feature is activated, the forced convection heat transfer coefficient is $h_{fo} = 27\ \mathrm{W/m^2 \cdot K}$. What is the heat flux at the batter or pan surface when the oven is operated in the convection mode? Assume a value of 0.97 for the emissivity of the cake batter and pan.

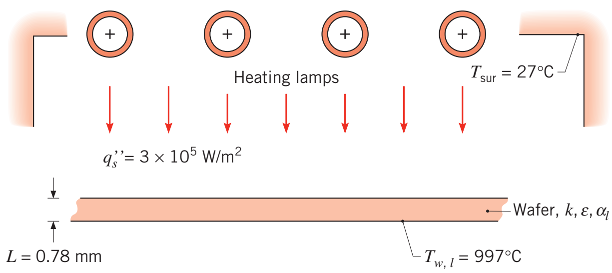
1.55 Annealing, an important step in semiconductor materials processing, can be accomplished by rapidly heating the silicon wafer to a high temperature for a short period of time. The schematic shows a method involving the use of a hot plate operating at an elevated temperature T_h. The wafer, initially at a temperature of $T_{w,i}$, is suddenly positioned at a gap separation distance L from the hot plate. The purpose of the analysis is to compare the heat fluxes by conduction through the gas within the gap and by radiation exchange between the hot plate and the cool wafer. The initial time rate of change in the temperature of the wafer, $(dT_w/dt)_i$, is also of interest. Approximating the surfaces of the hot plate and the wafer as blackbodies and assuming their diameter D to be much larger than the spacing L, the radiative heat flux may be expressed as $q''_{rad} = \sigma(T_h^4 - T_w^4)$. The silicon wafer has a thickness of $d = 0.78$ mm, a density of 2700 kg/m^3, and a specific heat of 875 J/kg·K. The thermal conductivity of the gas in the gap is 0.0436 W/m·K.



(a) For $T_h = 600°\mathrm{C}$ and $T_{w,i} = 20°\mathrm{C}$, calculate the radiative heat flux and the heat flux by conduction across a gap distance of $L = 0.2$ mm. Also determine the value of $(dT_w/dt)_i$, resulting from each of the heating modes.

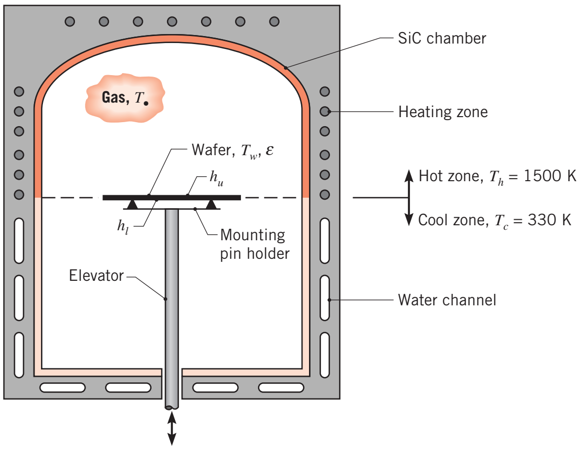
$\boxed{\text{(b)}}$ For gap distances of 0.2, 0.5, and 1.0 mm, determine the heat fluxes and temperature-time change as a function of the hot plate temperature for $300 \le T_h \le 1300°\mathrm{C}$. Display your results graphically. Comment on the relative importance of the two heat transfer modes and the effect of the gap distance on the heating process. Under what conditions could a wafer be heated to 900°C in less than 10 s?

1.56 In the thermal processing of semiconductor materials, annealing is accomplished by heating a silicon wafer according to a temperature-time recipe and then maintaining a fixed elevated temperature for a prescribed period of time. For the process tool arrangement shown as follows, the wafer is in an evacuated chamber whose walls are maintained at 27°C and within which heating lamps maintain a radiant flux q''_s at its upper surface. The wafer is 0.78 mm thick, has a thermal conductivity of 30 W/m·K, and an emissivity that equals its absorptivity to the radiant flux ($\varepsilon = \alpha_l = 0.65$). For $q''_s = 3.0 \times 10^5\ \mathrm{W/m^2}$, the temperature on its lower surface is measured by a radiation thermometer and found to have a value of $T_{w,l} = 997°\mathrm{C}$.



To avoid warping the wafer and inducing slip planes in the crystal structure, the temperature difference across the thickness of the wafer must be less than 2°C. Is this condition being met?

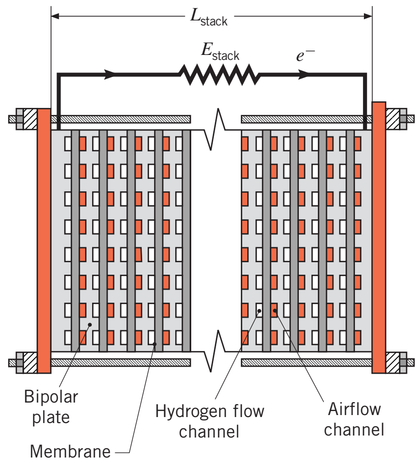
1.57 A furnace for processing semiconductor materials is formed by a silicon carbide chamber that is zone-heated on the top section and cooled on the lower section. With the elevator in the lowest position, a robot arm inserts the silicon wafer on the mounting pins. In a production operation, the wafer is rapidly moved toward the hot zone to achieve the temperature-time history required for the process recipe. In this position, the top and bottom surfaces of the wafer exchange radiation with the hot and cool zones, respectively, of the chamber. The zone temperatures are $T_h = 1500$ K and $T_c = 330$ K, and the emissivity and thickness of the wafer are $\varepsilon = 0.65$ and $d = 0.78$ mm, respectively. With the ambient gas at $T_\infty = 700$ K, convection coefficients at the upper and lower surfaces of the wafer are 8 and 4 W/m^2·K, respectively. The silicon wafer has a density of 2700 kg/m^3 and a specific heat of 875 J/kg·K.



(a) For an initial condition corresponding to a wafer temperature of $T_{w,i} = 300$ K and the position of the wafer shown schematically, determine the corresponding time rate of change of the wafer temperature, $(dT_w/dt)_i$.

(b) Determine the steady-state temperature reached by the wafer if it remains in this position. How significant is convection heat transfer for this situation? Sketch how you would expect the wafer temperature to vary as a function of vertical distance.

1.58 Single fuel cells such as the one of Example 1.5 can be scaled up by arranging them into a *fuel cell stack.* A stack consists of multiple electrolytic membranes that are sandwiched between electrically conducting *bipolar plates.* Air and hydrogen are fed to each membrane through *ow channels* within each bipolar plate, as shown in the sketch. With this stack arrangement, the individual fuel cells are connected in series, electrically, producing a stack voltage of $E_{\text{stack}} = N \times E_c$, where E_c is the voltage produced across each membrane and N is the number of membranes in the stack. The electrical current is the same for each membrane. The cell voltage, E_c, as well as the cell efficiency, increases with temperature (the air and hydrogen fed to the stack are humidified to allow operation at temperatures greater than in Example 1.5), but the membranes will fail at temperatures exceeding $T \approx 85°\text{C}$. Consider $L \times w$ membranes, where $L = w = 100$ mm, of thickness $t_m = 0.43$ mm, that each produce $E_c = 0.6$ V at $I = 60$ A, and $\dot{E}_{c,g} = 45$ W of thermal energy when operating at $T = 80°\text{C}$. The external surfaces of the stack are exposed to air at $T_\infty = 25°\text{C}$ and surroundings at $T_{\text{sur}} = 30°\text{C}$, with $\varepsilon = 0.88$ and $h = 150\ \text{W/m}^2 \cdot \text{K}$.



(a) Find the electrical power produced by a stack that is $L_{\text{stack}} = 200$ mm long, for bipolar plate thickness in the range 1 mm $< t_{\text{bp}} <$ 10 mm. Determine the total thermal energy generated by the stack.

(b) Calculate the surface temperature and explain whether the stack needs to be internally heated or cooled to operate at the optimal internal temperature of 80°C for various bipolar plate thicknesses.

(c) Identify how the internal stack operating temperature might be lowered or raised for a given bipolar plate thickness, and discuss design changes that would promote a more uniform temperature distribution within the stack. How would changes in the external air and surroundings temperature affect your answer? Which membrane in the stack is most likely to fail due to high operating temperature?

1.59 Consider the wind turbine of Example 1.3. To reduce the nacelle temperature to $T_s = 30°\text{C}$, the nacelle is vented and a fan is installed to force ambient air into and out of the nacelle enclosure. What is the minimum mass flow rate of air required if the air temperature increases to the nacelle surface temperature before exiting the nacelle? The specific heat of air is 1007 J/kg·K.

1.60 Consider the conducting rod of Example 1.4 under steady-state conditions. As suggested in Comment 3, the temperature of the rod may be controlled by varying the speed of airflow over the rod, which, in turn, alters the convection heat transfer coefficient. To consider the effect of the convection coefficient, generate plots of T versus I for values of $h = 50$, 100, and 250 $\text{W/m}^2 \cdot \text{K}$. Would variations in the surface emissivity have a significant effect on the rod temperature?

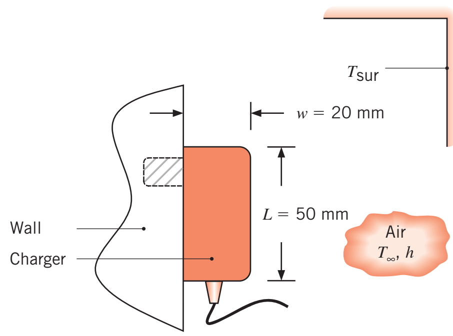
1.61 A long bus bar (cylindrical rod used for making electrical connections) of diameter D is installed in a large conduit having a surface temperature of 30°C and in which the ambient air temperature is $T_\infty =$ 30°C. The electrical resistivity, $\rho_e(\mu\Omega \cdot \text{m})$, of the bar material is a function of temperature, $\rho_{e,o} = \rho_e [1 + \alpha(T - T_o)]$, where $\rho_{e,o} = 0.0171\ \mu\Omega \cdot \text{m}$, $T_o =$ 25°C, and $\alpha = 0.00396\ \text{K}^{-1}$. The bar experiences free convection in the ambient air, and the convection coefficient depends on the bar diameter, as well as on the difference between the surface and ambient temperatures. The governing relation is of the form, $h = CD^{-0.25}\ (T - T_\infty)^{0.25}$, where $C = 1.21\ \text{W} \cdot \text{m}^{-1.75} \cdot \text{K}^{-1.25}$. The emissivity of the bar surface is $\varepsilon = 0.85$.

(a) Recognizing that the electrical resistance per unit length of the bar is $R'_e = \rho_e / A_c$, where A_c is its cross-sectional area, calculate the current-carrying capacity of a 20-mm-diameter bus bar if its temperature is not to exceed 65°C. Compare the relative importance of heat transfer by free convection and radiation exchange.

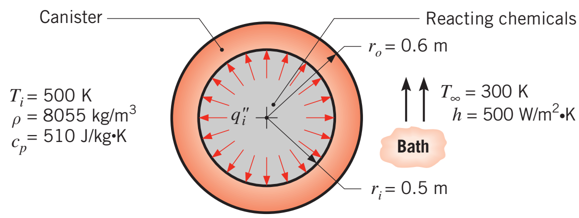
(b) To assess the trade-off between current-carrying capacity, operating temperature, and bar diameter, for diameters of 10, 20, and 40 mm, plot the bar temperature T as a function of current for the range $100 \le I \le 5000$ A. Also plot the ratio of the heat transfer by convection to the total heat transfer.

1.62 A small sphere of reference-grade iron with a specific heat of 447 J/kg·K and a mass of 0.515 kg is suddenly immersed in a water–ice mixture. Fine thermocouple wires suspend the sphere, and the temperature is observed to change from 15 to 14°C in 6.35 s. The experiment is repeated with a metallic sphere of the same diameter, but of unknown composition with a mass of 1.263 kg. If the same observed temperature change occurs in 4.59 s, what is the specific heat of the unknown material?

1.63 A 50 mm × 45 mm × 20 mm cell phone charger has a surface temperature of $T_s = 33$°C when plugged into an electrical wall outlet but not in use. The surface of the charger is of emissivity $\varepsilon = 0.92$ and is subject to a free convection heat transfer coefficient of $h = 4.5\ \text{W/m}^2 \cdot \text{K}$. The room air and wall temperatures are $T_\infty = 22$°C and $T_{sur} = 20$°C, respectively. If electricity costs $C = \$0.18/\text{kW} \cdot \text{h}$, determine the daily cost of leaving the charger plugged in when not in use.



1.64 A spherical, stainless steel (AISI 302) canister is used to store reacting chemicals that provide for a uniform heat flux q''_i to its inner surface. The canister is suddenly submerged in a liquid bath of temperature $T_\infty < T_i$, where T_i is the initial temperature of the canister wall.



(a) Assuming negligible temperature gradients in the canister wall and a constant heat flux q''_i, develop an equation that governs the variation of the wall temperature with time during the transient process. What is the initial rate of change of the wall temperature if $q''_i = 10^5\ \text{W/m}^2$?

(b) What is the steady-state temperature of the wall?

$\boxed{\text{(c)}}$ The convection coefficient depends on the velocity associated with fluid flow over the canister and whether the wall temperature is large enough to induce boiling in the liquid. Compute and plot the steady-state temperature as a function of h for the range $100 \le h \le 10{,}000\ \text{W/m}^2 \cdot \text{K}$. Is there a value of h below which operation would be unacceptable?

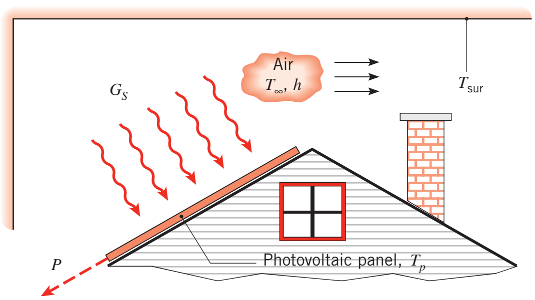
1.65 A freezer compartment is covered with a 2-mm-thick layer of frost at the time it malfunctions. If the compartment is in ambient air at 20°C and a coefficient of $h = 2\ \text{W/m}^2 \cdot \text{K}$ characterizes heat transfer by natural convection from the exposed surface of the layer, estimate the time required to completely melt the frost. The frost may be assumed to have a mass density of 700 kg/m^3 and a latent heat of fusion of 334 kJ/kg.

1.66 A vertical slab of Wood's metal is joined to a substrate on one surface and is melted as it is uniformly irradiated by a laser source on the opposite surface. The metal is initially at its fusion temperature of $T_f = 72°C$, and the melt runs off by gravity as soon as it is formed. The absorptivity of the metal to the laser radiation is $\alpha_1 = 0.4$, and its latent heat of fusion is $h_{sf} = 33$ kJ/kg.

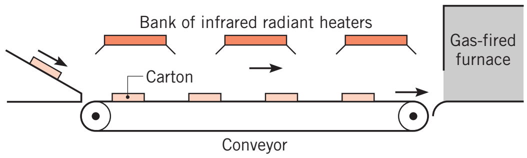
(a) Neglecting heat transfer from the irradiated surface by convection or radiation exchange with the surroundings, determine the instantaneous rate of melting in kg/s·m² if the laser irradiation is 5 kW/m². How much material is removed if irradiation is maintained for a period of 2 s?

(b) Allowing for convection to ambient air, with $T_\infty = 20°C$ and $h = 15$ W/m²·K, and radiation exchange with large surroundings ($\varepsilon = 0.4$, $T_{sur} = 20°C$), determine the instantaneous rate of melting during irradiation.

1.67 A photovoltaic panel of dimension 2 m × 4 m is installed on the roof of a home. The panel is irradiated with a solar flux of $G_S = 700$ W/m², oriented normal to the top panel surface. The absorptivity of the panel to the solar irradiation is $\alpha_S = 0.83$, and the efficiency of conversion of the absorbed flux to electrical power is $\eta = P/\alpha_S G_S A = 0.553 - 0.001\ \mathrm{K}^{-1} T_p$, where T_p is the panel temperature expressed in kelvins and A is the solar panel area. Determine the electrical power generated for (a) a still summer day, in which $T_{sur} = T_\infty = 35°C$, $h = 10$ W/m²·K, and (b) a breezy winter day, for which $T_{sur} = T_\infty = -15°C$, $h = 30$ W/m²·K. The panel emissivity is $\varepsilon = 0.90$.

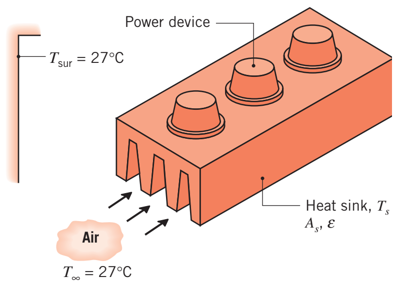


1.68 Following the hot vacuum forming of a paper-pulp mixture, the product, an egg carton, is transported on a conveyor for 18 s toward the entrance of a gas-fired oven where it is dried to a desired final water content. Very little water evaporates during the travel time. So, to increase the productivity of the line, it is proposed that a bank of infrared radiation heaters, which provide a uniform radiant flux of 5000 W/m², be installed over the conveyor. The carton has an exposed area of 0.0625 m² and a mass of 0.220 kg, 75% of which is water after the forming process.

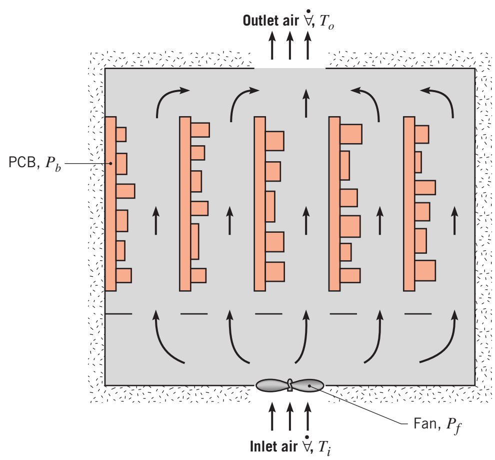


The chief engineer of your plant will approve the purchase of the heaters if they can reduce the water content by 10% of the total mass. Would you recommend the purchase? Assume the heat of vaporization of water is $h_{fg} = 2400$ kJ/kg.

1.69 Electronic power devices are mounted to a heat sink having an exposed surface area of 0.045 m² and an emissivity of 0.80. When the devices dissipate a total power of 20 W and the air and surroundings are at 27°C, the average sink temperature is 42°C. What average temperature will the heat sink reach when the devices dissipate 30 W for the same environmental condition?



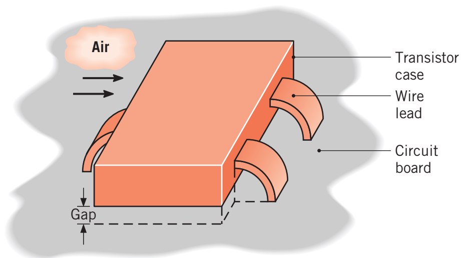
1.70 A computer consists of an array of five printed circuit boards (PCBs), each dissipating $P_b = 20$ W of power. Cooling of the electronic components on a board is provided by the forced flow of air, equally distributed in passages formed by adjoining boards, and the convection coefficient associated with heat transfer from the components to the air is approximately $h = 200$ W/m²·K. Air enters the computer console at a temperature of $T_i = 20°C$, and flow is driven by a fan whose power consumption is $P_f = 25$ W.



(a) If the temperature rise of the airflow, $(T_o - T_i)$, is not to exceed 15°C, what is the minimum allowable volumetric flow rate $\forall$ of the air? The density and specific heat of the air may be approximated as $\rho = 1.161$ kg/m^3 and $c_p = 1007$ J/kg·K, respectively.

(b) The component that is most susceptible to thermal failure dissipates 1 W/cm^2 of surface area. To minimize the potential for thermal failure, where should the component be installed on a PCB? What is its surface temperature at this location?

1.71 Consider a surface-mount type transistor on a circuit board whose temperature is maintained at 35°C. Air at 20°C flows over the upper surface of dimensions 4 mm × 8 mm with a convection coefficient of 50 W/m^2·K. Three wire leads, each of cross section 1 mm × 0.25 mm and length 4 mm, conduct heat from the case to the circuit board. The gap between the case and the board is 0.2 mm.



(a) Assuming the case is isothermal and neglecting radiation, estimate the case temperature when 150 mW is dissipated by the transistor and (i) stagnant air or (ii) a conductive paste fills the gap. The thermal conductivities of the wire leads, air, and conductive paste are 25, 0.0263, and 0.12 W/m·K, respectively.

$\boxed{\text{(b)}}$ Using the conductive paste to fill the gap, we wish to determine the extent to which increased heat dissipation may be accommodated, subject to the constraint that the case temperature not exceed 40°C. Options include increasing the air speed to achieve a larger convection coefficient h and/or changing the lead wire material to one of larger thermal conductivity. Independently considering leads fabricated from materials with thermal conductivities of 200 and 400 W/m·K, compute and plot the maximum allowable heat dissipation for variations in h over the range $50 \leq h \leq 250$ W/m^2·K.

1.72 The roof of a car in a parking lot absorbs a solar radiant flux of 800 W/m^2, and the underside is perfectly insulated. The convection coefficient between the roof and the ambient air is 12 W/m^2·K.

(a) Neglecting radiation exchange with the surroundings, calculate the temperature of the roof under steady-state conditions if the ambient air temperature is 20°C.

(b) For the same ambient air temperature, calculate the temperature of the roof if its surface emissivity is 0.8.

$\boxed{\text{(c)}}$ The convection coefficient depends on airflow conditions over the roof, increasing with increasing air speed. Compute and plot the roof temperature as a function of h for $2 \leq h \leq 200$ W/m^2·K.

1.73 Consider the conditions of Problem 1.22, but the surroundings temperature is 25°C and radiation exchange with the surroundings is not negligible. If the convection coefficient is 6.4 W/m^2·K and the emissivity of the plate is $\varepsilon = 0.42$, determine the time rate of change of the plate temperature, dT/dt, when the plate temperature is 225°C. Evaluate the heat loss by convection and the heat loss by radiation.

1.74 Most of the energy we consume as food is converted to thermal energy in the process of performing all our bodily functions and is ultimately lost as heat from our bodies. Consider a person who consumes 2100 kcal per day (note that what are commonly referred to as food calories are actually kilocalories), of which 2000 kcal is converted to thermal energy. (The remaining 100 kcal is used to do work on the environment.) The person has a surface area of 1.8 m^2 and is dressed in a bathing suit.

(a) The person is in a room at 20°C, with a convection heat transfer coefficient of 3 W/m^2·K. At this air temperature, the person is not perspiring much. Estimate the person's average skin temperature.

(b) If the temperature of the environment were 33°C, what rate of perspiration would be needed to maintain a comfortable skin temperature of 33°C?

1.75 Consider Problem 1.1.

(a) If the exposed cold surface of the insulation is at $T_2 = 20$°C, what is the value of the convection heat transfer coefficient on the cold side of the insulation if the surroundings temperature is $T_{sur} = 320$ K, the ambient temperature is $T_\infty = 5$°C, and the emissivity is $\varepsilon = 0.95$? Express your results in units of W/m^2·K and W/m^2·°C.

$\boxed{\text{(b)}}$ Using the convective heat transfer coefficient you calculated in part (a), determine the surface temperature, T_2, as the emissivity of the surface is varied over the range $0.05 \leq \varepsilon \leq 0.95$. The hot wall temperature of the insulation remains fixed at $T_1 = 30$°C. Display your results graphically.

1.76 The wall of an oven used to cure plastic parts is of thickness $L = 0.05$ m and is exposed to large surroundings and air at its outer surface. The air and the surroundings are at 300 K.

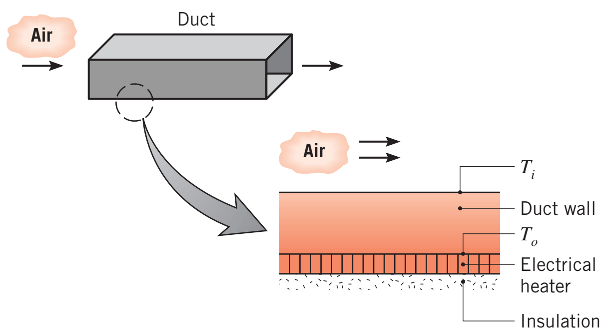
(a) If the temperature of the outer surface is 400 K and its convection coefficient and emissivity are

$h = 20\ \mathrm{W/m^2 \cdot K}$ and $\varepsilon = 0.8$, respectively, what is the temperature of the inner surface if the wall has a thermal conductivity of $k = 0.7\ \mathrm{W/m^2 \cdot K}$?

(b) Consider conditions for which the temperature of the inner surface is maintained at 600 K, while the air and large surroundings to which the outer surface is exposed are maintained at 300 K. Explore the effects of variations in k, h, and ε on (i) the temperature of the outer surface, (ii) the heat flux through the wall, and (iii) the heat fluxes associated with convection and radiation heat transfer from the outer surface. Specifically, compute and plot the foregoing dependent variables for parametric variations about baseline conditions of $k = 10\ \mathrm{W/m \cdot K}$, $h = 20\ \mathrm{W/m^2 \cdot K}$, and $\varepsilon = 0.5$. The suggested ranges of the independent variables are $0.1 \le k \le 400\ \mathrm{W/m \cdot K}$, $2 \le h \le 200\ \mathrm{W/m^2 \cdot K}$, and $0.05 \le \varepsilon \le 1$. Discuss the physical implications of your results. Under what conditions will the temperature of the outer surface be less than 45°C, which is a reasonable upper limit to avoid burn injuries if contact is made?

1.77 An experiment to determine the convection coefficient associated with airflow over the surface of a thick stainless steel casting involves the insertion of thermocouples into the casting at distances of 10 and 20 mm from the surface along a hypothetical line normal to the surface. The steel has a thermal conductivity of 15 W/m·K. If the thermocouples measure temperatures of 50 and 40°C in the steel when the air temperature is 100°C, what is the convection coefficient?

1.78 A thin electrical heating element provides a uniform heat flux q''_o to the outer surface of a duct through which airflows. The duct wall has a thickness of 10 mm and a thermal conductivity of 20 W/m·K.



(a) At a particular location, the air temperature is 30°C and the convection heat transfer coefficient between the air and inner surface of the duct is $100\ \mathrm{W/m^2 \cdot K}$. What heat flux q''_o is required to maintain the inner surface of the duct at $T_i = 85°\mathrm{C}$?

(b) For the conditions of part (a), what is the temperature (T_o) of the duct surface next to the heater?

(c) With $T_i = 85°\mathrm{C}$, compute and plot q''_o and T_o as a function of the air-side convection coefficient h for the range $10 \le h \le 200\ \mathrm{W/m^2 \cdot K}$. Briefly discuss your results.

1.79 A rectangular forced air heating duct is suspended from the ceiling of a basement whose air and walls are at a temperature of $T_\infty = T_{sur} = 5°\mathrm{C}$. The duct is 15 m long, and its cross section is 350 mm × 200 mm.

(a) For an uninsulated duct whose average surface temperature is 50°C, estimate the rate of heat loss from the duct. The surface emissivity and convection coefficient are approximately 0.5 and $4\ \mathrm{W/m^2 \cdot K}$, respectively.

(b) If heated air enters the duct at 58°C and a velocity of 4 m/s and the heat loss corresponds to the result of part (a), what is the outlet temperature? The density and specific heat of the air may be assumed to be $\rho = 1.10\ \mathrm{kg/m^3}$ and $c_p = 1008\ \mathrm{J/kg \cdot K}$, respectively.

1.80 Consider the steam pipe of Example 1.2. The facilities manager wants you to recommend methods for reducing the heat loss to the room, and two options are proposed. The first option would restrict air movement around the outer surface of the pipe and thereby reduce the convection coefficient by a factor of two. The second option would coat the outer surface of the pipe with a low emissivity ($\varepsilon = 0.4$) paint.

(a) Which of the foregoing options would you recommend?

(b) To prepare for a presentation of your recommendation to management, generate a graph of the heat loss q' as a function of the convection coefficient for $2 \le h \le 20\ \mathrm{W/m^2 \cdot K}$ and emissivities of 0.2, 0.4, and 0.8. Comment on the relative efficacy of reducing heat losses associated with convection and radiation.

1.81 During its manufacture, plate glass at 600°C is cooled by passing air over its surface such that the convection heat transfer coefficient is $h = 5\ \mathrm{W/m^2 \cdot K}$. To prevent cracking, it is known that the temperature gradient must not exceed 15°C/mm at any point in the glass during the cooling process. If the thermal conductivity of the glass is 1.4 W/m·K and its surface emissivity is 0.8, what is the lowest temperature of the air that can initially be used for the cooling? Assume that the temperature of the air equals that of the surroundings.

1.82 The curing process of Example 1.9 involves exposure of the plate to irradiation from an infrared lamp and attendant cooling by convection and radiation exchange with the surroundings. Alternatively, in lieu of the lamp, heating may be achieved by inserting the plate in an oven whose walls (the surroundings) are maintained at an elevated temperature.

(a) Consider conditions for which the oven walls are at 200°C, airflow over the plate is characterized by $T_\infty = 20°C$ and $h = 15\ W/m^2 \cdot K$, and the coating has an emissivity of $\varepsilon = 0.5$. What is the temperature of the plate?

$\boxed{\text{(b)}}$ For ambient air temperatures of 20, 40, and 60°C, determine the plate temperature as a function of the oven wall temperature over the range from 150 to 250°C. Plot your results, and identify conditions for which acceptable curing temperatures between 100 and 110°C may be maintained.

1.83 The diameter and surface emissivity of an electrically heated plate are $D = 300$ mm and $\varepsilon = 0.80$, respectively.

(a) Estimate the power needed to maintain a surface temperature of 200°C in a room for which the air and the walls are at 25°C. The coefficient characterizing heat transfer by natural convection depends on the surface temperature and, in units of $W/m^2 \cdot K$, may be approximated by an expression of the form $h = 0.80(T_s - T_\infty)^{1/3}$.

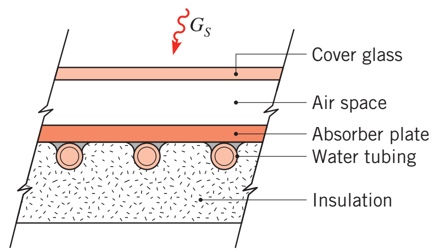
$\boxed{\text{(b)}}$ Assess the effect of surface temperature on the power requirement, as well as on the relative contributions of convection and radiation to heat transfer from the surface.

1.84 Bus bars proposed for use in a power transmission station have a rectangular cross section of height $H = 600$ mm and width $W = 200$ mm. The electrical resistivity, $\rho_e(\mu\Omega \cdot m)$, of the bar material is a function of temperature, $\rho_e = \rho_{e,o}[1 + \alpha(T - T_o)]$, where $\rho_{e,o} = 0.0828\ \mu\Omega \cdot m$, $T_o = 25°C$, and $\alpha = 0.0040\ K^{-1}$. The emissivity of the bar's painted surface is 0.8, and the temperature of the surroundings is 30°C. The convection coefficient between the bar and the ambient air at 30°C is 10 $W/m^2 \cdot K$.

(a) Assuming the bar has a uniform temperature T, calculate the steady-state temperature when a current of 60,000 A passes through the bar.

$\boxed{\text{(b)}}$ Compute and plot the steady-state temperature of the bar as a function of the convection coefficient for $10 \le h \le 100\ W/m^2 \cdot K$. What minimum convection coefficient is required to maintain a safe-operating temperature below 120°C? Will increasing the emissivity significantly affect this result?

1.85 A solar flux of 700 W/m^2 is incident on a flat-plate solar collector used to heat water. The area of the collector is 3 m^2, and 90% of the solar radiation passes through the cover glass and is absorbed by the absorber plate. The remaining 10% is reflected away from the collector. Water flows through the tube passages on the back side of the absorber plate and is heated from an inlet temperature T_i to an outlet temperature T_o. The cover glass, operating at a temperature of 30°C, has an emissivity of 0.94 and experiences radiation exchange with the sky at −10°C. The convection coefficient between the cover glass and the ambient air at 25°C is 10 $W/m^2 \cdot K$.



(a) Perform an overall energy balance on the collector to obtain an expression for the rate at which useful heat is collected per unit area of the collector, q''_u. Determine the value of q''_u.

(b) Calculate the temperature rise of the water, $T_o - T_i$, if the flow rate is 0.01 kg/s. Assume the specific heat of the water to be 4179 J/kg·K.

(c) The collector efficiency η is defined as the ratio of the useful heat collected to the rate at which solar energy is incident on the collector. What is the value of η?

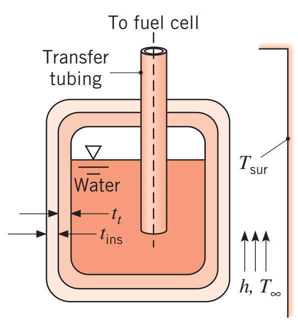
Process Identification

1.86 In analyzing the performance of a thermal system, the engineer must be able to identify the relevant heat transfer processes. Only then can the system behavior be properly quantified. For the following systems, identify the pertinent processes, designating them by appropriately labeled arrows on a sketch of the system. Answer additional questions that appear in the problem statement.

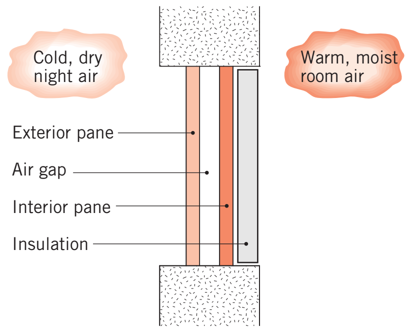
(a) Identify the heat transfer processes that determine the temperature of an asphalt pavement on a summer day. Write an energy balance for the surface of the pavement.

(b) Microwave radiation is known to be transmitted by plastics, glass, and ceramics but to be absorbed by materials having polar molecules such as water. Water molecules exposed to microwave radiation align and reverse alignment with the microwave radiation at frequencies up to $10^9\ s^{-1}$, causing heat to be generated. Contrast cooking in a microwave oven with cooking in a conventional radiant or convection oven. In each case, what is the physical mechanism responsible for heating the food? Which oven has the greater energy utilization efficiency? Why? Microwave heating is being considered for drying clothes. How would the operation of a microwave clothes dryer differ from a conventional dryer? Which is likely to have the greater energy utilization efficiency? Why?

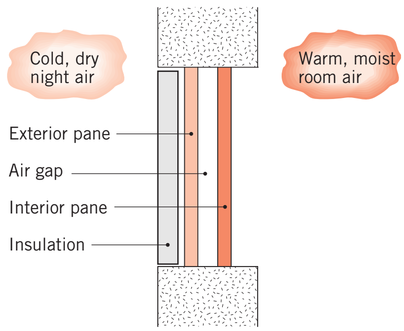
(c) To prevent freezing of the liquid water inside the fuel cell of an automobile, the water is drained to an onboard storage tank when the automobile is not in use. (The water is transferred from the tank back to the fuel cell when the automobile is turned on.) Consider a fuel cell–powered automobile that is parked outside on a very cold evening with $T_\infty = -20°C$. The storage tank is initially empty at $T_{i,t} = -20°C$, when liquid water, at atmospheric pressure and temperature $T_{i,w} = 50°C$, is introduced into the tank. The tank has a wall thickness t_t and is blanketed with insulation of thickness t_{ins}. Identify the heat transfer processes that will promote freezing of the water. Will the likelihood of freezing change as the insulation thickness is modified? Will the likelihood of freezing depend on the tank wall's thickness and material? Would freezing of the water be more likely if plastic (low thermal conductivity) or stainless steel (moderate thermal conductivity) tubing is used to transfer the water to and from the tank? Is there an optimal tank shape that would minimize the probability of the water freezing? Would freezing be more likely or less likely to occur if a thin sheet of aluminum foil (high thermal conductivity, low emissivity) is applied to the outside of the insulation?



(d) Your grandmother is concerned about reducing her winter heating bills. Her strategy is to loosely fit rigid polystyrene sheets of insulation over her double-pane windows right after the first freezing weather arrives in the autumn. Identify the relevant heat transfer processes on a cold winter night when the foamed insulation sheet is placed (i) on the inner surface and (ii) on the outer surface of her window. To avoid condensation damage, which configuration is preferred? Condensation on the window pane does not occur when the foamed insulation is not in place.

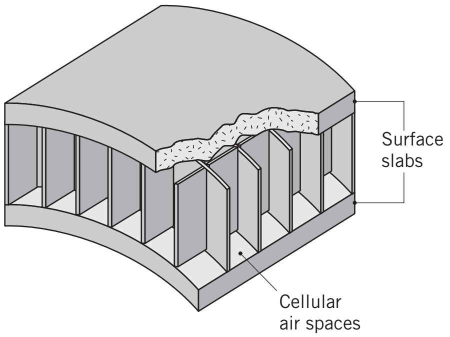


Insulation on inner surface

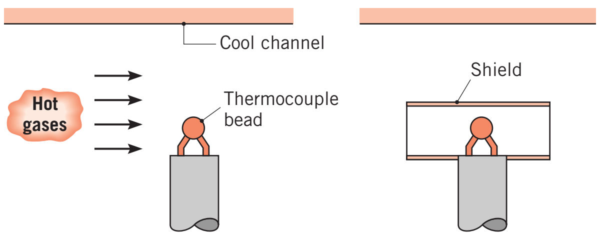


Insulation on outer surface

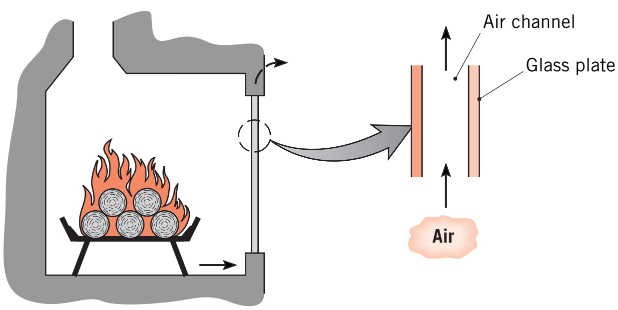
(e) There is considerable interest in developing building materials with improved insulating qualities. The development of such materials would do much to enhance energy conservation by reducing space heating requirements. It has been suggested that superior structural and insulating qualities could be obtained by using the composite shown. The material consists of a honeycomb, with cells of square cross section, sandwiched between solid slabs. The cells are filled with air, and the slabs, as well as the honeycomb matrix, are fabricated from plastics of low thermal conductivity. For heat transfer normal to the slabs, identify all heat transfer processes pertinent to the performance of the composite. Suggest ways in which this performance could be enhanced.



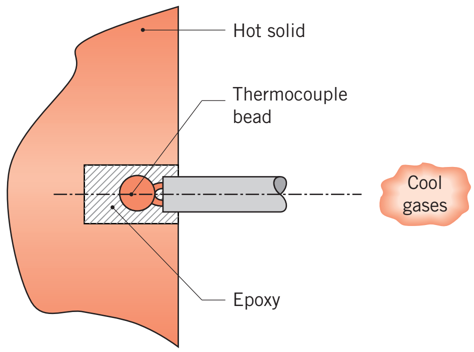
(f) A thermocouple junction (bead) is used to measure the temperature of a hot gas stream flowing through a channel by inserting the junction into the mainstream of the gas. The surface of the channel is cooled such that its temperature is well below that of the gas. Identify the heat transfer processes associated with the junction surface. Will the junction sense a temperature that is less than, equal to, or greater than the gas temperature? A radiation shield is a small, open-ended tube that encloses the thermocouple junction, yet allows for passage of the gas through the tube. How does use of such a shield improve the accuracy of the temperature measurement?



(g) A double-glazed, glass fire screen is inserted between a wood-burning fireplace and the interior of a room. The screen consists of two vertical glass plates that are separated by a space through which room air may flow (the space is open at the top and bottom). Identify the heat transfer processes associated with the fire screen.

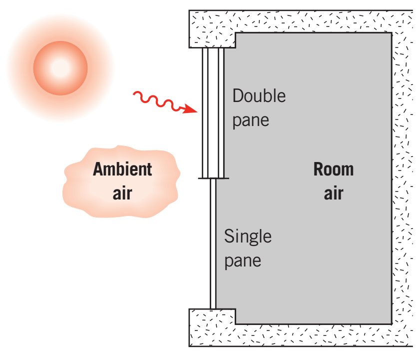


(h) A thermocouple junction is used to measure the temperature of a solid material. The junction is inserted into a small circular hole and is held in place by epoxy. Identify the heat transfer processes associated with the junction. Will the junction sense a temperature less than, equal to, or greater than the solid temperature? How will the thermal conductivity of the epoxy affect the junction temperature?



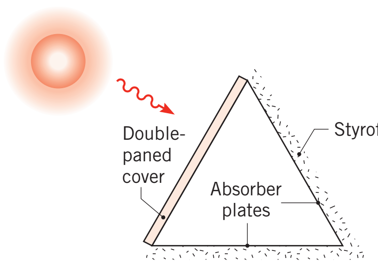
1.87 In considering the following problems involving heat transfer in the natural environment (outdoors), recognize that solar radiation is comprised of long and short wavelength components. If this radiation is incident on a *semi-transparent medium*, such as water or glass, two things will happen to the nonreflected portion of the radiation. The long wavelength component will be absorbed at the surface of the medium, whereas the short wavelength component will be transmitted by the surface.

(a) The number of panes in a window can strongly influence the heat loss from a heated room to the outside ambient air. Compare the single- and double-paned units shown by identifying relevant heat transfer processes for each case.



(b) In a typical flat-plate solar collector, energy is collected by a working fluid that is circulated through tubes that are in good contact with the back face of an absorber plate. The back face is insulated from the surroundings, and the absorber plate receives solar radiation on its front face, which is typically covered by one or more transparent plates. Identify the relevant heat transfer processes, first for the absorber plate with no cover plate and then for the absorber plate with a single cover plate.

(c) The solar energy collector design shown in the schematic has been used for agricultural applications. Air is blown through a long duct whose cross section is in the form of an equilateral triangle. One side of the triangle is comprised of a double-paned, semitransparent cover; the other two sides are constructed from aluminum sheets painted flat black on the inside and covered on the outside with a layer of styrofoam insulation. During sunny periods, air entering the system is heated for delivery to either a greenhouse, grain drying unit, or storage system.



Identify all heat transfer processes associated with the cover plates, the absorber plate(s), and the air.

(d) Evacuated-tube solar collectors are capable of improved performance relative to flat-plate collectors. The design consists of an inner tube enclosed in an outer tube that is transparent to solar radiation. The annular space between the tubes is evacuated. The outer, opaque surface of the inner tube absorbs solar radiation, and a working fluid is passed through the tube to collect the solar energy. The collector design generally consists of a row of such tubes arranged in front of a reflecting panel. Identify all heat transfer processes relevant to the performance of this device.