ME225 - Heat Transfer
A core curriculum of Mechanical Engineering

- Lecture Slides for ME225 Course
- Date: 2010 Spring
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Chapter 1
Introduction

Basic Concepts

Heat: Energy that can be transferred from one system to another as a result of temp. difference.

Thermodynamics is concerned with the amount of heat transfer as a system undergoes a process from one *equilibrium* state to another. However, in practice we are more concerned about the rate of heat transfer (heat transfer per unit time) than we are with the amount of it.

Example: How long will it take for hot coffee to cool down from 90 °C to 80 °C?

Heat Transfer

- Determination of rates of heat transfer (cooling/heat time)
- Variation of temperature

In contrast to thermodynamics, which deals with equilibrium states, *heat transfer* deals with the systems that lack thermal equilibrium and thus it is a *non-equilibrium* phenomenon.

Some Applications of Heat Transfer

Daily life

Solar energy

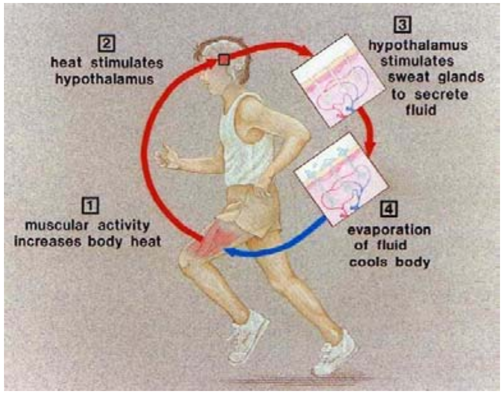
Buildings

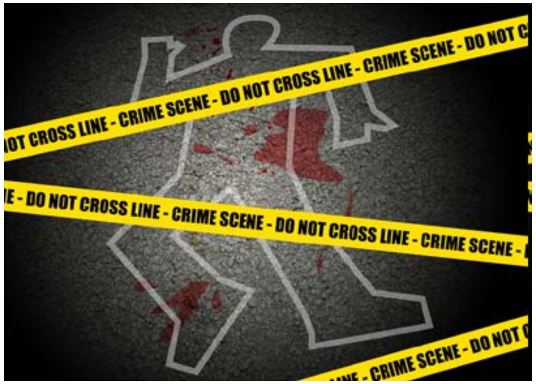
Electronics

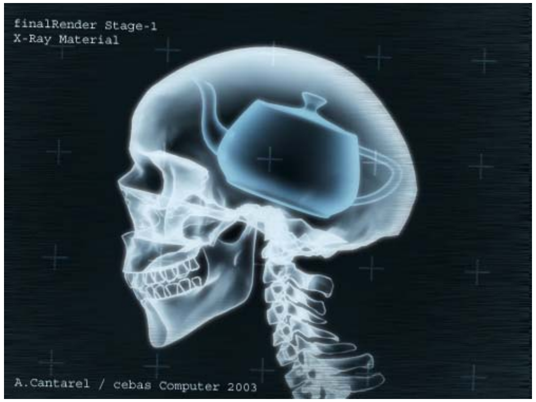
Heat exchangers

Combustion

Human Body


2
heat stimulates
hypothalamus
3
hypothalamus
stimulates
sweat glands
to secrete
fluid
1
muscular activity
increases body heat
4
evaporation
of fluid
cools body


CRIME SCENE - DO NOT CROSS LINE - CRIME SCENE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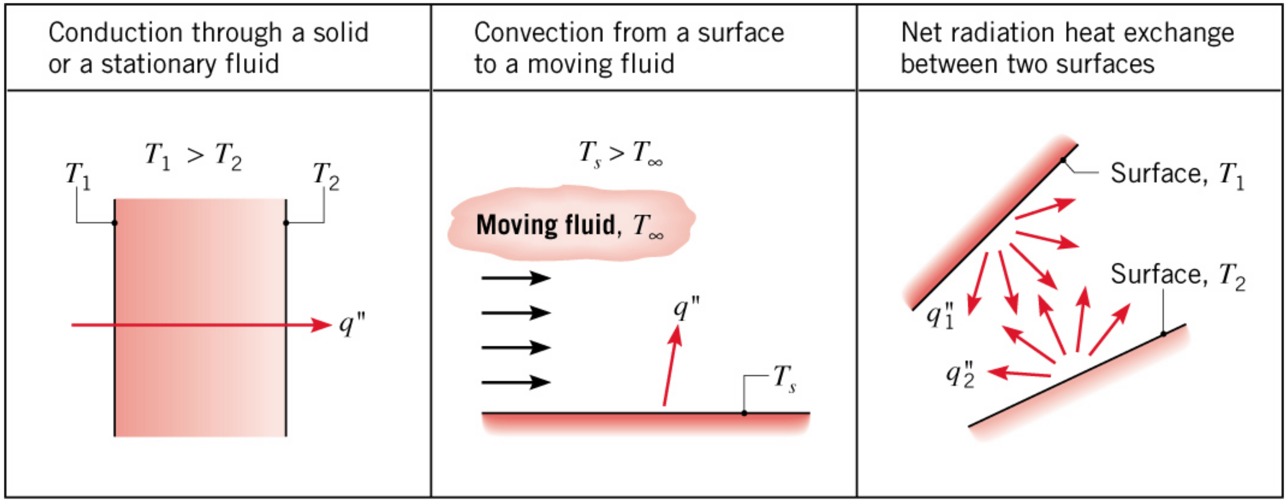
Modes of Heat Transfer



Conduction: Heat transfer in a solid or a stationary fluid (gas or liquid) due to the random motion of its constituent atoms, molecules and /or electrons.

Convection: Heat transfer due to the combined influence of bulk and random motion for fluid flow over a surface.

Radiation: Energy that is emitted by matter due to changes in the electron configurations of its atoms or molecules and is transported as electromagnetic waves (or photons).

- Conduction and convection require the presence of temperature variations in a material medium.
- Although radiation originates from matter, its transport does not require a material medium and occurs most efficiently in a vacuum.

Conduction

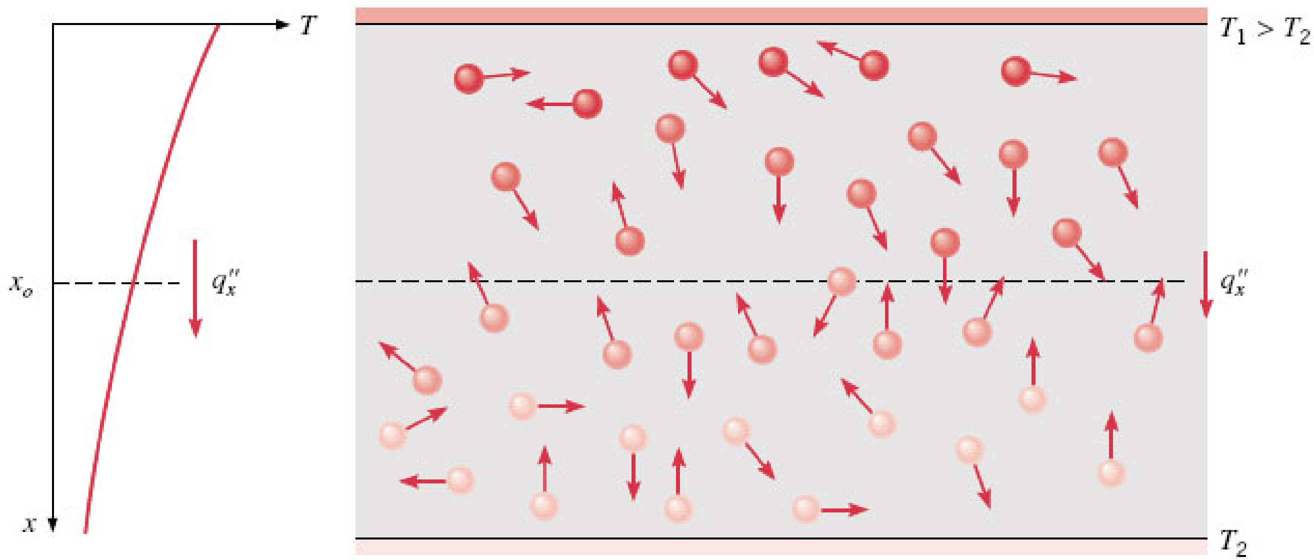
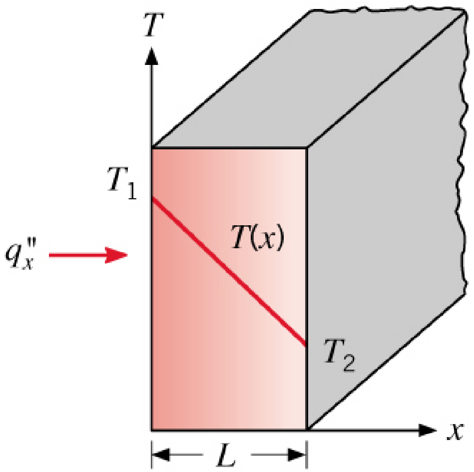


FIGURE 1.2 Association of conduction heat transfer with diffusion of energy due to molecular activity.



Fourier's Law:

$$q''_x = -k\frac{dT}{dx} = -k\frac{T_2 - T_1}{L}$$

Convection

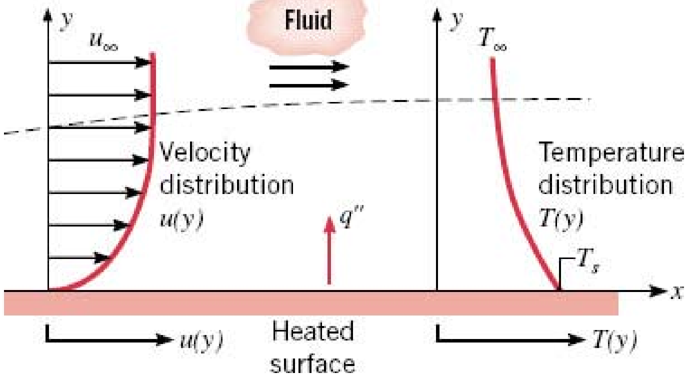


FIGURE 1.4 Boundary layer development in convection heat transfer.

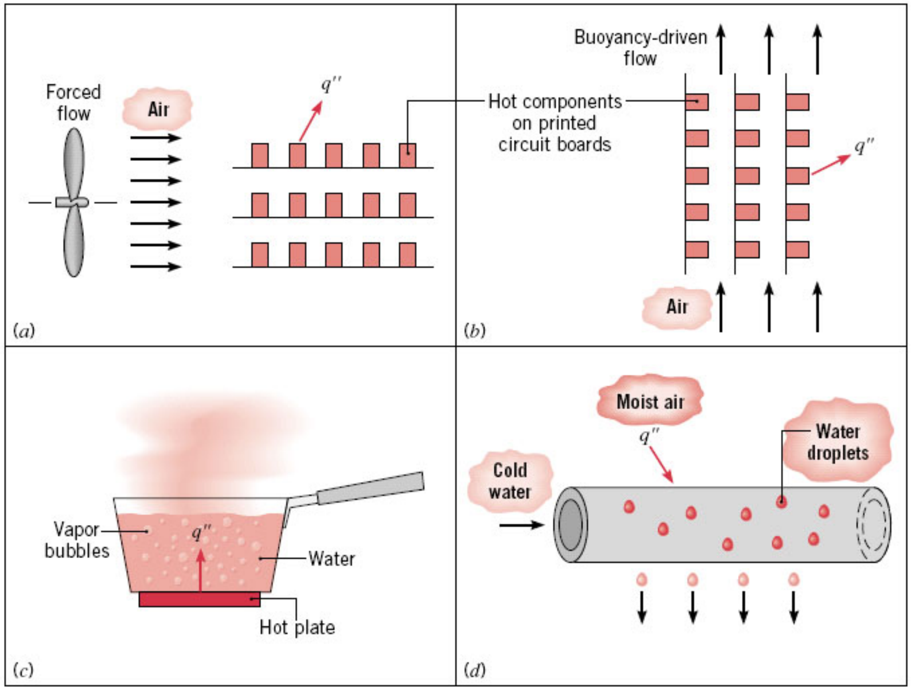


FIGURE 1.5 Convection heat transfer processes. (*a*) Forced convection. (*b*) Natural convection. (*c*) Boiling. (*d*) Condensation.

Newton's law: $q'' = \mathrm{h}\left(T_s - T_\infty\right)$

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

Radiation

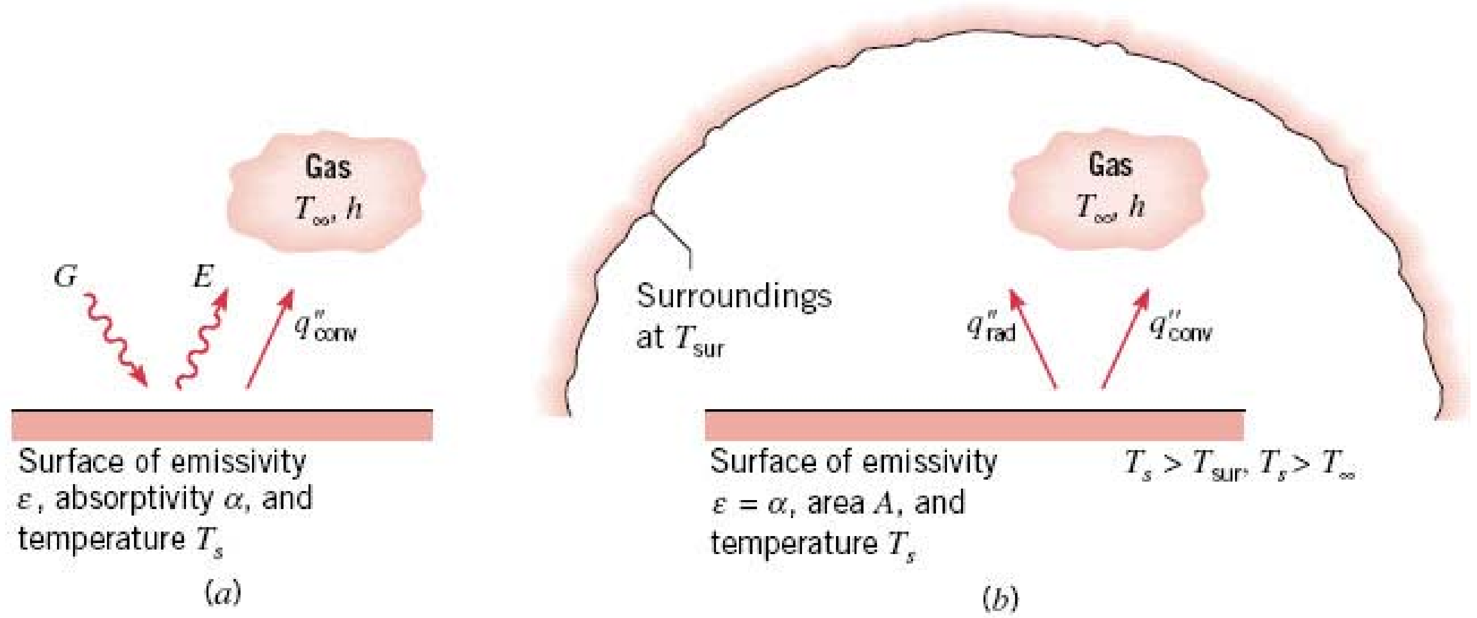


FIGURE 1.6 Radiation exchange: (*a*) at a surface and (*b*) between a surface and large surroundings.

Stefan-Boltzmann Law:

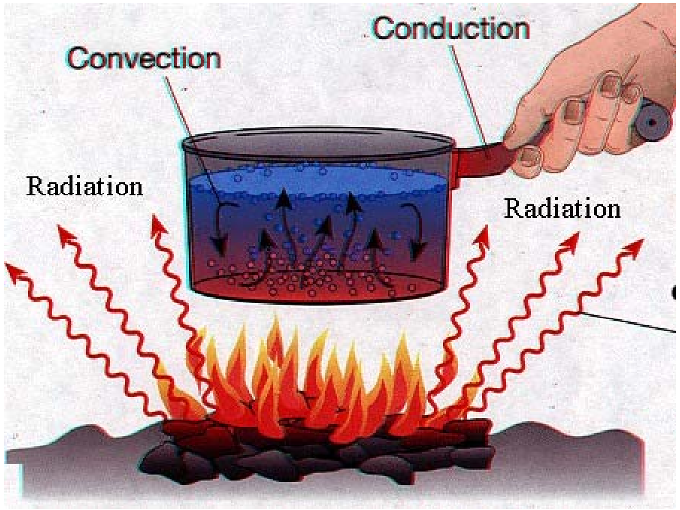
$$E = \varepsilon E_b = \varepsilon \sigma T_s^4$$

If $\alpha = \varepsilon$, the net radiation heat flux from the surface due to exchange with the surroundings is:

$$q''_{rad} = \varepsilon E_b(T_s) - \alpha G = \varepsilon\sigma\left(T_s^4 - T_{sur}^4\right)$$

TABLE 1.5 Summary of heat transfer processes

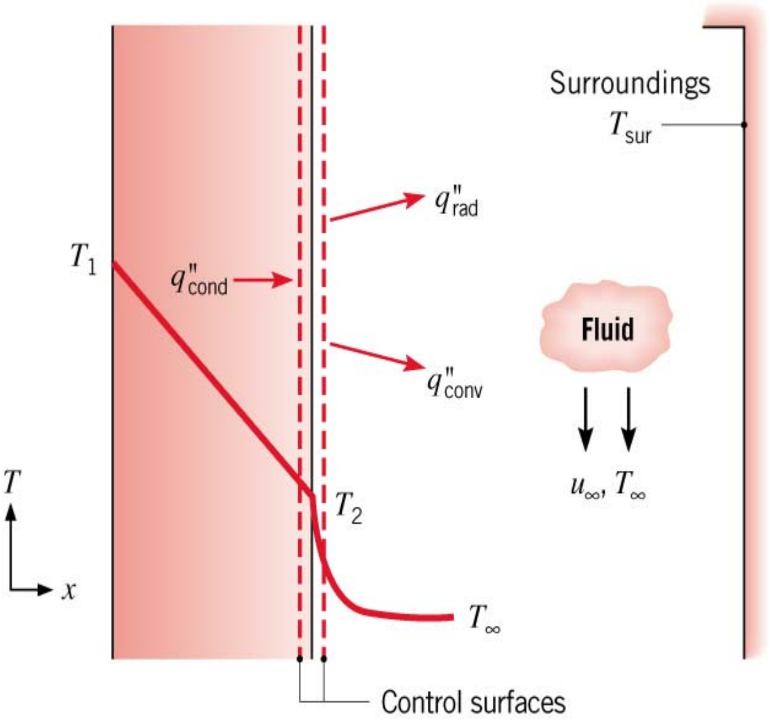
Mode	Mechanism(s)	Rate Equation	Equation Number	Transport Property or Coefficient
Conduction	Diffusion of energy due to random molecular motion	$q''_x\ (\mathrm{W/m^2}) = -k\dfrac{dT}{dx}$	(1.1)	$k\ (\mathrm{W/m \cdot K})$
Convection	Diffusion of energy due to random molecular motion plus energy transfer due to bulk motion (advection)	$q''(\mathrm{W/m^2}) = h(T_s - T_\infty)$	(1.3a)	$h\ (\mathrm{W/m^2 \cdot K})$
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 $h_r\ (\mathrm{W/m^2 \cdot K})$



Surface Energy Balance

- With no mass and volume, energy storage and generation are not pertinent to the energy balance of surface.

Consider the surface of a wall with heat transfer by conduction, convection and radiation.



Conservation of Energy:

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

$$k\frac{T_1 - T_2}{L} = h\left(T_2 - T_\infty\right) + \varepsilon_2\sigma\left(T_2^4 - T_{sur}^4\right)$$

Chapter 2
Introduction to Conduction

Fourier's Law

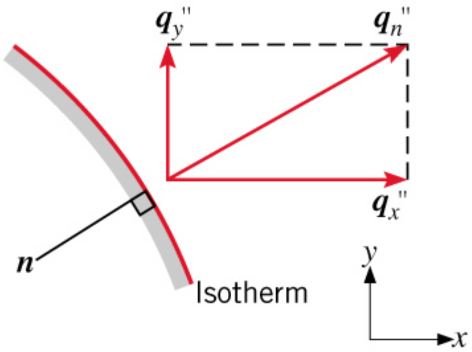
- A rate equation that allows determination of the conduction heat flux from knowledge of the temperature distribution in a medium

- Its most general (vector) form for multidimensional conduction is:

$$\vec{q}'' = -k\,\vec{\nabla}\,T$$

Implications:

- Heat transfer is in the direction of decreasing temperature (basis for minus sign).

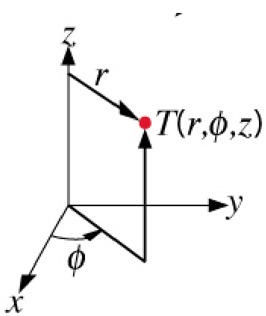


- Direction of heat transfer is perpendicular to lines of constant temperature (isotherms).

- Heat flux vector may be resolved into orthogonal components

- Cartesian Coordinates: $T(x,y,z)$

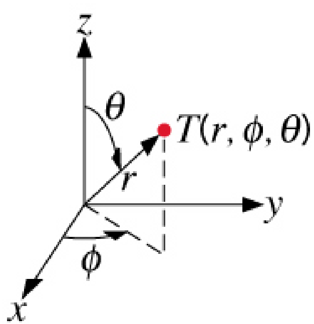
$$\vec{q''} = \underbrace{-k\frac{\partial T}{\partial x}\vec{i}}_{q''_x} \underbrace{-k\frac{\partial T}{\partial y}\vec{j}}_{q''_y} \underbrace{-k\frac{\partial T}{\partial z}\vec{k}}_{q''_z} \qquad (2.3)$$



- Cylindrical Coordinates: $T(r,\phi,z)$

$$\vec{q''} = \underbrace{-k\frac{\partial T}{\partial r}\vec{i}}_{q''_r} \underbrace{-k\frac{\partial T}{r\partial \phi}\vec{j}}_{q''_\phi} \underbrace{-k\frac{\partial T}{\partial z}\vec{k}}_{q''_z} \qquad (2.22)$$

$$q_r = A_r q''_r = 2\pi r L q''_r$$



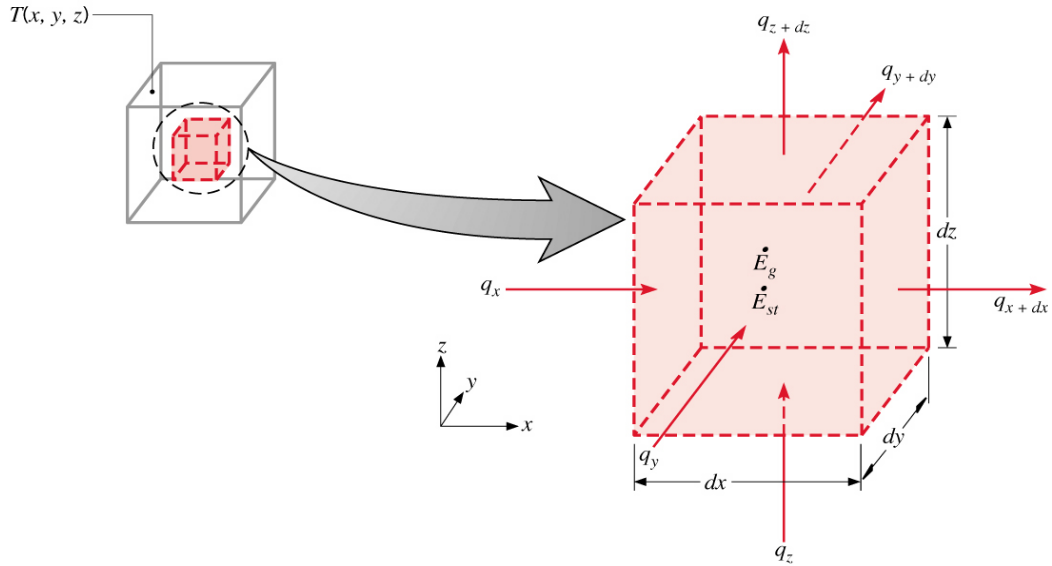
- Spherical Coordinates: $T(r,\phi,\theta)$

$$\vec{q''} = \underbrace{-k\frac{\partial T}{\partial r}\vec{i}}_{q''_r} \underbrace{-k\frac{\partial T}{r\partial \theta}\vec{j}}_{q''_\theta} \underbrace{-k\frac{\partial T}{r\sin\theta\,\partial \phi}\vec{k}}_{q''_\phi} \qquad (2.25)$$

$$q_r = A_r q''_r = 4\pi r^2 q''_r$$

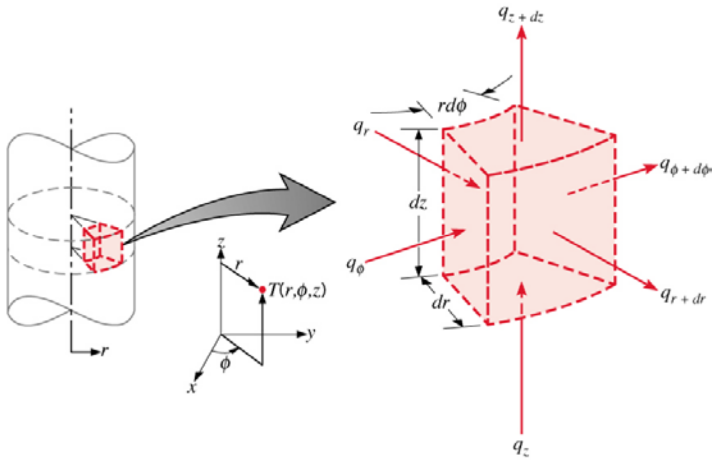
The Heat Equation

- A differential equation whose solution provides the temperature distribution in a stationary medium.
- Based on applying conservation of energy to a differential control volume through which energy transfer is exclusively by conduction.
- Cartesian Coordinates:



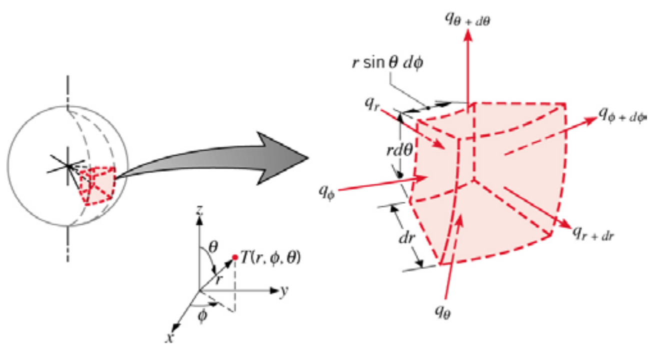
$$\underbrace{\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)}_{\text{Net transfer of thermal energy into the control volume (inflow-outflow)}}+\underbrace{\dot{q}}_{\text{Thermal energy generation}}=\underbrace{\rho c_p\frac{\partial T}{\partial t}}_{\text{Change in thermal energy storage}} \qquad (2.17)$$

- Cylindrical Coordinates:



$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial}{\partial \phi}\left(k\frac{\partial T}{\partial \phi}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{q}=\rho c_p\frac{\partial T}{\partial t} \qquad (2.24)$$

- Spherical Coordinates:

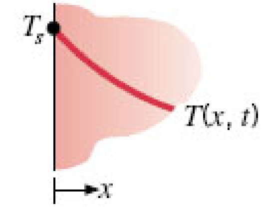


$$\frac{1}{r^2}\frac{\partial}{\partial r}\left(kr^2\frac{\partial T}{\partial r}\right)+\frac{1}{r^2\sin^2\theta}\frac{\partial}{\partial \phi}\left(k\frac{\partial T}{\partial \phi}\right)+\frac{1}{r^2\sin\theta}\frac{\partial}{\partial \theta}\left(k\sin\theta\frac{\partial T}{\partial \theta}\right)+\dot{q}=\rho c_p\frac{\partial T}{\partial t} \qquad (2.27)$$

TABLE 2.2 Boundary conditions for the heat diffusion equation at the surface ($x = 0$)

1. Constant surface temperature

$$T(0, t) = T_s \tag{2.29}$$



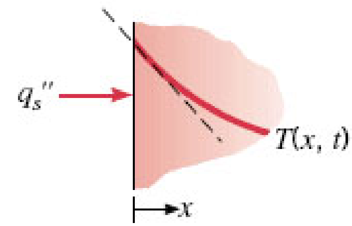
Example:

Phase change

2. Constant surface heat flux

(a) Finite heat flux

$$-k \frac{\partial T}{\partial x}\bigg|_{x=0} = q_s'' \tag{2.30}$$

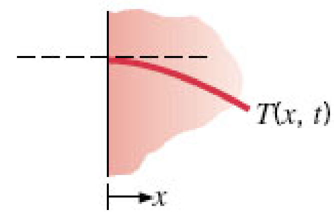


Example:

Heat tape

(b) Adiabatic or insulated surface

$$\frac{\partial T}{\partial x}\bigg|_{x=0} = 0 \tag{2.31}$$

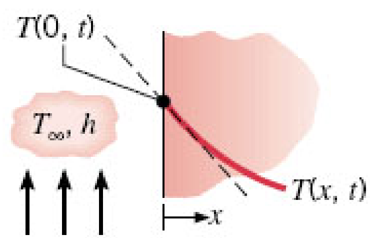


Example:

Insulation

3. Convection surface condition

$$-k \frac{\partial T}{\partial x}\bigg|_{x=0} = h[T_\infty - T(0, t)] \tag{2.32}$$



Example:

Car window

Thermophysical Properties

Thermal conductivity, *k*: A measure of a material's ability to transfer thermal energy by conduction.

Thermal diffusivity, α: A measure of a material's ability to respond to changes in its thermal environment.

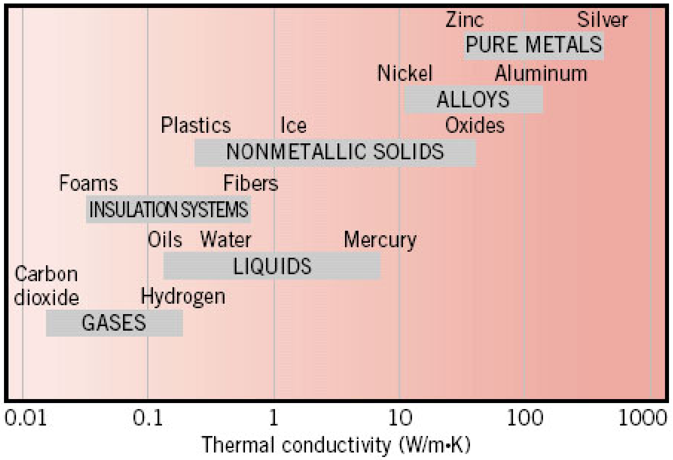


FIGURE 2.4 Range of thermal conductivity for various states of matter at normal temperatures and pressure.

Property Tables:

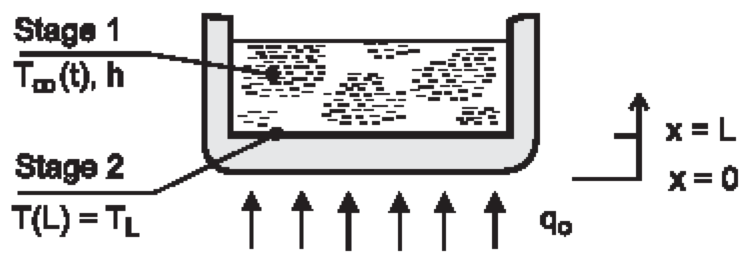
Solids:	Tables A.1 – A.3
Gases:	Table A.4
Liquids:	Tables A.5 – A.7

PROBLEM 2.21

KNOWN: Diameter D, thickness L and initial temperature T_i of pan. Heat rate from stove to bottom of pan. Convection coefficient h and variation of water temperature $T_\infty(t)$ during Stage 1. Temperature T_L of pan surface in contact with water during Stage 2.

FIND: Form of heat equation and boundary conditions associated with the two stages.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional conduction in pan bottom, (2) Heat transfer from stove is uniformly distributed over surface of pan in contact with the stove, (3) Constant properties.

Chapter 5
Transient Conduction

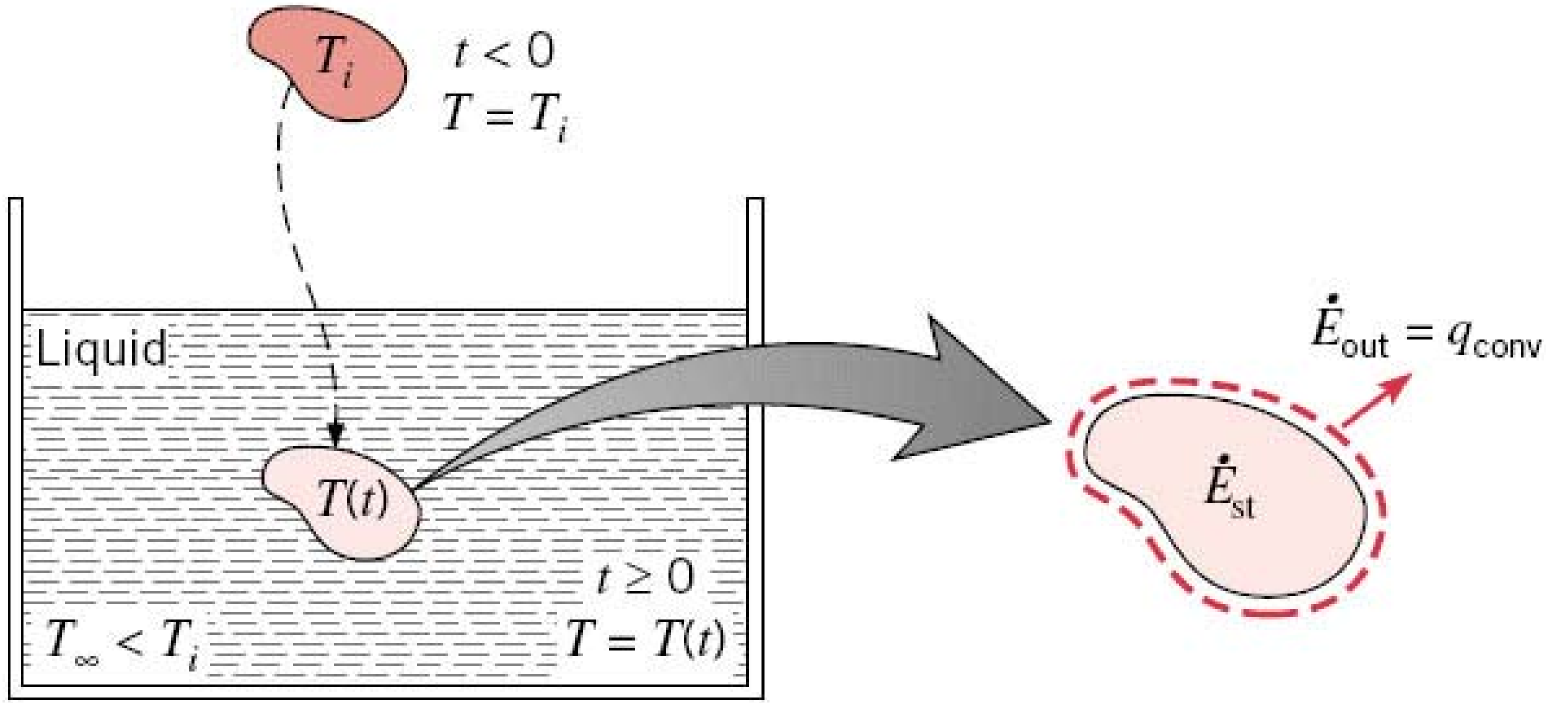


FIGURE 5.1 Cooling of a hot metal forging.

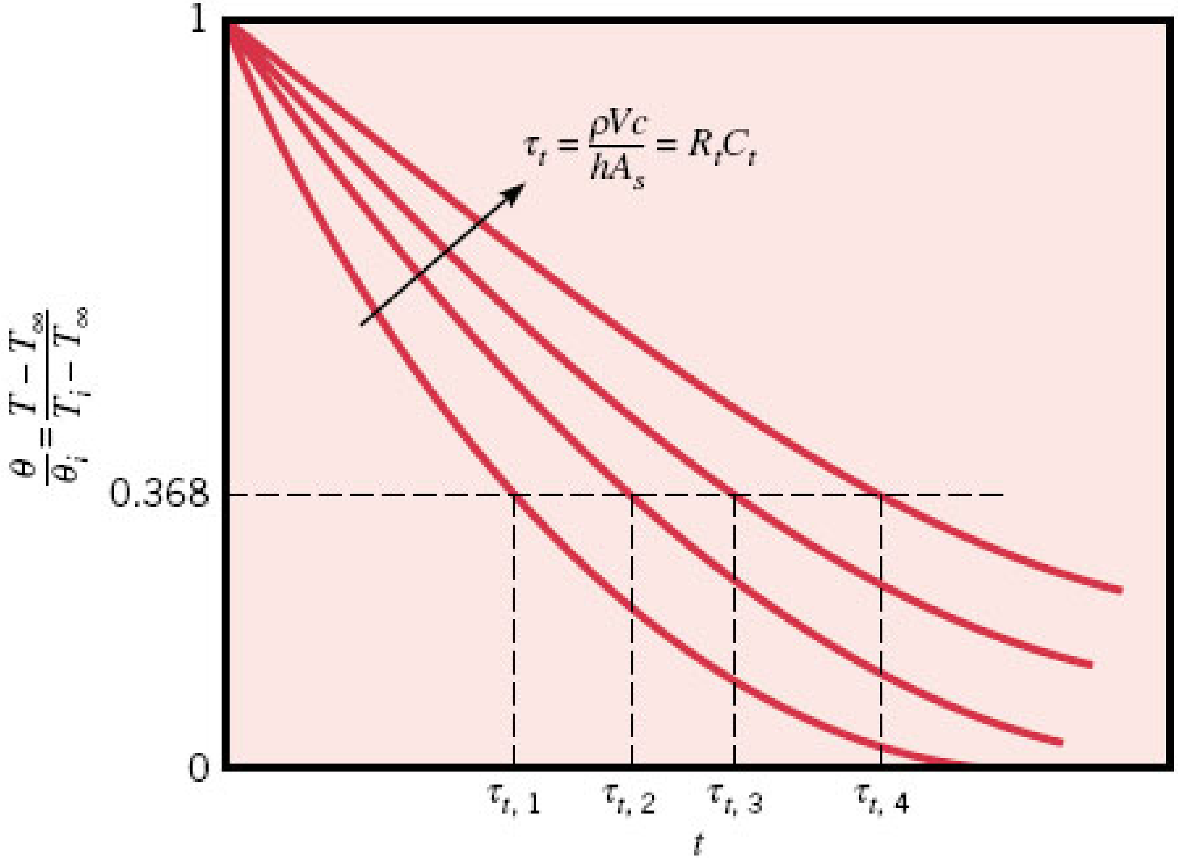


FIGURE 5.2 Transient temperature response of lumped capacitance solids for different thermal time constants τ_t.

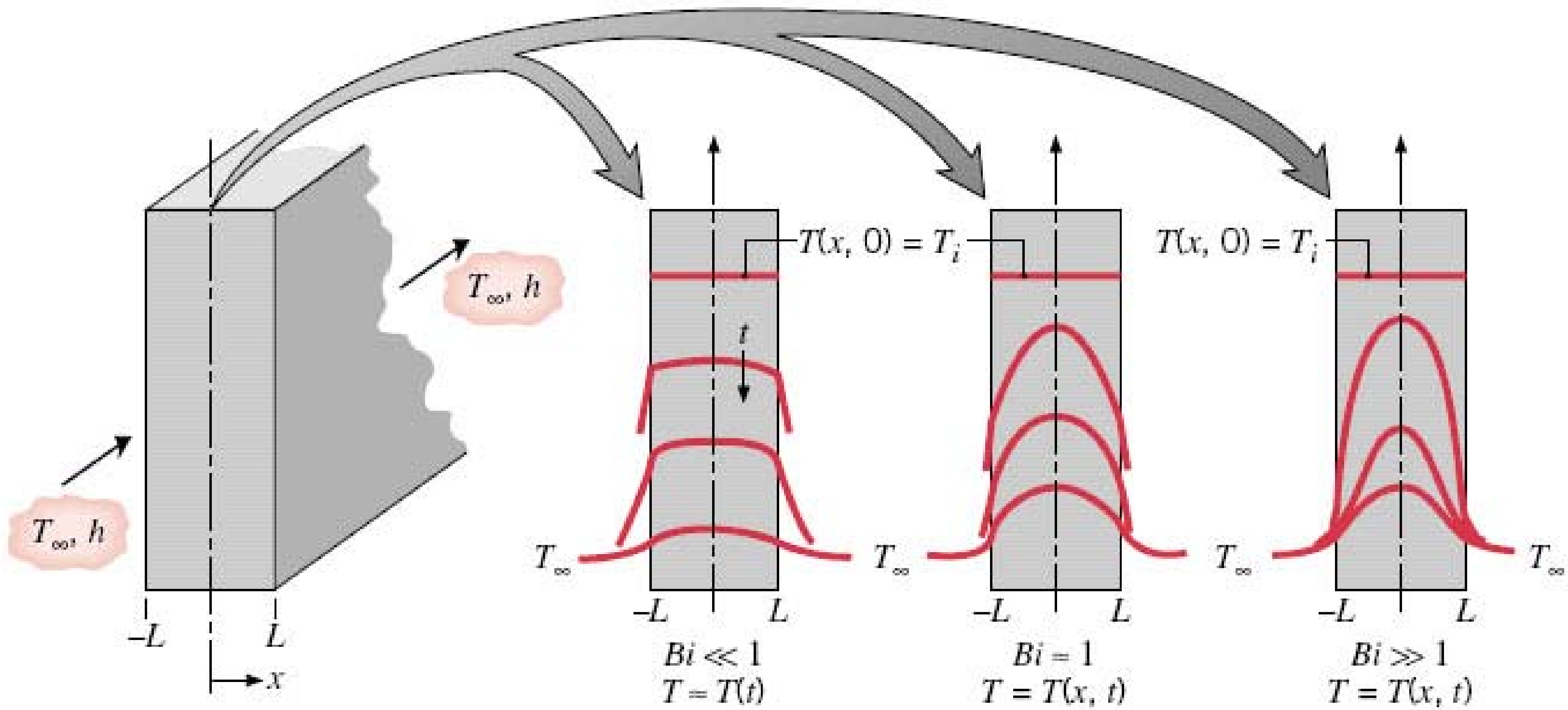


FIGURE 5.4 Transient temperature distributions for different Biot numbers in a plane wall symmetrically cooled by convection.

Solution to the Heat Equation for a Plane Wall with Symmetrical Convection Conditions

- If the lumped capacitance approximation can not be made, consideration must be given to spatial, as well as temporal, variations in temperature during the transient process.

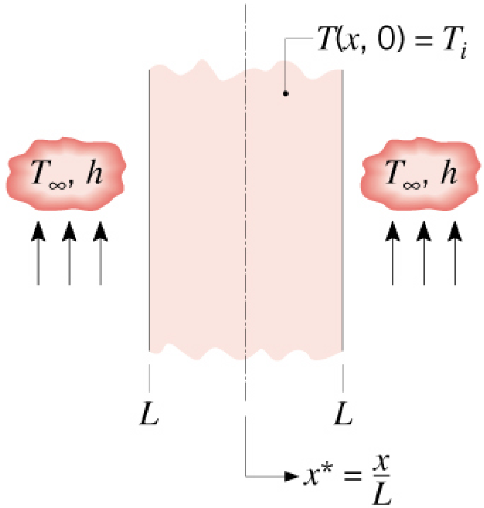
  - For a plane wall with symmetrical convection conditions and constant properties, the heat equation and initial/boundary conditions are:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \qquad (5.26)$$

$$T(x,0) = T_i \qquad (5.27)$$

$$\left.\frac{\partial T}{\partial x}\right|_{x=0} = 0 \qquad (5.28)$$

$$-k\left.\frac{\partial T}{\partial x}\right|_{x=L} = h\left[T(L,t) - T_\infty\right] \qquad (5.29)$$



- Existence of seven independent variables:

$$T = T\left(x,t,T_i,T_\infty,k,\alpha,h\right) \qquad (5.30)$$

How may the functional dependence be simplified?

- Non-dimensionalization of Heat Equation and Initial/Boundary Conditions:

Dimensionless temperature difference: $\theta^* \equiv \frac{\theta}{\theta_i} = \frac{T - T_\infty}{T_i - T_\infty}$

Dimensionless coordinate: $x^* \equiv \frac{x}{L}$

Dimensionless time: $t^* \equiv \frac{\alpha t}{L^2} \equiv Fo$ The Fourier Number

The Biot Number: $Bi \equiv \frac{hL}{k_{solid}}$

$$\theta^* = f\left(x^*, Fo, Bi\right)$$

- Exact Solution:

$$\theta^* = \sum_{n=1}^{\infty} C_n \exp\left(-\zeta_n^2 Fo\right)\cos\left(\zeta_n x^*\right) \quad (5.39a)$$

$$C_n = \frac{4\sin\zeta_n}{2\zeta_n + \sin\left(2\zeta_n\right)} \qquad \zeta_n \tan\zeta_n = Bi \quad (5.39b,c)$$

See Appendix B.3 for first four roots (Eigenvalues $\zeta_1, \ldots, \zeta_4$) of Eq. (5.39c)

- The One-Term Approximation $\left(Fo > 0.2\right)$:
  - Variation of midplane temperature ($x^*= 0$) with time$\left(Fo\right)$:

$$\theta_o^* \equiv \frac{\left(T_o - T_\infty\right)}{\left(T_i - T_\infty\right)} \approx C_1 \exp\left(-\zeta_1^2 Fo\right) \tag{5.41}$$

Table 5.1 $\rightarrow$ C_1 and ζ_1 as a function of Bi

  - Variation of temperature with location (x^*) and time$\left(Fo\right)$:

$$\theta^* = \theta_o^* \cos\left(\zeta_1 x^*\right) \tag{5.40b}$$

  - Change in thermal energy storage with time:

$$\Delta E_{st} = -Q \tag{5.43a}$$

$$Q = Q_o\left(1 - \frac{\sin\zeta_1}{\zeta_1}\theta_o^*\right) \tag{5.46}$$

$$Q_o = \rho c V\left(T_i - T_\infty\right) \tag{5.44}$$

Can the foregoing results be used for a plane wall that is well insulated on one side and convectively heated or cooled on the other?

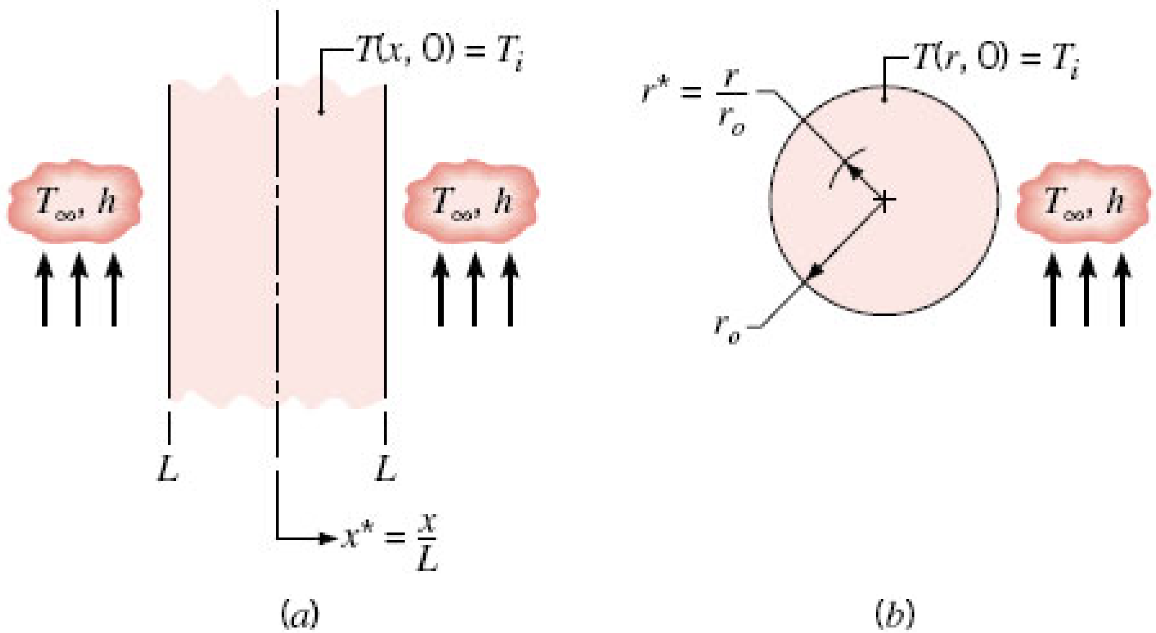


FIGURE 5.6 One-dimensional systems with an initial uniform temperature subjected to sudden convection conditions. (*a*) Plane wall. (*b*) Infinite cylinder or sphere.

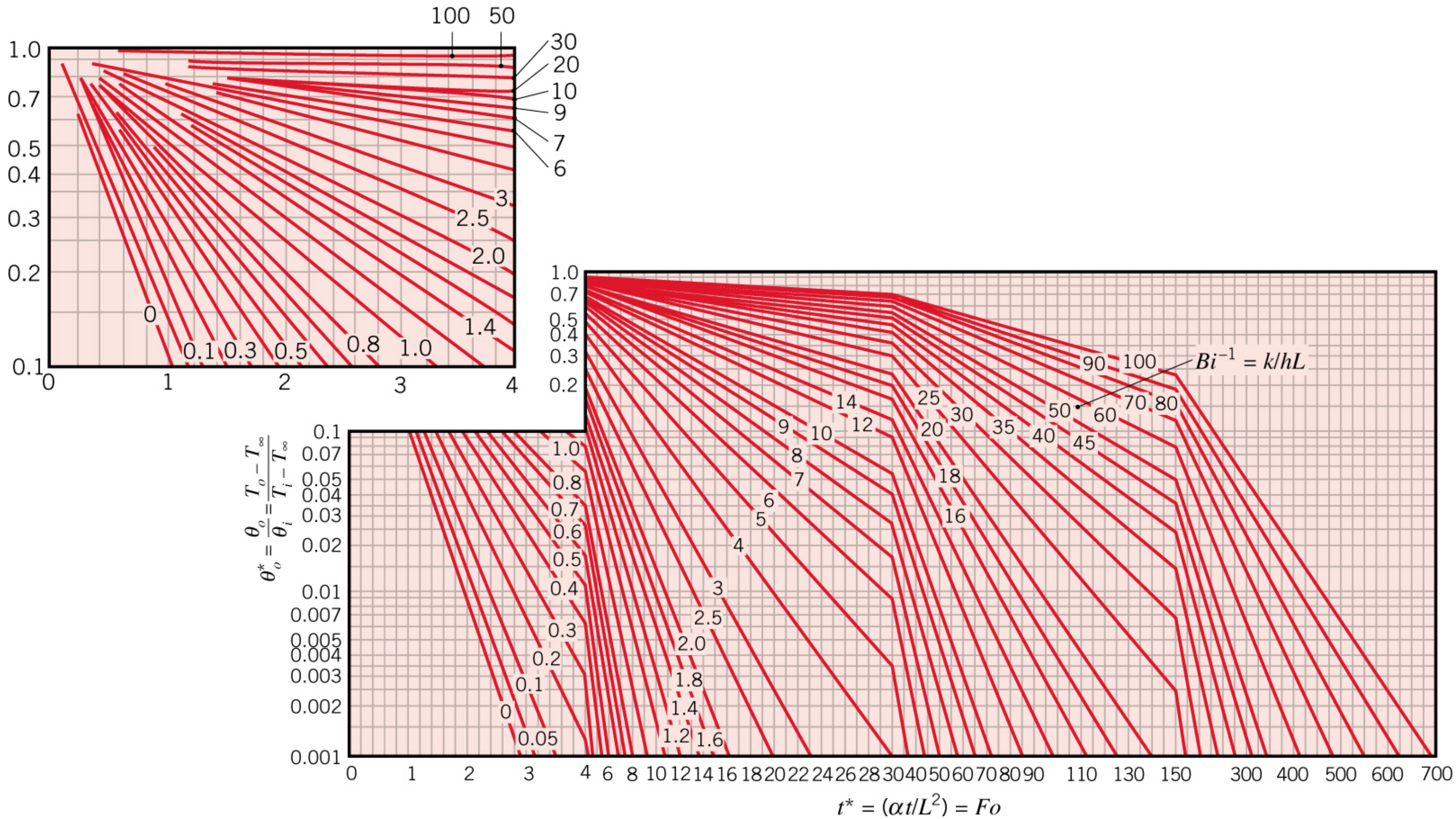
TABLE 5.1 Coefficients used in the one-term approximation to the series solutions for transient one-dimensional conduction

	Plane Wall		Infinite Cylinder		Sphere	
Bi^a	ζ_1 (rad)	C_1	ζ_1 (rad)	C_1	ζ_1 (rad)	C_1
0.01	0.0998	1.0017	0.1412	1.0025	0.1730	1.0030
0.02	0.1410	1.0033	0.1995	1.0050	0.2445	1.0060
0.03	0.1723	1.0049	0.2440	1.0075	0.2991	1.0090
0.04	0.1987	1.0066	0.2814	1.0099	0.3450	1.0120
0.05	0.2218	1.0082	0.3143	1.0124	0.3854	1.0149
0.06	0.2425	1.0098	0.3438	1.0148	0.4217	1.0179
0.07	0.2615	1.0114	0.3709	1.0173	0.4551	1.0209
0.08	0.2791	1.0130	0.3960	1.0197	0.4860	1.0239
0.09	0.2956	1.0145	0.4195	1.0222	0.5150	1.0268
0.10	0.3111	1.0161	0.4417	1.0246	0.5423	1.0298
0.15	0.3779	1.0237	0.5376	1.0365	0.6609	1.0445
0.20	0.4328	1.0311	0.6170	1.0483	0.7593	1.0592
0.25	0.4801	1.0382	0.6856	1.0598	0.8447	1.0737
0.30	0.5218	1.0450	0.7465	1.0712	0.9208	1.0880
0.4	0.5932	1.0580	0.8516	1.0932	1.0528	1.1164
0.5	0.6533	1.0701	0.9408	1.1143	1.1656	1.1441
0.6	0.7051	1.0814	1.0184	1.1345	1.2644	1.1713
0.7	0.7506	1.0919	1.0873	1.1539	1.3525	1.1978
0.8	0.7910	1.1016	1.1490	1.1724	1.4320	1.2236
0.9	0.8274	1.1107	1.2048	1.1902	1.5044	1.2488
1.0	0.8603	1.1191	1.2558	1.2071	1.5708	1.2732
2.0	1.0769	1.1785	1.5994	1.3384	2.0288	1.4793
3.0	1.1925	1.2102	1.7887	1.4191	2.2889	1.6227
4.0	1.2646	1.2287	1.9081	1.4698	2.4556	1.7202
5.0	1.3138	1.2402	1.9898	1.5029	2.5704	1.7870
6.0	1.3496	1.2479	2.0490	1.5253	2.6537	1.8338
7.0	1.3766	1.2532	2.0937	1.5411	2.7165	1.8673
8.0	1.3978	1.2570	2.1286	1.5526	1.7654	1.8920
9.0	1.4149	1.2598	2.1566	1.5611	2.8044	1.9106
10.0	1.4289	1.2620	2.1795	1.5677	2.8363	1.9249
20.0	1.4961	1.2699	2.2881	1.5919	2.9857	1.9781
30.0	1.5202	1.2717	2.3261	1.5973	3.0372	1.9898
40.0	1.5325	1.2723	2.3455	1.5993	3.0632	1.9942
50.0	1.5400	1.2727	2.3572	1.6002	3.0788	1.9962
100.0	1.5552	1.2731	2.3809	1.6015	3.1102	1.9990
∞	1.5708	1.2733	2.4050	1.6018	3.1415	2.0000

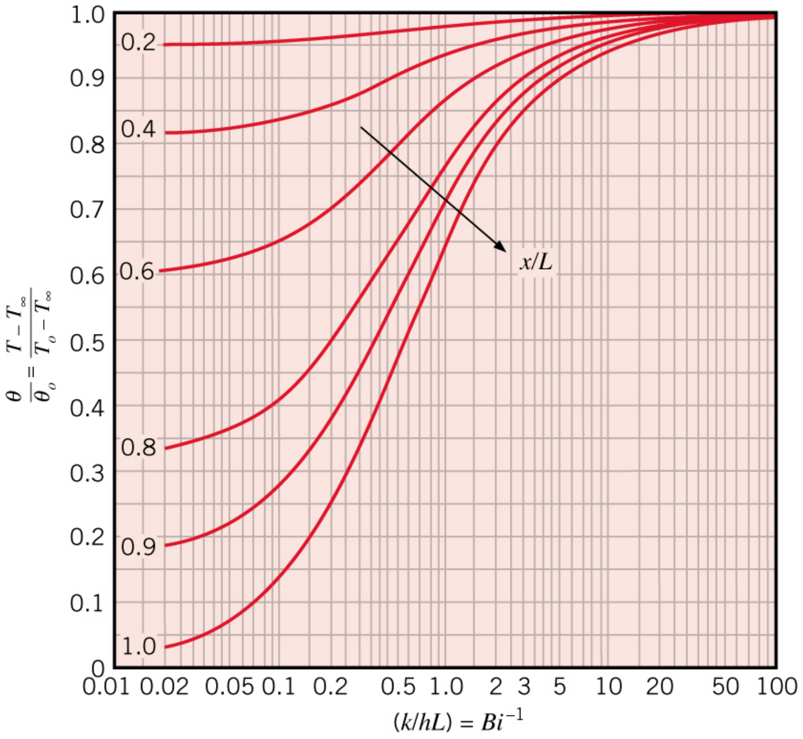
[a] $Bi = hL/k$ for the plane wall and hr_o/k for the infinite cylinder and sphere. See Figure 5.6.

Graphical Representation of the One-Term Approximation The Heisler Charts, Section 5 S.1

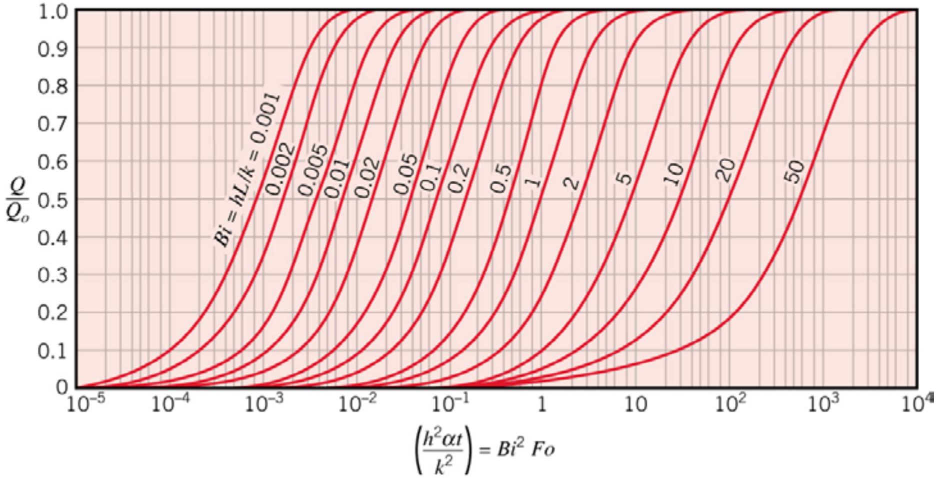
- Midplane Temperature:



- Temperature Distribution:

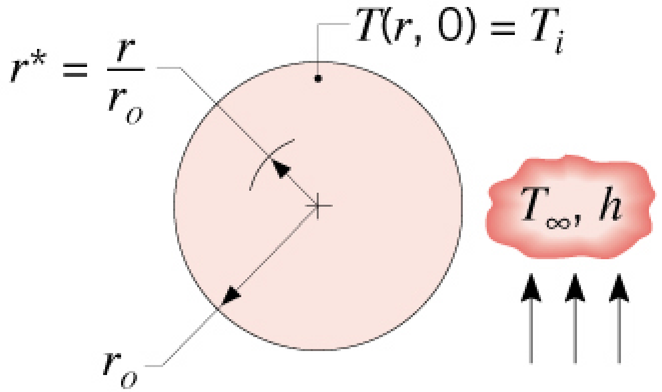


- Change in Thermal Energy Storage:



Radial Systems

- Long Rods or Spheres Heated or Cooled by Convection.



- One-Term Approximations:

  Long Rod: Eqs. (5.49) and (5.51)

  Sphere: Eqs. (5.50) and (5.52)

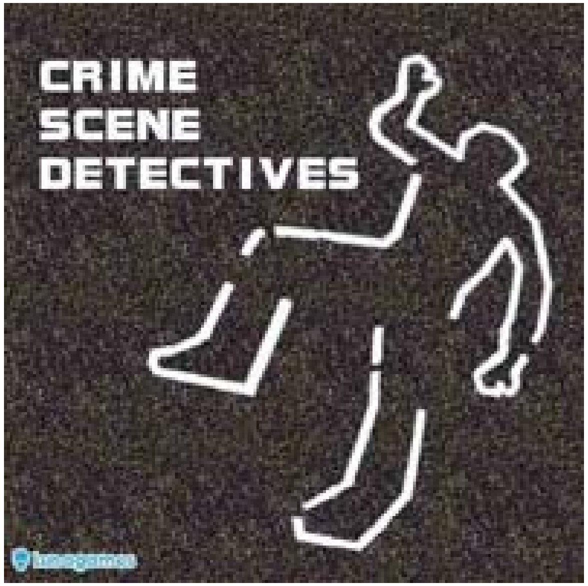
- Graphical Representations:

  Long Rod: Figs. 5 S.4 – 5 S.6

  Sphere: Figs. 5 S.7 – 5 S.9

CSI Rolla!

A person is found dead with a body center temperature of 25 °C in a room at 20 °C. The heat transfer coefficient is estimated to be h = 8 W/m^2. °C. Rolla Police Department asked you to predict the time of death. Modeling the body as a 30-cm-diameter and 1.70-m-long cylinder, and assuming the body to have the properties of water, estimate how long the person has been dead. Note: The temperature of a healthy body is 37 °C.



Chapter 6
Introduction to Convection

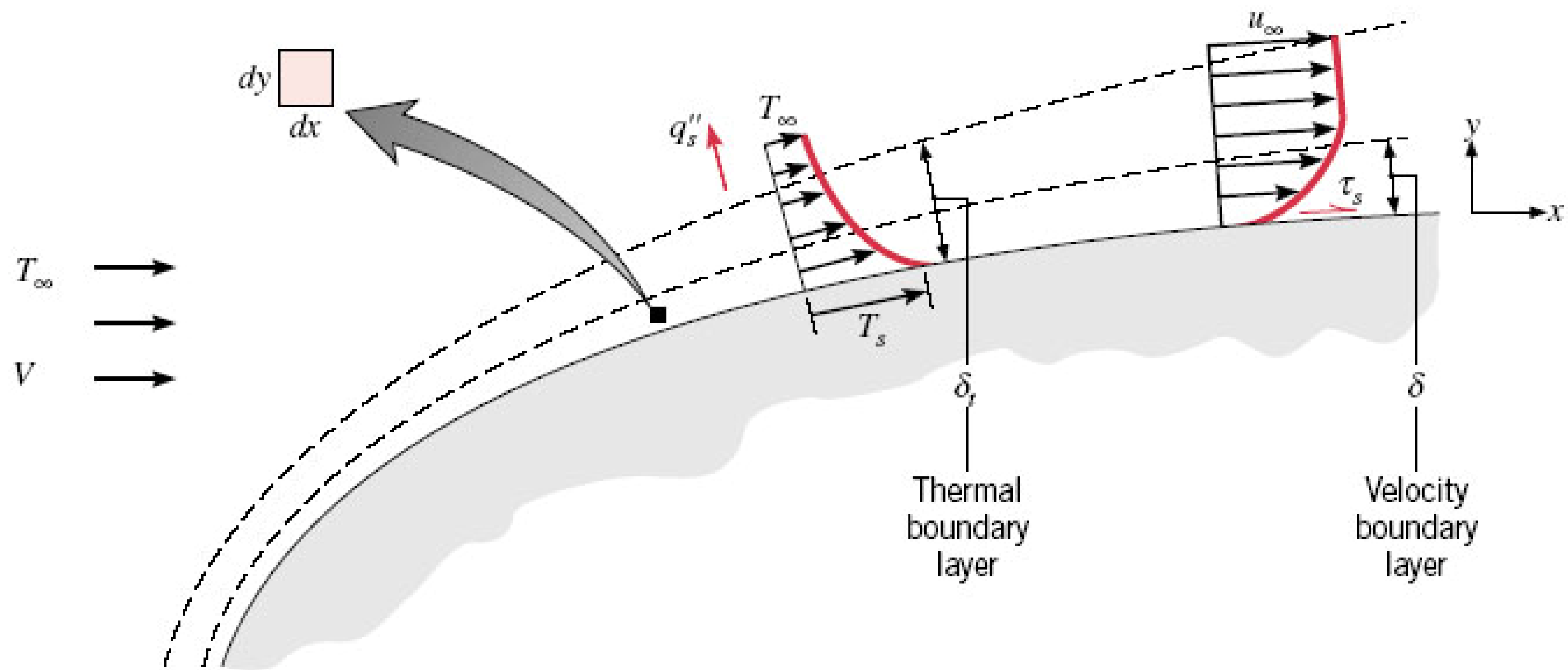
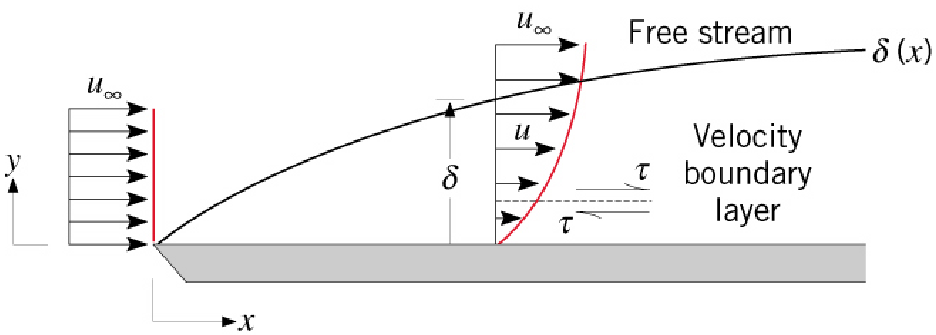


FIGURE 6.7 Development of the velocity and thermal boundary layers for an arbitrary surface.

Boundary Layers: Physical Features

- Velocity Boundary Layer



  – A consequence of viscous effects associated with relative motion between a fluid and a surface.

  – A flow region characterized by shear stresses and velocity gradients.

$$\delta \rightarrow \frac{u(y)}{u_\infty} = 0.99$$

  – A region between the surface and the free stream whose thickness δ increases in the flow direction.

$$\tau_s = \mu \frac{\partial u}{\partial y}\bigg|_{y=0}$$

  – Manifested by a surface shear stress τ_s that provides a drag force, F_D.

$$F_D = \int_{A_s} \tau_s dA_s$$

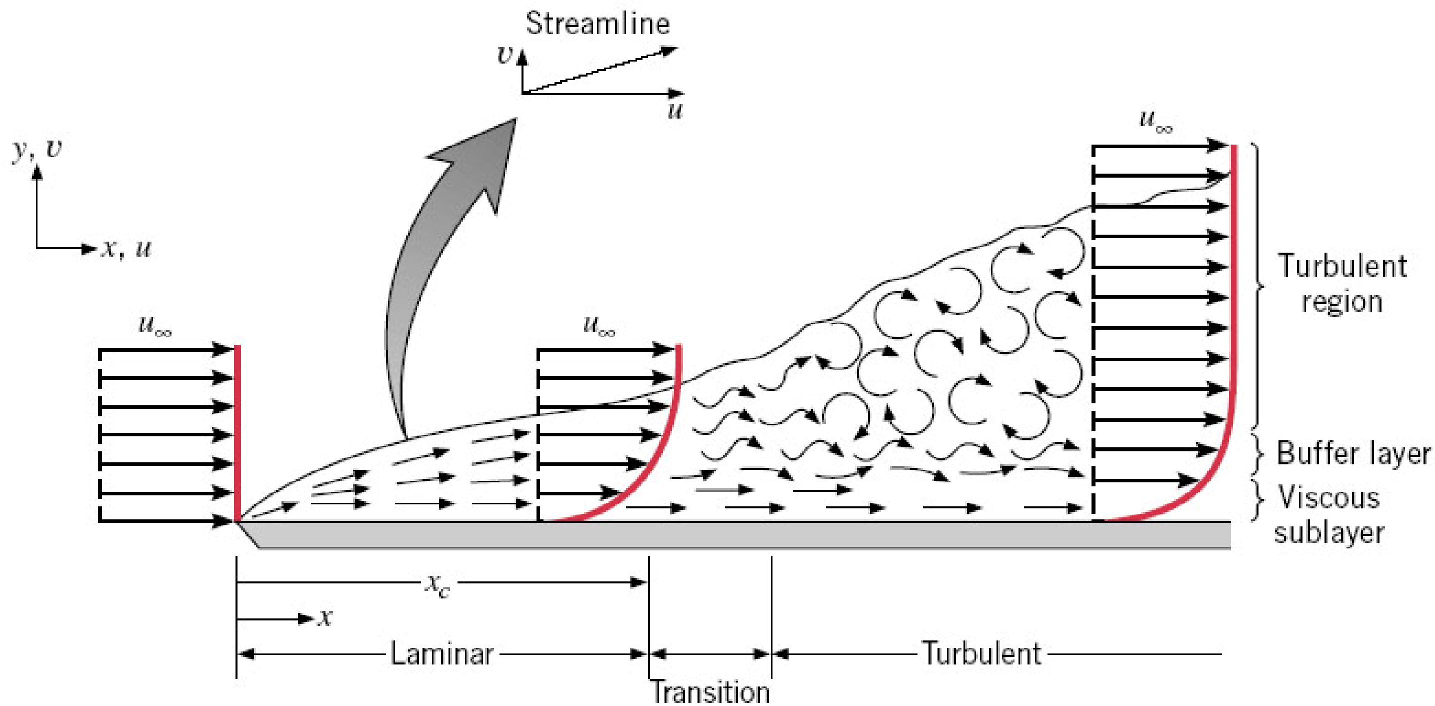


FIGURE 6.4 Velocity boundary layer development on a flat plate.

$$\mathrm{Re}_{x,c} \equiv \frac{\rho u_\infty x_c}{\mu} \rightarrow \text{ critical Reynolds number}$$

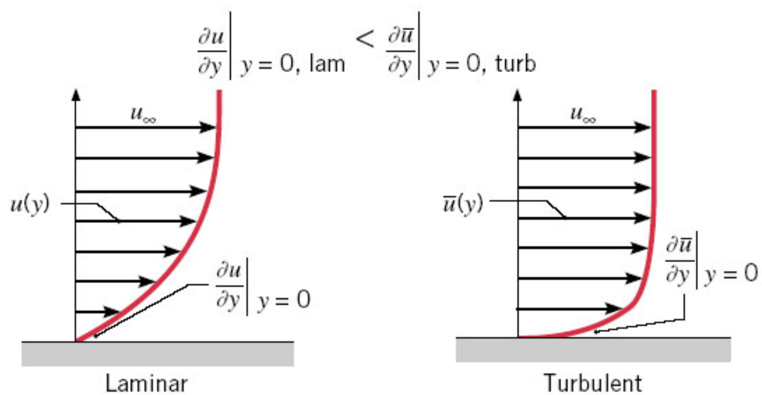
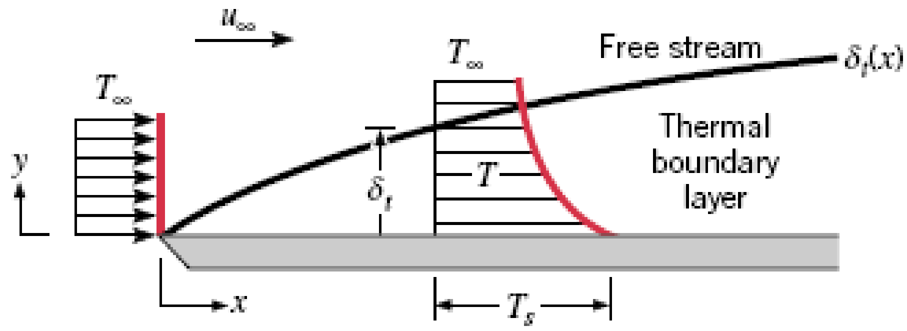


FIGURE 6.5 Comparison of laminar and turbulent velocity boundary layer profiles for the same free stream velocity.[1]

- Thermal Boundary Layer



  - A consequence of heat transfer between the surface and fluid.

  - A region of the flow characterized by temperature gradients and heat fluxes.

$$\delta_t \rightarrow \frac{T_s - T(y)}{T_s - T_\infty} = 0.99$$

  - A region between the surface and the free stream whose thickness δ_t increases in the flow direction.

$$q_s'' = -k_f \frac{\partial T}{\partial y}\bigg|_{y=0}$$

  - Manifested by a surface heat flux q_s'' and a convection heat transfer coefficient h .

$$h \equiv \frac{-k_f \partial T / \partial y\big|_{y=0}}{T_s - T_\infty}$$

PROBLEM 6.25

KNOWN: Air, water, engine oil or mercury at 300K in laminar, parallel flow over a flat plate.

FIND: Sketch of velocity and thermal boundary layer thickness.

ASSUMPTIONS: (1) Laminar flow.

PROPERTIES: For the fluids at 300K:

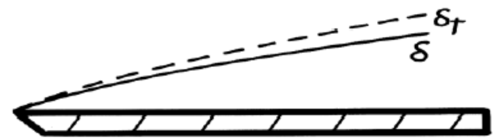
Fluid	Table	Pr
Air	A.4	0.71
Water	A.6	5.83
Engine Oil	A.5	6400
Mercury	A.5	0.025

ANALYSIS: For laminar, boundary layer flow over a flat plate.

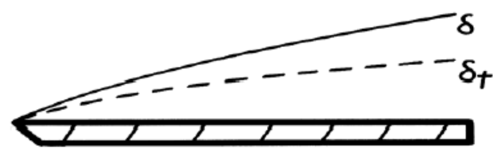
$$\frac{\delta}{\delta_t} \sim \mathrm{Pr}^n$$

where n > 0. Hence, the boundary layers appear as shown below.

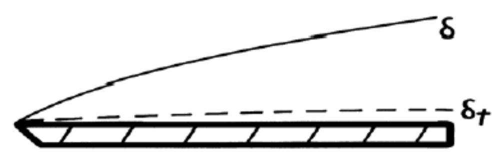
Air:



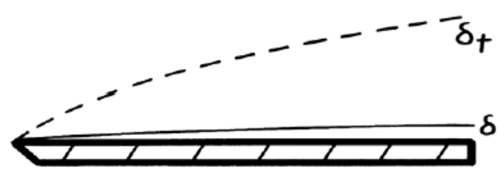
Water:



Engine Oil:



Mercury:



COMMENTS: Although Pr strongly influences relative boundary layer development in laminar flow, its influence is weak for turbulent flow.

Boundary Layer Similarity

- Dependent boundary layer variables of interest are: τ_s and q'' or h

- For a prescribed geometry, the corresponding independent variables are:

Geometrical: Size (L), Location (x,y)

Hydrodynamic: Velocity (V)

Fluid Properties: Hydrodynamic: ρ, μ

Thermal: c_p, k

Hence,

$$u = f\left(x, y, L, V, \rho, \mu\right)$$

$$\tau_s = f\left(x, L, V, \rho, \mu\right)$$

and

$$T = f\left(x, y, L, V, \rho, \mu, c_p, k\right)$$

$$h = f\left(x, L, V, \rho, \mu, c_p, k\right)$$

- Recast the boundary layer equations by introducing dimensionless forms of the independent and dependent variables.

$$x^* \equiv \frac{x}{L} \qquad y^* \equiv \frac{y}{L}$$

$$u^* \equiv \frac{u}{V} \qquad v^* \equiv \frac{v}{V}$$

$$T^* \equiv \frac{T - T_s}{T_\infty - T_s}$$

- Neglecting viscous dissipation, the following normalized forms of the x-momentum and energy equations are obtained:

$$u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} = -\frac{dp^*}{dx^*} + \frac{1}{\mathrm{Re}_L} \frac{\partial^2 u^*}{\partial y^{*2}}$$

$$u^* \frac{\partial T^*}{\partial x^*} + v^* \frac{\partial T^*}{\partial y^*} = \frac{1}{\mathrm{Re}_L \,\mathrm{Pr}} \frac{\partial^2 T^*}{\partial y^{*2}}$$

- For a prescribed geometry (dP/dx is known):

$$u^* = f\left(x^*, y^*, \mathrm{Re}_L\right)$$

$$T^* = f\left(x^*, y^*, \mathrm{Re}_L, \mathrm{Pr}\right)$$

$$\tau_s = \mu \frac{\partial u}{\partial y}\bigg|_{y=0} = \left(\frac{\mu V}{L}\right)\frac{\partial u^*}{\partial y^*}\bigg|_{y^*=0}$$

$$h = \frac{-k_f \partial T / \partial y\big|_{y=0}}{T_s - T_\infty} = -\frac{k_f}{L}\frac{\left(T_\infty - T_s\right)}{\left(T_s - T_\infty\right)}\frac{\partial T^*}{\partial y^*}\bigg|_{y^*=0} = +\frac{k_f}{L}\frac{\partial T^*}{\partial y^*}\bigg|_{y^*=0}$$

$$C_f \equiv \frac{\tau_s}{\rho V^2/2} = \frac{2}{\mathrm{Re}_L}\frac{\partial u^*}{\partial y^*}\bigg|_{y^*=0}$$

$$Nu \equiv \frac{hL}{k_f} = \frac{\partial T^*}{\partial y^*}\bigg|_{y^*=0}$$

$$\frac{\partial u^*}{\partial y^*}\bigg|_{y^*=0} = f\left(x^*, \mathrm{Re}_L\right)$$

$$\frac{\partial T^*}{\partial y^*}\bigg|_{y^*=0} = f\left(x^*, \mathrm{Re}_L, \mathrm{Pr}\right)$$

$$C_{f,x} = \frac{2}{\mathrm{Re}_L} f\left(x^*, \mathrm{Re}_L\right) = f\left(x^*, \mathrm{Re}_L\right)$$

$$Nu_x = f\left(x^*, \mathrm{Re}_L, \mathrm{Pr}\right)$$

$$\overline{C_f} = f\left(\mathrm{Re}_L\right)$$

$$\overline{Nu} = f\left(\mathrm{Re}_L, \mathrm{Pr}\right)$$

The Reynolds Analogy

- Equivalence of dimensionless momentum and energy equations for negligible pressure gradient (dp*/dx*~0) and Pr~1:

$$\underbrace{u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*}}_{\text{Advection terms}} = \underbrace{\frac{1}{\text{Re}} \frac{\partial^2 u^*}{\partial y^{*2}}}_{\text{Diffusion}}$$

$$\overbrace{u^* \frac{\partial T^*}{\partial x^*} + v^* \frac{\partial T^*}{\partial y^*}}^{\text{Advection terms}} = \overbrace{\frac{1}{\text{Re}} \frac{\partial^2 T^*}{\partial y^{*2}}}^{\text{Diffusion}}$$

- Hence, for equivalent boundary conditions, the solutions are of the same form:

$$u^* = T^*$$

$$\left.\frac{\partial u^*}{\partial y^*}\right|_{y^*=0} = \left.\frac{\partial T^*}{\partial y^*}\right|_{y^*=0}$$

$$C_f \frac{\text{Re}}{2} = Nu$$

or, with the Stanton number defined as,

$$St \equiv \frac{h}{\rho V c_p} = \frac{Nu}{Re\,Pr}$$

With Pr = 1, the Reynolds analogy, which relates important parameters of the velocity and thermal boundary layers, is

$$\frac{C_f}{2} = St$$

- Modified Reynolds (Chilton-Colburn) Analogy:

  – An empirical result that extends applicability of the Reynolds analogy:

$$\frac{C_f}{2} = St\,\mathrm{Pr}^{2/3} = \frac{Nu}{RePr^{1/3}} \quad 0.6 < \mathrm{Pr} < 60$$

  – Applicable to laminar flow if dp*/dx* ~ 0.

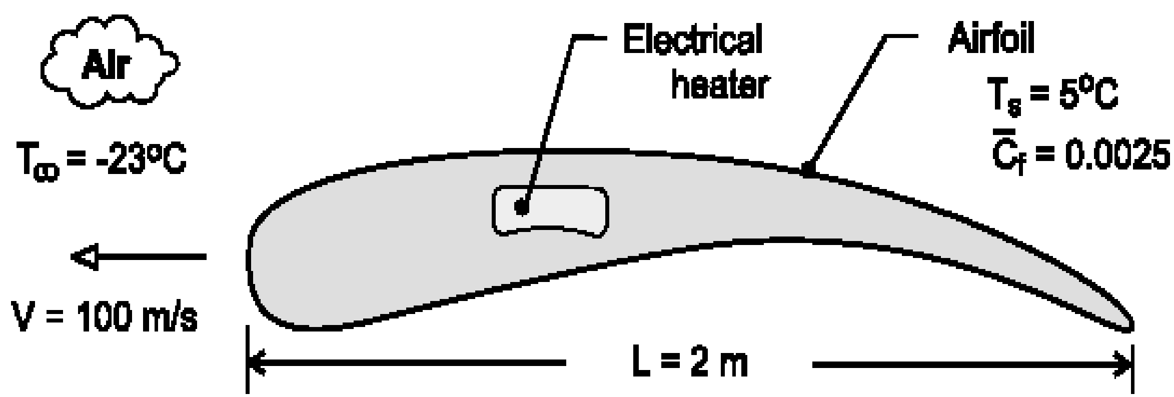
  – Generally applicable to turbulent flow without restriction on dp*/dx*.

PROBLEM 6.43

KNOWN: Nominal operating conditions of aircraft and characteristic length and average friction coefficient of wing.

FIND: Average heat flux needed to maintain prescribed surface temperature of wing.

SCHEMATIC:



ASSUMPTIONS: (1) Applicability of modified Reynolds analogy, (2) Constant properties.

PROPERTIES: Prescribed, Air: $\nu = 16.3 \times 10^{-6}$ m^2/s, k = 0.022 W/m·K, Pr = 0.72.

Chapter 7
External Flows

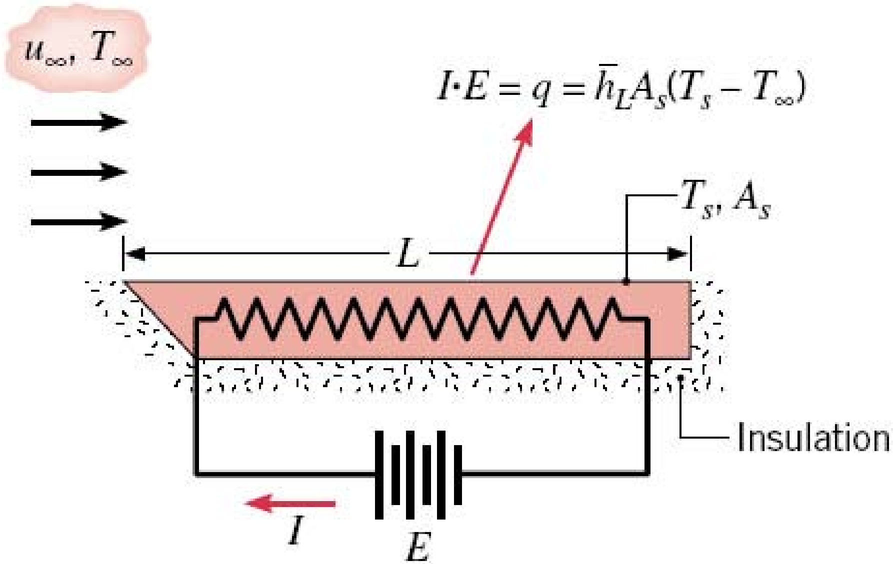


FIGURE 7.1 Experiment for measuring the average convection heat transfer coefficient $\bar{h}_L$.

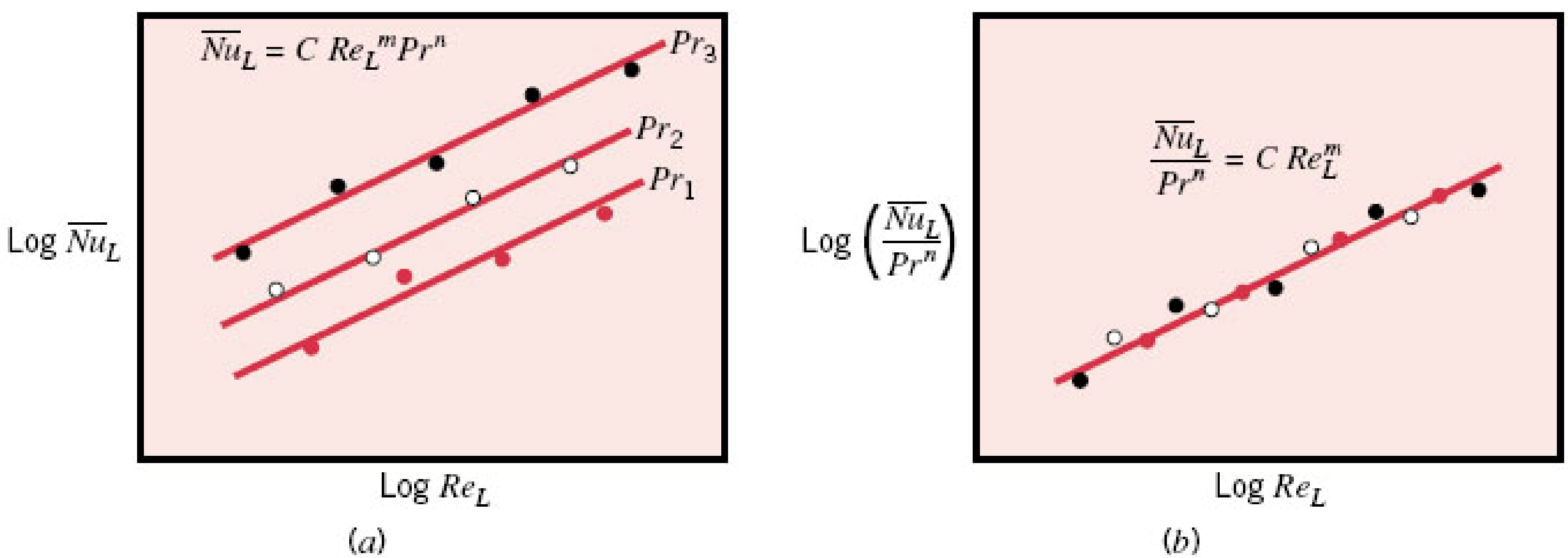
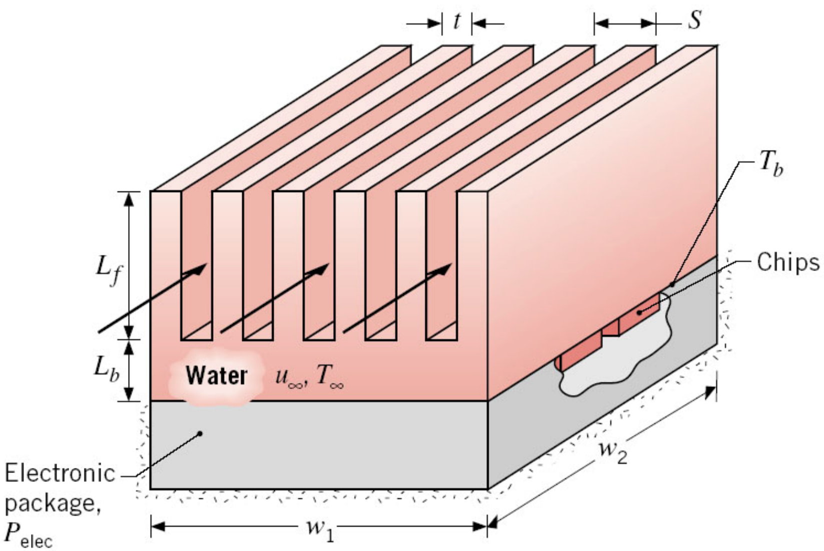
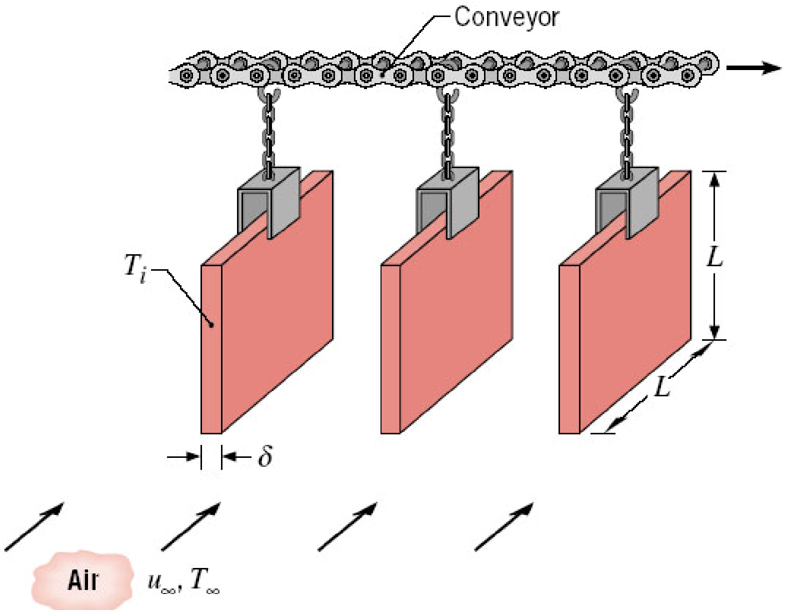
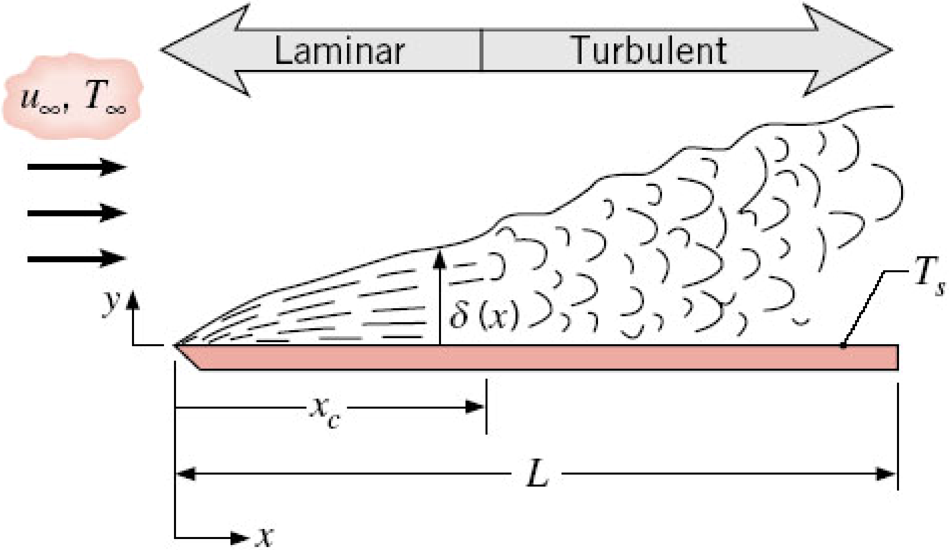


FIGURE 7.2 Dimensionless representation of convection heat transfer measurements.

Flow Over Flat Plate







$$\mathrm{Re}_{x,c} \approx 5 \times 10^5$$

FIGURE 7.3
The flat plate in parallel flow.

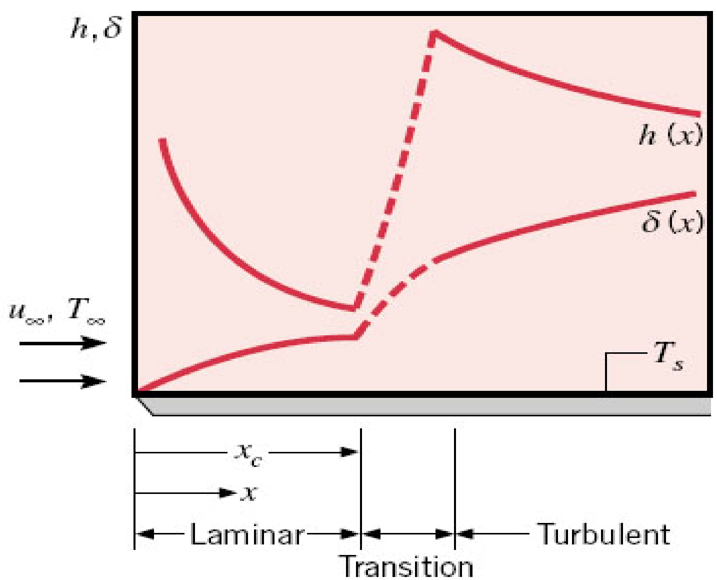
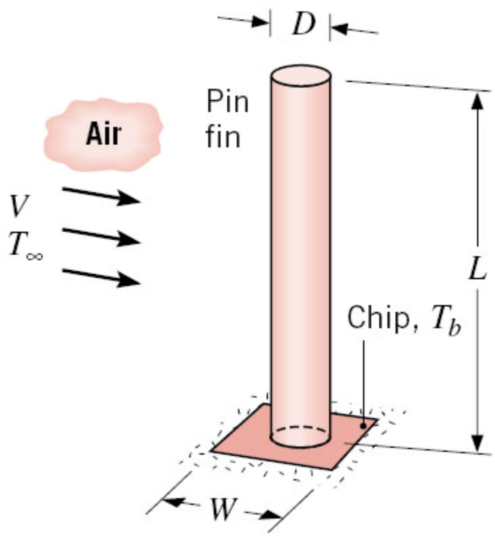
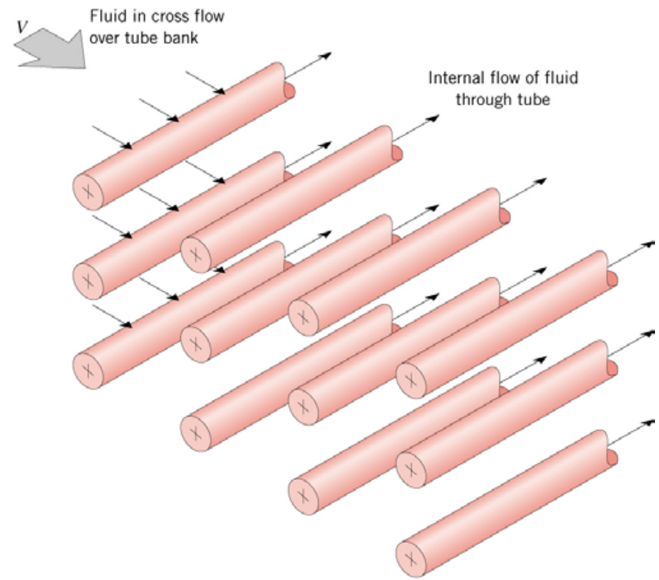


FIGURE 6.6
Variation of velocity boundary layer thickness δ and the local heat transfer coefficient h for flow over an isothermal flat plate.

Cylinder/Sphere in Cross Flow





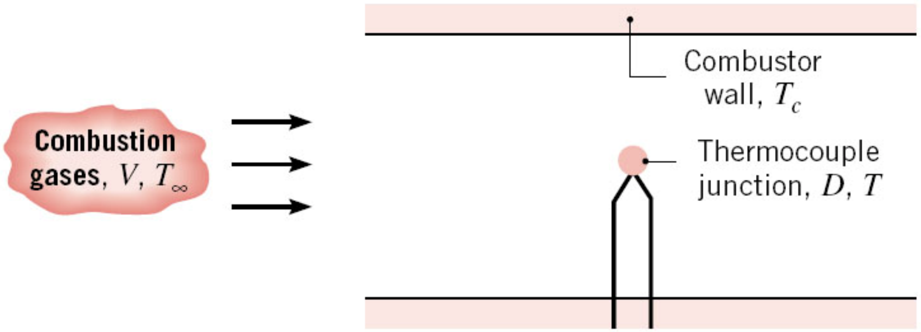


TABLE 7.9 Summary of convection heat transfer correlations for external flow[a]

Correlation		Geometry	Conditions[b]
$\delta = 5x\, Re_x^{-1/2}$	(7.17)	Flat plate	Laminar, T_f
$C_{f,x} = 0.664 Re_x^{-1/2}$	(7.18)	Flat plate	Laminar, local, T_f
$Nu_x = 0.332 Re_x^{1/2}\, Pr^{1/3}$	(7.21)	Flat plate	Laminar, local, T_f, $Pr \gtrsim 0.6$
$\delta_t = \delta Pr^{-1/3}$	(7.22)	Flat plate	Laminar, T_f
$\overline{C}_{f,x} = 1.328 Re_x^{-1/2}$	(7.24)	Flat plate	Laminar, average, T_f
$\overline{Nu}_x = 0.664 Re_x^{1/2}\, Pr^{1/3}$	(7.25)	Flat plate	Laminar, average, T_f, $Pr \gtrsim 0.6$
$Nu_x = 0.565 Pe_x^{1/2}$	(7.26)	Flat plate	Laminar, local, T_f, $Pr \lesssim 0.05$, $Pe_x \gtrsim 100$
$C_{f,x} = 0.0592 Re_x^{-1/5}$	(7.28)	Flat plate	Turbulent, local, T_f, $Re_x \lesssim 10^8$
$\delta = 0.37x\, Re_x^{-1/5}$	(7.29)	Flat plate	Turbulent, T_f, $Re_x \lesssim 10^8$
$Nu_x = 0.0296 Re_x^{4/5}\, Pr^{1/3}$	(7.30)	Flat plate	Turbulent, local, T_f, $Re_x \lesssim 10^8$, $0.6 \lesssim Pr \lesssim 60$
$\overline{C}_{f,L} = 0.074 Re_L^{-1/5} - 1742 Re_L^{-1}$	(7.33)	Flat plate	Mixed, average, T_f, $Re_{x,c} = 5 \times 10^5$, $Re_L \lesssim 10^8$
$\overline{Nu}_L = (0.037 Re_L^{4/5} - 871) Pr^{1/3}$	(7.31)	Flat plate	Mixed, average, T_f, $Re_{x,c} = 5 \times 10^5$, $Re_L \lesssim 10^8$, $0.6 \lesssim Pr \lesssim 60$
$\overline{Nu}_D = C\, Re_D^m\, Pr^{1/3}$ (Table 7.2)	(7.44)	Cylinder	Average, T_f, $0.4 \lesssim Re_D \lesssim 4 \times 10^5$, $Pr \gtrsim 0.7$
$\overline{Nu}_D = C\, Re_D^m\, Pr^n (Pr/Pr_s)^{1/4}$ (Table 7.4)	(7.45)	Cylinder	Average, T_∞, $1 \lesssim Re_D \lesssim 10^6$, $0.7 \lesssim Pr \lesssim 500$
$\overline{Nu}_D = 0.3 + [0.62 Re_D^{1/2}\, Pr^{1/3} \times [1 + (0.4/Pr)^{2/3}]^{-1/4}] \times [1 + (Re_D/282{,}000)^{5/8}]^{4/5}$	(7.46)	Cylinder	Average, T_f, $Re_D\, Pr \gtrsim 0.2$
$\overline{Nu}_D = 2 + (0.4 Re_D^{1/2} + 0.06 Re_D^{2/3}) Pr^{0.4} \times (\mu/\mu_s)^{1/4}$	(7.48)	Sphere	Average, T_∞, $3.5 \lesssim Re_D \lesssim 7.6 \times 10^4$, $0.71 \lesssim Pr \lesssim 380$
$\overline{Nu}_D = 2 + 0.6 Re_D^{1/2}\, Pr^{1/3}$	(7.49)	Falling drop	Average, T_∞
$\overline{Nu}_D = 1.13 C_1 C_2\, Re_{D,\max}^m\, Pr^{1/3}$ (Tables 7.5, 7.6)	(7.52), (7.53)	Tube bank[c]	Average, $\overline{T}_f$, $2000 \lesssim Re_{D,\max} \lesssim 4 \times 10^4$, $Pr \gtrsim 0.7$
$\overline{Nu}_D = C C_2\, Re_{D,\max}^m\, Pr^{0.36} (Pr/Pr_s)^{1/4}$ (Tables 7.7, 7.8)	(7.56), (7.57)	Tube bank[c]	Average, $\overline{T}$, $1000 \lesssim Re_D \lesssim 2 \times 10^6$, $0.7 \lesssim Pr \lesssim 500$

TABLE 7.2 Constants of Equation 7.44 for the circular cylinder in cross flow [11, 12]

Re_D	C	m
0.4–4	0.989	0.330
4–40	0.911	0.385
40–4000	0.683	0.466
4000–40,000	0.193	0.618
40,000–400,000	0.027	0.805

TABLE 7.3 Constants of Equation 7.44 for noncircular cylinders in cross flow of a gas [13]

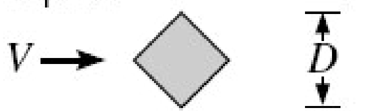
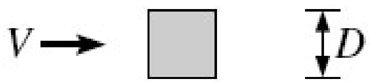
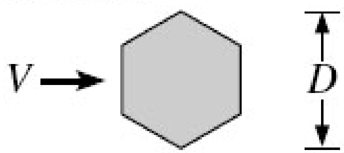
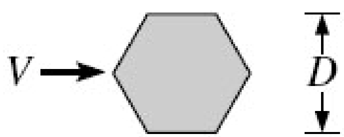
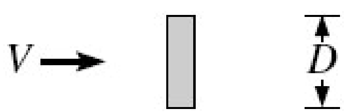
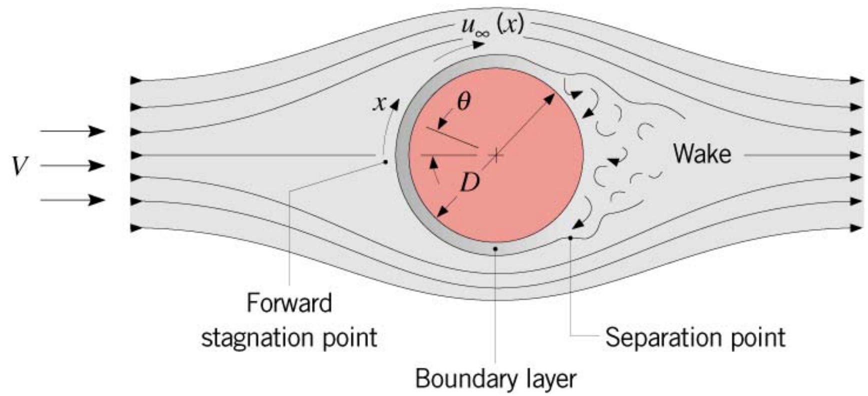
Geometry	Re_D	C	m
Square			
	$5 \times 10^3–10^5$	0.246	0.588
	$5 \times 10^3–10^5$	0.102	0.675
Hexagon			
	$5 \times 10^3–1.95 \times 10^4$	0.160	0.638
	$1.95 \times 10^4–10^5$	0.0385	0.782
	$5 \times 10^3–10^5$	0.153	0.638
Vertical plate			
	$4 \times 10^3–1.5 \times 10^4$	0.228	0.731

TABLE 7.4 Constants of Equation 7.45 for the circular cylinder in cross flow [16]

Re_D	C	m
1–40	0.75	0.4
40–1000	0.51	0.5
10^3–2×10^5	0.26	0.6
2×10^5–10^6	0.076	0.7

- Conditions depend on special features of boundary layer development, including onset at a stagnation point and separation, as well as transition to turbulence.

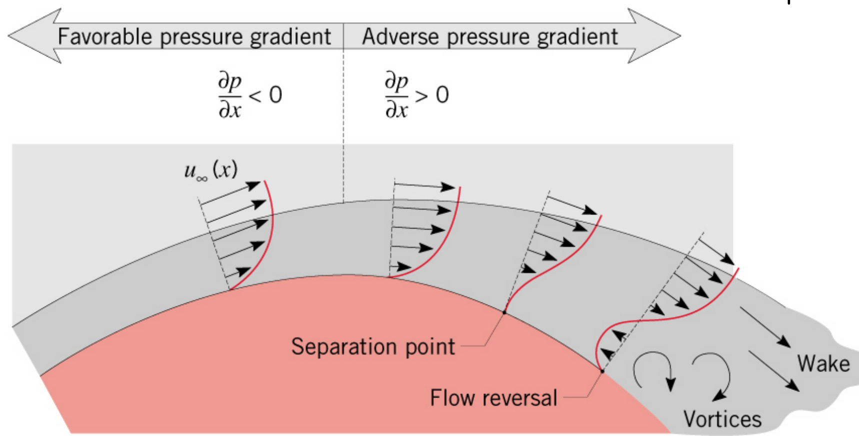


- Stagnation point: Location of zero velocity $(u_\infty = 0)$ and maximum pressure.

- Followed by boundary layer development under a favorable pressure gradient $(dp/dx < 0)$ and hence acceleration of the free stream flow $(du_\infty/dx > 0)$.

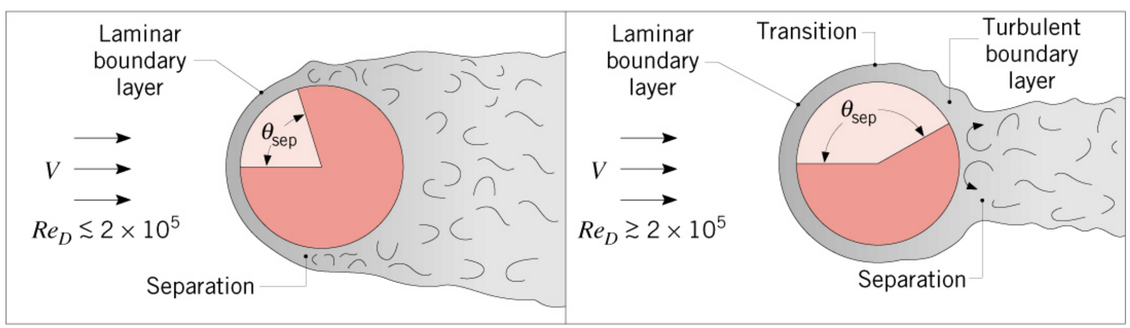
- As the rear of the cylinder is approached, the pressure must begin to increase. Hence, there is a minimum in the pressure distribution, *p(x),* after which boundary layer development occurs under the influence of an adverse pressure gradient $(dp/dx > 0, du_\infty/dx < 0)$.

– Separation occurs when the velocity gradient $du/dy\big|_{y=0}$ reduces to zero.



and is accompanied by flow reversal and a downstream wake.

– Location of separation depends on boundary layer transition.



$$\mathrm{Re}_D \equiv \frac{\rho V D}{\mu} = \frac{VD}{\nu}$$

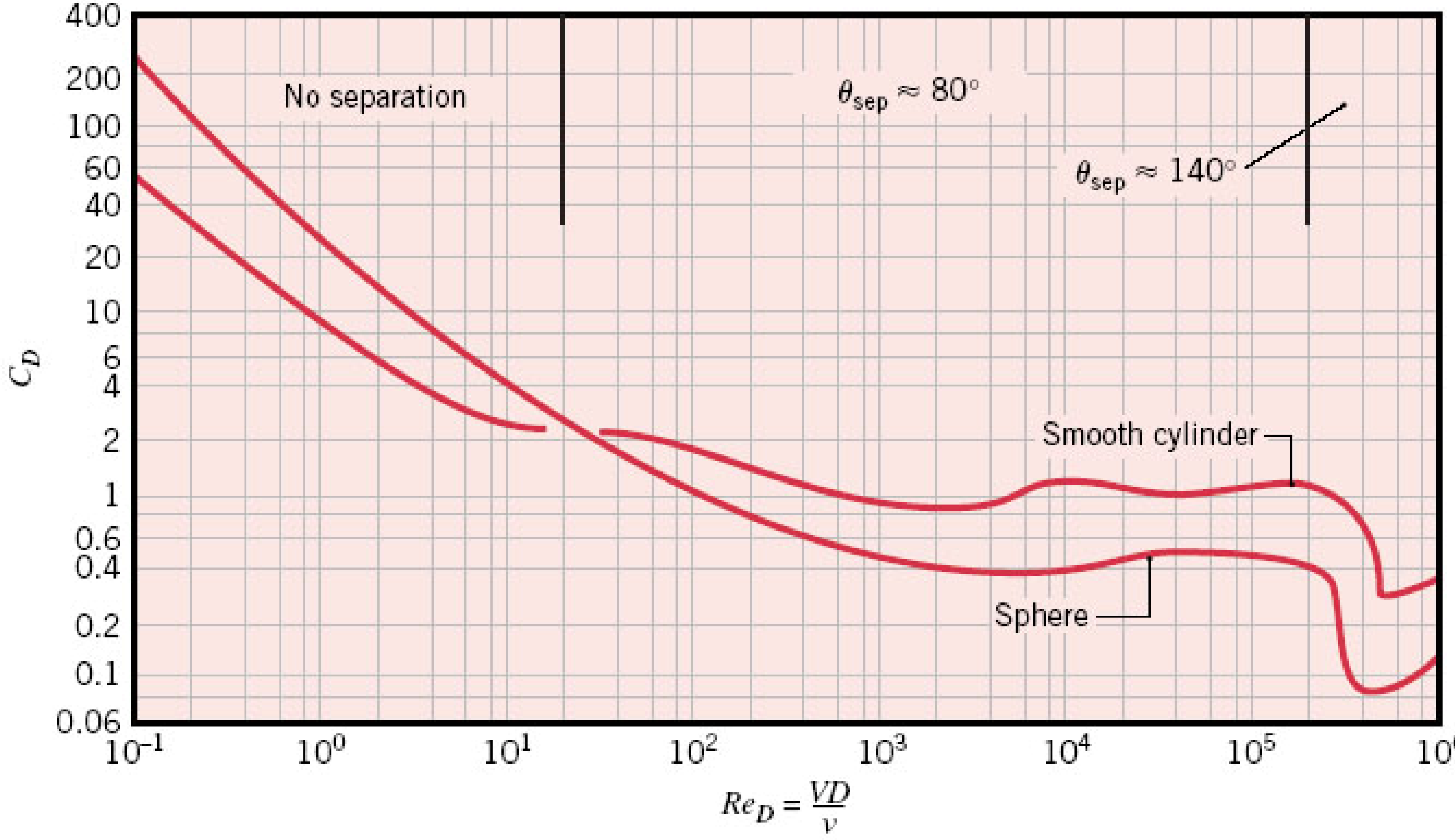


FIGURE 7.8 Drag coefficients for a smooth circular cylinder in cross flow and for a sphere [2]. Boundary layer separation angles are for a cylinder. Adapted with permission.

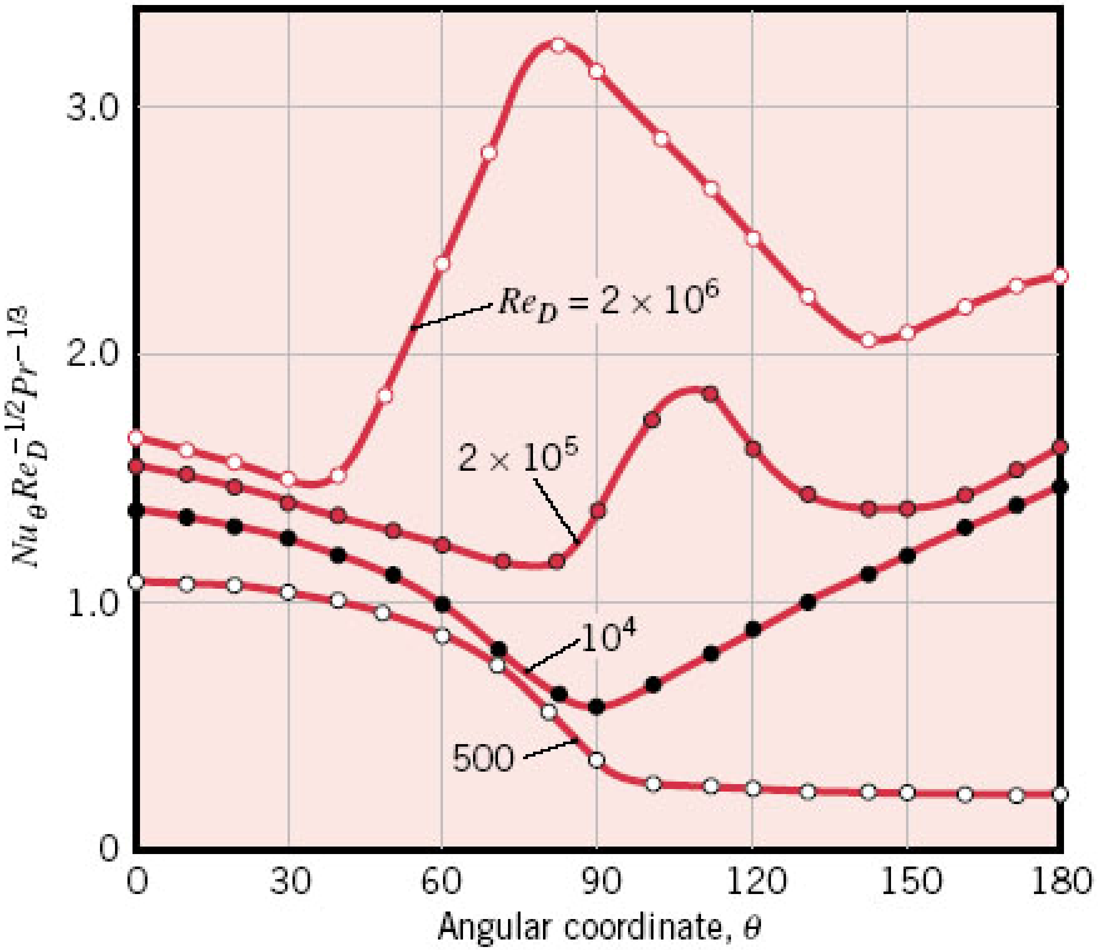


FIGURE 7.9 Local Nusselt number for airflow normal to a circular cylinder. Adapted with permission from Zukauskas, A., "Convective Heat Transfer in Cross Flow," in S. Kakac, R. K. Shah, and W. Aung, Eds., *Handbook of Single-Phase Convective Heat Transfer*, Wiley, New York, 1987.

Chapter 8
Internal Flows

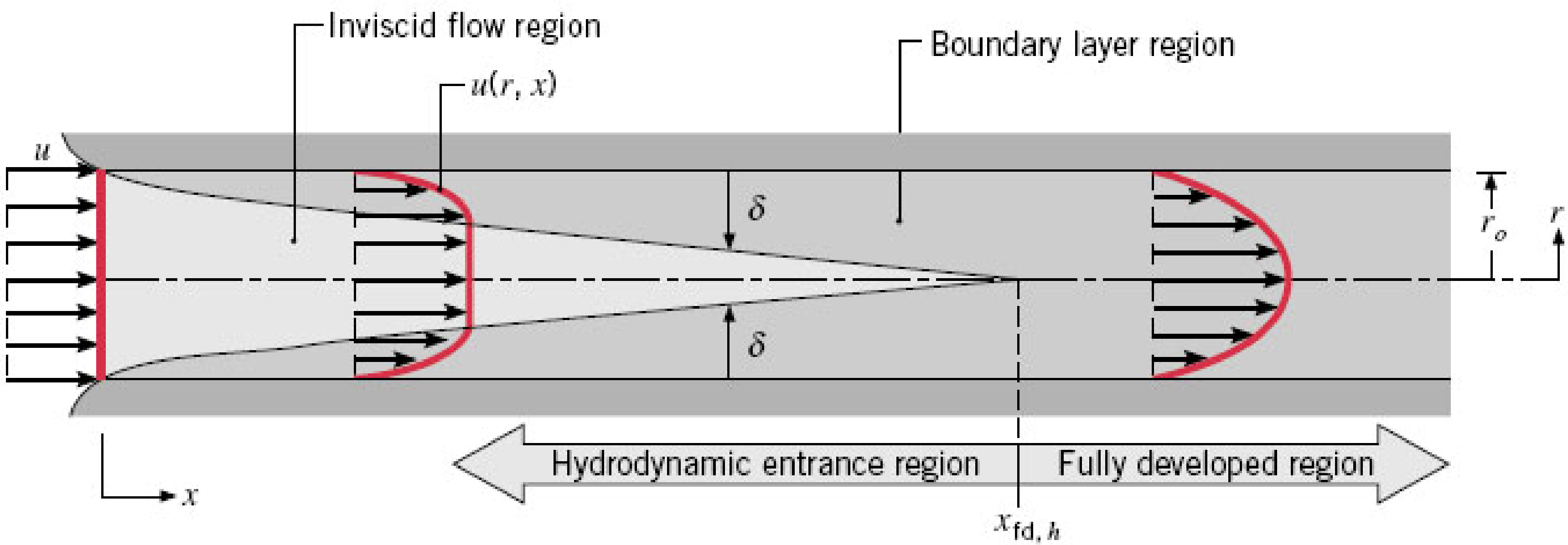


FIGURE 8.1 Laminar, hydrodynamic boundary layer development in a circular tube.

– Velocity boundary layer develops on surface of tube and thickens with increasing *x*.

– Inviscid region of uniform velocity shrinks as boundary layer grows.

– Subsequent to boundary layer merger at the centerline, the velocity profile becomes parabolic and invariant with *x* (hydrodynamically fully-developed flow).

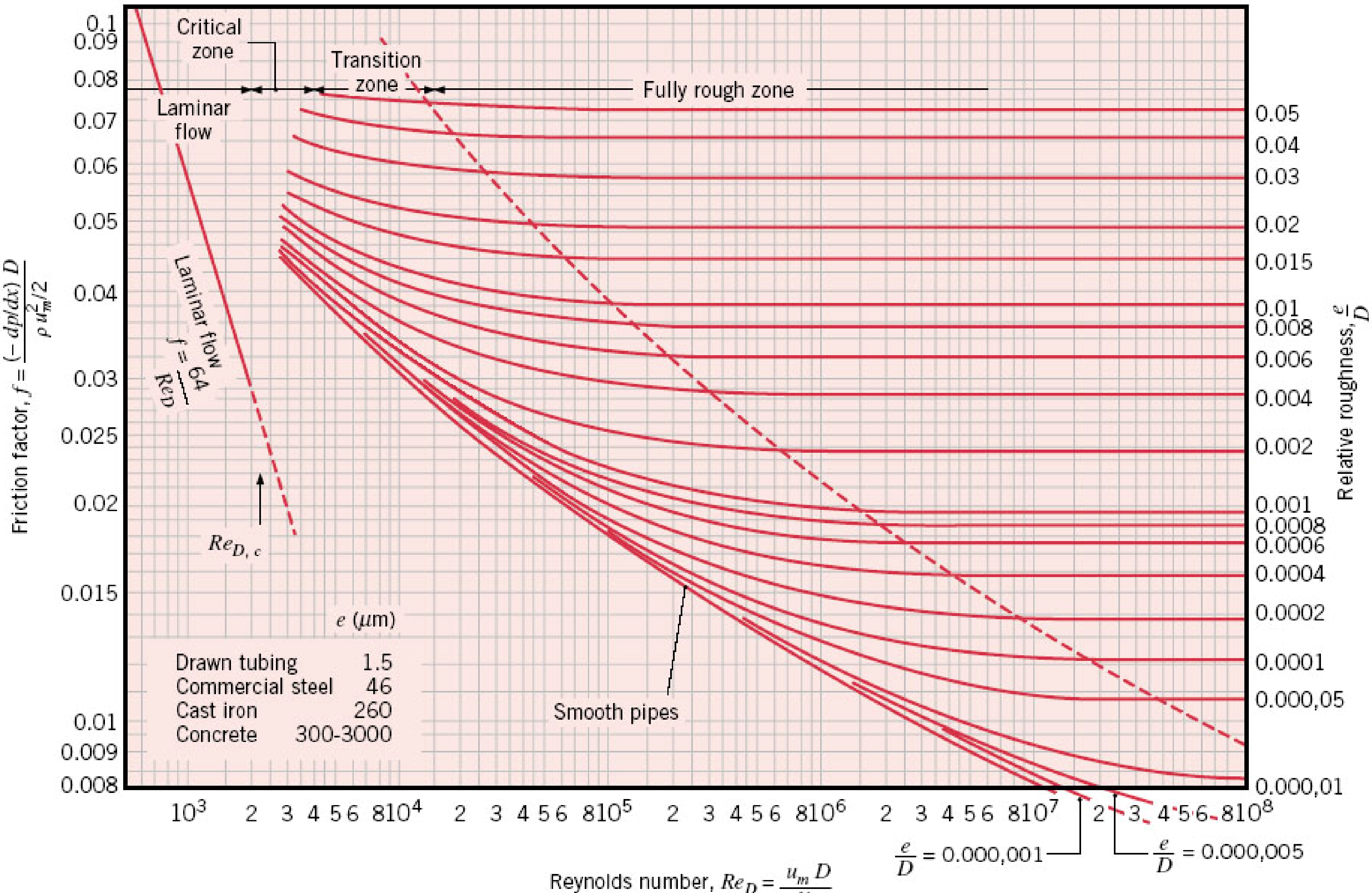


Figure 8.3 Friction factor for fully developed flow in a circular tube [3]. Used with permission.

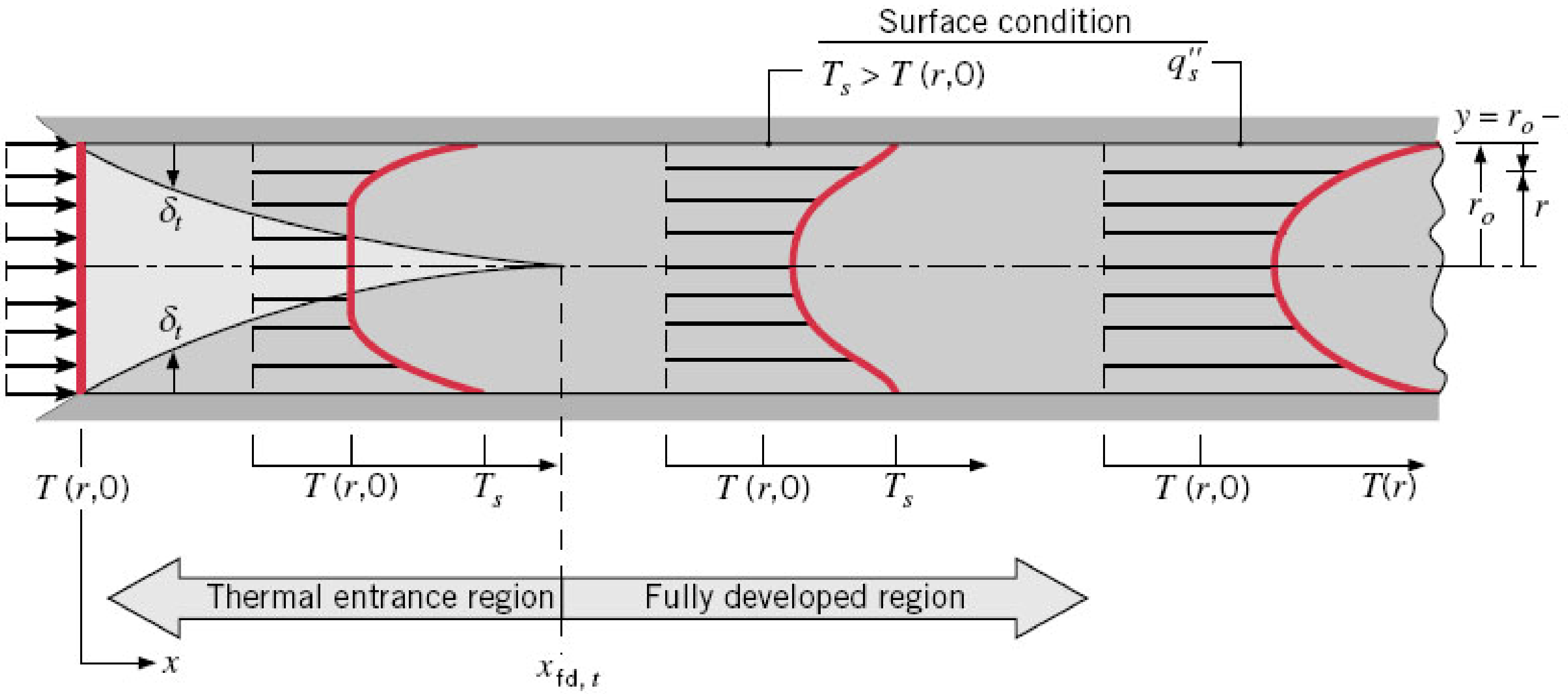


FIGURE 8.4 Thermal boundary layer development in a heated circular tube.

- Thermal boundary layer develops on surface of tube and thickens with increasing *x*.
- Isothermal core shrinks as boundary layer grows.
- Subsequent to boundary layer merger, dimensionless forms of the temperature profile become independent of x (thermally-fully developed flow).

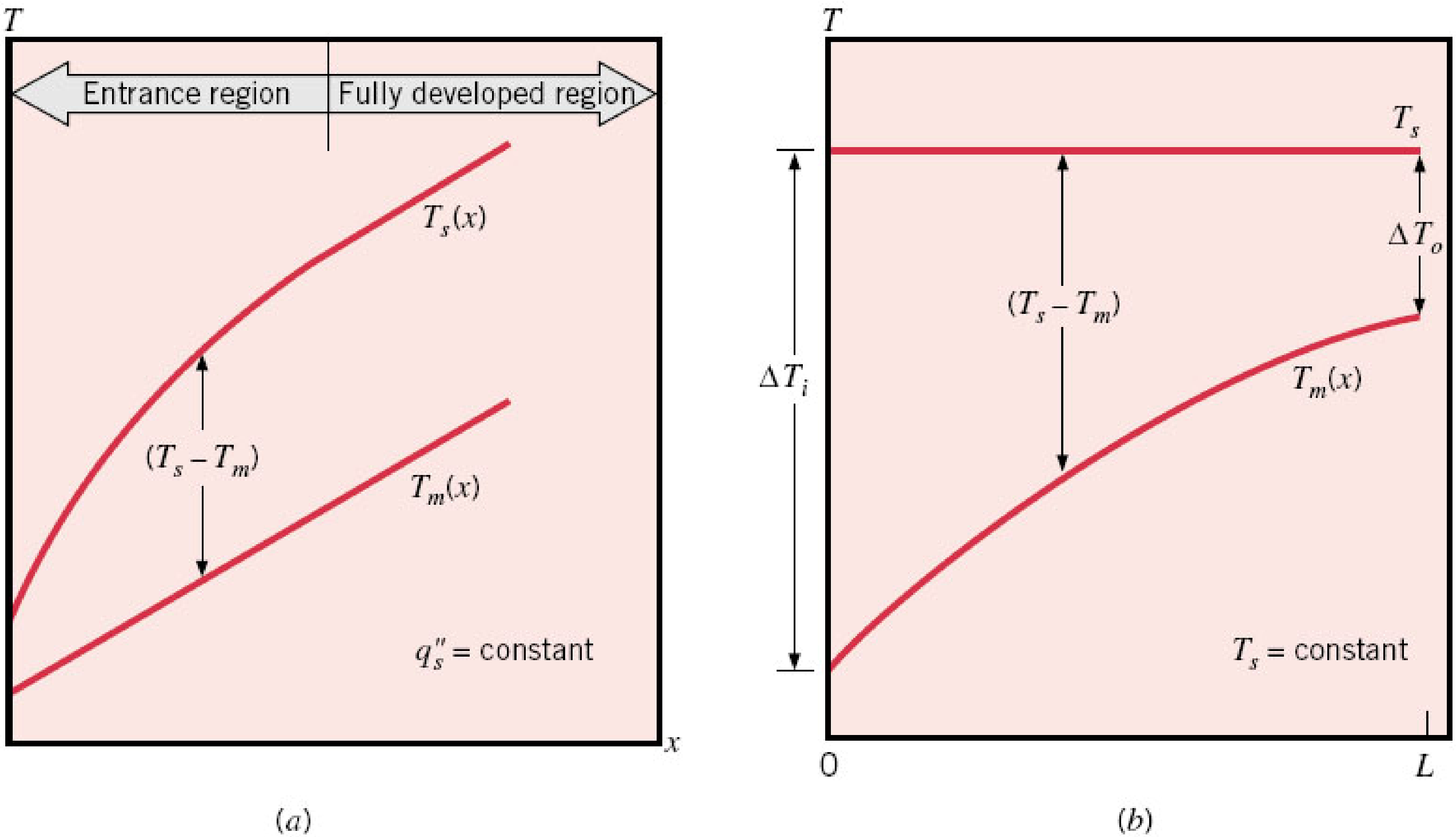
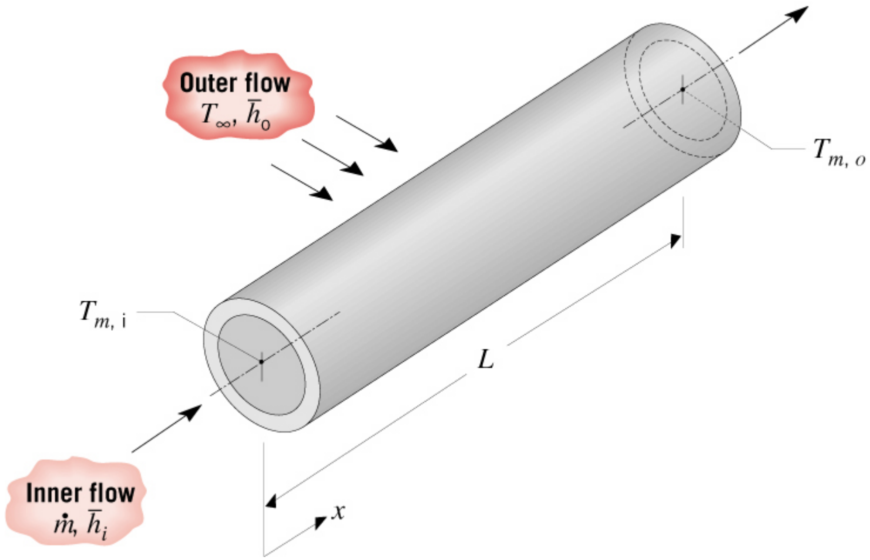


FIGURE 8.7 Axial temperature variations for heat transfer in a tube. (*a*) Constant surface heat flux. (*b*) Constant surface temperature.

- Special Case: Uniform External Fluid Temperature



$$\frac{\Delta T_o}{\Delta T_i} = \frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}} = \exp\left(-\frac{\bar{U} A_s}{\dot{m} c_p}\right) = \exp\left(-\frac{1}{\dot{m} c_p R_{tot}}\right)$$

$$q = \bar{U} A_s \, \Delta T_{\ell m} = \frac{\Delta T_{\ell m}}{R_{tot}}$$

$\Delta T_{\ell m} \rightarrow \quad T_s$ replaced by T_∞.

Note: Replacement of T_∞ by $T_{s,o}$ if outer surface temperature is uniform.

TABLE 8.4 Summary of convection correlations for flow in a circular tube[a,d]

Correlation		Conditions
$f = 64/Re_D$	(8.19)	Laminar, fully developed
$Nu_D = 4.36$	(8.53)	Laminar, fully developed, uniform q_s''
$Nu_D = 3.66$	(8.55)	Laminar, fully developed, uniform T_s
$\overline{Nu}_D = 3.66 + \dfrac{0.0668(D/L)Re_D\,Pr}{1 + 0.04[(D/L)Re_D\,Pr]^{2/3}}$	(8.56)	Laminar, thermal entry (or combined entry with $Pr \gtrsim 5$), uniform T_s
or		
$\overline{Nu}_D = 1.86\left(\dfrac{Re_D\,Pr}{L/D}\right)^{1/3}\left(\dfrac{\mu}{\mu_s}\right)^{0.14}$	(8.57)	Laminar, combined entry, $0.6 \lesssim Pr \lesssim 5$, $0.0044 \lesssim (\mu/\mu_s) \lesssim 9.75$, uniform T_s
$f = 0.316\,Re_D^{-1/4}$	(8.20a)[b]	Turbulent, fully developed, $Re_D \lesssim 2 \times 10^4$
$f = 0.184\,Re_D^{-1/5}$	(8.20b)[b]	Turbulent, fully developed, $Re_D \gtrsim 2 \times 10^4$
or		
$f = (0.790 \ln Re_D - 1.64)^{-2}$	(8.21)[b]	Turbulent, fully developed, $3000 \lesssim Re_D \lesssim 5 \times 10^6$
$Nu_D = 0.023Re_D^{4/5}\,Pr^n$	(8.60)[c]	Turbulent, fully developed, $0.6 \lesssim Pr \lesssim 160$, $Re_D \gtrsim 10{,}000$, $(L/D) \gtrsim 10$, $n = 0.4$ for $T_s > T_m$ and $n = 0.3$ for $T_s < T_m$
or		
$Nu_D = 0.027Re_D^{4/5}\,Pr^{1/3}\left(\dfrac{\mu}{\mu_s}\right)^{0.14}$	(8.61)[c]	Turbulent, fully developed, $0.7 \lesssim Pr \lesssim 16{,}700$, $Re_D \gtrsim 10{,}000$, $L/D \gtrsim 10$
or		
$Nu_D = \dfrac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$	(8.62)[c]	Turbulent, fully developed, $0.5 \lesssim Pr \lesssim 2000$, $3000 \lesssim Re_D \lesssim 5 \times 10^6$, $(L/D) \gtrsim 10$
$Nu_D = 4.82 + 0.0185(Re_D\,Pr)^{0.827}$	(8.64)	Liquid metals, turbulent, fully developed, uniform q_s'', $3.6 \times 10^3 \lesssim Re_D \lesssim 9.05 \times 10^5$, $10^2 \lesssim Pe_D \lesssim 10^4$
$Nu_D = 5.0 + 0.025(Re_D\,Pr)^{0.8}$	(8.65)	Liquid metals, turbulent, fully developed, uniform T_s, $Pe_D \gtrsim 100$

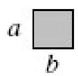
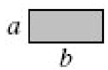
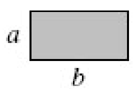
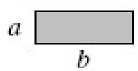
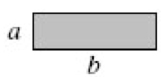
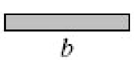
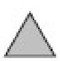
[a]Properties in Equations 8.53, 8.55, 8.60, 8.61, 8.62, 8.64, and 8.65 are based on T_m; properties in Equations 8.19, 8.20, and 8.21 are based on $T_f \equiv (T_s + T_m)/2$; properties in Equations 8.56 and 8.57 are based on $\overline{T}_m \equiv (T_{m,i} + T_{m,o})/2$.

[b]Equations 8.20 and 8.21 pertain to smooth tubes. For rough tubes, Equation 8.62 should be used with the results of Figure 8.3.

[c]As a first approximation, Equations 8.60, 8.61, or 8.62 may be used to evaluate the average Nusselt number $\overline{Nu}_D$ over the entire tube length, if $(L/D) \gtrsim 10$. The properties should then be evaluated at the average of the mean temperature, $\overline{T}_m \equiv (T_{m,i} + T_{m,o})/2$.

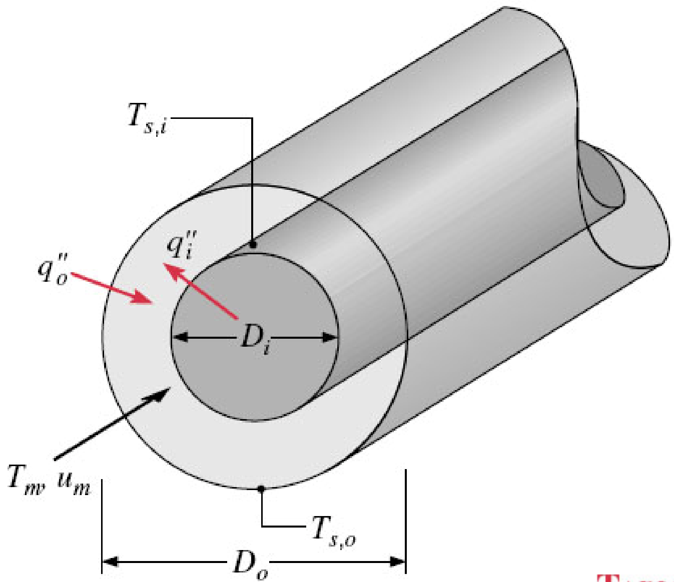
[d]For tubes of noncircular cross section, $Re_D \equiv D_h u_m/\nu$, $D_h \equiv 4A_c/P$, and $u_m = \dot{m}/\rho A_c$. Results for fully developed laminar flow are provided in Table 8.1. For turbulent flow, Equation 8.60 may be used as a first approximation.

TABLE 8.1 Nusselt numbers and friction factors for fully developed laminar flow in tubes of differing cross section

Cross Section	$\frac{b}{a}$	$Nu_D \equiv \frac{hD_h}{k}$ (Uniform q_s'')	$Nu_D \equiv \frac{hD_h}{k}$ (Uniform T_s)	$f\,Re_{D_h}$
	—	4.36	3.66	64
	1.0	3.61	2.98	57
	1.43	3.73	3.08	59
	2.0	4.12	3.39	62
	3.0	4.79	3.96	69
	4.0	5.33	4.44	73
	8.0	6.49	5.60	82
	∞	8.23	7.54	96
 Heated / Insulated	∞	5.39	4.86	96
	—	3.11	2.49	53

Used with permission from W. M. Kays and M. E. Crawford, *Convection Heat and Mass Transfer*, 3rd ed. McGraw-Hill, New York, 1993.

Concentric Tube Annulus



$$q_i'' = h_i\left(T_{s,i} - T_m\right)$$
$$q_o'' = h_o\left(T_{s,o} - T_m\right)$$

$$Nu_i \equiv \frac{h_i D_h}{k} \qquad Nu_o \equiv \frac{h_o D_h}{k}$$

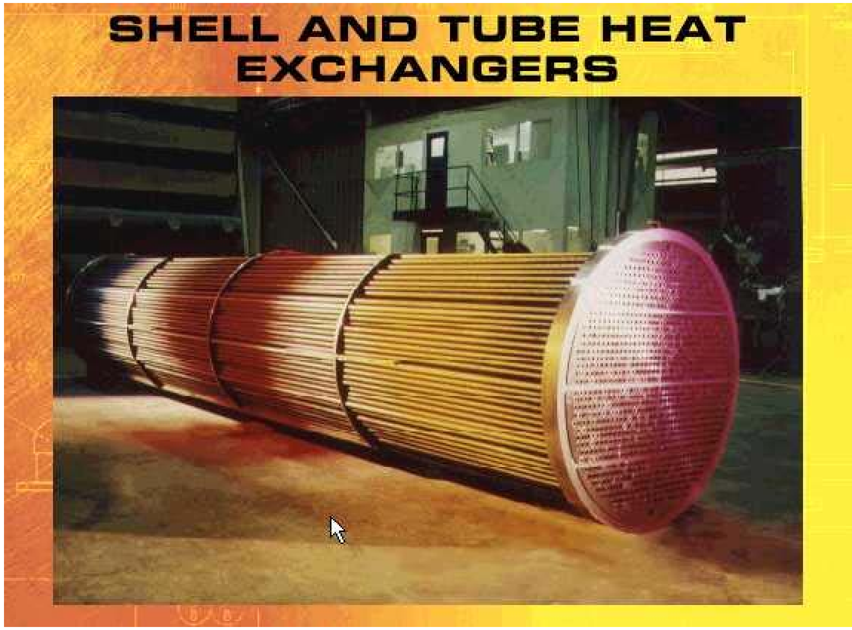
FIGURE 8.11

TABLE 8.2 Nusselt number for fully developed laminar flow in a circular tube annulus with one surface insulated and the other at constant temperature

D_i/D_o	Nu_i	Nu_o	*Comments*
0	—	3.66	See Equation 8.55
0.05	17.46	4.06	
0.10	11.56	4.11	
0.25	7.37	4.23	
0.50	5.74	4.43	
$\approx$1.00	4.86	4.86	See Table 8.1, $b/a \rightarrow \infty$

Used with permission from W. M. Kays and H. C. Perkins, in W. M. Rohsenow and J. P. Hartnett, Eds., *Handbook of Heat Transfer,* Chap. 7, McGraw-Hill, New York, 1972.

Chapter 11
Heat Exchangers



Car Radiator

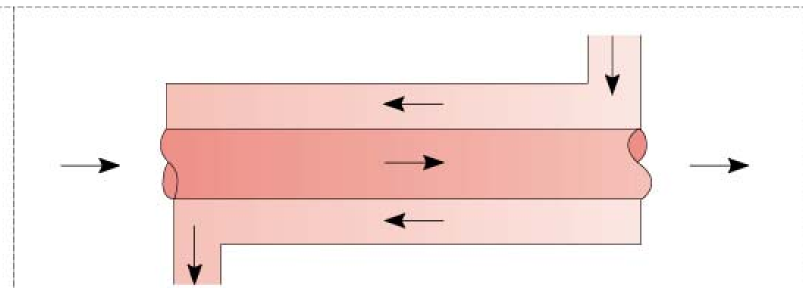
Air conditioning

Heat Exchanger Types

Heat exchangers are ubiquitous to energy conversion and utilization. They involve heat exchange between two fluids separated by a solid and encompass a wide range of flow configurations.

- Concentric-Tube Heat Exchangers

Parallel Flow

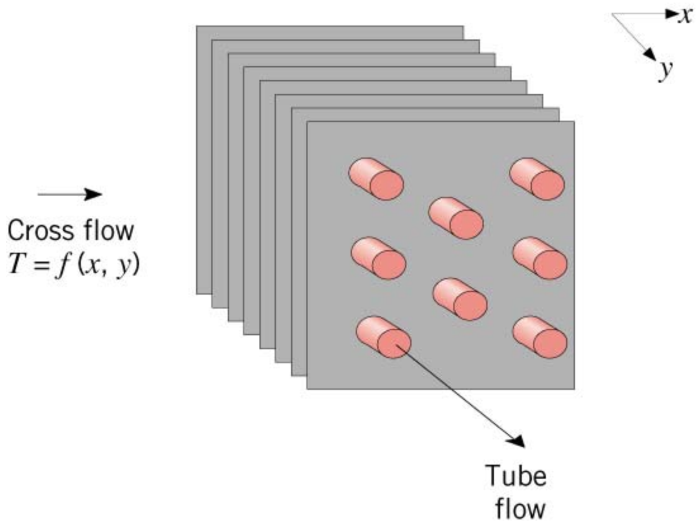


Counterflow

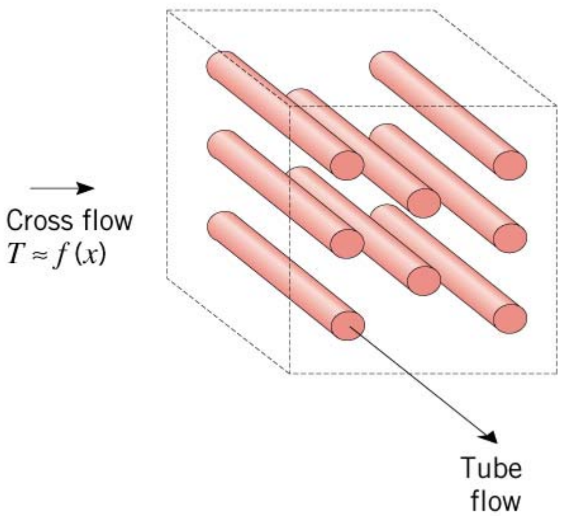
➢ Simplest configuration

➢ Superior performance associated with counter flow

- Cross-flow Heat Exchangers



Finned-Both Fluids
Unmixed

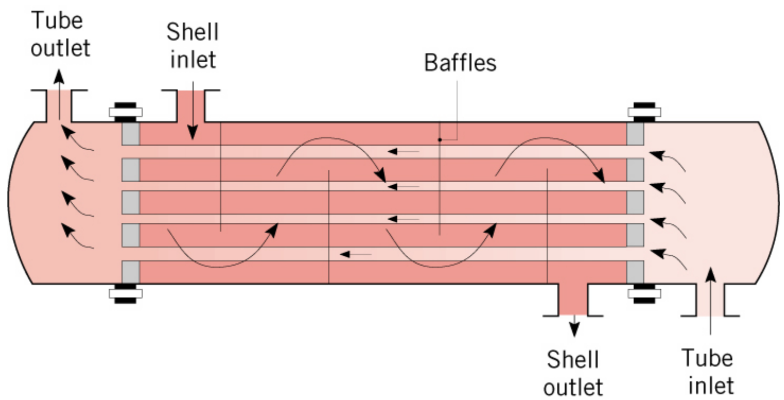


Unfinned-One Fluid Mixed
the Other Unmixed

- For cross-flow over the tubes, fluid motion, and hence mixing, in the transverse direction (y) is prevented for the finned tubes, but occurs for the unfinned condition.

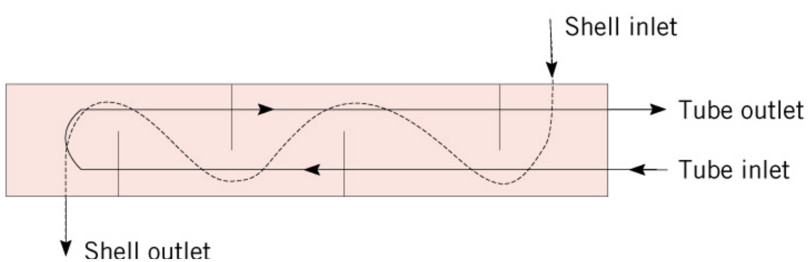
- Heat exchanger performance is influenced by mixing.

- Shell-and-Tube Heat Exchangers

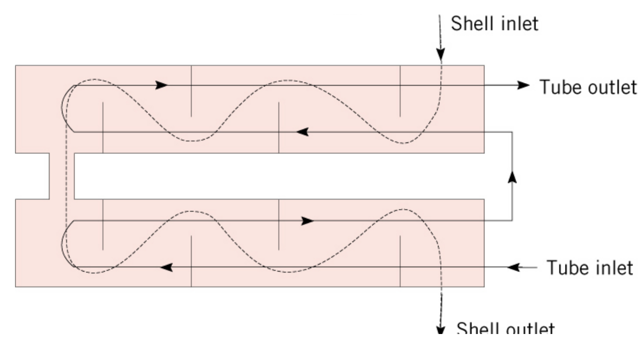


One Shell Pass and One Tube Pass

- ➢ Baffles are used to establish a cross-flow and to induce turbulent mixing of the shell-side fluid, both of which enhance convection.
- ➢ The number of tube and shell passes may be varied, e.g.:

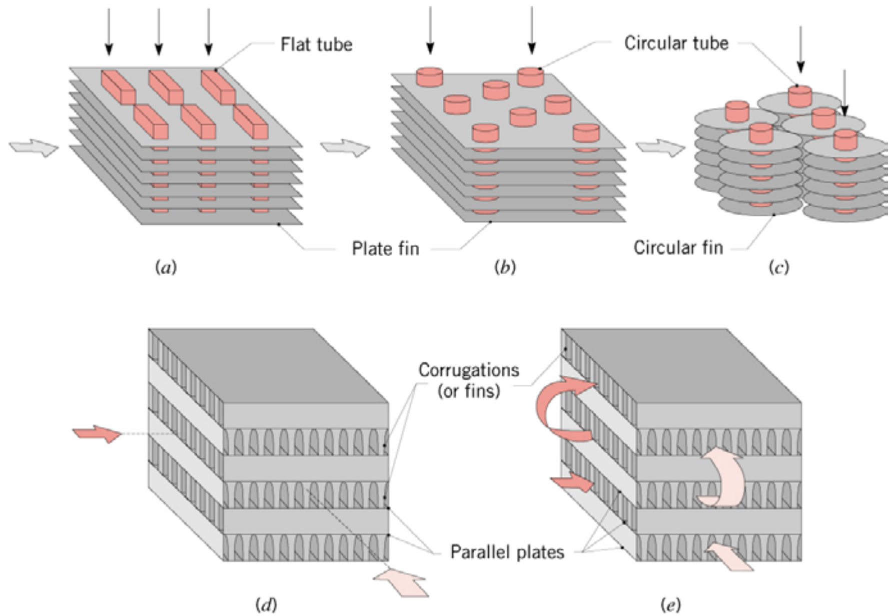


One Shell Pass,
Two Tube Passes



Two Shell Passes,
Four Tube Passes

- Compact Heat Exchangers
  - Widely used to achieve large heat rates per unit volume, particularly when one or both fluids is a gas.
  - Characterized by large heat transfer surface areas per unit volume, small flow passages, and laminar flow.



(a) Fin-tube (flat tubes, continuous plate fins)
(b) Fin-tube (circular tubes, continuous plate fins)
(c) Fin-tube (circular tubes, circular fins)
(d) Plate-fin (single pass)
(e) Plate-fin (multipass)

Table 11.1 Representative Fouling Factors [1]

Fluid	R''_f ($m^2 \cdot K/W$)
Seawater and treated boiler feedwater (below 50°C)	0.0001
Seawater and treated boiler feedwater (above 50°C)	0.0002
River water (below 50°C)	0.0002–0.001
Fuel oil	0.0009
Refrigerating liquids	0.0002
Steam (nonoil bearing)	0.0001

Table 11.2 Representative Values of the Overall Heat Transfer Coefficient

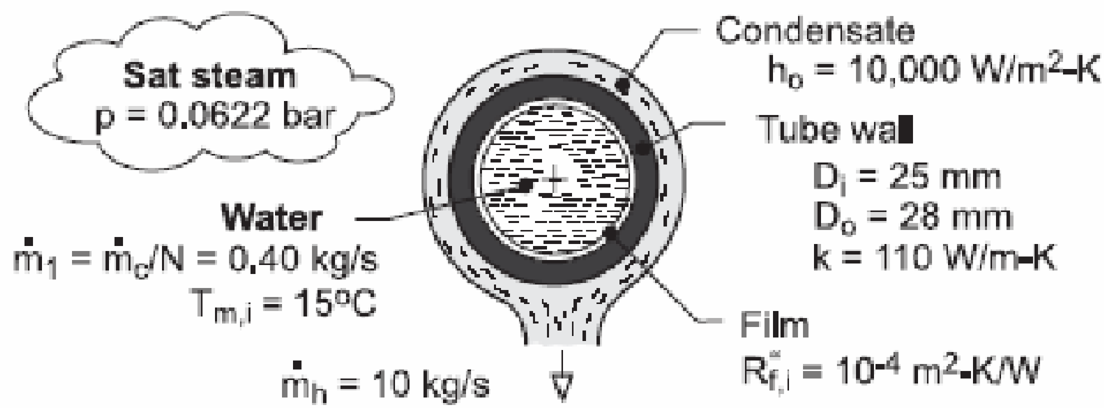
Fluid Combination	U ($W/m^2 \cdot K$)
Water to water	850–1700
Water to oil	110–350
Steam condenser (water in tubes)	1000–6000
Ammonia condenser (water in tubes)	800–1400
Alcohol condenser (water in tubes)	250–700
Finned-tube heat exchanger (water in tubes, air in cross flow)	25–50

PROBLEM 11.7

KNOWN: Number, inner and outer diameters, and thermal conductivity of condenser tubes. Convection coefficient at outer surface. Overall flow rate, inlet temperature and properties of water flow through the tubes. Flow rate and pressure of condensing steam. Fouling factor for inner surface.

FIND: (a) Overall coefficient based on outer surface area, U_o, without fouling, (b) Overall coefficient with fouling, (c) Temperature of water leaving the condenser.

SCHEMATIC:



ASSUMPTIONS: (1) Water is incompressible with negligible viscous dissipation, (2) Fully-developed flow in tubes, (3) Negligible effect of fouling on D_i.

PROPERTIES: Water (Given): c_p = 4180 J/kg·K, $\mu = 9.6 \times 10^{-4}$ N·s/m², k = 0.60 W/m·K, Pr = 6.6. *Table A-6*, Water, saturated vapor (p = 0.0622 bars): T_{sat} = 310 K, $h_{fg} = 2.414 \times 10^{6}$ J/kg.

ANALYSIS: (a) Without fouling, Eq. 11.5 yields

$$\frac{1}{U_o} = \frac{1}{h_i}\left(\frac{D_o}{D_i}\right) + \frac{D_o \ln(Do/Di)}{2k_t} + \frac{1}{h_o}$$

With $Re_{D_i} = 4\dot{m}_1/\pi D_i \mu = 1.60\,\text{kg/s}/\left(\pi \times 0.025\text{m} \times 9.6 \times 10^{-4}\,\text{N}\cdot\text{s/m}^2\right) = 21,220$, flow in the tubes is turbulent, and from Eq. 8.60

$$h_i = \left(\frac{k}{D_i}\right)0.023\,Re_{D_i}^{4/5}\,Pr^{0.4} = \left(\frac{0.60\,\text{W/m}\cdot\text{K}}{0.025\text{m}}\right)0.023(21,200)^{4/5}(6.6)^{0.4} = 3400\,\text{W/m}^2\cdot\text{K}$$

$$U_o = \left[\frac{1}{3400}\left(\frac{28}{25}\right) + \frac{0.028\ln(28/25)}{2\times 110} + \frac{1}{10,000}\right]^{-1}\text{W/m}^2\cdot\text{K} =$$

$$\left(3.29\times 10^{-4} + 1.44\times 10^{-5} + 10^{-4}\right)^{-1}\text{W/m}^2\cdot\text{K} = 2255\,\text{W/m}^2\cdot\text{K}$$

(b) With fouling, Eq. 11.5 yields

$$U_o = \left[4.43\times 10^{-4} + (Do/Di)R''_{f,i}\right]^{-1} = \left(5.55\times 10^{-4}\right)^{-1} = 1800\,\text{W/m}^2\cdot\text{K}$$

(c) The rate at which energy is extracted from the steam equals the rate of heat transfer to the water, $\dot{m}_h h_{fg} = \dot{m}_c c_p (T_{m,o} - T_{m,i})$, in which case

$$T_{m,o} = T_{m,i} + \frac{\dot{m}_h h_{fg}}{\dot{m}_c c_p} = 15°\text{C} + \frac{10\,\text{kg/s}\times 2.414\times 10^6\,\text{J/kg}}{400\,\text{kg/s}\times 4180\,\text{J/kg}\cdot\text{K}} = 29.4°\text{C} \qquad <$$

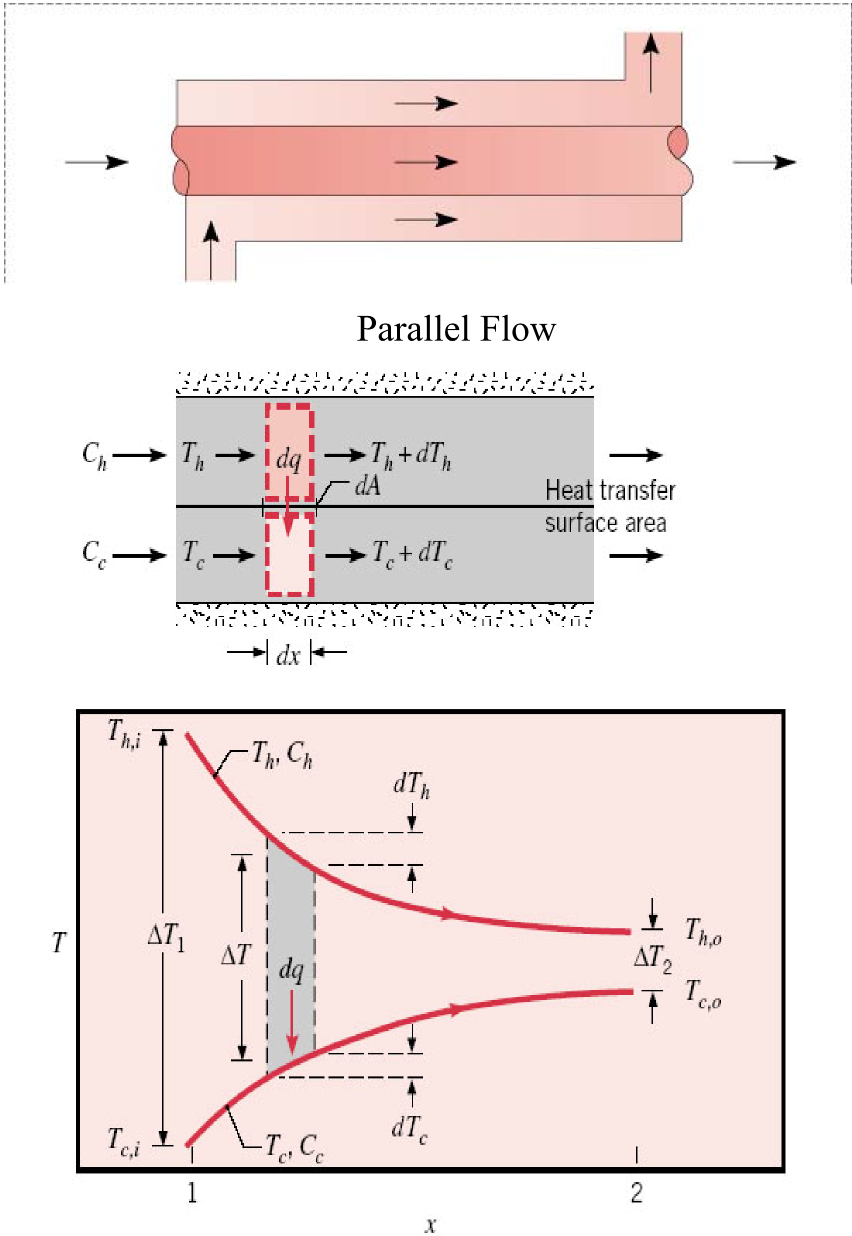


FIGURE 11.7 Temperature distributions for a parallel-flow heat exchanger.

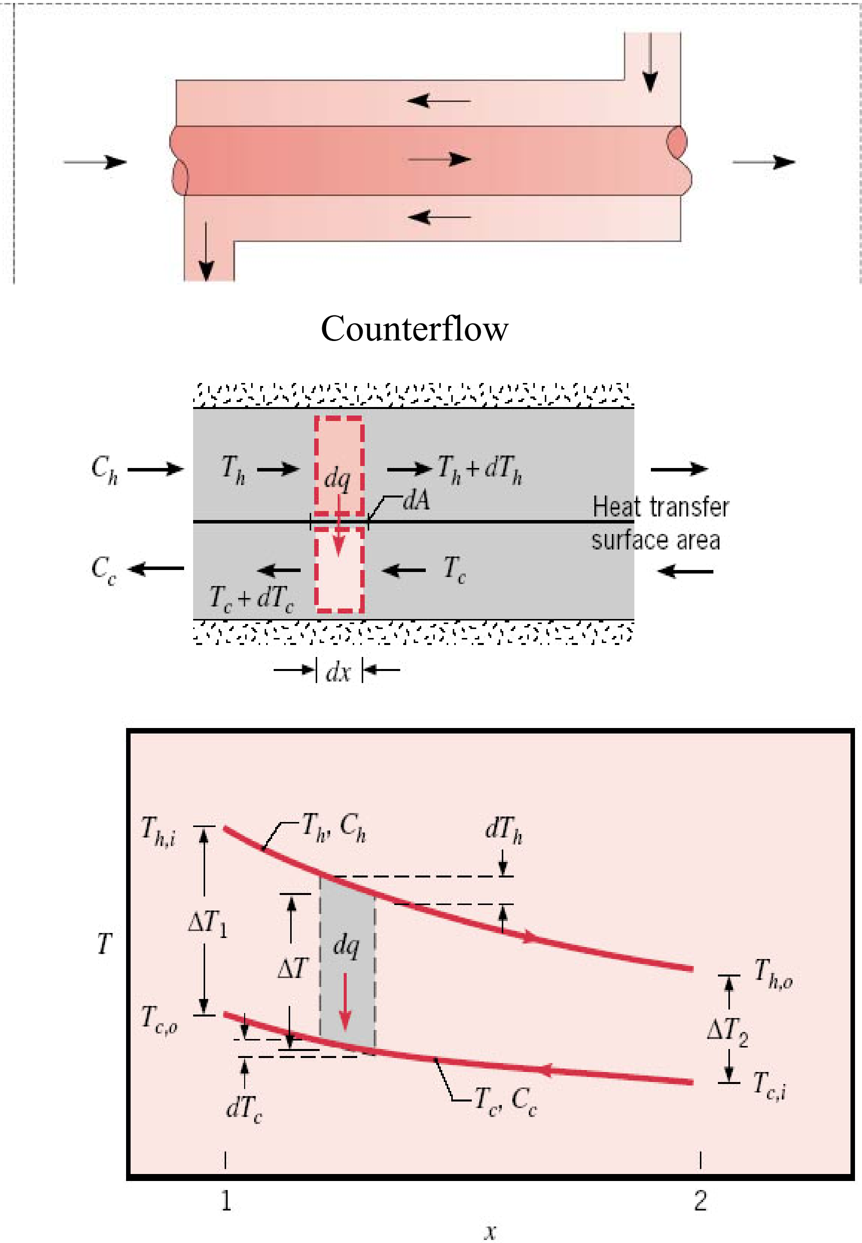
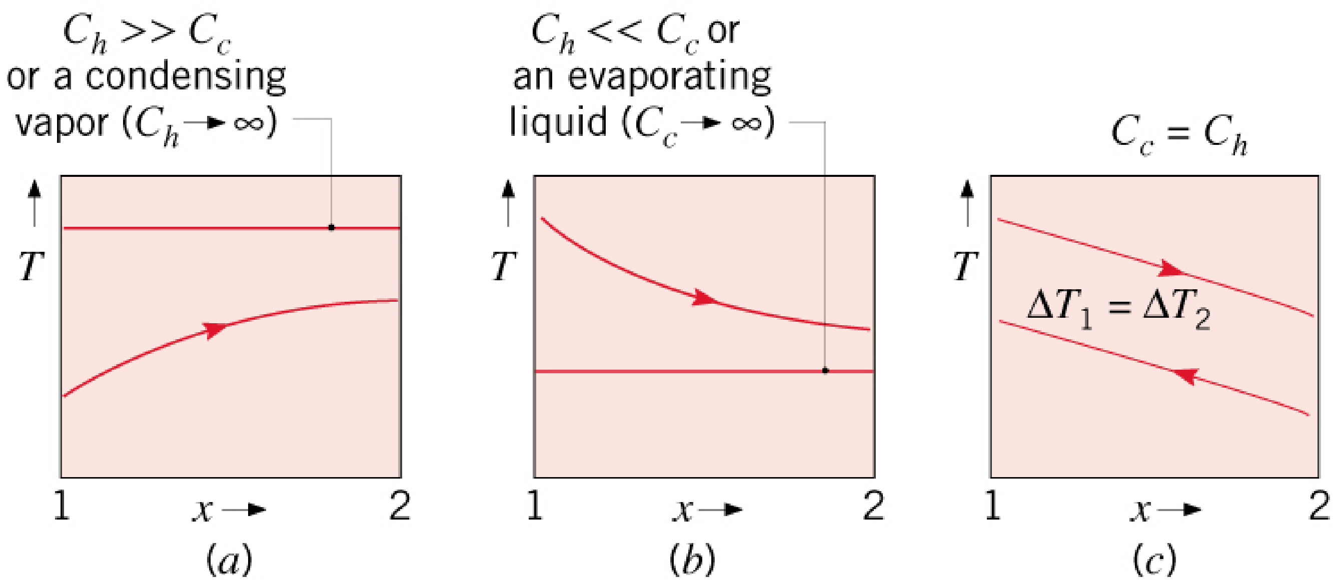


FIGURE 11.8 Temperature distributions for a counterflow heat exchanger.

Special Operating Conditions



- Case (a): $C_h >> C_c$ or h is a condensing vapor $\left(C_h \rightarrow \infty\right)$.
  - Negligible or no change in $T_h \left(T_{h,o} = T_{h,i}\right)$.
- Case (b): $C_c >> C_h$ or c is an evaporating liquid $\left(C_c \rightarrow \infty\right)$.
  - Negligible or no change in $T_c \left(T_{c,o} = T_{c,i}\right)$.
- Case (c): $C_h = C_c$.
  - $\Delta T_1 = \Delta T_2 = \Delta T_{1m}$

Table 11.3 Heat Exchanger Effectiveness Relations [5]

Flow Arrangement	Relation		
Concentric tube			
Parallel flow	$\varepsilon = \dfrac{1 - \exp[-\text{NTU}(1 + C_r)]}{1 + C_r}$		(11.28a)
Counterflow	$\varepsilon = \dfrac{1 - \exp[-\text{NTU}(1 - C_r)]}{1 - C_r \exp[-\text{NTU}(1 - C_r)]}$	$(C_r < 1)$	
	$\varepsilon = \dfrac{\text{NTU}}{1 + \text{NTU}}$	$(C_r = 1)$	(11.29a)
Shell-and-tube			
One shell pass (2, 4, . . . tube passes)	$\varepsilon_1 = 2\left\{1 + C_r + (1 + C_r^2)^{1/2} \times \dfrac{1 + \exp[-(\text{NTU})_1(1 + C_r^2)^{1/2}]}{1 - \exp[-(\text{NTU})_1(1 + C_r^2)^{1/2}]}\right\}^{-1}$		(11.30a)
n Shell passes ($2n$, $4n$, . . . tube passes)	$\varepsilon = \left[\left(\dfrac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1}\right)^n - 1\right]\left[\left(\dfrac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1}\right)^n - C_r\right]^{-1}$		(11.31a)
Cross-flow (single pass)			
Both fluids unmixed	$\varepsilon = 1 - \exp\left[\left(\dfrac{1}{C_r}\right)(\text{NTU})^{0.22}\{\exp[-C_r(\text{NTU})^{0.78}] - 1\}\right]$		(11.32)
C_{max} (mixed), C_{min} (unmixed)	$\varepsilon = \left(\dfrac{1}{C_r}\right)(1 - \exp\{-C_r[1 - \exp(-\text{NTU})]\})$		(11.33a)
C_{min} (mixed), C_{max} (unmixed)	$\varepsilon = 1 - \exp(-C_r^{-1}\{1 - \exp[-C_r(\text{NTU})]\})$		(11.34a)
All exchangers ($C_r = 0$)	$\varepsilon = 1 - \exp(-\text{NTU})$		(11.35a)

TABLE 11.4 Heat Exchanger NTU Relations

Flow Arrangement	Relation	
Concentric tube		
Parallel flow	$\mathrm{NTU} = -\dfrac{\ln[1 - \varepsilon(1 + C_r)]}{1 + C_r}$	(11.28b)
Counterflow	$\mathrm{NTU} = \dfrac{1}{C_r - 1}\ln\left(\dfrac{\varepsilon - 1}{\varepsilon C_r - 1}\right) \quad (C_r < 1)$	
	$\mathrm{NTU} = \dfrac{\varepsilon}{1 - \varepsilon} \quad (C_r = 1)$	(11.29b)
Shell-and-tube		
One shell pass (2, 4, . . . tube passes)	$(\mathrm{NTU})_1 = -(1 + C_r^2)^{-1/2}\ln\left(\dfrac{E - 1}{E + 1}\right)$	(11.30b)
	$E = \dfrac{2/\varepsilon_1 - (1 + C_r)}{(1 + C_r^2)^{1/2}}$	(11.30c)
n Shell passes ($2n$, $4n$, . . . tube passes)	Use Equations 11.30b and 11.30c with $\varepsilon_1 = \dfrac{F - 1}{F - C_r} \quad F = \left(\dfrac{\varepsilon C_r - 1}{\varepsilon - 1}\right)^{1/n} \quad \mathrm{NTU} = n(\mathrm{NTU})_1$	(11.31b, c, d)
Cross-flow (single pass)		
C_{max} (mixed), C_{min} (unmixed)	$\mathrm{NTU} = -\ln\left[1 + \left(\dfrac{1}{C_r}\right)\ln(1 - \varepsilon C_r)\right]$	(11.33b)
C_{min} (mixed), C_{max} (unmixed)	$\mathrm{NTU} = -\left(\dfrac{1}{C_r}\right)\ln[C_r\ln(1 - \varepsilon) + 1]$	(11.34b)
All exchangers ($C_r = 0$)	$\mathrm{NTU} = -\ln(1 - \varepsilon)$	(11.35b)

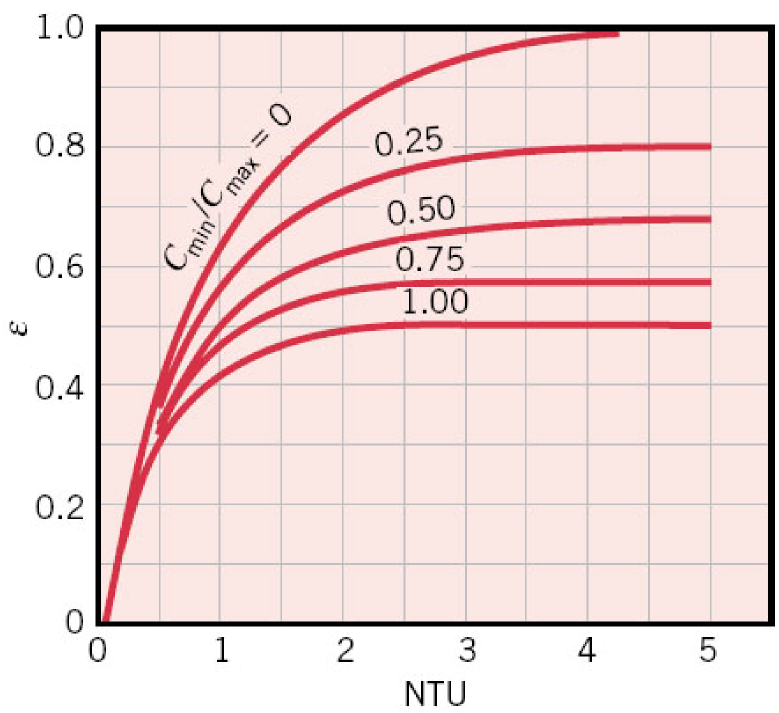


FIGURE 11.10 Effectiveness of a parallel-flow heat exchanger (Equation 11.28).

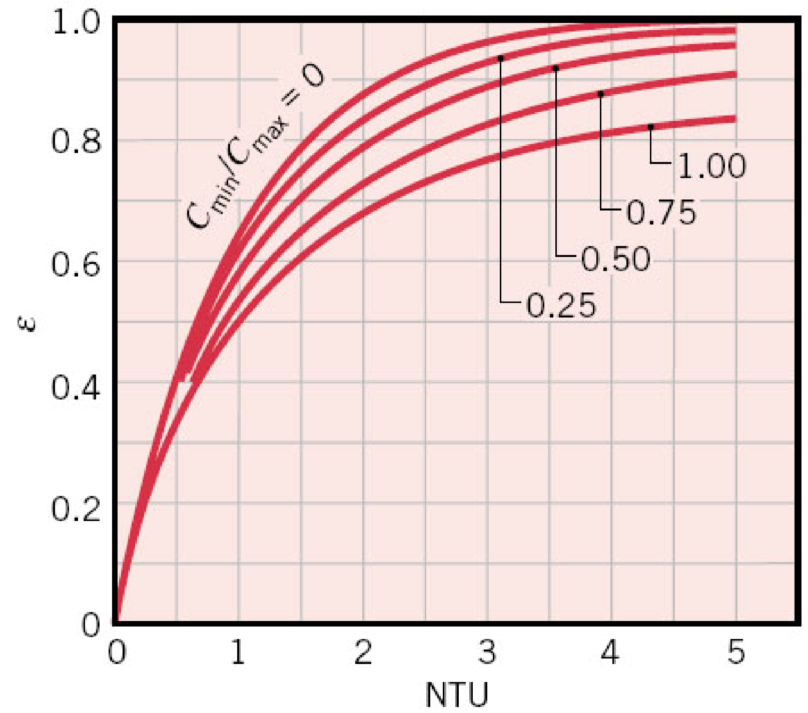


FIGURE 11.11 Effectiveness of a counterflow heat exchanger (Equation 11.29).

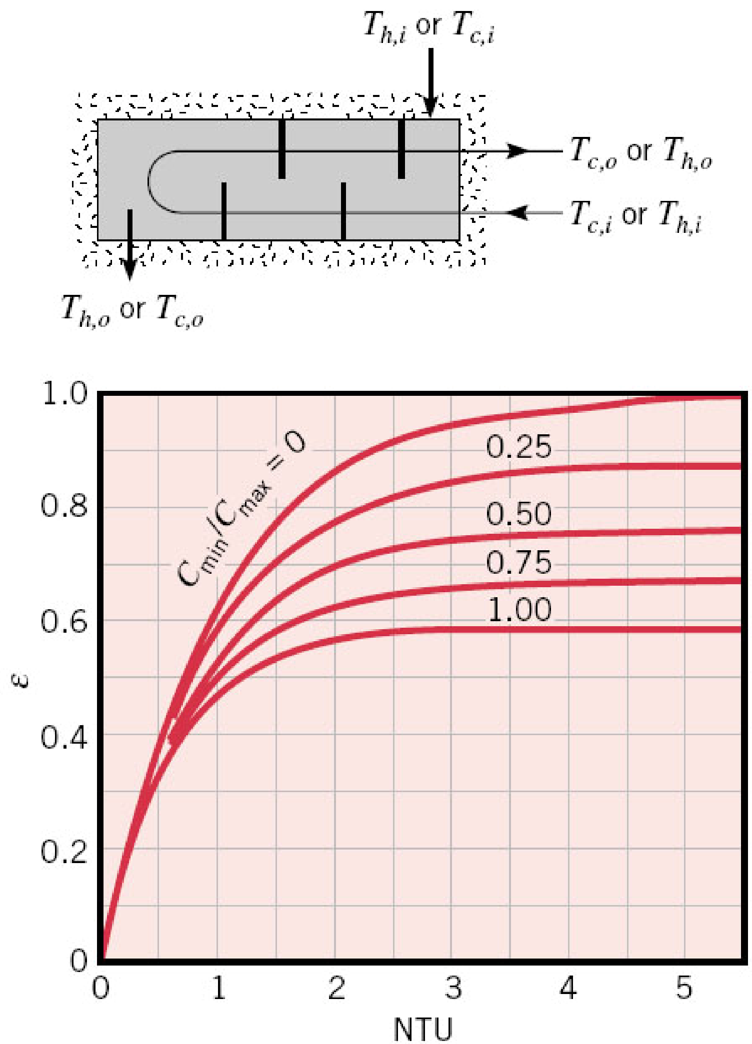


FIGURE 11.12 Effectiveness of a shell-and-tube heat exchanger with one shell and any multiple of two tube passes (two, four, etc. tube passes) (Equation 11.30).

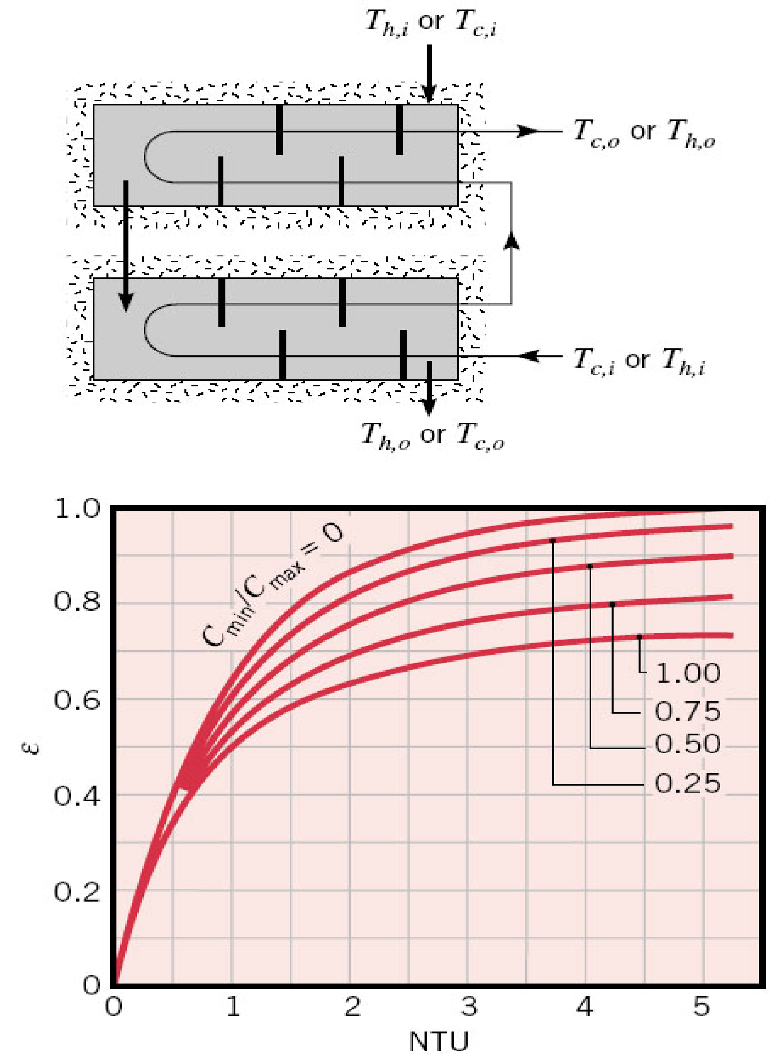


FIGURE 11.13 Effectiveness of a shell-and-tube heat exchanger with two shell passes and any multiple of four tube passes (four, eight, etc. tube passes) (Equation 11.31 with $n = 2$).

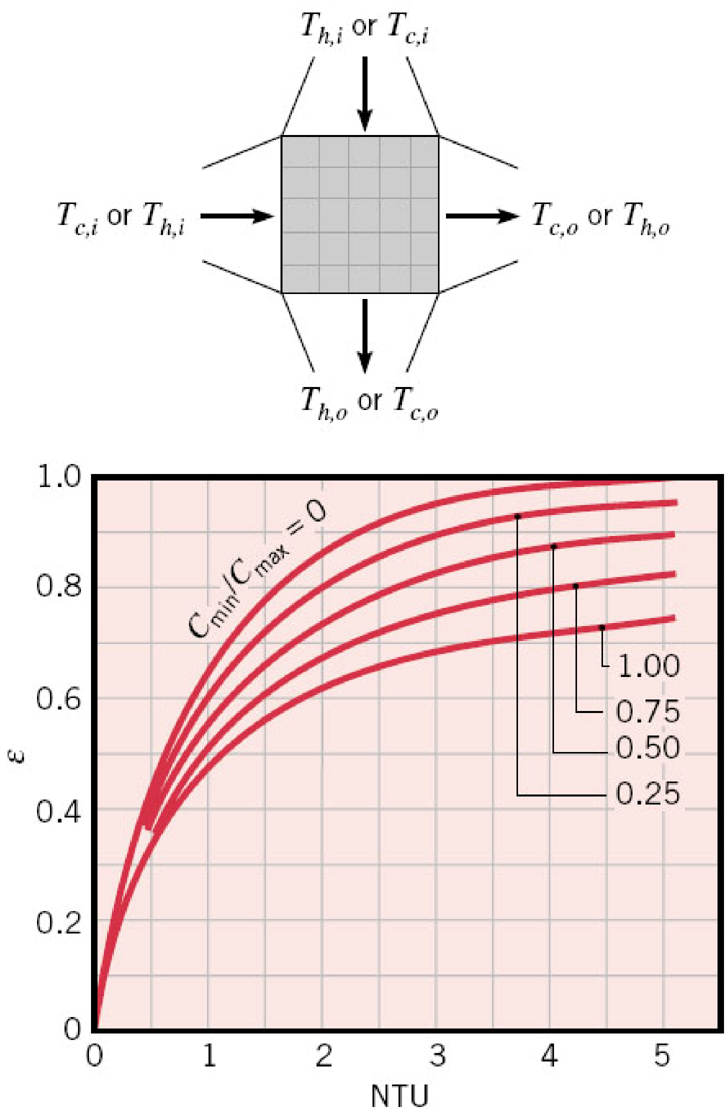


FIGURE 11.14 Effectiveness of a single-pass, cross-flow heat exchanger with both fluids unmixed (Equation 11.32).

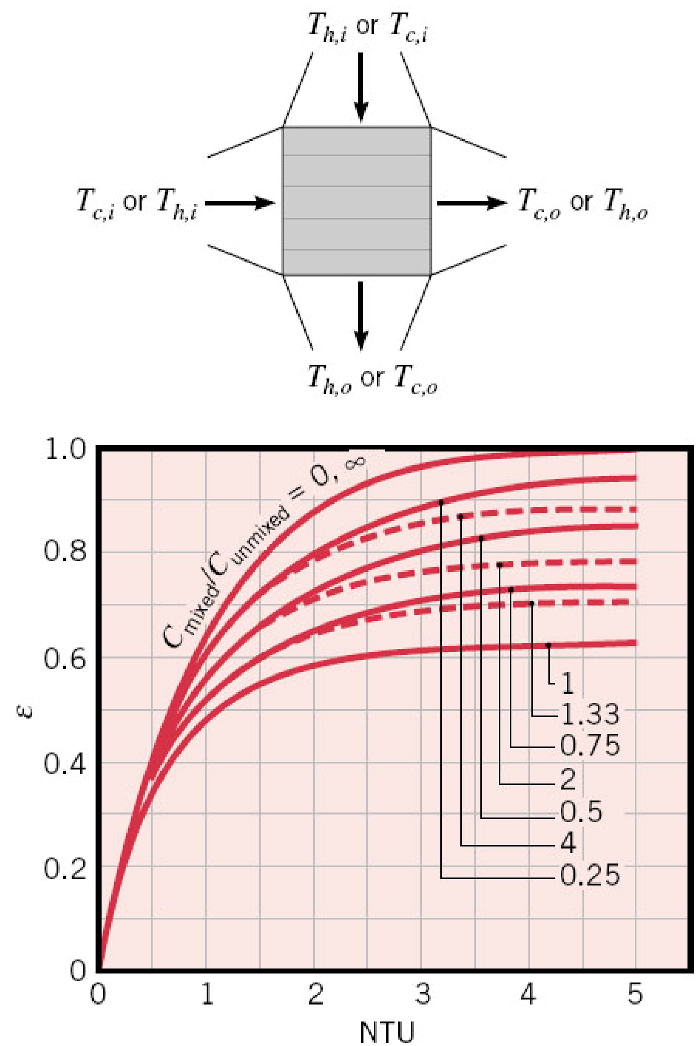


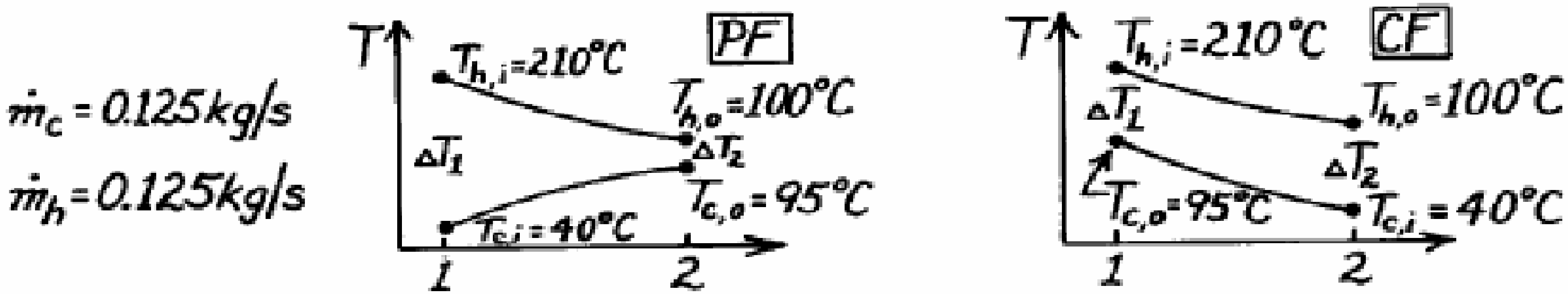
FIGURE 11.15 Effectiveness of a single-pass, cross-flow heat exchanger with one fluid mixed and the other unmixed (Equations 11.33, 11.34).

PROBLEM 11.63

KNOWN: Concentric tube heat exchanger with prescribed conditions.

FIND: (a) Maximum possible heat transfer, (b) Effectiveness, (c) Whether heat exchanger should be run in PF or CF to minimize size or weight; determine ratio of required areas for the two flow conditions.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat loss to surroundings, (2) Constant properties, (3) Overall heat transfer coefficient remains unchanged for PF or CF conditions.

PROPERTIES: Hot fluid (given): c = 2100 J/kg·K; Cold fluid (given): c = 4200 J/kg·K.

ANALYSIS: (a) The maximum possible heat transfer rate is given by Eq. 11.18.

$$q_{max} = C_{min}\left(T_{h,i} - T_{c,o}\right).$$

The minimum capacity fluid is the hot fluid with $C_{min} = \dot{m}_h\ c_h$, giving

$$q_{max} = \dot{m}_h\, c_h\left(T_{h,i} - T_{c,o}\right) = 0.125\frac{\text{kg}}{\text{s}}\times 2100\frac{\text{J}}{\text{kg}\cdot\text{K}}(210-40)\text{K} = 44,625\,\text{W}.$$

(b) The effectiveness is defined by Eq. 11.19 and the heat rate, q, can be determined from an energy balance on the cold fluid.

$$\varepsilon = q/q_{max} = \dot{m}_c\, c_c\left(T_{c,o} - T_{c,i}\right)/q_{max}$$

$$\varepsilon = 0.125\ \text{kg/s}\times 4200\ \text{J/kg}\cdot\text{K}(95-40)\text{K}/44,625\,\text{W} = 0.65.$$ <

(c) Operating the heat exchanger under CF conditions will require a smaller heat transfer area than for PF conditions. The ratio of the areas is

$$\frac{A_{CF}}{A_{PF}} = \frac{q/U\Delta T_{\ell m,CF}}{q/U\Delta T_{\ell m,PF}} = \frac{\Delta T_{\ell m,PF}}{\Delta T_{\ell m,CF}}.$$

To calculate the LMTD, first find $T_{h,o}$ from overall energy balances on the two fluids.

$$T_{h,o} = T_{h,i} - \frac{\dot{m}_c\, c_c}{\dot{m}_h\, c_h}\left(T_{c,o} - T_{c,i}\right) = 210°\text{C} - \frac{0.125\times 4200}{0.125\times 2100}(95-40)°\text{C} = 100°\text{C}.$$

Using Eq. 11.15 with ΔT_1 and ΔT_2 as shown below, find $\Delta T_{\ell m} = (\Delta T_1 - \Delta T_2)/\ell n(\Delta T_1/\Delta T_2)$.

Substituting values, find

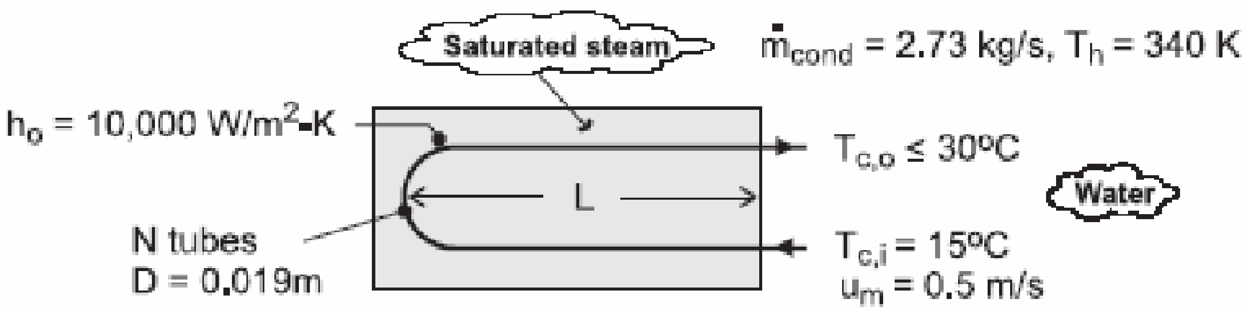
$$\frac{A_{CF}}{A_{PF}} = \frac{\left[(210-40)-(100-95)\right]/\ell n(170/5)}{\left[(210-95)-(100-40)\right]\ell n(115/60)} = \frac{46.8°\text{C}}{84.5°\text{C}} = 0.55.$$ <

PROBLEM 11.38

KNOWN: Temperature, convection coefficient and condensation rate of saturated steam. Tube diameter for shell-and-tube heat exchanger with one shell pass and two tube passes. Velocity and inlet and maximum allowable exit temperatures of cooling water.

FIND: (a) Minimum number of tubes and tube length per pass, (b) Effect of tube-side heat transfer enhancement on tube length.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat exchange with surroundings, (2) Negligible tube wall conduction and fouling resistance, (3) Constant properties, (4) Fully developed internal flow throughout.

PROPERTIES: *Table A-6*, Sat. water (340 K): $h_{fg} = 2.342 \times 10^6$ J/kg; Sat. water $\left(\overline{T}_c = 22.5°C \approx 295\ K\right)$: ρ=998 kg/m^3, c_p = 4181 J/kg·K, $\mu = 959 \times 10^{-6}$ N·s/m^2, k = 0.606 W/m·K, Pr = 6.62.

ANALYSIS: (a) The required heat rate and the maximum allowable temperature rise of the water determine the minimum allowable flow rate. That is, with

$$q = q_{cond} = \dot{m}_{cond} h_{fg} = 2.73\ \text{kg/s} \times 2.342\times10^{6}\ \text{J/kg} = 6.39\times10^{6}\ \text{W}$$

$$\dot{m}_{c,min} = \frac{q}{c_{p,c}\left(T_{c,o} - T_{c,i}\right)} = \frac{6.39\times10^{6}\ \text{W}}{4181\ \text{J/kg}\cdot\text{K}\left(15°\text{C}\right)} = 101.9\ \text{kg/s}$$

With a specified flow rate per tube of $\dot{m}_{c,1} = \rho u_m \pi D^2/4 = 998\ \text{kg/m}^3 \times 0.5\ \text{m/s} \times \pi\,(0.019\text{m})^2/4 =$ 0.141 kg/s, the minimum number of tubes is

$$N_{min} = \frac{\dot{m}_{c,min}}{\dot{m}_{c,1}} = \frac{101.9\ \text{kg/s}}{0.141\ \text{kg/s}} = 720$$ <

To determine the corresponding tube length, we must first find the required heat transfer surface area. With $Re_D = \rho u_m D/\mu = 998\ \text{kg/m}^3\ (0.5\ \text{m/s})\ 0.019\text{m}/959\times10^{-6}\ \text{N}\cdot\text{s/m}^2 = 9{,}886$, the Dittus-Boelter equation yields

$$\overline{h}_i = \left(k/D\right)0.023\ Re_D^{4/5}\ Pr^{0.4} = \left(0.606\ \text{W/m}\cdot\text{K}/0.019\text{m}\right)0.023\left(9886\right)^{4/5}\left(6.62\right)^{0.4} = 2454\ \text{W/m}^2\cdot\text{K}$$

Hence, $U = \left[\overline{h}_i^{-1} + h_o^{-1}\right]^{-1} = 1970\ \text{W/m}^2\cdot\text{K}$

With $C_r = 0$, $C_{min} = \dot{m}_c\, c_{p,c} = 101.9$ kg/s $\times$ 4181 J/kg-K $= 4.26 \times 10^5$ W/K, $q_{max} = C_{min}(T_{h,i} - T_{c,i})$ $= 4.26 \times 10^5$ W/K (340 – 288) K $= 2.215 \times 10^7$ W and $\varepsilon = q/q_{max} = 0.289$, Eq. 11.35b yields NTU = – ln (1 – ε) = – ln (1 – 0.289) = 0.341. Hence the tube length per pass is

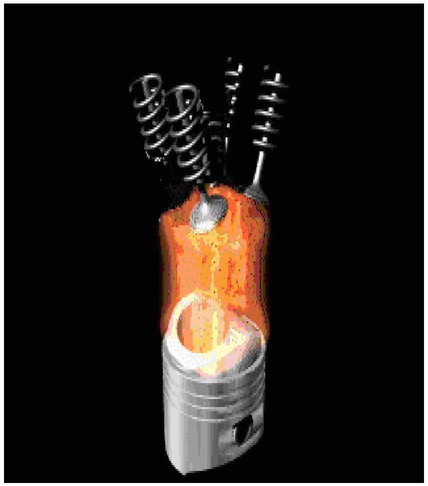
$$L = \frac{A}{2N\pi D} = \frac{NTU \times C_{min}}{2N\pi DU} = \frac{0.341 \times 4.26 \times 10^5 \text{ W/K}}{2 \times 720 \times \pi(0.019\text{m})1970 \text{ W/m}^2 \cdot \text{K}} = 0.858\text{m} \qquad <$$

(b) If the tube-side convection coefficient is doubled, $\overline{h}_i = 4908 \text{ W/m}^2 \cdot \text{K}$ and U = 3292 $\text{W/m}^2 \cdot \text{K}$. Since q, C_r, C_{min}, q_{max} and hence ε are unchanged, the number of transfer units is still NTU = 0.341. Hence, the tube length per pass is now

$$L = \frac{NTU \times C_{min}}{2N\pi DU} = \frac{0.341 \times 4.26 \times 10^5 \text{ W/K}}{2 \times 720 \times \pi(0.019\text{m})3292 \text{ W/m}^2 \cdot \text{K}} = 0.513\text{m} \qquad <$$

Chapter 12
Radiation: Processes and Properties

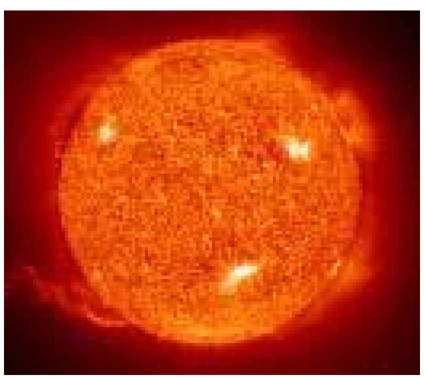
Radiation heat transfer mode is important in applications involving high temperatures.



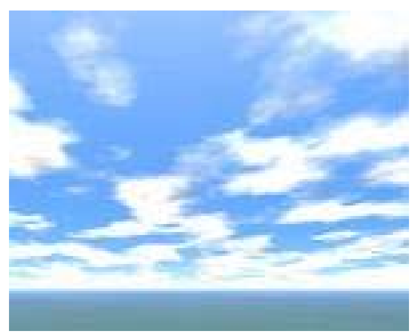
Engines

Rocket nozzles

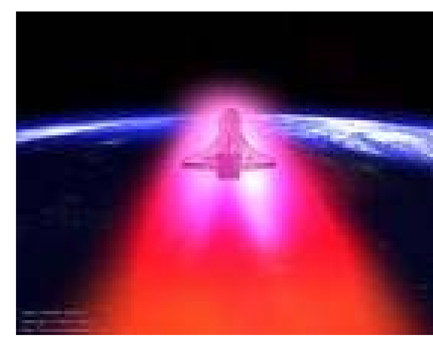
Fires



Sun



Atmospheric radiation



Space vehicles

- Unlike conduction and convection, radiation requires no medium.
- Attention is focused on thermal radiation, whose origins are associated with emission from matter at an absolute temperature.
- Emission is due to oscillations and transitions of the many electrons that comprise matter, which are, in turn, sustained by the thermal energy of the matter.
- Emission corresponds to heat transfer from the matter and hence to a reduction in thermal energy stored by the matter.

Radiation heat transfer can take place between two bodies separated by a medium colder than both bodies.

- Radiation may also be intercepted and absorbed by matter.
- Absorption results in heat transfer to the matter and hence an increase in thermal energy stored by the matter.

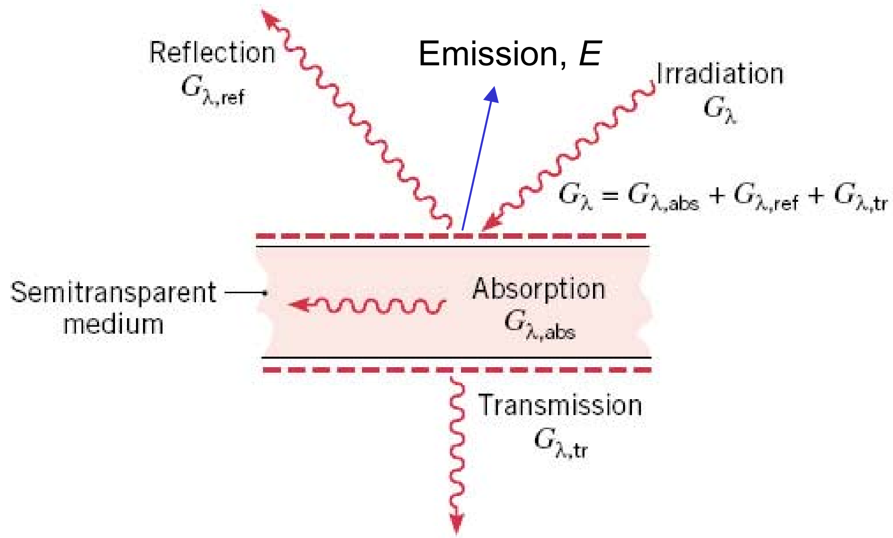


FIGURE 12.20
Absorption, reflection, and transmission processes associated with a semitransparent medium.

- The dual nature of radiation:
  - In some cases, the physical manifestations of radiation may be explained by viewing it as particles (aka photons or quanta).
  - In other cases, radiation behaves as an electromagnetic wave.

– In all cases, radiation is characterized by a wavelength λ and frequency ν, which are related through the speed at which radiation propagates in the medium of interest:

$$\lambda = \frac{c}{\nu}$$

For propagation in a vacuum, $c_0 = 2.998 \times 10^8$ m/s

$c = c_0/n$, where n is index of refraction of the medium

$n \approx 1$ for air and most gases; $n \approx 1.5$ for water and glass

Quantum theory views electromagnetic radiation as the propagation of discrete packets of energy called photons. Each photon of frequency ν has the energy of:

$E\ (\text{eV}) = h\,\nu = hc/\lambda = 1.24/\lambda\ (\mu\text{m})$

$h = 6.625 \times 10^{-34}$ J.s ; $\text{eV} = 1.6022 \times 10^{-19}$ J

Analysis of thermal radiation is much more complex compared to conduction and convection.

Conduction and convection

- Short-range phenomena
  (~ mean free path, ca. 10^{-10} m)

- Properties (k, μ, ρ, etc.) are easily measured.

- Energy balance on an infinitesimal volume, leading to partial differential equation with 4 variables (x, y, z, t)

Radiation

- Long-range phenomena
  ($10^{-10} – 10^{+10}$ m)

- Properties are difficult to measure (wavelength dependence).

- Energy balance over the entire volume, leading to integral equation with 7 variables ($x, y, z, t, \theta, \phi, \lambda$)

Electromagnetic Spectrum

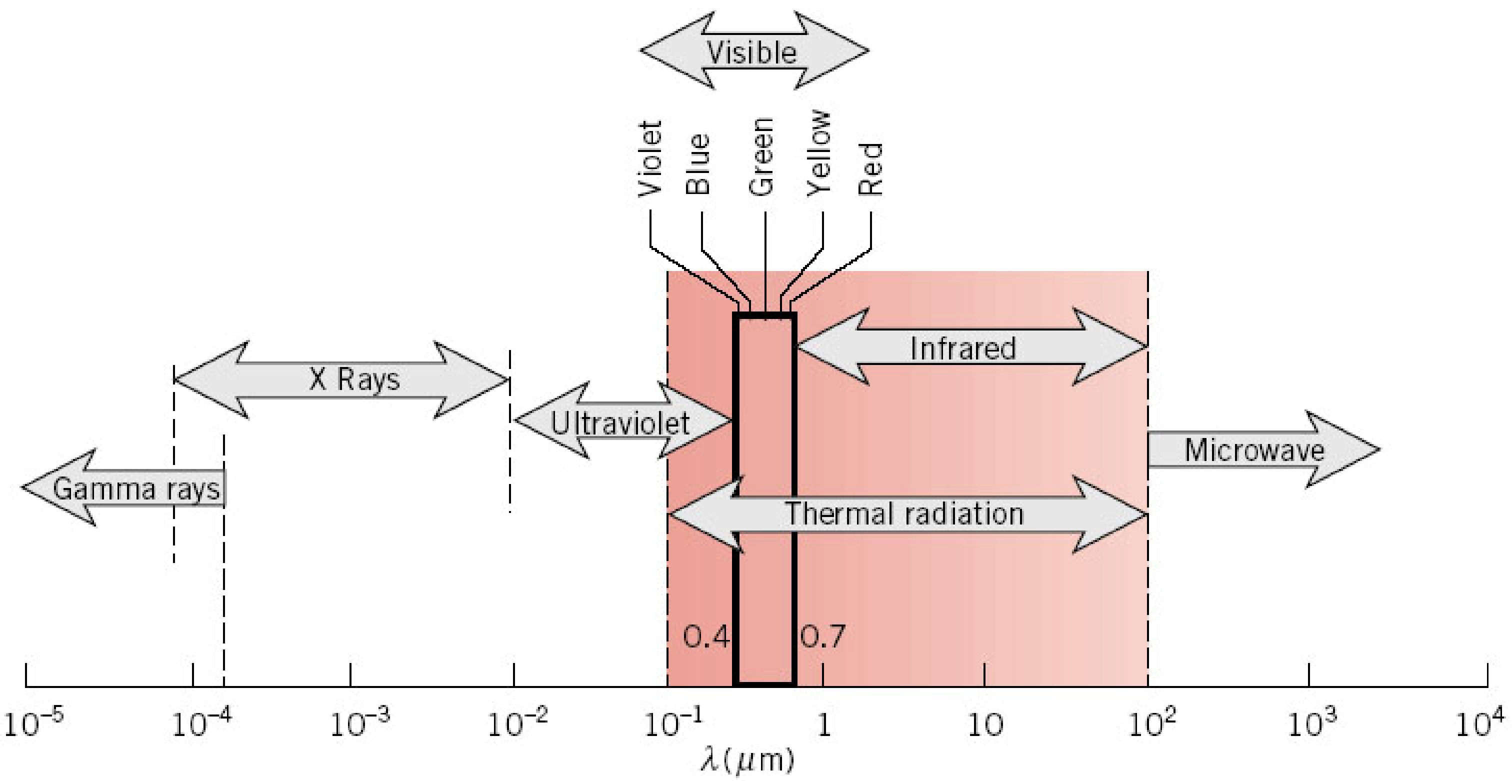


FIGURE 12.3 Spectrum of electromagnetic radiation.

- Thermal radiation is confined to the infrared, visible and ultraviolet regions of the spectrum $(0.1 < \lambda < 100\mu m)$.

Blackbody Radiation

- Fact: Every object emits radiation in all directions over all wavelengths depending on the material, surface properties, and temperature.
- Question: What is the maximum amount of radiation that can be emitted by a surface at a given temperature?
- Answer: Define an idealized body, called "blackbody" to serve as a standard.

Blackbody = perfect emitter and absorber of radiation

- ➢At a given T and λ, no surface can emit more radiation than a blackbody.
- ➢Regardless of λ and direction, a blackbody absorbs all incident radiation.
- ➢A blackbody is a diffuse emitter, i.e., emits radiation uniformly in all directions.

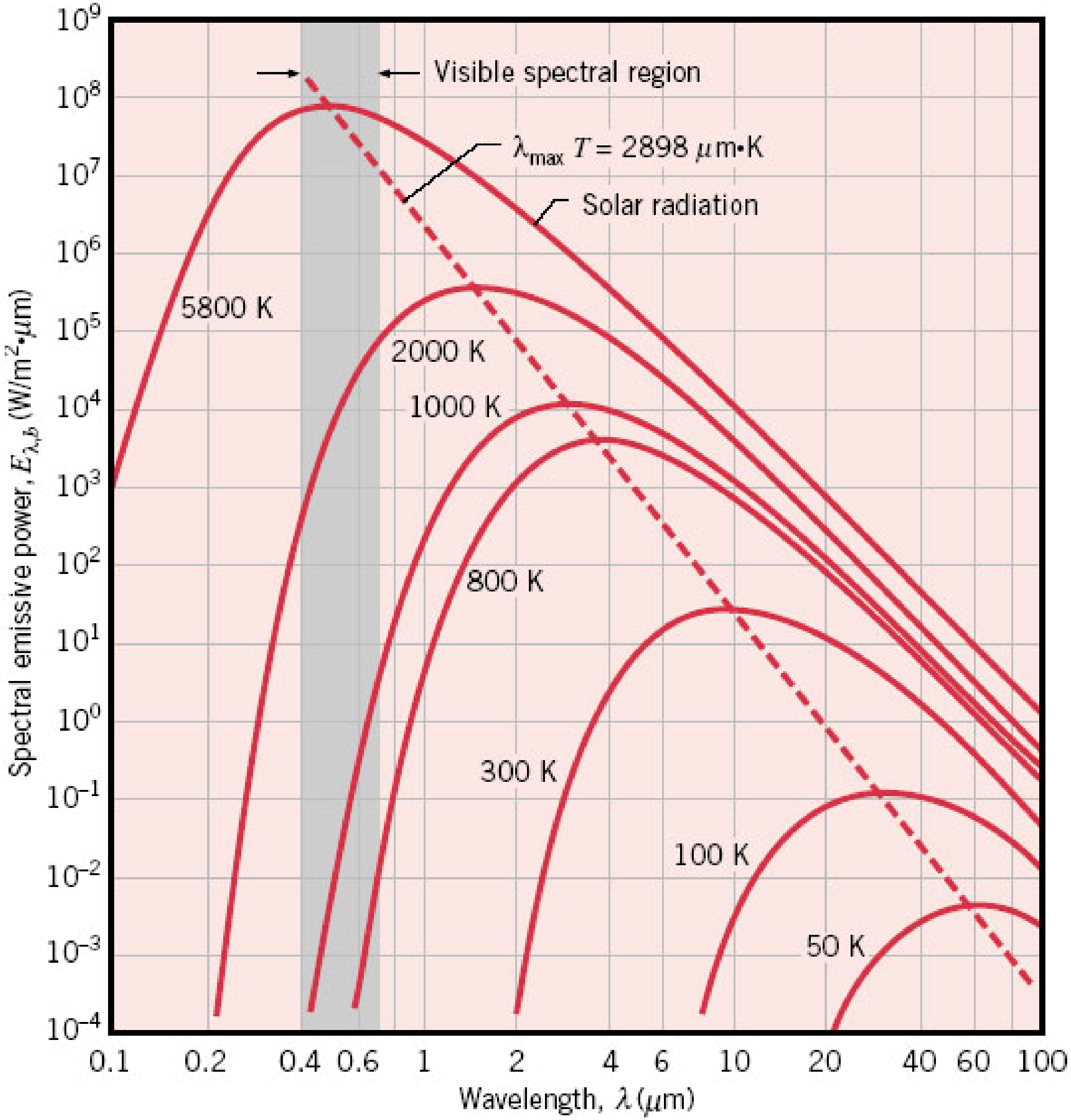


FIGURE 12.12 Spectral blackbody emissive power.

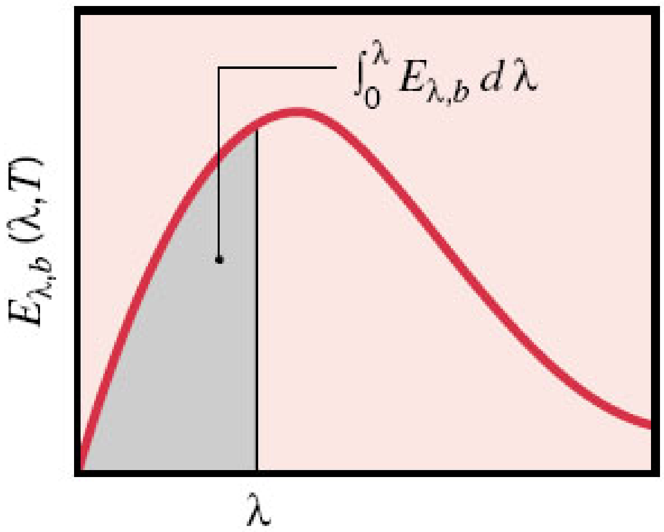


FIGURE 12.13 Radiation emission from a blackbody in the spectral band 0 to $\boldsymbol{\lambda}$.

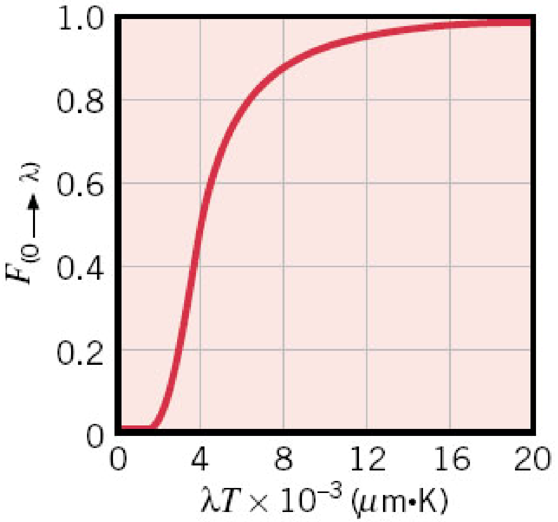


FIGURE 12.14
Fraction of the total blackbody emission in the spectral band from 0 to $\boldsymbol{\lambda}$ as a function of $\boldsymbol{\lambda} T$.

TABLE 12.1 Blackbody Radiation Functions

λT (μm · K)	$F_{(0 \rightarrow \lambda)}$	$I_{\lambda,b}(\lambda, T)/\sigma T^5$ (μm · K · sr)$^{-1}$	$\dfrac{I_{\lambda,b}(\lambda, T)}{I_{\lambda,b}(\lambda_{max}, T)}$
200	0.000000	0.375034×10^{-27}	0.000000
400	0.000000	0.490335×10^{-13}	0.000000
600	0.000000	0.104046×10^{-8}	0.000014
800	0.000016	0.991126×10^{-7}	0.001372
1,000	0.000321	0.118505×10^{-5}	0.016406
1,200	0.002134	0.523927×10^{-5}	0.072534
1,400	0.007790	0.134411×10^{-4}	0.186082
1,600	0.019718	0.249130	0.344904
1,800	0.039341	0.375568	0.519949
2,000	0.066728	0.493432	0.683123
2,200	0.100888	0.589649×10^{-4}	0.816329
2,400	0.140256	0.658866	0.912155
2,600	0.183120	0.701292	0.970891
2,800	0.227897	0.720239	0.997123
2,898	0.250108	0.722318×10^{-4}	1.000000
3,000	0.273232	0.720254×10^{-4}	0.997143
3,200	0.318102	0.705974	0.977373
3,400	0.361735	0.681544	0.943551
3,600	0.403607	0.650396	0.900429
3,800	0.443382	0.615225×10^{-4}	0.851737
4,000	0.480877	0.578064	0.800291
4,200	0.516014	0.540394	0.748139
4,400	0.548796	0.503253	0.696720
4,600	0.579280	0.467343	0.647004
4,800	0.607559	0.433109	0.599610
5,000	0.633747	0.400813	0.554898
5,200	0.658970	0.370580×10^{-4}	0.513043
5,400	0.680360	0.342445	0.474092
5,600	0.701046	0.316376	0.438002
5,800	0.720158	0.292301	0.404671
6,000	0.737818	0.270121	0.373965
6,200	0.754140	0.249723×10^{-4}	0.345724
6,400	0.769234	0.230985	0.319783
6,600	0.783199	0.213786	0.295973
6,800	0.796129	0.198008	0.274128
7,000	0.808109	0.183534	0.254090
7,200	0.819217	0.170256×10^{-4}	0.235708
7,400	0.829527	0.158073	0.218842
7,600	0.839102	0.146891	0.203360
7,800	0.848005	0.136621	0.189143
8,000	0.856288	0.127185	0.176079
8,500	0.874608	0.106772×10^{-4}	0.147819
9,000	0.890029	0.901463×10^{-5}	0.124801

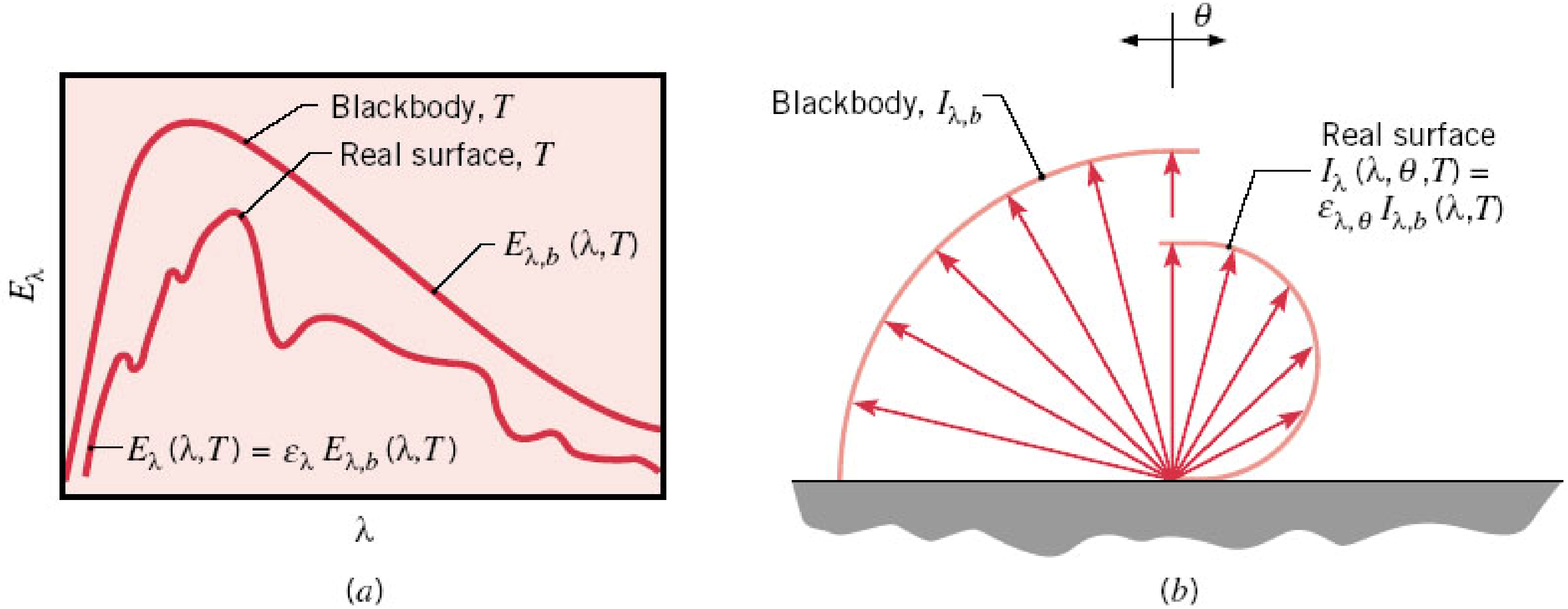


FIGURE 12.15 Comparison of blackbody and real surface emission. (a) Spectral distribution. (b) Directional distribution.

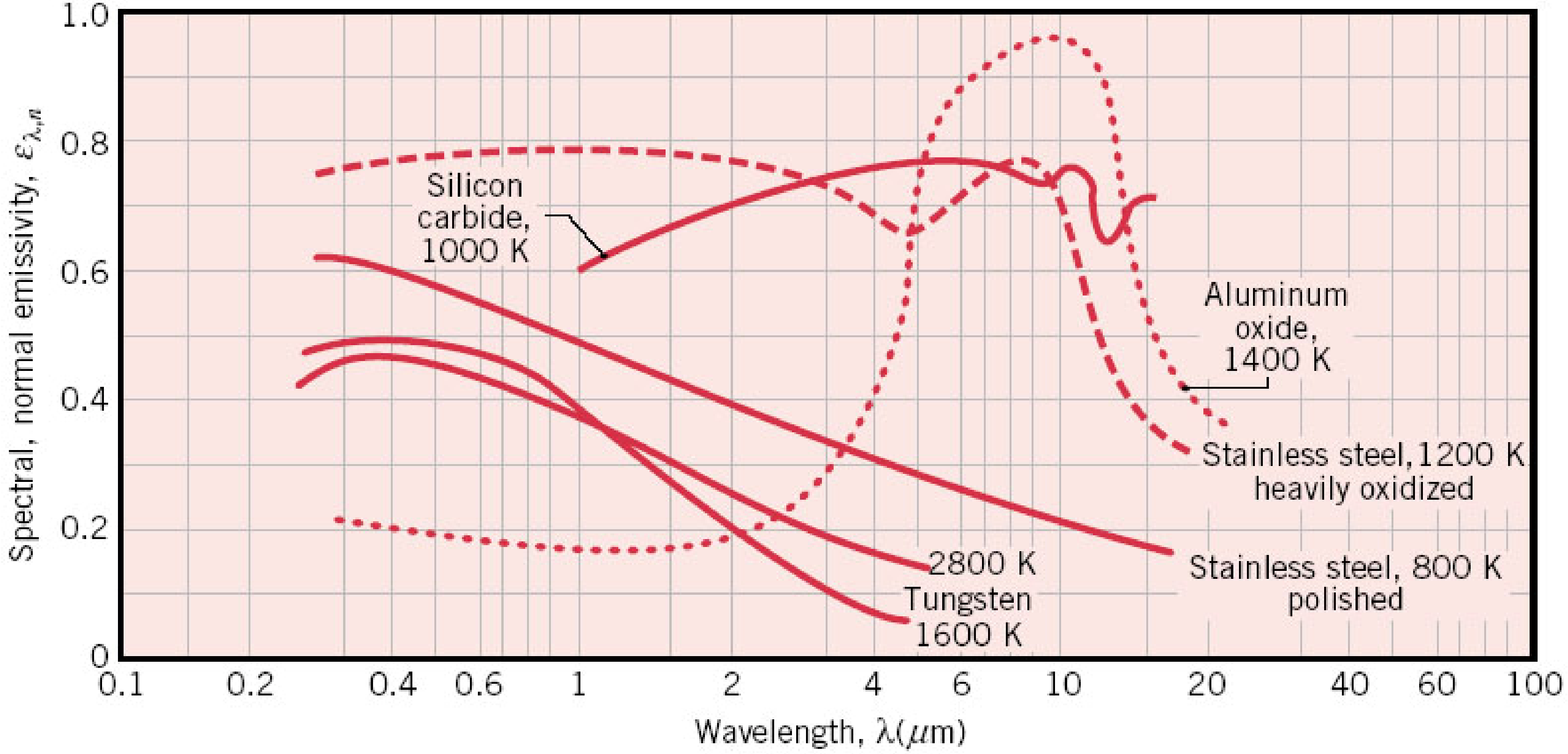


FIGURE 12.17 Spectral dependence of the spectral, normal emissivity $\varepsilon_{\lambda,n}$ of selected materials.

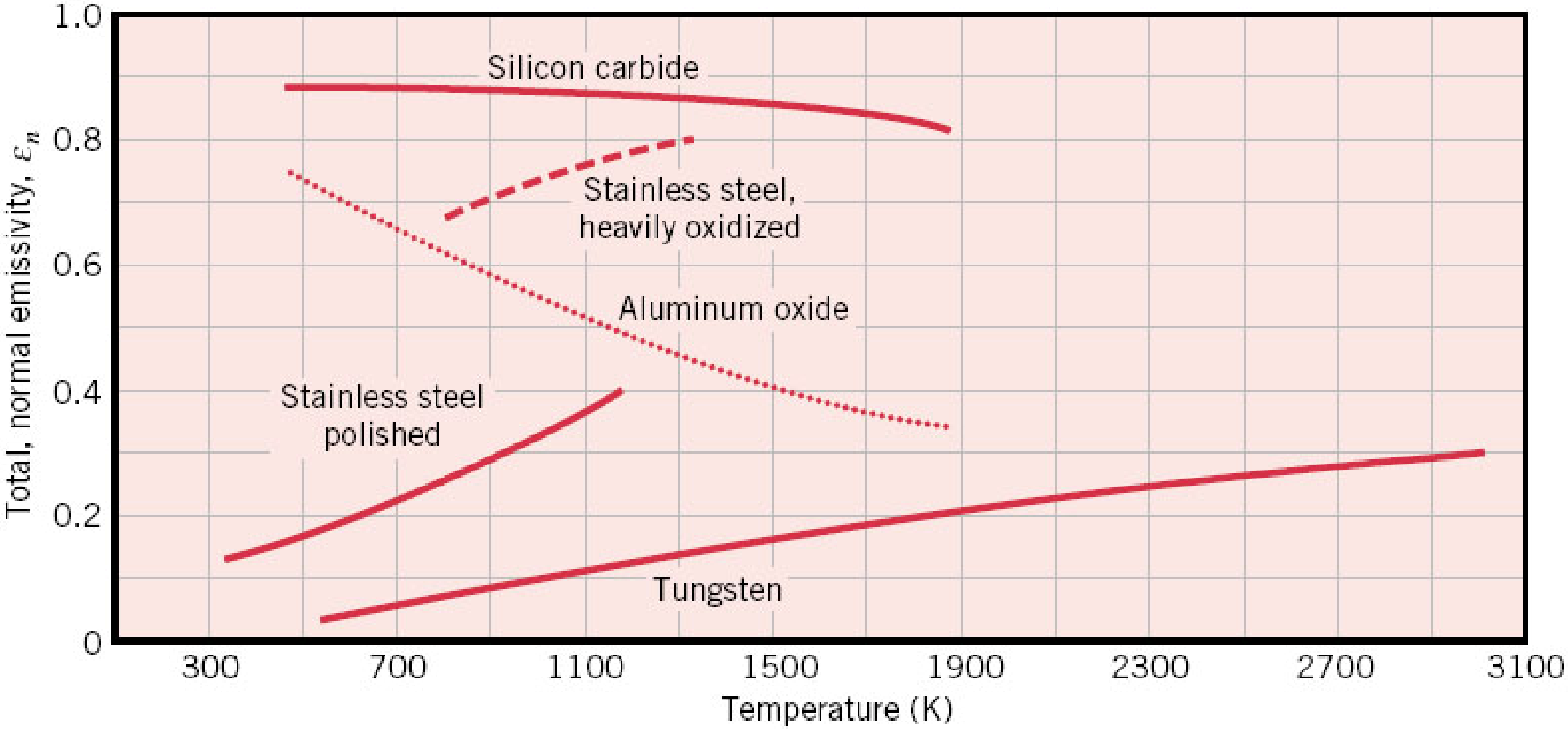
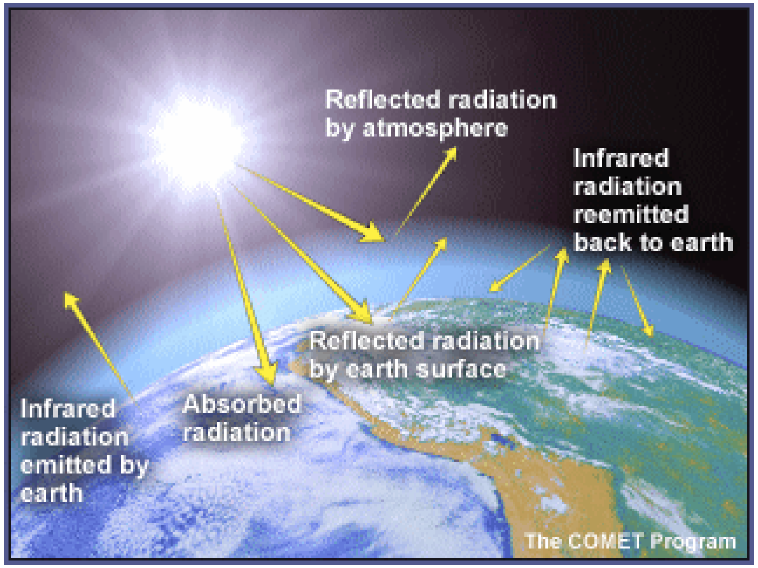


FIGURE 12.18 Temperature dependence of the total, normal emissivity ε_n of selected materials.



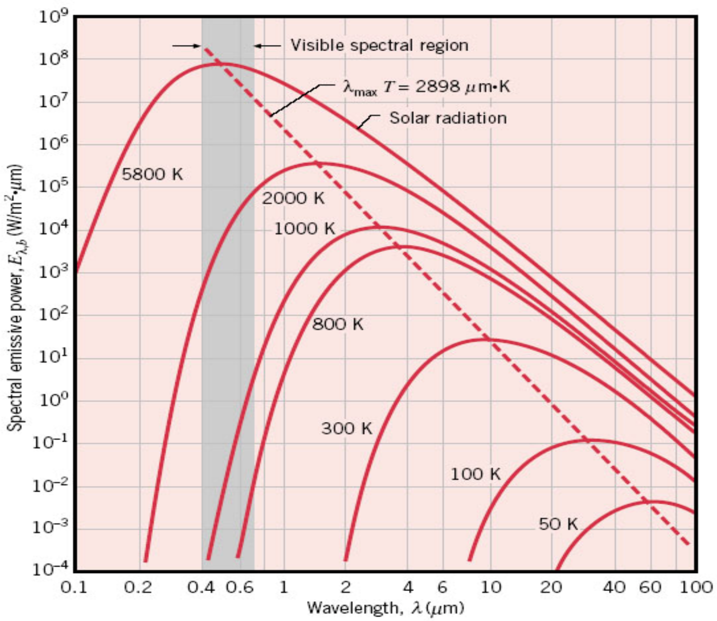


FIGURE 12.12 Spectral blackbody emissive power.

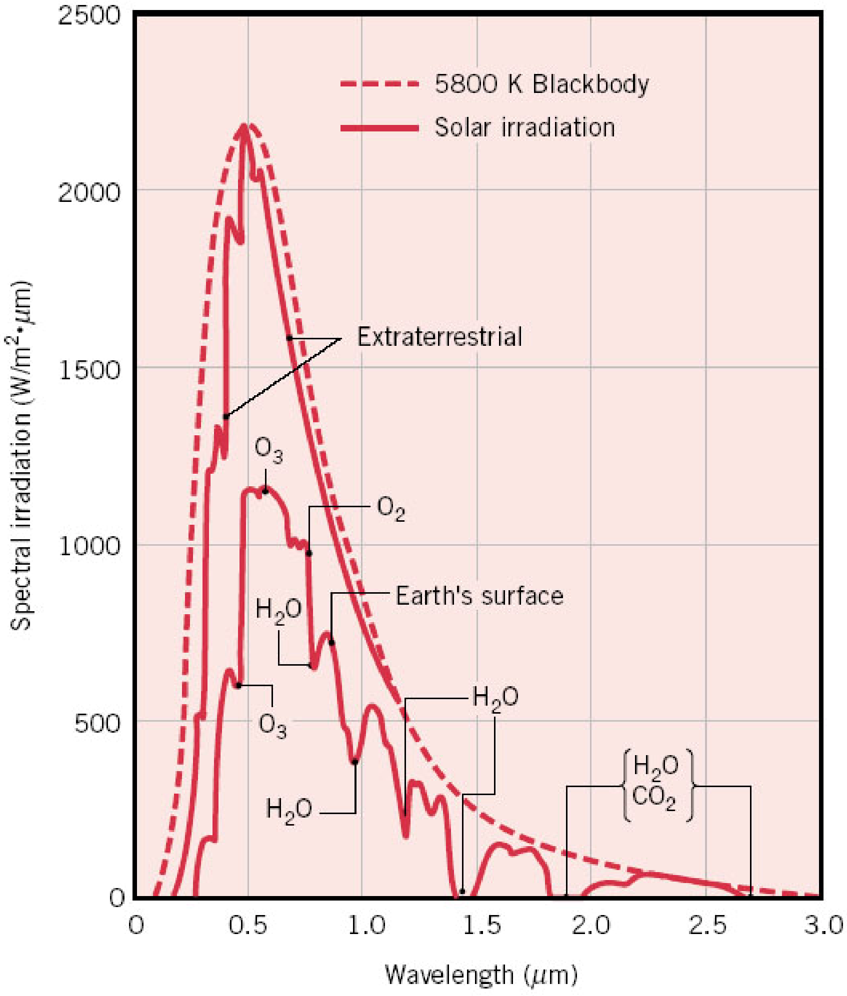


FIGURE 12.28 Spectral distribution of solar radiation.

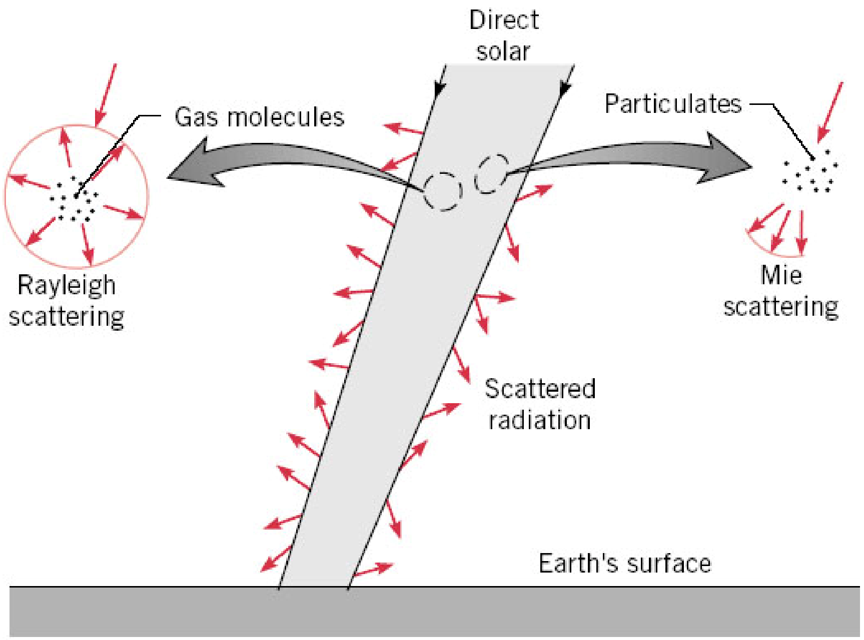


FIGURE 12.29 Scattering of solar radiation in the earth's atmosphere.

- Interaction of solar radiation with earth's atmosphere:
  - Absorption by aerosols over the entire spectrum.
  - Absorption by gases (CO_2, H_2O, O_3) in discrete wavelength bands.
  - Scattering by gas molecules and aerosols.

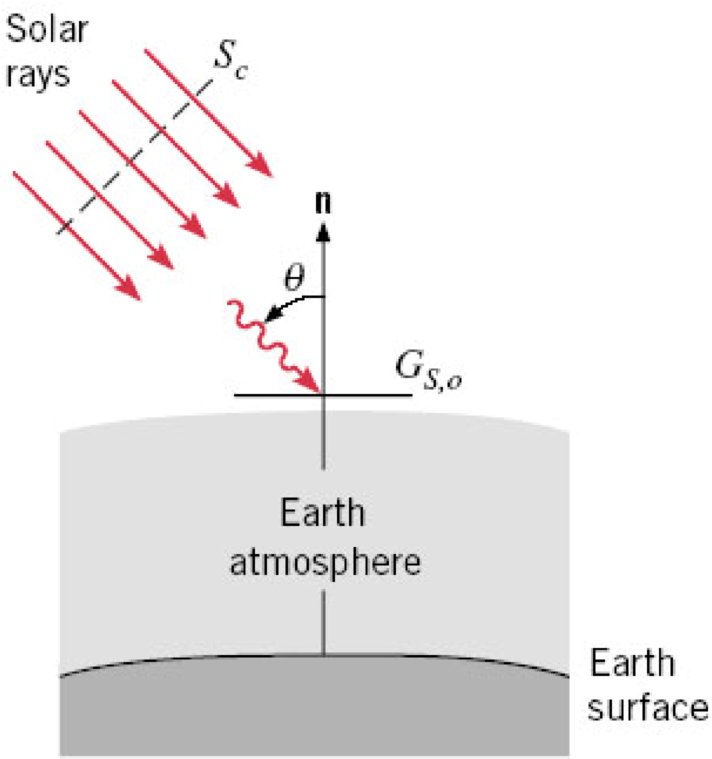
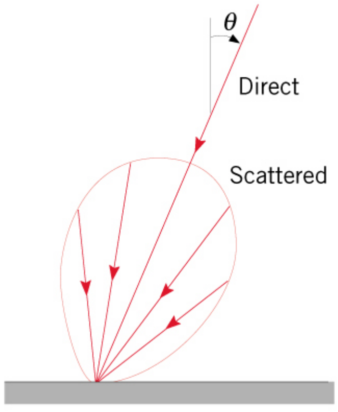


FIGURE 12.27
Directional nature of solar radiation outside the earth's atmosphere.

$S_c \rightarrow$ the solar constant or heat flux $\left(1353\ \mathrm{W/m^2}\right)$
when the earth is at its mean distance from the sun.

$$G_{S,o} = f \times S_c \times \cos\theta$$

- Effect of Atmosphere on Directional Distribution of Solar Radiation:
  - Rayleigh scattering is approximately uniform in all directions (isotropic scattering), while Mie scattering is primarily in the direction of the sun's rays (forward peaked).

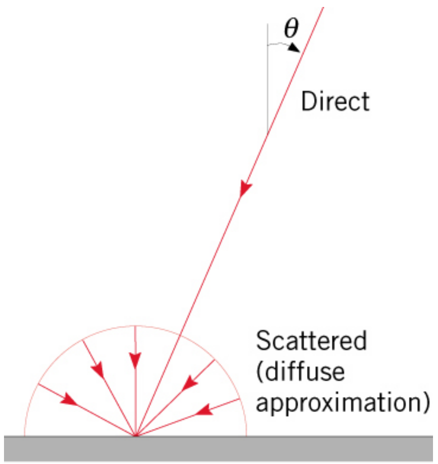


  - Directional distribution of radiation at the earth's surface has two components.
    - Direct radiation: Unscattered and in the direction θ of the sun's rays.
    - Diffuse radiation: Scattered radiation strongly peaked in the forward direction.
  - Calculation of solar irradiation for a horizontal surface often presumes the scattered component to be isotropic.

$$G_S = G_{S,dir} + G_{S,dif} = q''_{dir} \cos\theta + \pi I_{dif}$$

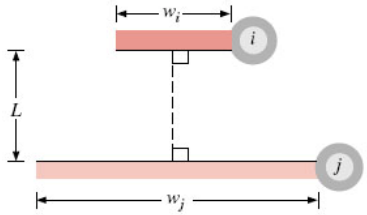
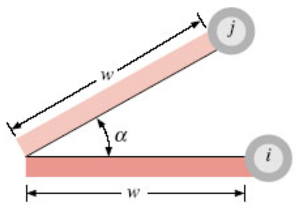
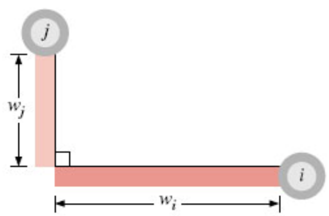
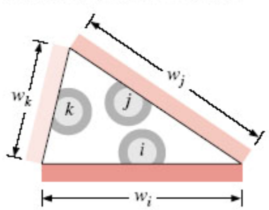
$$0.1 < \left(G_{S,dif} / G_S\right) < 1.0$$

Clear skies — Completely overcast



Chapter 13
Radiation Exchange Between Surfaces

TABLE 13.1 View Factors for Two-Dimensional Geometries [4]

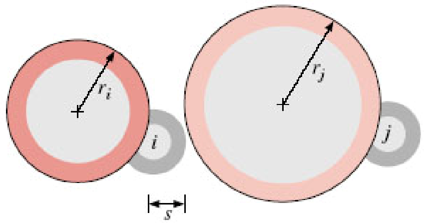
Geometry	Relation
Parallel Plates with Midlines Connected by Perpendicular 	$F_{ij} = \dfrac{[(W_i + W_j)^2 + 4]^{1/2} - [(W_j - W_i)^2 + 4]^{1/2}}{2W_i}$ $W_i = w_i/L,\ W_j = w_j/L$
Inclined Parallel Plates of Equal Width and a Common Edge 	$F_{ij} = 1 - \sin\left(\dfrac{\alpha}{2}\right)$
Perpendicular Plates with a Common Edge 	$F_{ij} = \dfrac{1 + (w_j/w_i) - [1 + (w_j/w_i)^2]^{1/2}}{2}$
Three-Sided Enclosure 	$F_{ij} = \dfrac{w_i + w_j - w_k}{2w_i}$

(continues)

TABLE 13.1 *Continued*

Geometry	Relation

Parallel Cylinders of Different Radii

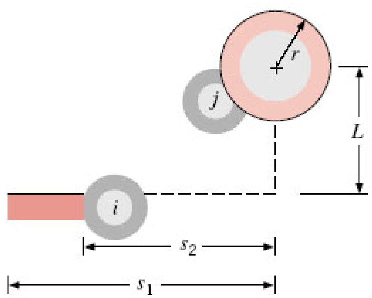


$$F_{ij} = \frac{1}{2\pi}\Big\{\pi + [C^2 - (R+1)^2]^{1/2} - [C^2 - (R-1)^2]^{1/2} + (R-1)\cos^{-1}\left[\left(\frac{R}{C}\right) - \left(\frac{1}{C}\right)\right] - (R+1)\cos^{-1}\left[\left(\frac{R}{C}\right) + \left(\frac{1}{C}\right)\right]\Big\}$$

$R = r_j/r_i,\ S = s/r_i$

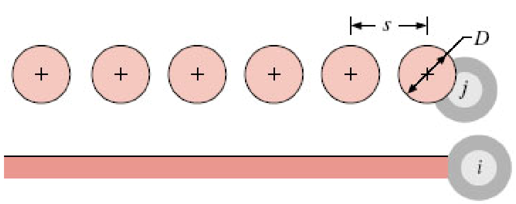
$C = 1 + R + S$

Cylinder and Parallel Rectangle



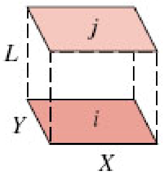
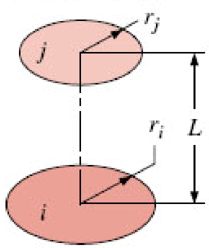
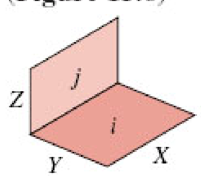
$$F_{ij} = \frac{r}{s_1 - s_2}\left[\tan^{-1}\frac{s_1}{L} - \tan^{-1}\frac{s_2}{L}\right]$$

Infinite Plane and Row of Cylinders



$$F_{ij} = 1 - \left[1 - \left(\frac{D}{s}\right)^2\right]^{1/2} + \left(\frac{D}{s}\right)\tan^{-1}\left[\left(\frac{s^2 - D^2}{D^2}\right)^{1/2}\right]$$

Table 13.2 View Factors for Three-Dimensional Geometries [4]

Geometry	Relation
Aligned Parallel Rectangles (Figure 13.4) 	$\bar{X} = X/L, \bar{Y} = Y/L$ $F_{ij} = \frac{2}{\pi \bar{X}\bar{Y}}\left\{ \ln\left[\frac{(1+\bar{X}^2)(1+\bar{Y}^2)}{1+\bar{X}^2+\bar{Y}^2}\right]^{1/2} + \bar{X}(1+\bar{Y}^2)^{1/2}\tan^{-1}\frac{\bar{X}}{(1+\bar{Y}^2)^{1/2}} + \bar{Y}(1+\bar{X}^2)^{1/2}\tan^{-1}\frac{\bar{Y}}{(1+\bar{X}^2)^{1/2}} - \bar{X}\tan^{-1}\bar{X} - \bar{Y}\tan^{-1}\bar{Y} \right\}$
Coaxial Parallel Disks (Figure 13.5) 	$R_i = r_i/L, R_j = r_j/L$ $S = 1 + \frac{1+R_j^2}{R_i^2}$ $F_{ij} = \frac{1}{2}\{S - [S^2 - 4(r_j/r_i)^2]^{1/2}\}$
Perpendicular Rectangles with a Common Edge (Figure 13.6) 	$H = Z/X, W = Y/X$ $F_{ij} = \frac{1}{\pi W}\left(W\tan^{-1}\frac{1}{W} + H\tan^{-1}\frac{1}{H} - (H^2+W^2)^{1/2}\tan^{-1}\frac{1}{(H^2+W^2)^{1/2}} + \frac{1}{4}\ln\left\{ \frac{(1+W^2)(1+H^2)}{1+W^2+H^2}\left[\frac{W^2(1+W^2+H^2)}{(1+W^2)(W^2+H^2)}\right]^{W^2} \times \left[\frac{H^2(1+H^2+W^2)}{(1+H^2)(H^2+W^2)}\right]^{H^2} \right\} \right)$

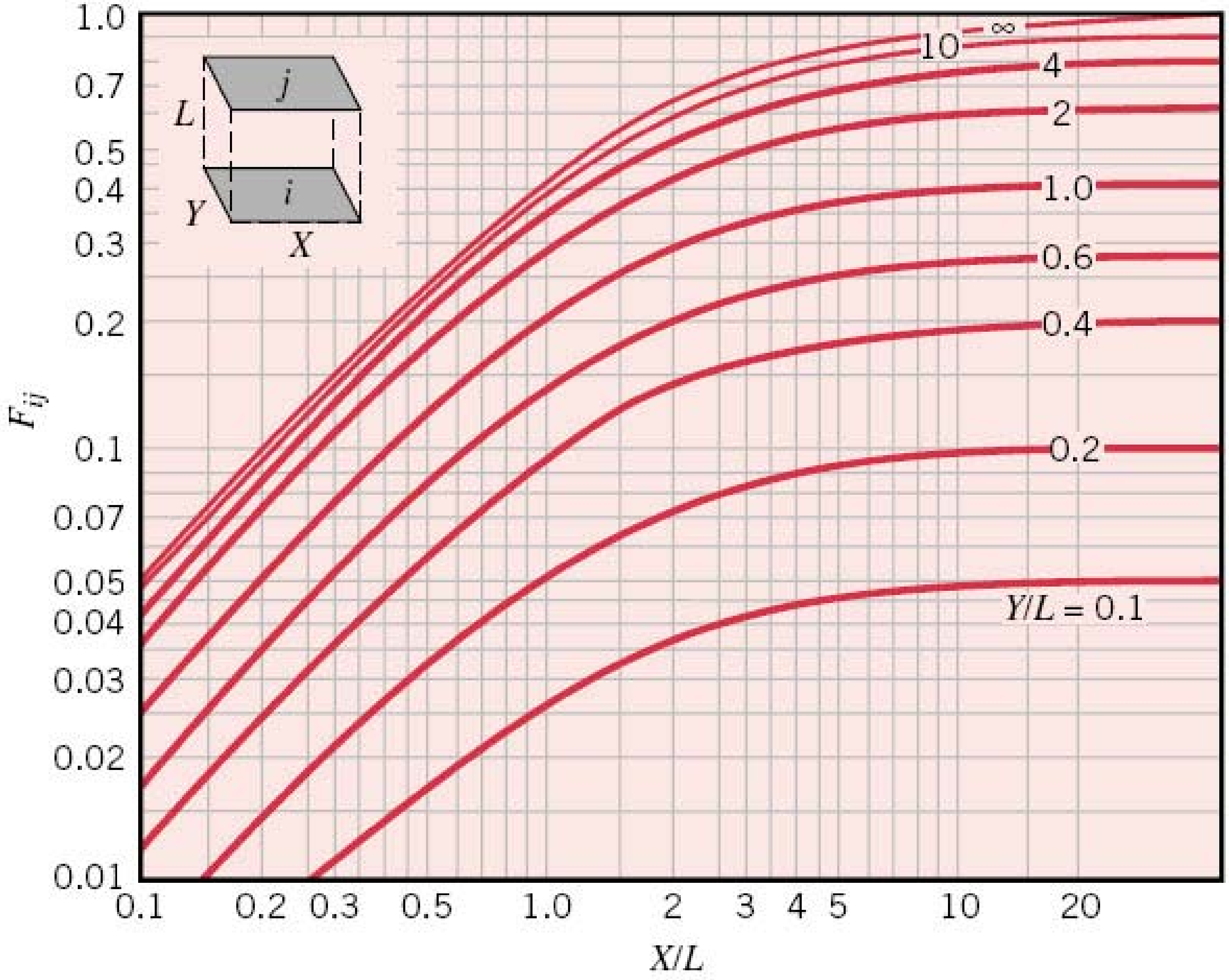


FIGURE 13.4 View factor for aligned parallel rectangles.

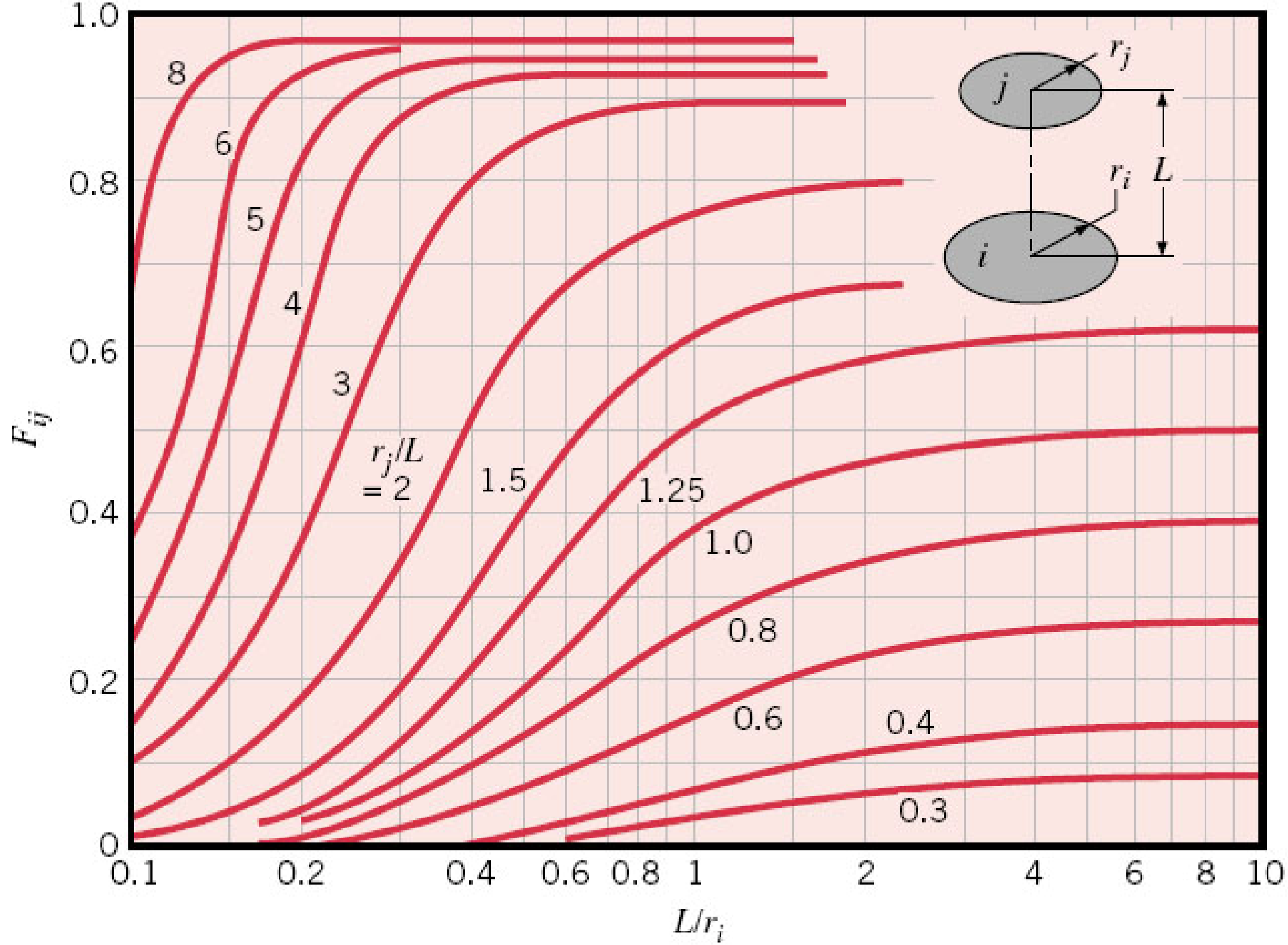


FIGURE 13.5 View factor for coaxial parallel disks.

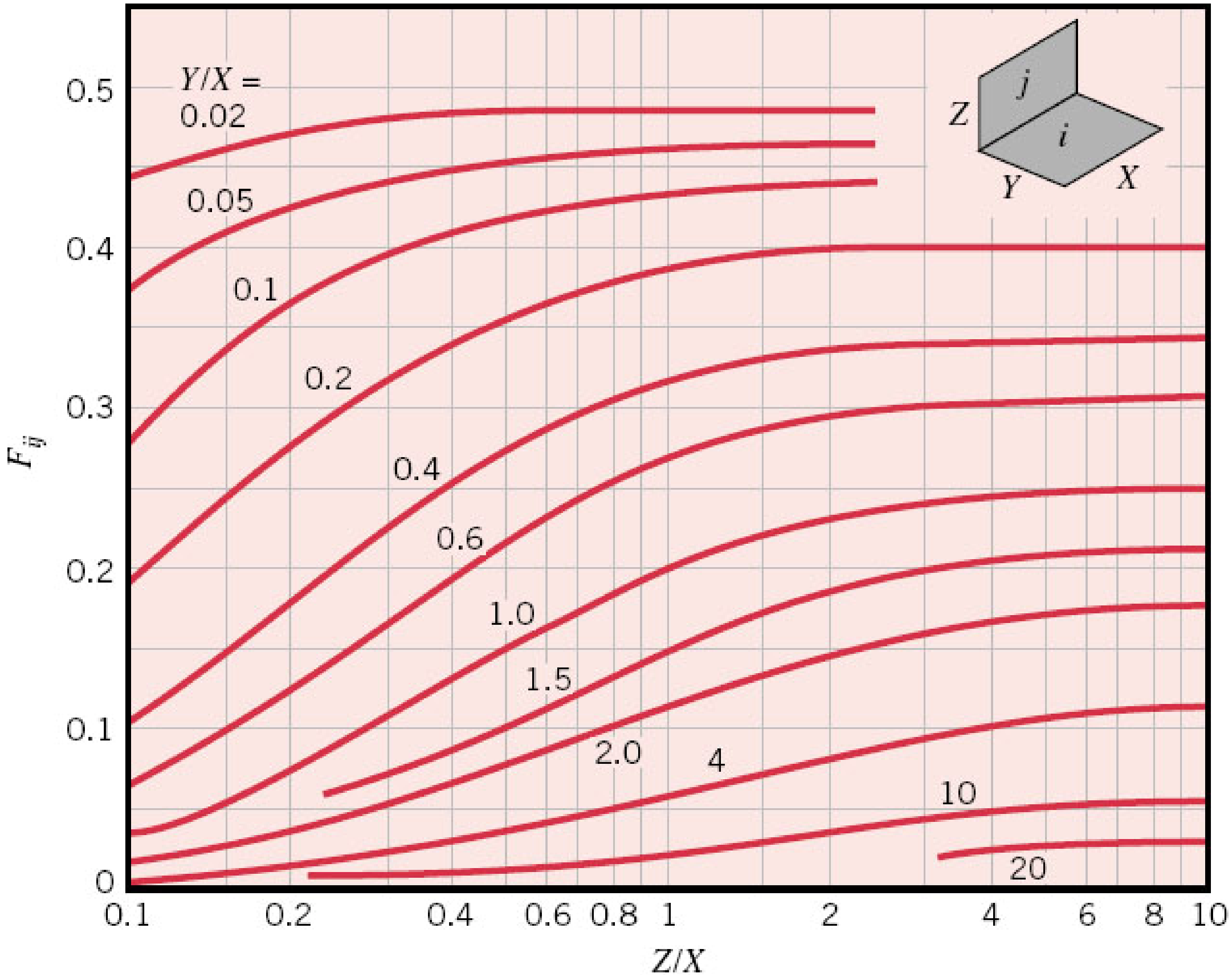
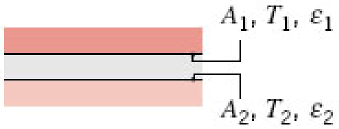


FIGURE 13.6 View factor for perpendicular rectangles with a common edge.

Table 13.3 Special Diffuse, Gray, Two-Surface Enclosures

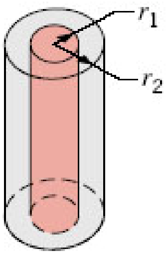
Large (Infinite) Parallel Planes



$A_1 = A_2 = A$
$F_{12} = 1$

$$q_{12} = \frac{A\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \tag{13.19}$$

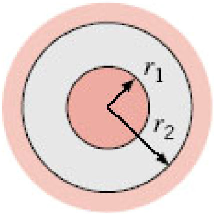
Long (Infinite) Concentric Cylinders



$\frac{A_1}{A_2} = \frac{r_1}{r_2}$
$F_{12} = 1$

$$q_{12} = \frac{\sigma A_1(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{r_1}{r_2}\right)} \tag{13.20}$$

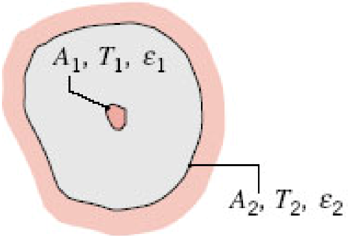
Concentric Spheres



$\frac{A_1}{A_2} = \frac{r_1^2}{r_2^2}$
$F_{12} = 1$

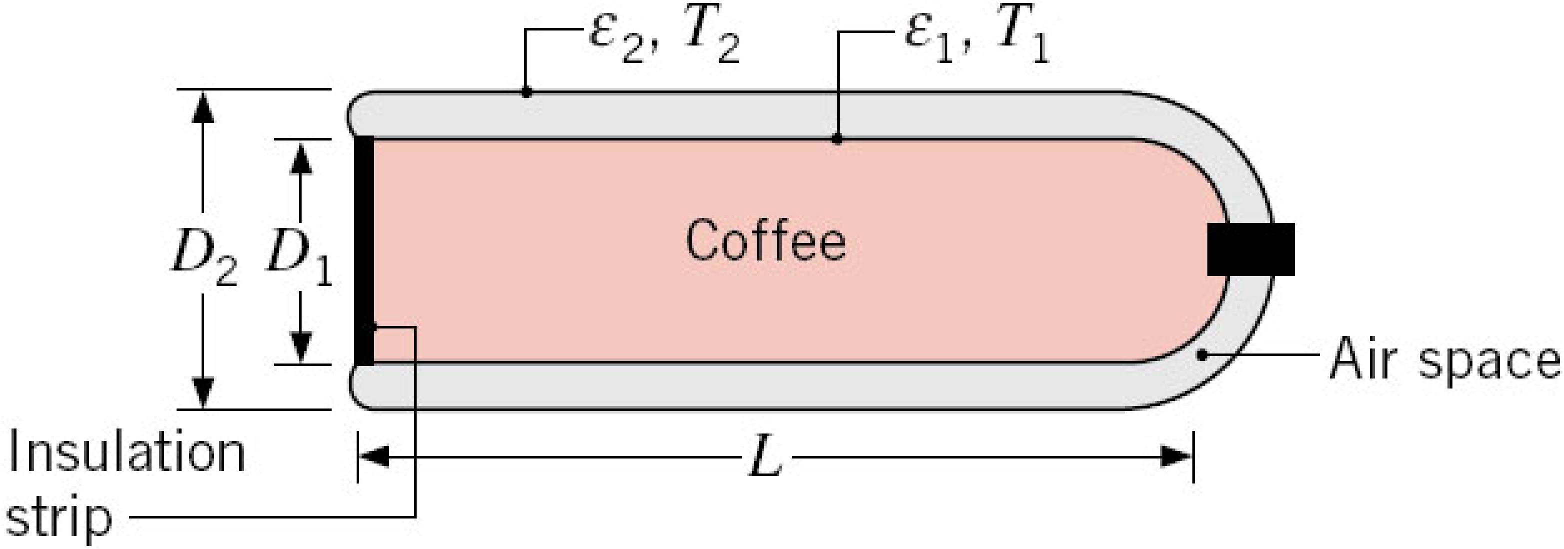
$$q_{12} = \frac{\sigma A_1(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{r_1}{r_2}\right)^2} \tag{13.21}$$

Small Convex Object in a Large Cavity



$\frac{A_1}{A_2} \approx 0$
$F_{12} = 1$

$$q_{12} = \sigma A_1 \varepsilon_1 (T_1^4 - T_2^4) \tag{13.22}$$



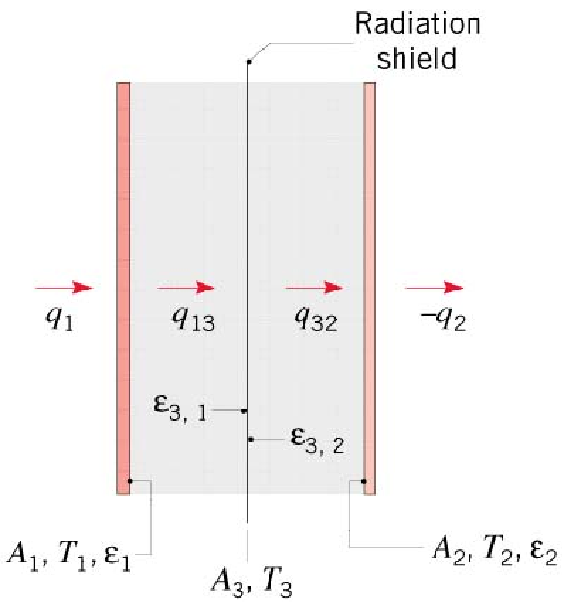
ε_2, T_2
ε_1, T_1
D_2
D_1
Coffee
Air space
Insulation strip
L

P. 13.106

Radiation Shields

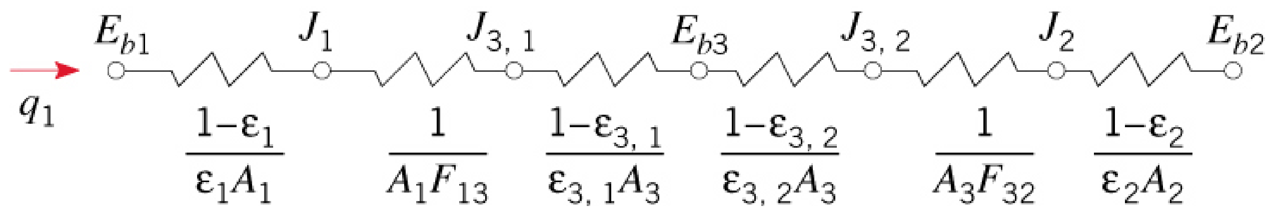
- High reflectivity (low $\alpha = \varepsilon$) surface(s) inserted between two surfaces for which a reduction in radiation exchange is desired.

- Consider use of a single shield in a two-surface enclosure, such as that associated with large parallel plates:



Note that, although rarely the case, emissivities may differ for opposite surfaces of the shield.

- Radiation Network:



$$q_{12} = q_1 = -q_2 = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1-\varepsilon_1}{\varepsilon_1 A_1} + \frac{1}{A_1 F_{13}} + \frac{1-\varepsilon_{3,1}}{\varepsilon_{3,1} A_3} + \frac{1-\varepsilon_{3,2}}{\varepsilon_{3,2} A_3} + \frac{1}{A_3 F_{32}} + \frac{1-\varepsilon_2}{\varepsilon_2 A_2}}$$

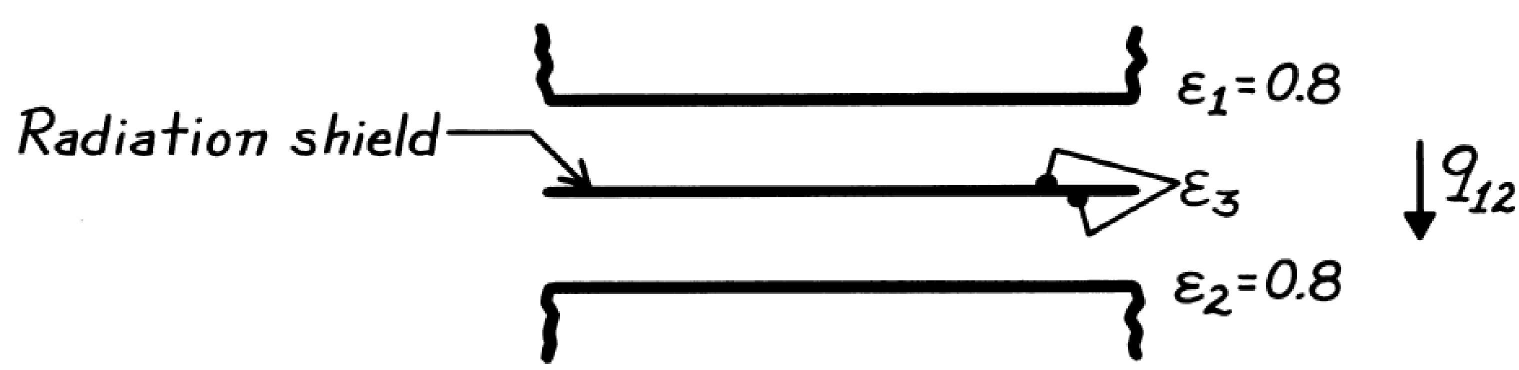
- The foregoing result may be readily extended to account for multiple shields and may be applied to long, concentric cylinders and concentric spheres, as well as large parallel plates.

PROBLEM 13.56

KNOWN: Emissivities of two large, parallel surfaces.

FIND: Heat shield emissivity needed to reduce radiation transfer by a factor of 10.

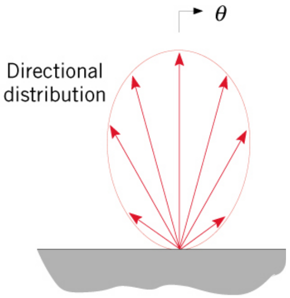
SCHEMATIC:



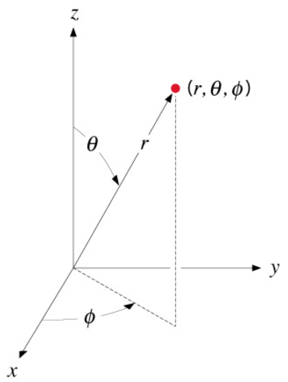
ASSUMPTIONS: (a) Diffuse-gray surface behavior, (b) Negligible conduction resistance for shield, (c) Same emissivity on opposite sides of shield.

Directional Considerations and the Concept of Radiation Intensity

- Radiation emitted by a surface will be in all directions associated with a hypothetical hemisphere about the surface and is characterized by a directional distribution.

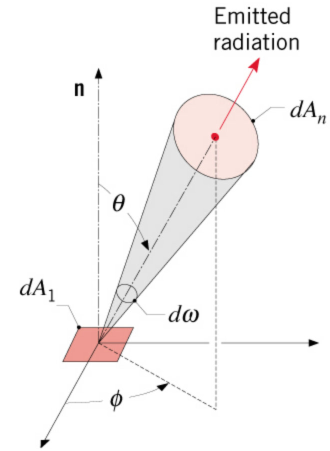


- Direction may be represented in a spherical coordinate system characterized by the zenith or polar angle θ and the azimuthal angle ϕ .



- The amount of radiation emitted from a surface, dA_1, and propagating in a particular direction, θ, ϕ, is quantified in terms of a differential solid angle associated with the direction.

$$d\omega \equiv \frac{dA_n}{r^2}$$



$dA_n \rightarrow$ unit element of surface on a hypothetical sphere and normal to the θ, ϕ direction.

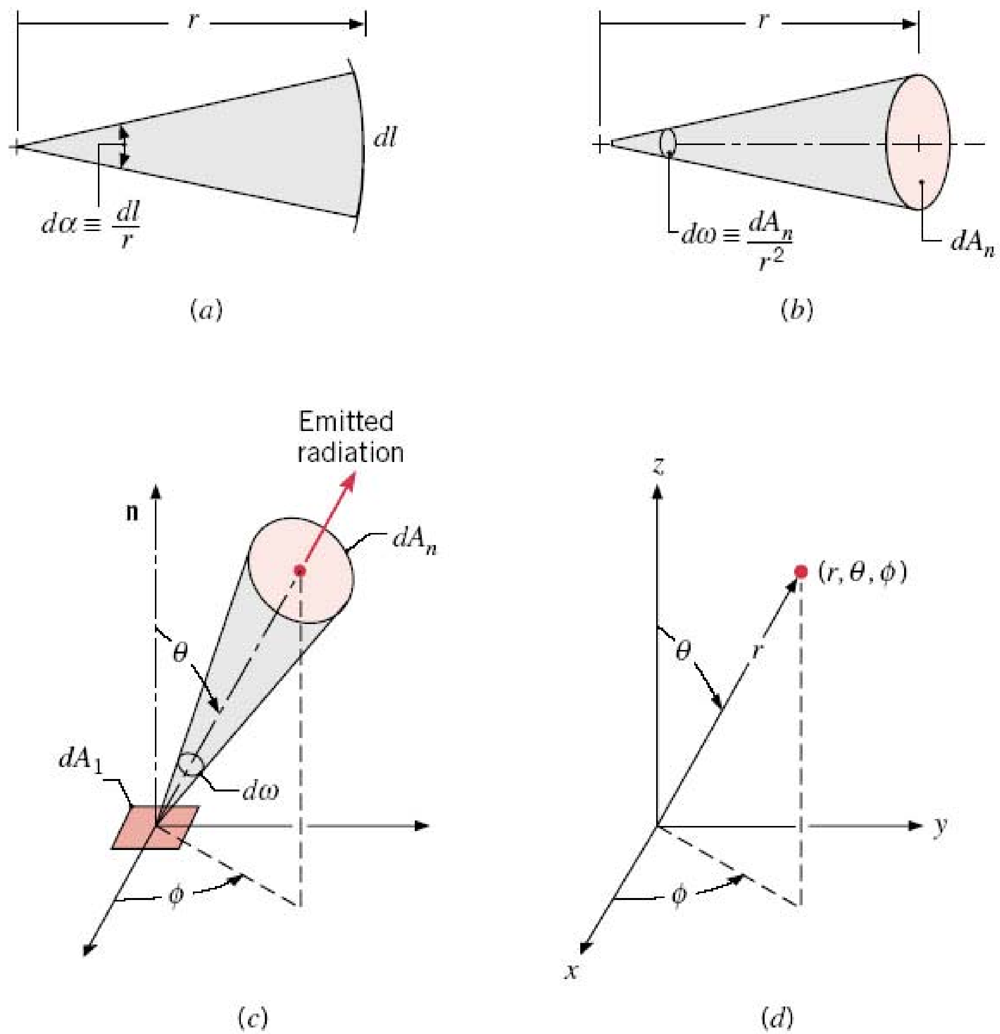
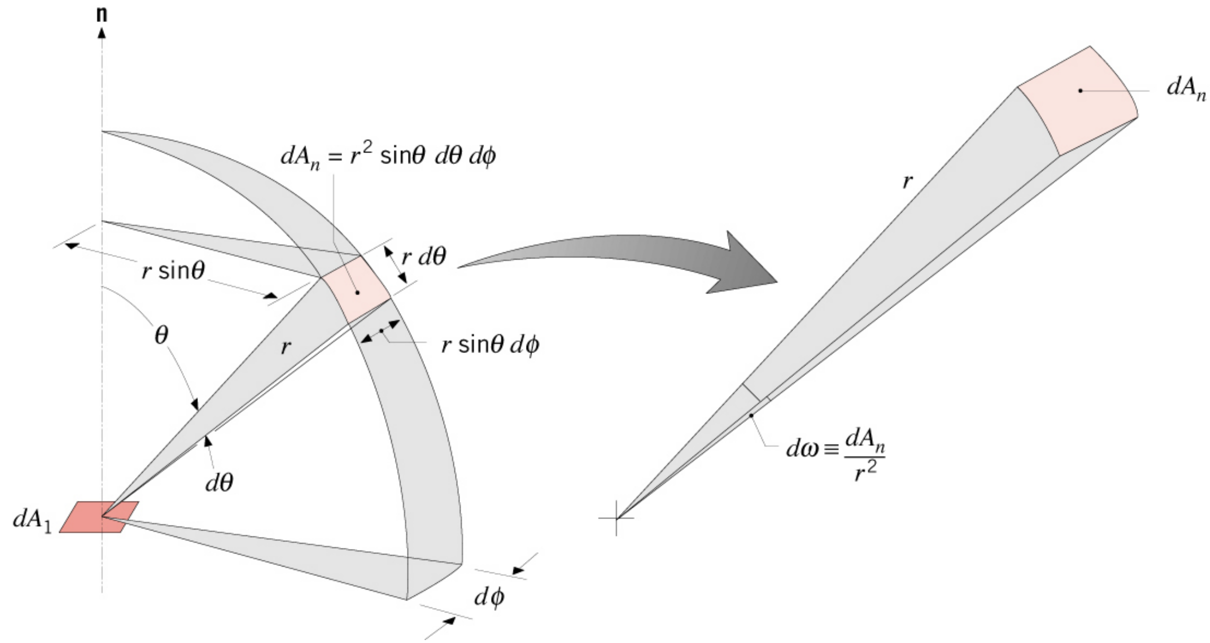


FIGURE 12.5 Mathematical definitions. (*a*) Plane angle. (*b*) Solid angle. (*c*) Emission of radiation from a differential area dA_1 into a solid angle $d\omega$ subtended by dA_n at a point on dA_1. (*d*) The spherical coordinate system.



$$dA_n = r^2 \sin\theta d\theta d\phi$$

$$d\omega = \frac{dA_n}{r^2} = \sin\theta d\theta d\phi$$

– The solid angle ω has units of steradians (sr).

– The solid angle associated with a complete hemisphere is

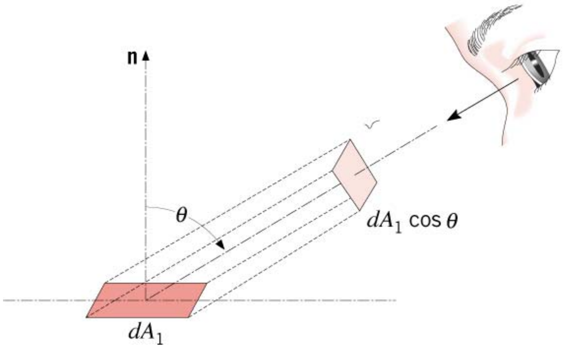
$$\omega_{hemi} = \int_0^{2\pi} \int_0^{\pi/2} \sin\theta d\theta d\phi = 2\pi \text{sr}$$

• Spectral Intensity: A quantity used to specify the radiant heat flux $\left(\text{W/m}^2\right)$ within a unit solid angle about a prescribed direction $\left(\text{W/m}^2 \cdot \text{sr}\right)$ and within a unit wavelength interval about a prescribed wavelength $\left(\text{W/m}^2 \cdot \text{sr} \cdot \mu\text{m}\right)$.

- The spectral intensity $I_{\lambda,e}$ associated with emission from a surface element in the solid angle $d\omega$ about θ,ϕ and the wavelength interval $d\lambda$ about is defined as:

$$I_{\lambda,e}(\lambda,\theta,\phi) \equiv \frac{dq}{(dA_1 \cos\theta)\cdot d\omega \cdot d\lambda}$$

- The rationale for defining the radiation flux in terms of the projected surface area $(dA_1 \cos\theta)$ stems from the existence of surfaces for which, to a good approximation, $I_{\lambda,e}$ is independent of direction. Such surfaces are termed diffuse, and the radiation is said to be isotropic.



  - The projected area is how dA_1 would appear if observed along θ,ϕ .
    - What is the projected area for $\theta = 0$?
    - What is the projected area for $\theta = \pi/2$?

- The spectral heat rate and heat flux associated with emission from dA_1 are, respectively,

$$dq_\lambda \equiv \frac{dq}{d\lambda} = I_{\lambda,e}(\lambda,\theta,\phi)\, dA_1 \cos\theta\, d\omega$$

$$dq''_\lambda = I_{\lambda,e}(\lambda,\theta,\phi)\cos\theta\, d\omega = I_{\lambda,e}(\lambda,\theta,\phi)\cos\theta \sin\theta\, d\theta\, d\phi$$

Relation of Intensity to Emissive Power, Irradiation, and Radiosity

- The spectral emissive power $\left(\mathrm{W/m^2 \cdot \mu m}\right)$ corresponds to spectral emission over all possible directions.

$$E_\lambda(\lambda) = \int_0^{2\pi} \int_0^{\pi/2} I_{\lambda,e}(\lambda,\theta,\phi) \cos\theta \sin\theta d\theta d\phi$$

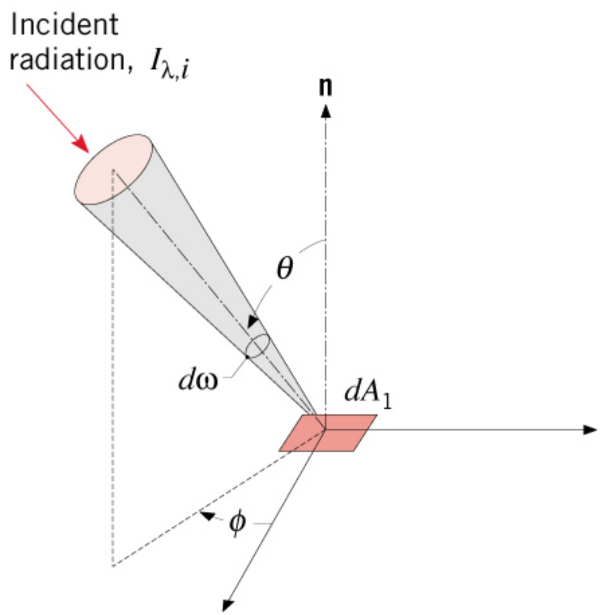
- The total emissive power $\left(\mathrm{W/m^2}\right)$ corresponds to emission over all directions and wavelengths.

$$E = \int_0^\infty E_\lambda(\lambda) d\lambda$$

- For a diffuse surface, emission is isotropic and

$$E_\lambda(\lambda) = \pi I_{\lambda,e}(\lambda) \qquad E = \pi I_e$$

- The spectral intensity of radiation incident on a surface, $I_{\lambda,i}$, is defined in terms of the unit solid angle about the direction of incidence, the wavelength interval $d\lambda$ about λ, and the projected area of the receiving surface, $dA_1 \cos\theta$.



- The spectral irradiation $\left(\mathrm{W/m^2 \cdot \mu m}\right)$ is then:

$$G_\lambda(\lambda) = \int_0^{2\pi} \int_0^{\pi/2} I_{\lambda,i}(\lambda,\theta,\phi) \cos\theta \sin\theta d\theta d\phi$$

and the total irradiation $\left(\mathrm{W/m^2}\right)$ is

$$G = \int_0^\infty G_\lambda(\lambda) d\lambda$$

- With $I_{\lambda,e+r}$ designating the spectral intensity associated with radiation emitted by the surface and the reflection of incident radiation, the spectral radiosity $\left(\mathrm{W/m^2 \cdot \mu m}\right)$ is:

$$J_\lambda(\lambda) = \int_0^{2\pi} \int_0^{\pi/2} I_{\lambda,e+r}(\lambda,\theta,\phi) \cos\theta \sin\theta d\theta d\phi$$

and the total radiosity $\left(\mathrm{W/m^2}\right)$ is

$$J = \int_0^\infty J_\lambda(\lambda) d\lambda$$

The View Factor (also Configuration or Shape Factor)

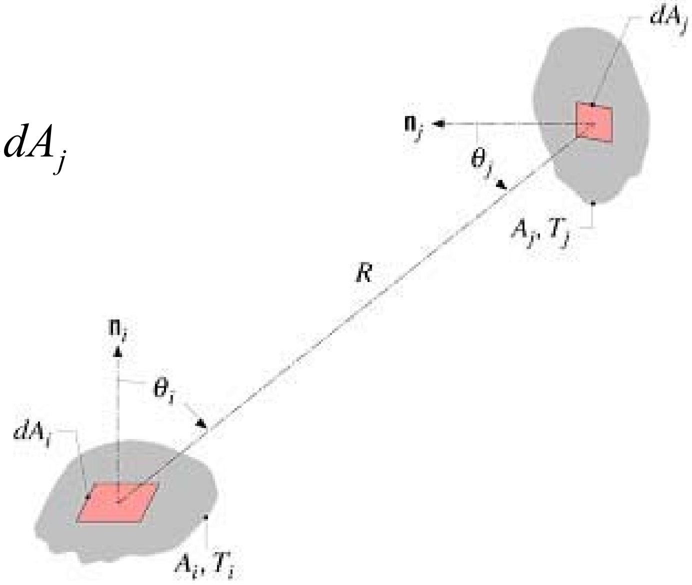
- The view factor, F_{ij}, is a geometrical quantity corresponding to the fraction of the radiation leaving surface *i* that is intercepted by surface *j*.

$$F_{ij} = \frac{q_{i\to j}}{A_i J_i}$$

- The view factor integral provides a general expression for F_{ij}. Consider exchange between areas dA_i *and* dA_j :

$$dq_{i\to j} = I_i \cos\theta_i dA_i d\omega_{j-i} = J_i \frac{\cos\theta_i \cos\theta_j}{\pi R^2} dA_i dA_j$$

$$F_{ij} = \frac{1}{A_i}\int_{A_i}\int_{A_j}\frac{\cos\theta_i \cos\theta_j}{\pi R^2} dA_i dA_j$$

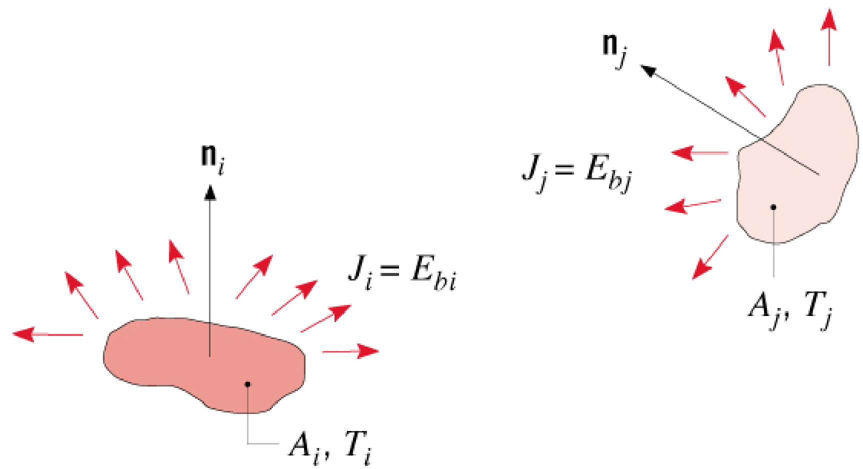


Blackbody Radiation Exchange

- For a blackbody, $J_i = E_{bi}$.

- Net radiative exchange between two surfaces that can be approximated as blackbodies $\longrightarrow$ *net* rate at which radiation leaves surface *i* due to its interaction with *j*

  or net rate at which surface *j* gains radiation due to its interaction with *i*



$$q_{ij} = q_{i\to j} - q_{j\to i}$$

$$q_{ij} = A_i F_{ij} E_{bi} - A_j F_{ji} E_{bj}$$

$$q_{ij} = A_i F_{ij} \sigma \left(T_i^4 - T_j^4 \right)$$

- Net radiation transfer from surface *i* due to exchange with all (N) surfaces of an enclosure:

$$q_i = \sum_{j=1}^{N} A_i F_{ij} \sigma \left(T_i^4 - T_j^4 \right)$$

General Radiation Analysis for Exchange between the *N* Opaque, Diffuse, Gray Surfaces of an Enclosure

$$\left(\varepsilon_i = \alpha_i = 1-\rho_i\right)$$

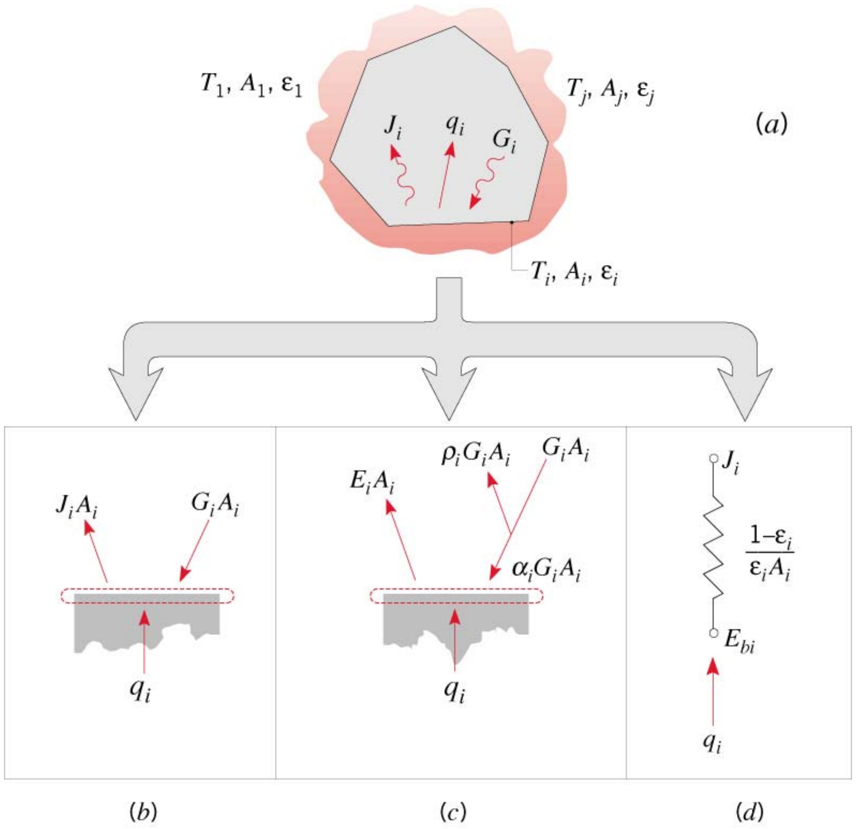
- Alternative expressions for net radiative transfer from surface *i*:

$$q_i = A_i\left(J_i - G_i\right) \rightarrow \text{ Fig. (b)} \qquad (1)$$

$$q_i = A_i\left(E_i - \alpha_i G_i\right) \rightarrow \text{ Fig. (c)} \qquad (2)$$

$$q_i = \frac{E_{bi} - J_i}{\left(1-\varepsilon_i\right)/\varepsilon_i A_i} \rightarrow \text{ Fig. (d)} \qquad (3)$$

Suggests a surface radiative resistance of the form: $\left(1-\varepsilon_i\right)/\varepsilon_i A_i$



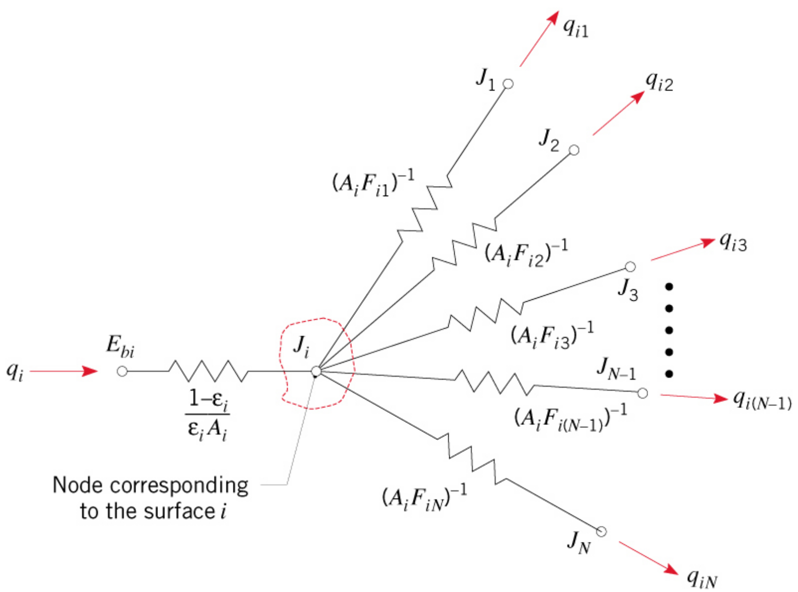
$$q_i = \sum_{j=1}^{N} A_i F_{ij} \left(J_i - J_j \right) = \sum_{j=1}^{N} \frac{J_i - J_j}{\left(A_i F_{ij} \right)^{-1}} \qquad (4)$$

Suggests a space or geometrical resistance of the form: $\left(A_i F_{ij} \right)^{-1}$

- Equating Eqs. (3) and (4) corresponds to a radiation balance on surface *i*:

$$\frac{E_{bi} - J_i}{\left(1 - \varepsilon_i\right) / \varepsilon_i A_i} = \sum_{j=1}^{N} \frac{J_i - J_j}{\left(A_i F_{ij} \right)^{-1}} \qquad (5)$$

which may be represented by a radiation network of the form



- Methodology of an Enclosure Analysis

  - Apply Eq. (4) to each surface for which the net radiation heat rate q_i is known.

  - Apply Eq. (5) to each of the remaining surfaces for which the temperature T_i, and hence E_{bi}, is known.

  - Evaluate all of the view factors appearing in the resulting equations.

  - Solve the system of N equations for the unknown radiosities, $J_1, J_2, ..., J_N$.

  - Use Eq. (3) to determine q_i for each surface of known T_i and T_i for each surface of known q_i.

- Treatment of the virtual surface corresponding to an opening (aperture) of area A_i, through which the interior surfaces of an enclosure exchange radiation with large surroundings at T_{sur}:

  - Approximate the opening as blackbody of area, A_i, temperature, $T_i = T_{sur}$, and properties, $\varepsilon_i = \alpha_i = 1$.

HEISLER CHARTS

1. Plane Wall

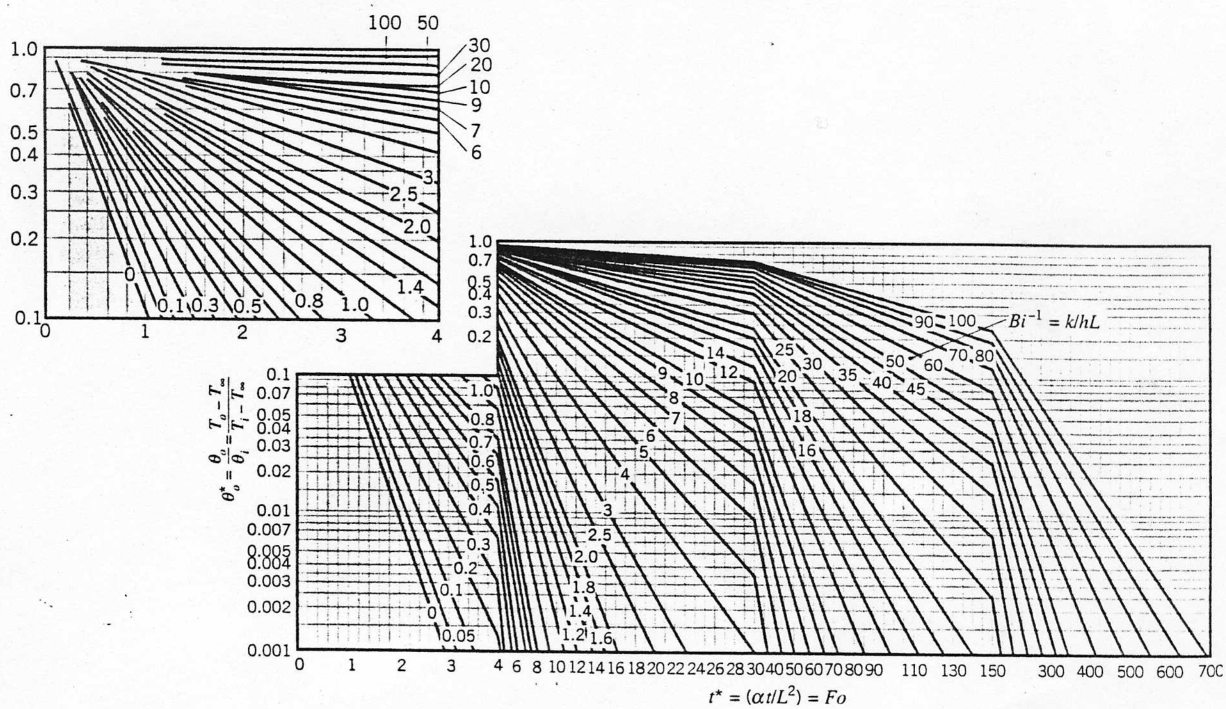


FIGURE D.1 Midplane temperature as a function of time for a plane wall of thickness $2L$ [1]. Used with permission.

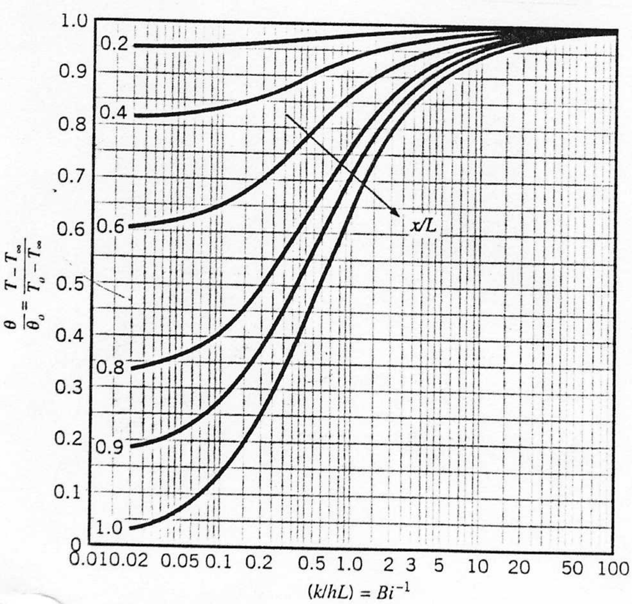
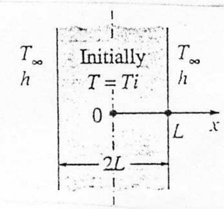


FIGURE D.2 Temperature distribution in a plane wall of thickness $2L$ [1]. Used with permission.



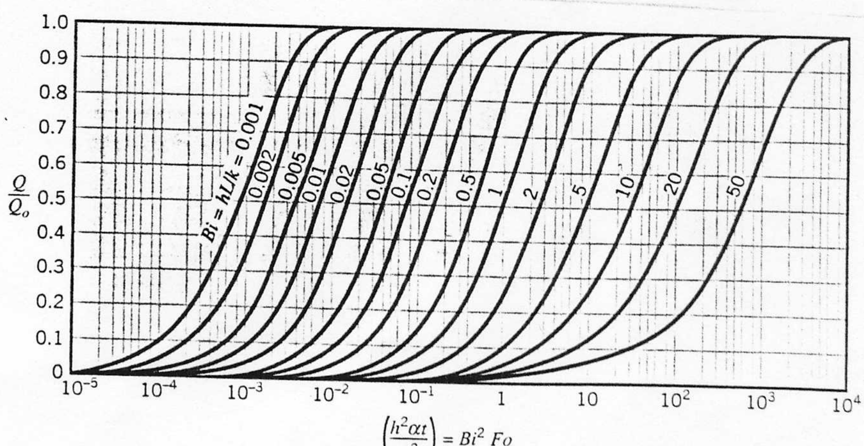


FIGURE D.3 Internal energy change as a function of time for a plane wall of thickness $2L$ [2]. Adapted with permission.

2. Long Cylinder

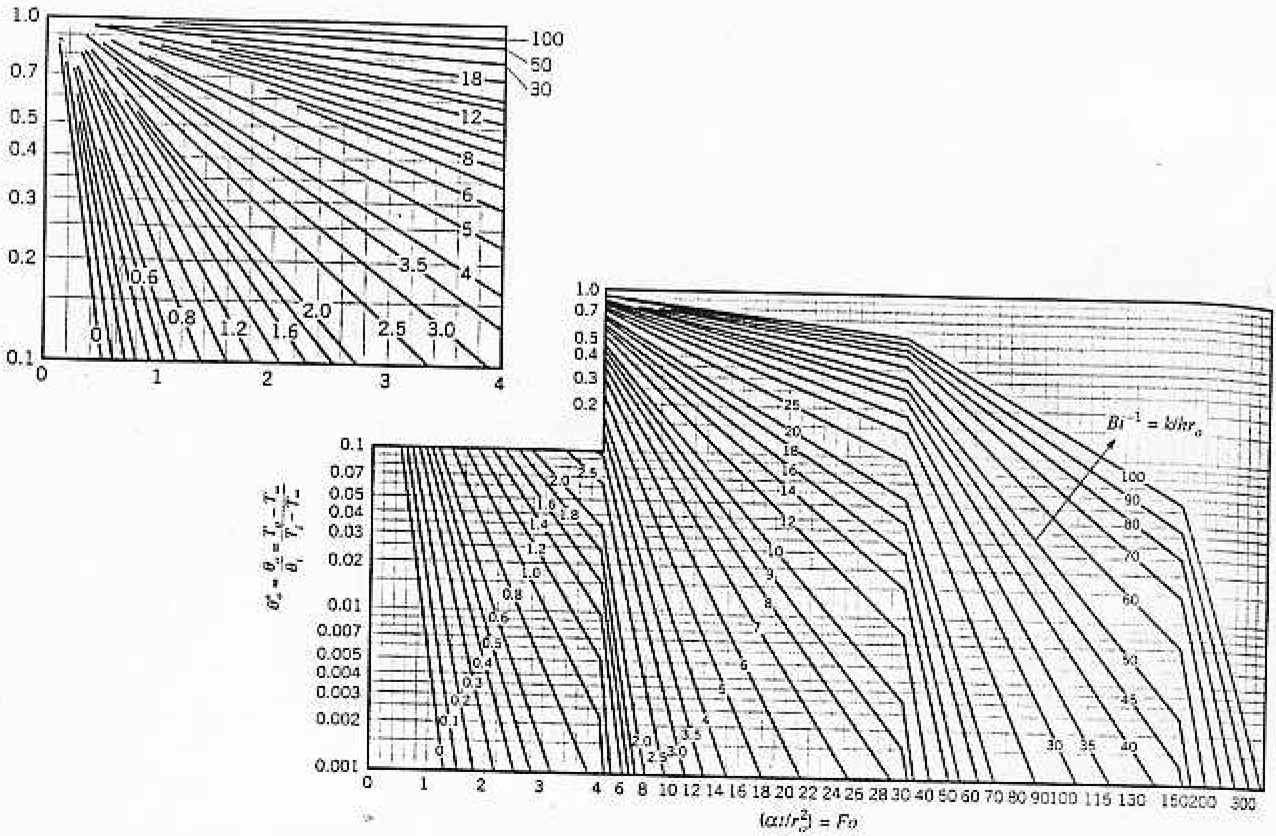


FIGURE D.4 Centerline temperature as a function of time for an infinite cylinder of radius r_o [1]. Used with permission.

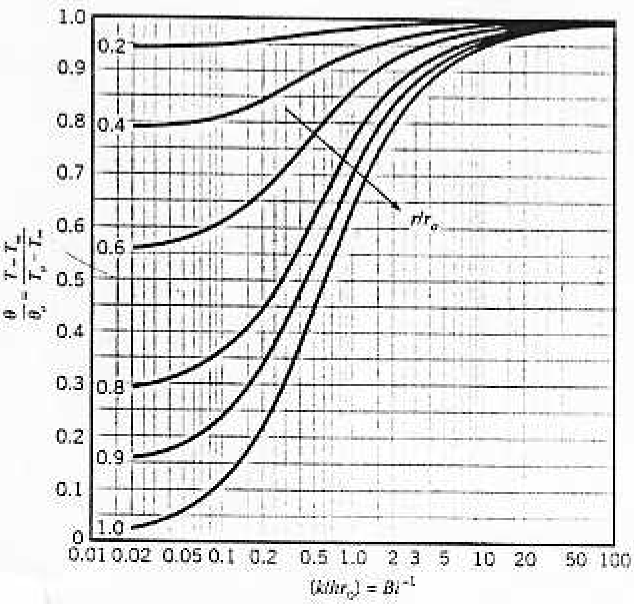


FIGURE D.5 Temperature distribution in an infinite cylinder of radius r_o [1]. Used with permission.

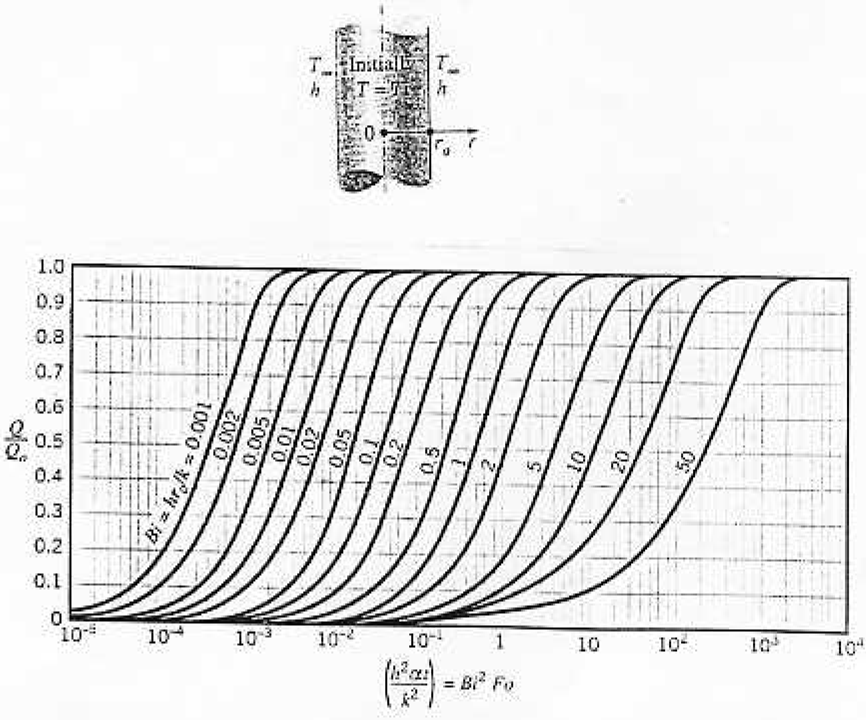


FIGURE D.6 Internal energy change as a function of time for an infinite cylinder of radius r_o [2]. Adapted with permission.

3. Sphere

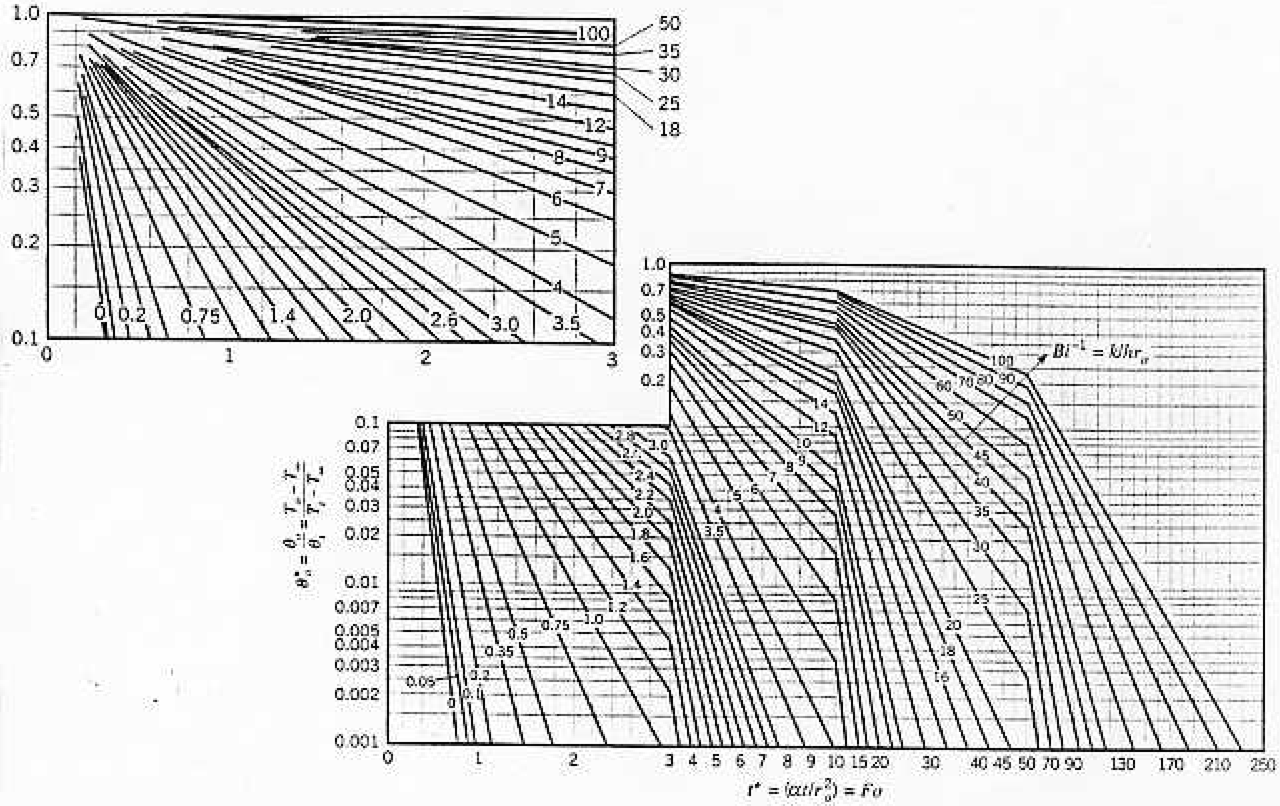
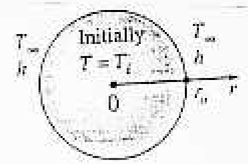


FIGURE D.7 Center temperature as a function of time in a sphere of radius r_o [1]. Used with permission.



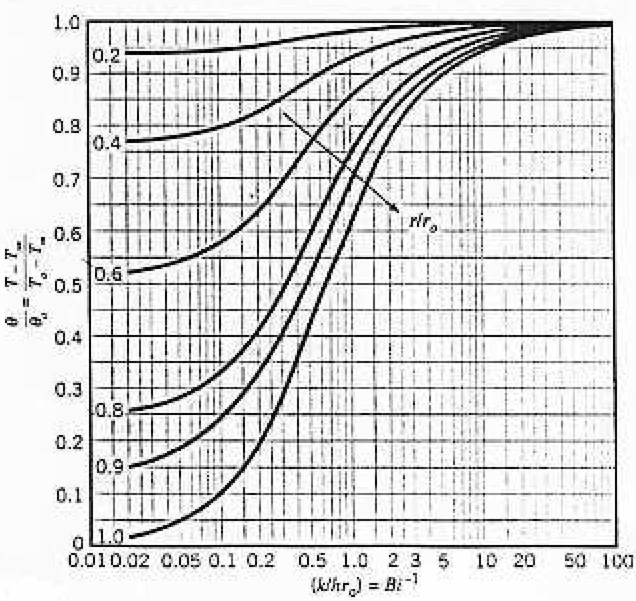


FIGURE D.8 Temperature distribution in a sphere of radius r_o [1]. Used with permission.

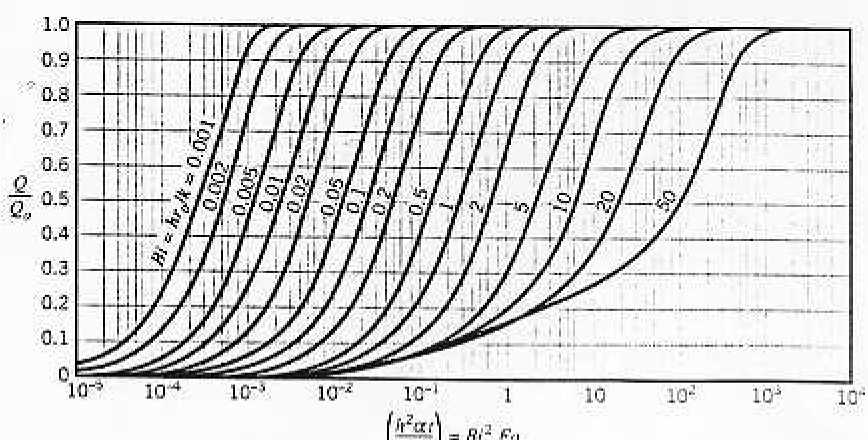


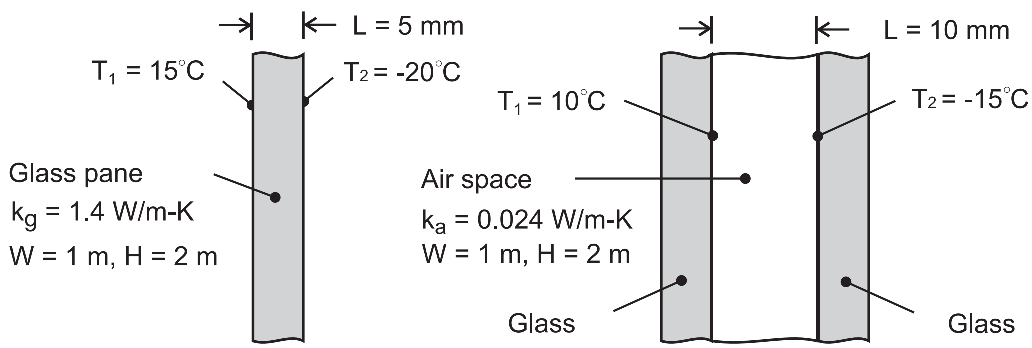
FIGURE D.9 Internal energy change as a function of time for a sphere of radius r_o [2]. Adapted with permission.

PROBLEM 1.6

KNOWN: Width, height, thickness and thermal conductivity of a single pane window and the air space of a double pane window. Representative winter surface temperatures of single pane and air space.

FIND: Heat loss through single and double pane windows.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional conduction through glass or air, (2) Steady-state conditions, (3) Enclosed air of double pane window is stagnant (negligible buoyancy induced motion).

ANALYSIS: From Fourier's law, the heat losses are

Single Pane: $$q_g = k_g A \frac{T_1 - T_2}{L} = 1.4\ \text{W/m}\cdot\text{K}\left(2\text{m}^2\right)\frac{35\ {}^\circ\text{C}}{0.005\text{m}} = 19,600\ \text{W}$$ <

Double Pane: $$q_a = k_a A \frac{T_1 - T_2}{L} = 0.024\left(2\text{m}^2\right)\frac{25\ {}^\circ\text{C}}{0.010\ \text{m}} = 120\ \text{W}$$ <

COMMENTS: Losses associated with a single pane are unacceptable and would remain excessive, even if the thickness of the glass were doubled to match that of the air space. The principal advantage of the double pane construction resides with the low thermal conductivity of air (~ 60 times smaller than that of glass). For a fixed ambient outside air temperature, use of the double pane construction would also increase the surface temperature of the glass exposed to the room (inside) air.

ME225 - Heat Transfer
A core curriculum of Mechanical Engineering

- Homework Solutions by Textbook
- Date: 2010.02
- By: David Malawey

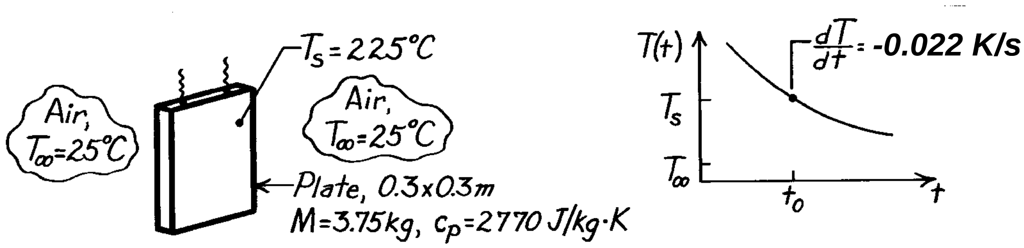
The following section of this PDF is a printout of raw values for homework solutions in the course text-book. These solutions are not worked out - they are simply an index for reference to the final solutions. This list has solutions for more problems than those assigned as homework in the course.

PROBLEM 1.22

KNOWN: Hot vertical plate suspended in cool, still air. Change in plate temperature with time at the instant when the plate temperature is 225°C.

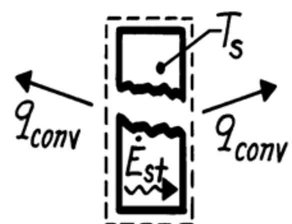
FIND: Convection heat transfer coefficient for this condition.

SCHEMATIC:



ASSUMPTIONS: (1) Plate is isothermal, (2) Negligible radiation exchange with surroundings, (3) Negligible heat lost through suspension wires.

ANALYSIS: As shown in the cooling curve above, the plate temperature decreases with time. The condition of interest is for time t_o. For a control surface about the plate, the conservation of energy requirement is



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

$$-2hA_s\left(T_s - T_\infty\right) = Mc_p\frac{dT}{dt}$$

where A_s is the surface area of one side of the plate. Solving for h, find

$$h = \frac{Mc_p}{2A_s\left(T_s - T_\infty\right)}\left(\frac{-dT}{dt}\right)$$

$$h = \frac{3.75\text{ kg} \times 2770\text{ J/kg}\cdot\text{K}}{2 \times \left(0.3 \times 0.3\right)\text{m}^2\left(225 - 25\right)\text{K}} \times 0.022\text{ K/s} = 6.3\text{ W/m}^2\cdot\text{K}$$ <

COMMENTS: (1) Assuming the plate is very highly polished with emissivity of 0.08, determine whether radiation exchange with the surroundings at 25°C is negligible compared to convection.

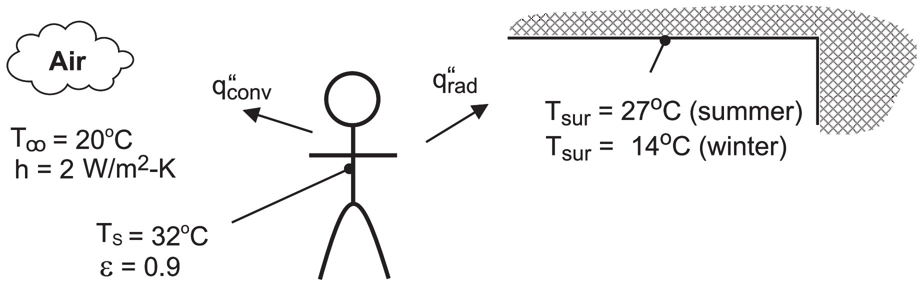
(2) We will later consider the criterion for determining whether the isothermal plate assumption is reasonable. If the thermal conductivity of the present plate were high (such as aluminum or copper), the criterion would be satisfied.

PROBLEM 1.24

KNOWN: Air and wall temperatures of a room. Surface temperature, convection coefficient and emissivity of a person in the room.

FIND: Basis for difference in comfort level between summer and winter.

SCHEMATIC:



ASSUMPTIONS: (1) Person may be approximated as a small object in a large enclosure.

ANALYSIS: Thermal comfort is linked to heat loss from the human body, and a *chilled* feeling is associated with excessive heat loss. Because the temperature of the room air is fixed, the different summer and winter comfort levels cannot be attributed to convection heat transfer from the body. In both cases, the heat flux is

Summer and Winter: $q''_{conv} = h\left(T_s - T_\infty\right) = 2\ \text{W/m}^2 \cdot \text{K} \times 12\ ^\circ\text{C} = 24\ \text{W/m}^2$

However, the heat flux due to radiation will differ, with values of

Summer: $q''_{rad} = \varepsilon\sigma\left(T_s^4 - T_{sur}^4\right) = 0.9 \times 5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4 \left(305^4 - 300^4\right)\text{K}^4 = 28.3\ \text{W/m}^2$

Winter: $q''_{rad} = \varepsilon\sigma\left(T_s^4 - T_{sur}^4\right) = 0.9 \times 5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4 \left(305^4 - 287^4\right)\text{K}^4 = 95.4\ \text{W/m}^2$

There is a significant difference between winter and summer radiation fluxes, and the chilled condition is attributable to the effect of the colder walls on radiation.

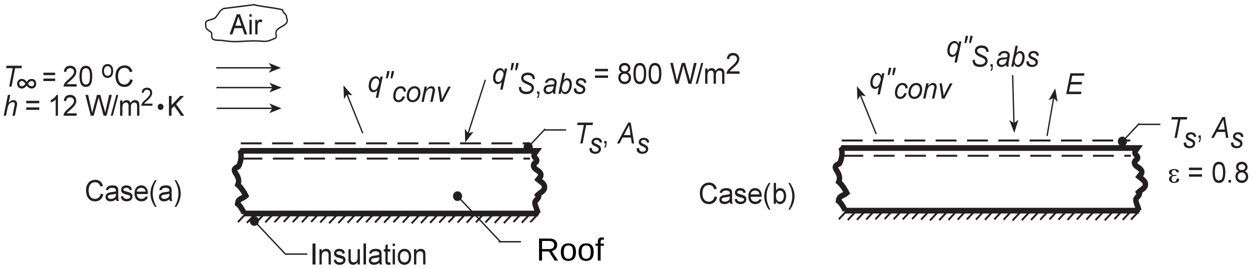
COMMENTS: For a representative surface area of A = 1.5 m^2, the heat losses are q_{conv} = 36 W, $q_{rad(summer)}$ = 42.5 W and $q_{rad(winter)}$ = 143.1 W. The winter time radiation loss is significant and if maintained over a 24 h period would amount to 2,950 kcal.

PROBLEM 1.55

KNOWN: Top surface of car roof absorbs solar flux, $q''_{S,abs}$, and experiences for case (a): convection with air at T_∞ and for case (b): the same convection process and radiation emission from the roof.

FIND: Temperature of the roof, T_s, for the two cases. Effect of airflow on roof temperature.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Negligible heat transfer to auto interior, (3) Negligible radiation from atmosphere.

ANALYSIS: (a) Apply an energy balance to the control surfaces shown on the schematic. For an instant of time, $\dot{E}_{in} - \dot{E}_{out} = 0$. Neglecting radiation emission, the relevant processes are convection between the plate and the air, q''_{conv}, and the absorbed solar flux, $q''_{S,abs}$. Considering the roof to have an area A_s,

$$q''_{S,abs} \cdot A_s - hA_s\left(T_s - T_\infty\right) = 0$$

$$T_s = T_\infty + q''_{S,abs}/h$$

$$T_s = 20^\circ C + \frac{800 W/m^2}{12 W/m^2 \cdot K} = 20^\circ C + 66.7^\circ C = 86.7^\circ C \qquad <$$

(b) With radiation emission from the surface, the energy balance has the form

$$q''_{S,abs} \cdot A_s - q_{conv} - E \cdot A_s = 0$$

$$q''_{S,abs} A_s - hA_s\left(T_s - T_\infty\right) - \varepsilon A_s \sigma T_s^4 = 0 .$$

Substituting numerical values, with temperature in absolute units (K),

$$800 \frac{W}{m^2} - 12 \frac{W}{m^2 \cdot K}\left(T_s - 293 K\right) - 0.8 \times 5.67 \times 10^{-8} \frac{W}{m^2 \cdot K^4} T_s^4 = 0$$

$$12 T_s + 4.536 \times 10^{-8} T_s^4 = 4316$$

It follows that T_s = 320 K = 47°C. $\qquad <$

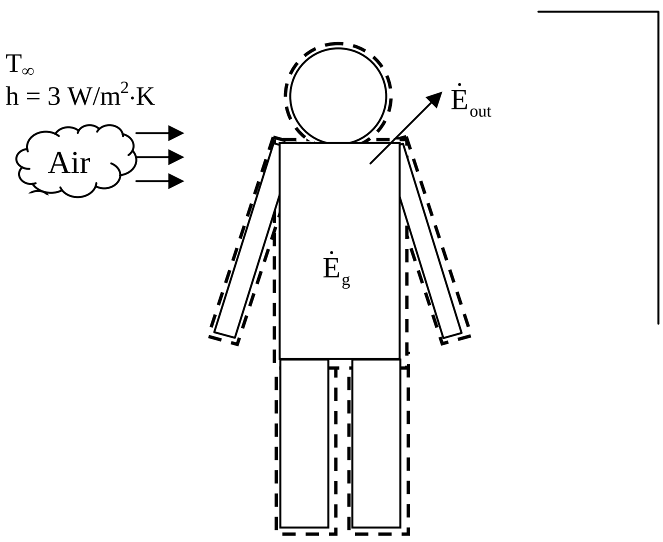
Continued.….

PROBLEM 1.57

KNOWN: Daily thermal energy generation, surface area, temperature of the environment, and heat transfer coefficient.

FIND: (a) Skin temperature when the temperature of the environment is 20ºC, and (b) Rate of perspiration to maintain skin temperature of 33ºC when the temperature of the environment is 33ºC.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Thermal energy is generated at a constant rate throughout the day, (3) Air and surrounding walls are at same temperature, (4) Skin temperature is uniform, (5) Bathing suit has no effect on heat loss from body, (6) Heat loss is by convection and radiation to the environment, and by perspiration in Part 2. (Heat loss due to respiration, excretion of waste, etc., is negligible.), (7) Large surroundings.

PROPERTIES: Table A.11, skin: $\varepsilon = 0.95$, Table A.6, water (306 K): $\rho = 994$ kg/m^3, $h_{fg} = 2421$ kJ/kg.

ANALYSIS:
(a) The rate of energy generation is:

$$\dot{E}_g = 2000 \times 10^3 \text{ cal/day} \big/ (0.239 \text{ cal/J} \times 86{,}400 \text{ s/day}) = 96.9 \text{ W}$$

Under steady-state conditions, an energy balance on the human body yields:

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

Thus $\dot{E}_{out}$ = q = 96.9 W. Energy outflow is due to convection and net radiation from the surface to the environment, Equations 1.3a and 1.7, respectively.

$$\dot{E}_{out} = hA(T_s - T_\infty) + \varepsilon\sigma A(T_s^4 - T_{sur}^4)$$

Substituting numerical values

Continued…

$$96.9\text{ W} = 3\text{ W/m}^2\cdot\text{K}\times 1.8\text{ m}^2\times(T_s - 293\text{ K})$$
$$+\,0.95\times 5.67\times 10^{-8}\text{ W/m}^2\cdot\text{K}^4\times 1.8\text{ m}^2\times(T_s^4 - (293\text{ K})^4)$$

and solving either by trial-and-error or using *IHT* or other equation solver, we obtain

$$T_s = 299\text{ K} = 26^{\circ}\text{C} \qquad <$$

Since the comfortable range of skin temperature is typically 32 – 35ºC, we usually wear clothing warmer than a bathing suit when the temperature of the environment is 20ºC.

(b) If the skin temperature is 33ºC when the temperature of the environment is 33ºC, there will be no heat loss due to convection or radiation. Thus, all the energy generated must be removed due to perspiration:

$$\dot{E}_{out} = \dot{m}h_{fg}$$

We find:

$$\dot{m} = \dot{E}_{out}/h_{fg} = 96.9\text{ W}/2421\text{ kJ/kg} = 4.0\times 10^{-5}\text{ kg/s}$$

This is the perspiration rate in mass per unit time. The volumetric rate is:

$$\dot{\forall} = \dot{m}/\rho = 4.0\times 10^{-5}\text{ kg/s} / 994\text{ kg/m}^3 = 4.0\times 10^{-8}\text{ m}^3\text{/s} = 4.0\times 10^{-5}\ \ell\text{/s} \qquad <$$

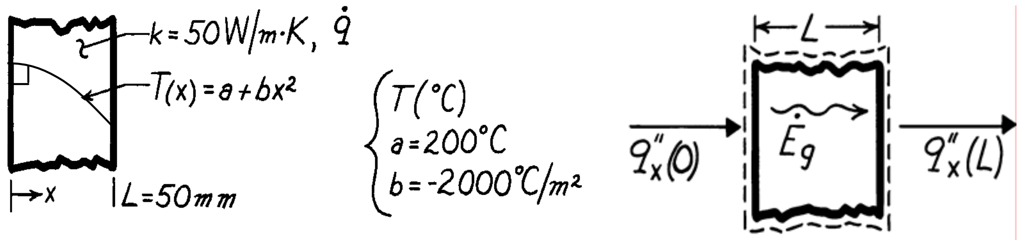
COMMENTS: (1) In Part 1, heat losses due to convection and radiation are 32.4 W and 60.4 W, respectively. Thus, it would not have been reasonable to neglect radiation. Care must be taken to include radiation when the heat transfer coefficient is small, even if the problem statement does not give any indication of its importance. (2) The rate of thermal energy generation is not constant throughout the day; it adjusts to maintain a constant core temperature. Thus, the energy generation rate may decrease when the temperature of the environment goes up, or increase (for example, by shivering) when the temperature of the environment is low. (3) The skin temperature is not uniform over the entire body. For example, the extremities are usually cooler. Skin temperature also adjusts in response to changes in the environment. As the temperature of the environment increases, more blood flow will be directed near the surface of the skin to increase its temperature, thereby increasing heat loss. (4) If the perspiration rate found in Part 2 was maintained for eight hours, the person would lose 1.2 liters of liquid. This demonstrates the importance of consuming sufficient amounts of liquid in warm weather.

PROBLEM 2.23

KNOWN: Temperature distribution in a one-dimensional wall with prescribed thickness and thermal conductivity.

FIND: (a) The heat generation rate, $\dot{q}$, in the wall, (b) Heat fluxes at the wall faces and relation to $\dot{q}$.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional heat flow, (3) Constant properties.

ANALYSIS: (a) The appropriate form of the heat equation for steady-state, one-dimensional conditions with constant properties is Eq. 2.19 re-written as

$$\dot{q}=-k\frac{d}{dx}\left[\frac{dT}{dx}\right]$$

Substituting the prescribed temperature distribution,

$$\dot{q}=-k\frac{d}{dx}\left[\frac{d}{dx}\left(a+bx^2\right)\right]=-k\frac{d}{dx}[2bx]=-2bk$$

$$\dot{q}=-2\left(-2000^\circ C/m^2\right)\times 50\ W/m\cdot K=2.0\times 10^5\ W/m^3. \qquad <$$

(b) The heat fluxes at the wall faces can be evaluated from Fourier's law,

$$q''_x(x)=-k\left.\frac{dT}{dx}\right]_x .$$

Using the temperature distribution T(x) to evaluate the gradient, find

$$q''_x(x)=-k\frac{d}{dx}\left[a+bx^2\right]=-2kbx.$$

The fluxes at x = 0 and x = L are then

$$q''_x(0)=0 \qquad <$$

$$q''_x(L)=-2kbL=-2\times 50W/m\cdot K\left(-2000^\circ C/m^2\right)\times 0.050m$$

$$q''_x(L)=10,000\ W/m^2. \qquad <$$

COMMENTS: From an overall energy balance on the wall, it follows that, for a unit area,

$$\dot{E}_{in}-\dot{E}_{out}+\dot{E}_g=0 \qquad q''_x(0)-q''_x(L)+\dot{q}L=0$$

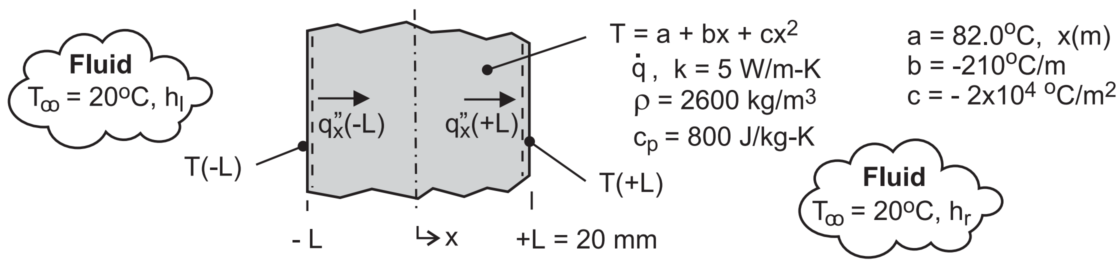
$$\dot{q}=\frac{q''_x(L)-q''_x(0)}{L}=\frac{10,000\ W/m^2-0}{0.050m}=2.0\times 10^5 W/m^3.$$

PROBLEM 2.25

KNOWN: Analytical expression for the steady-state temperature distribution of a plane wall experiencing uniform volumetric heat generation $\dot{q}$ while convection occurs at both of its surfaces.

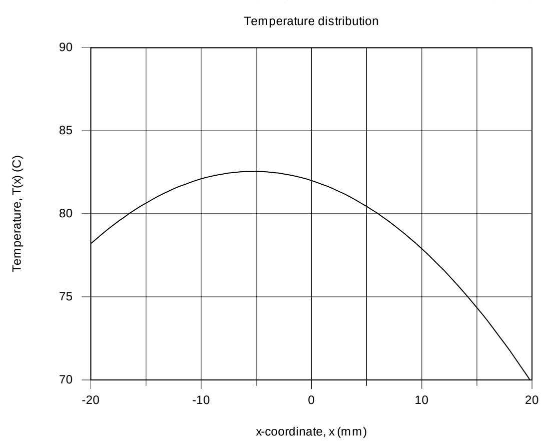
FIND: (a) Sketch the temperature distribution, T(x), and identify significant physical features, (b) Determine $\dot{q}$, (c) Determine the surface heat fluxes, $q''_x(-L)$ and $q''_x(+L)$; how are these fluxes related to the generation rate; (d) Calculate the convection coefficients at the surfaces x = L and x = +L, (e) Obtain an expression for the heat flux distribution, $q''_x(x)$; explain significant features of the distribution; (f) If the source of heat generation is suddenly deactivated ($\dot{q}$ = 0), what is the rate of change of energy stored at this instant; (g) Determine the temperature that the wall will reach eventually with $\dot{q} = 0$; determine the energy that must be removed by the fluid per unit area of the wall to reach this state.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Uniform volumetric heat generation, (3) Constant properties.

ANALYSIS: (a) Using the analytical expression in the Workspace of IHT, the temperature distribution appears as shown below. The significant features include (1) parabolic shape, (2) maximum does not occur at the mid-plane, T(-5.25 mm) = 83.3°C, (3) the gradient at the x = +L surface is greater than at x = -L. Find also that T(-L) = 78.2°C and T(+L) = 69.8°C for use in part (d).



(b) Substituting the temperature distribution expression into the appropriate form of the heat diffusion equation, Eq. 2.19, the rate of volumetric heat generation can be determined.

$$\frac{d}{dx}\left(\frac{dT}{dx}\right)+\frac{\dot{q}}{k}=0 \qquad \text{where} \qquad T(x)=a+bx+cx^2$$

$$\frac{d}{dx}(0+b+2cx)+\frac{\dot{q}}{k}=(0+2c)+\frac{\dot{q}}{k}=0$$

Continued …..

$$\dot{q} = -2ck = -2\left(-2\times10^{4}{}^{\circ}\mathrm{C/m^2}\right)5\,\mathrm{W/m\cdot K} = 2\times10^{5}\ \mathrm{W/m^3}$$ <

(c) The heat fluxes at the two boundaries can be determined using Fourier's law and the temperature distribution expression.

$$q''_x(x) = -k\frac{dT}{dx} \qquad \text{where} \qquad T(x) = a + bx + cx^2$$

$$q''_x(-L) = -k\left[0 + b + 2cx\right]_{x=-L} = -\left[b - 2cL\right]k$$

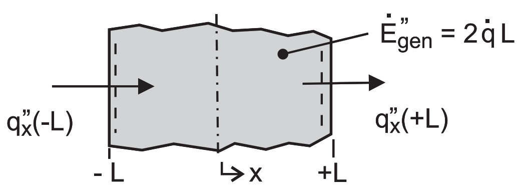
$$q''_x(-L) = -\left[-210^{\circ}\mathrm{C/m} - 2\left(-2\times10^{4}{}^{\circ}\mathrm{C/m^2}\right)0.020\mathrm{m}\right]\times 5\,\mathrm{W/m\cdot K} = -2950\,\mathrm{W/m^2}$$ <

$$q''_x(+L) = -\left(b + 2cL\right)k = +5050\,\mathrm{W/m^2}$$ <

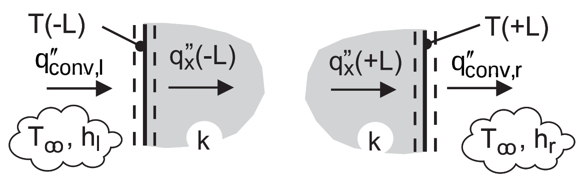
From an overall energy balance on the wall as shown in the sketch below, $\dot{E}_{in} - \dot{E}_{out} + \dot{E}_{gen} = 0$,

$$+q''_x(-L) - q''_x(+L) + 2\dot{q}L \overset{?}{=} 0 \quad \text{or} \quad -2950\,\mathrm{W/m^2} - 5050\,\mathrm{W/m^2} + 8000\,\mathrm{W/m^2} = 0$$

where $2\dot{q}L = 2\times2\times10^{5}\ \mathrm{W/m^3}\times0.020\ \mathrm{m} = 8000\,\mathrm{W/m^2}$, so the equality is satisfied



Part (c) Overall energy balance



Part (d) Surface energy balances

(d) The convection coefficients, h_l and h_r, for the left- and right-hand boundaries (x = -L and x= +L, respectively), can be determined from the convection heat fluxes that are equal to the conduction fluxes at the boundaries. See the surface energy balances in the sketch above. See also part (a) result for T(-L) and T(+L).

$$q''_{conv,\ell} = q''_x(-L)$$

$$h_l\left[T_\infty - T(-L)\right] = h_l\left[20 - 78.2\right]\mathrm{K} = -2950\,\mathrm{W/m^2} \qquad h_l = 51\,\mathrm{W/m^2\cdot K}$$ <

$$q''_{conv,r} = q''_x(+L)$$

$$h_r\left[T(+L) - T_\infty\right] = h_r\left[69.8 - 20\right]\mathrm{K} = +5050\,\mathrm{W/m^2} \qquad h_r = 101\,\mathrm{W/m^2\cdot K}$$ <

(e) The expression for the heat flux distribution can be obtained from Fourier's law with the temperature distribution

$$q''_x(x) = -k\frac{dT}{dx} = -k\left[0 + b + 2cx\right]$$

$$q''_x(x) = -5\,\mathrm{W/m\cdot K}\left[-210^{\circ}\mathrm{C/m} + 2\left(-2\times10^{4}{}^{\circ}\mathrm{C/m^2}\right)\right]x = 1050 + 2\times10^{5}x$$ <

Continued …..

PROBLEM 2.25 (Cont.)

The distribution is linear with the x-coordinate. The maximum temperature will occur at the location where $q''_x(x_{max}) = 0$,

$$x_{max} = -\frac{1050\,\text{W/m}^2}{2\times10^5\ \text{W/m}^3} = -5.25\times10^{-3}\text{m} = -5.25\,\text{mm} \qquad <$$

(f) If the source of the heat generation is suddenly deactivated so that $\dot{q} = 0$, the appropriate form of the heat diffusion equation for the ensuing transient conduction is

$$k\frac{\partial}{\partial x}\left(\frac{\partial T}{\partial x}\right) = \rho c_p \frac{\partial T}{\partial t}$$

At the instant this occurs, the temperature distribution is still $T(x) = a + bx + cx^2$. The right-hand term represents the rate of energy storage per unit volume,

$$\dot{E}''_{st} = k\frac{\partial}{\partial x}\left[0 + b + 2cx\right] = k\left[0 + 2c\right] = 5\,\text{W/m}\cdot\text{K}\times2\left(-2\times10^{4}\,°\text{C/m}^2\right) = -2\times10^5\ \text{W/m}^3 \qquad <$$

(g) With no heat generation, the wall will eventually ($t \to \infty$) come to equilibrium with the fluid, $T(x,\infty) = T_\infty = 20°C$. To determine the energy that must be removed from the wall to reach this state, apply the conservation of energy requirement over an interval basis, Eq. 1.11b. The "initial" state is that corresponding to the steady-state temperature distribution, T_i, and the "final" state has $T_f = 20°C$. We've used T_∞ as the reference condition for the energy terms.

$$E''_{in} - E''_{out} = \Delta E''_{st} = E''_f - E''_i \qquad \text{with} \qquad E''_{in} = 0.$$

$$E''_{out} = c_p\int_{-L}^{+L}\left(T_i - T_\infty\right)dx$$

$$E''_{out} = \rho c_p\int_{-L}^{+L}\left[a + bx + cx^2 - T_\infty\right]dx = \rho c_p\left[ax + bx^2/2 + cx^3/3 - T_\infty x\right]_{-L}^{+L}$$

$$E''_{out} = \rho c_p\left[2aL + 0 + 2cL^3/3 - 2T_\infty L\right]$$

$$E''_{out} = 2600\,\text{kg/m}^3\times800\,\text{J/kg}\cdot\text{K}\left[2\times82°\text{C}\times0.020\text{m} + 2\left(-2\times10^4\,°\text{C/m}^2\right)\right.$$
$$\left.\left(0.020\text{m}\right)^3/3 - 2\left(20°\text{C}\right)0.020\text{m}\right]$$

$$E''_{out} = 4.94\times10^6\ \text{J/m}^2 \qquad <$$

COMMENTS: (1) In part (a), note that the temperature gradient is larger at x = + L than at x = - L. This is consistent with the results of part (c) in which the conduction heat fluxes are evaluated.

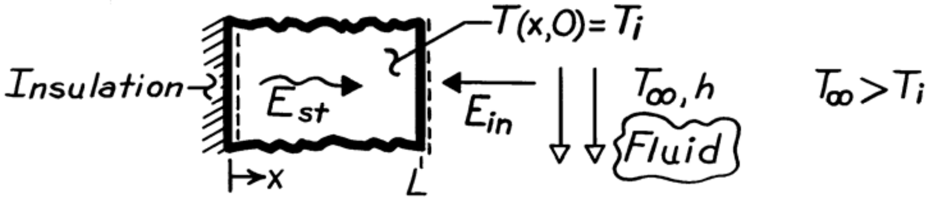
Continued …..

PROBLEM 2.46

KNOWN: Plane wall, initially at a uniform temperature, is suddenly exposed to convective heating.

FIND: (a) Differential equation and initial and boundary conditions which may be used to find the temperature distribution, T(x,t); (b) Sketch T(x,t) for these conditions: initial ($t \leq 0$), steady-state, $t \to \infty$, and two intermediate times; (c) Sketch heat fluxes as a function of time for surface locations; (d) Expression for total energy transferred to wall per unit volume (J/m^3).

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional conduction, (2) Constant properties, (3) No internal heat generation.

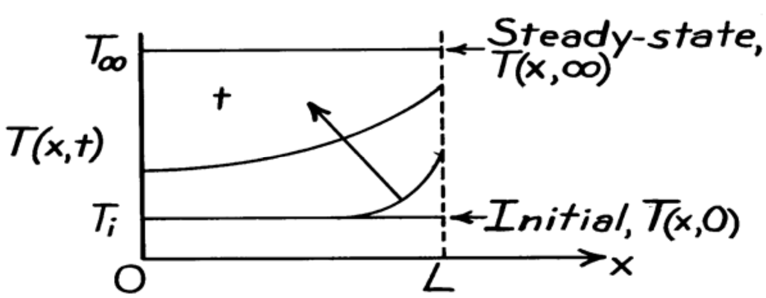
ANALYSIS: (a) For one-dimensional conduction with constant properties, the heat equation has the form,

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t}$$

and the conditions are:

$$\begin{cases} \text{Initial, } t \leq 0: \quad T(x,0) = T_i & \text{uniform} \\ \text{Boundaries:} \quad x=0 \quad \partial T/\partial x)_0 = 0 & \text{adiabatic} \\ \qquad\qquad x=L \quad -k\partial T/\partial x)_L = h\left[T(L,t) - T_\infty\right] & \text{convection} \end{cases}$$

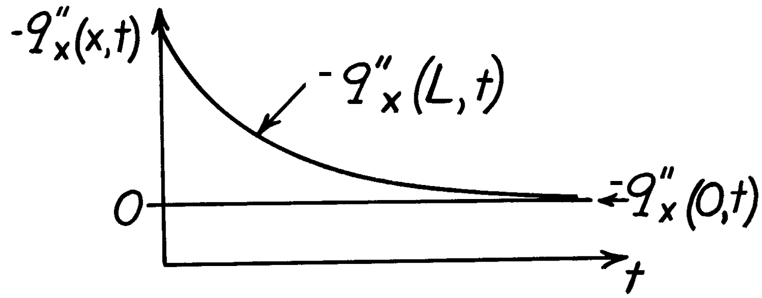
(b) The temperature distributions are shown on the sketch.



Note that the gradient at x = 0 is always zero, since this boundary is adiabatic. Note also that the gradient at x = L decreases with time.

(c) The heat flux, $q''_x(x,t)$, as a function of time, is shown on the sketch for the surfaces x = 0 and x = L.

Continued …..



For the surface at $x = 0$, $q''_x(0,t) = 0$ since it is adiabatic. At x = L and t = 0, $q''_x(L,0)$ is a maximum (in magnitude)

$$\left|q''_x(L,0)\right| = h\left|T(L,0) - T_\infty\right|$$

where $T(L,0) = T_i$. The temperature difference, and hence the flux, decreases with time.

(d) The total energy transferred to the wall may be expressed as

$$E_{in} = \int_0^\infty q''_{conv} A_s dt$$
$$E_{in} = hA_s \int_0^\infty \left(T_\infty - T(L,t)\right) dt$$

Dividing both sides by $A_s L$, the energy transferred per unit volume is

$$\frac{E_{in}}{V} = \frac{h}{L}\int_0^\infty \left[T_\infty - T(L,t)\right] dt \qquad \left[J/m^3\right]$$

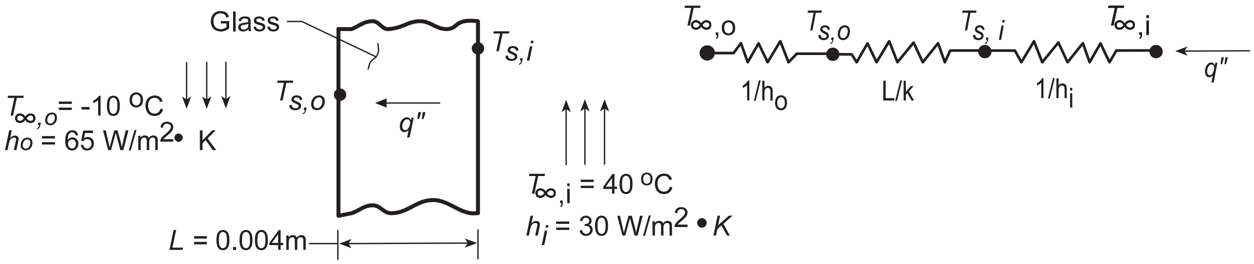
COMMENTS: Note that the heat flux at x = L is into the wall and is hence in the negative x direction.

PROBLEM 3.2

KNOWN: Temperatures and convection coefficients associated with air at the inner and outer surfaces of a rear window.

FIND: (a) Inner and outer window surface temperatures, $T_{s,i}$ and $T_{s,o}$, and (b) $T_{s,i}$ and $T_{s,o}$ as a function of the outside air temperature $T_{\infty,o}$ and for selected values of outer convection coefficient, h_o.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional conduction, (3) Negligible radiation effects, (4) Constant properties.

PROPERTIES: *Table A-3*, Glass (300 K): k = 1.4 W/m·K.

ANALYSIS: (a) The heat flux may be obtained from Eqs. 3.11 and 3.12,

$$q'' = \frac{T_{\infty,i} - T_{\infty,o}}{\frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_i}} = \frac{40^\circ\mathrm{C} - \left(-10^\circ\mathrm{C}\right)}{\frac{1}{65\,\mathrm{W/m^2\cdot K}} + \frac{0.004\,\mathrm{m}}{1.4\,\mathrm{W/m\cdot K}} + \frac{1}{30\,\mathrm{W/m^2\cdot K}}}$$

$$q'' = \frac{50^\circ\mathrm{C}}{\left(0.0154 + 0.0029 + 0.0333\right)\mathrm{m^2\cdot K/W}} = 969\,\mathrm{W/m^2}\,.$$

Hence, with $q'' = h_i\left(T_{\infty,i} - T_{\infty,o}\right)$, the inner surface temperature is

$$T_{s,i} = T_{\infty,i} - \frac{q''}{h_i} = 40^\circ\mathrm{C} - \frac{969\,\mathrm{W/m^2}}{30\,\mathrm{W/m^2\cdot K}} = 7.7^\circ\mathrm{C} \qquad <$$

Similarly for the outer surface temperature with $q'' = h_o\left(T_{s,o} - T_{\infty,o}\right)$ find

$$T_{s,o} = T_{\infty,o} + \frac{q''}{h_o} = -10^\circ\mathrm{C} + \frac{969\,\mathrm{W/m^2}}{65\,\mathrm{W/m^2\cdot K}} = 4.9^\circ\mathrm{C} \qquad <$$

(b) Using the same analysis, $T_{s,i}$ and $T_{s,o}$ have been computed and plotted as a function of the outside air temperature, $T_{\infty,o}$, for outer convection coefficients of h_o = 2, 65, and 100 W/m^2·K. As expected, $T_{s,i}$ and $T_{s,o}$ are linear with changes in the outside air temperature. The difference between $T_{s,i}$ and $T_{s,o}$ increases with increasing convection coefficient, since the heat flux through the window likewise increases. This difference is larger at lower outside air temperatures for the same reason. Note that with h_o = 2 W/m^2·K, $T_{s,i}$ - $T_{s,o}$, is too small to show on the plot.

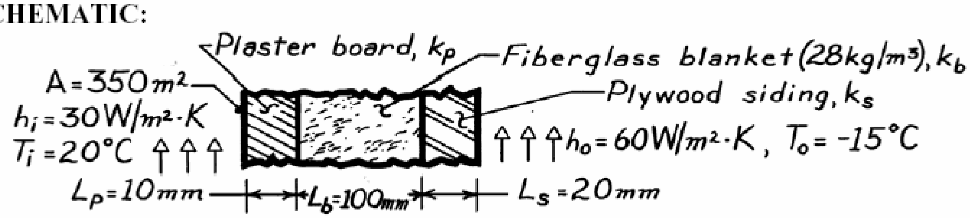
Continued …..

PROBLEM 3.13

KNOWN: Composite wall of a house with prescribed convection processes at inner and outer surfaces.

FIND: (a) Expression for thermal resistance of house wall, R_{tot}; (b) Total heat loss, q(W); (c) Effect on heat loss due to increase in outside heat transfer convection coefficient, h_o; and (d) Controlling resistance for heat loss from house.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional conduction, (2) Steady-state conditions, (3) Negligible contact resistance.

PROPERTIES: *Table A-3*, $\left(\overline{T} = \left(T_i + T_o\right)/2 = \left(20-15\right)^\circ C/2 = 2.5^\circ C \approx 300K\right)$: Fiberglass blanket, 28 kg/m^3, k_b = 0.038 W/m·K; Plywood siding, k_s = 0.12 W/m·K; Plasterboard, k_p = 0.17 W/m·K.

ANALYSIS: (a) The expression for the total thermal resistance of the house wall follows from Eq. 3.18.

$$R_{tot} = \frac{1}{h_i A} + \frac{L_p}{k_p A} + \frac{L_b}{k_b A} + \frac{L_s}{k_s A} + \frac{1}{h_o A}. \qquad <$$

(b) The total heat loss through the house wall is

$$q = \Delta T / R_{tot} = \left(T_i - T_o\right)/R_{tot}.$$

Substituting numerical values, find

$$R_{tot} = \frac{1}{30 W/m^2 \cdot K \times 350 m^2} + \frac{0.01 m}{0.17 W/m \cdot K \times 350 m^2} + \frac{0.10 m}{0.038 W/m \cdot K \times 350 m^2}$$

$$+ \frac{0.02 m}{0.12 W/m \cdot K \times 350 m^2} + \frac{1}{60 W/m^2 \cdot K \times 350 m^2}$$

$$R_{tot} = \left[9.52 + 16.8 + 752 + 47.6 + 4.76\right] \times 10^{-5}\ ^\circ C/W = 831 \times 10^{-5}\ ^\circ C/W$$

The heat loss is then,

$$q = \left[20 - (-15)\right]^\circ C / 831 \times 10^{-5}\ ^\circ C/W = 4.21\ kW. \qquad <$$

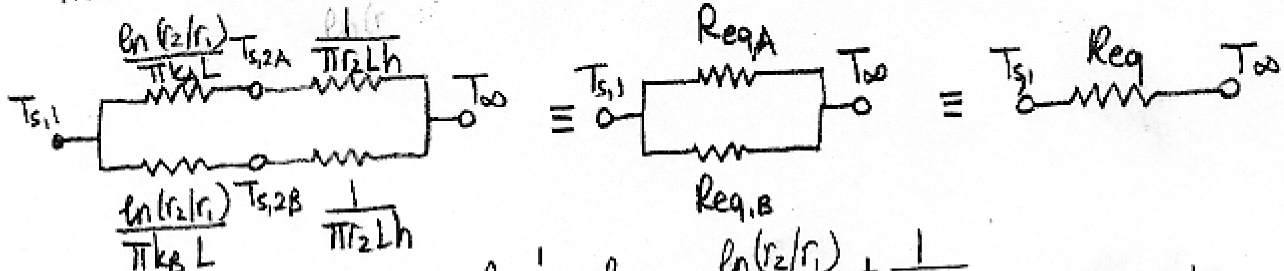
(c) If h_o changes from 60 to 300 $W/m^2 \cdot K$, $R_o = 1/h_o A$ changes from 4.76×10^{-5} °C/W to 0.95×10^{-5} °C/W. This reduces R_{tot} to 826×10^{-5} °C/W, which is a 0.6% decrease and hence a 0.6% increase in q.

(d) From the expression for R_{tot} in part (b), note that the insulation resistance, $L_b/k_b A$, is $752/830 \approx 90\%$ of the total resistance. Hence, this material layer controls the resistance of the wall. From part (c) note that a 5-fold decrease in the outer convection resistance due to an increase in the wind velocity has a negligible effect on the heat loss.

3.52

This problem can be solved in two different ways depending on the assumption made since it is not a purely 1-D geometry.

(i) Assume infinite contact resistance between the materials so that there is no heat crossing between materials A and B at their interface (otherwise, not only radial but also circumferential heat transfer in r and ϕ directions respectively). Then,



$$Req'_{,A} = \frac{Req_{,A}}{L} = \frac{\ln(r_2/r_1)}{\pi k_A} + \frac{1}{\pi r_2 h}$$

$$Req'_{,B} = \frac{Req_{,B}}{L} = \frac{\ln(r_2/r_1)}{\pi k_B} + \frac{1}{\pi r_2 h}$$

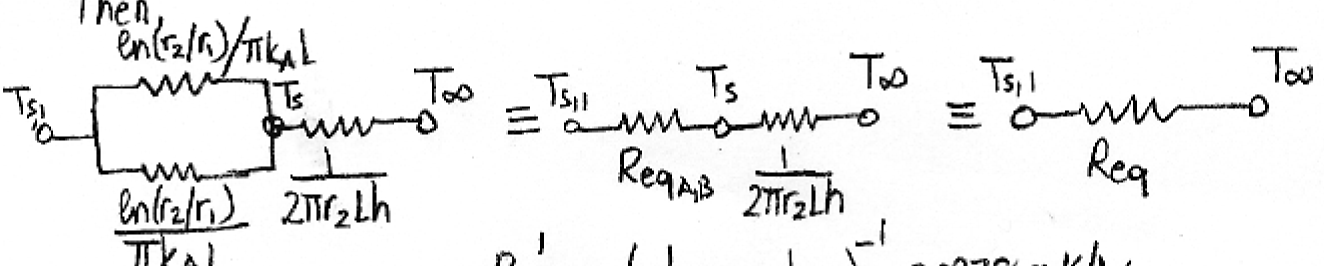
$$\frac{1}{R'_{eq}} = \frac{1}{Req'_{,A}} + \frac{1}{Req'_{,B}}$$

$$Req'_{,A} = \frac{\ln(0.1/0.05)}{(\pi)(2)} + \frac{1}{(\pi)(0.1)(25)} = 0.110 + 0.127 = 0.237 \text{ m.K/W}$$

$$Req'_{,B} = \frac{\ln(0.1/0.05)}{(\pi)(0.25)} + \frac{1}{(\pi)(0.1)(25)} = 0.882 + 0.127 = 1.009 \text{ m.K/W}$$

$$R'_{eq} = \left(\frac{1}{0.237} + \frac{1}{1.009}\right)^{-1} = 0.192 \text{ m.K/W} \Rightarrow q' = \frac{\Delta T}{R'_{eq}} = \frac{500-300}{0.192} = 1042 \text{ W/m}$$

(ii) Alternatively, assume isothermal outer surface, which is less likely because materials A and B have very different thermal conductivities. Then,



$$Req'_{,AB} = \left(\frac{1}{0.110} + \frac{1}{0.882}\right)^{-1} = 0.0978 \text{ m.K/W}$$

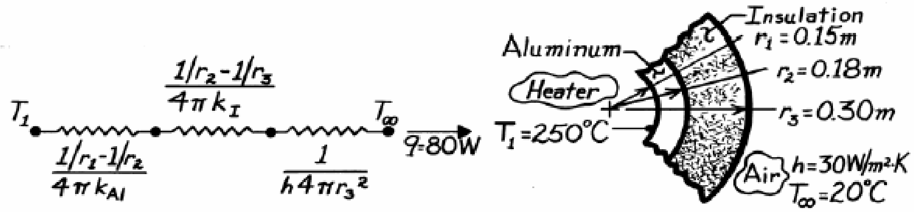
$$R'_{eq} = 0.0978 + \frac{1}{(2\pi)(0.1)(25)} = 0.0978 + 0.0635 = 0.161 \text{ m.K} \Rightarrow q' = \frac{200}{0.161} = 1240 \text{ W/m}$$

PROBLEM 3.57

KNOWN: Thickness of hollow aluminum sphere and insulation layer. Heat rate and inner surface temperature. Ambient air temperature and convection coefficient.

FIND: Thermal conductivity of insulation.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional radial conduction, (3) Constant properties, (4) Negligible contact resistance, (5) Negligible radiation exchange at outer surface.

PROPERTIES: *Table A-1*, Aluminum (523K): k ≈ 230 W/m·K.

ANALYSIS: From the thermal circuit,

$$q=\frac{T_1-T_\infty}{R_{tot}}=\frac{T_1-T_\infty}{\frac{1/r_1-1/r_2}{4\pi k_{A1}}+\frac{1/r_2-1/r_3}{4\pi k_I}+\frac{1}{h4\pi r_3^2}}$$

$$q=\frac{(250-20)^\circ C}{\left[\frac{1/0.15-1/0.18}{4\pi(230)}+\frac{1/0.18-1/0.30}{4\pi k_I}+\frac{1}{30(4\pi)(0.3)^2}\right]\frac{K}{W}}=80\text{ W}$$

or

$$3.84\times10^{-4}+\frac{0.177}{k_I}+0.0029=\frac{230}{80}=2.875.$$

Solving for the unknown thermal conductivity, find

$$k_I = 0.062\text{ W/m·K.}$$ <

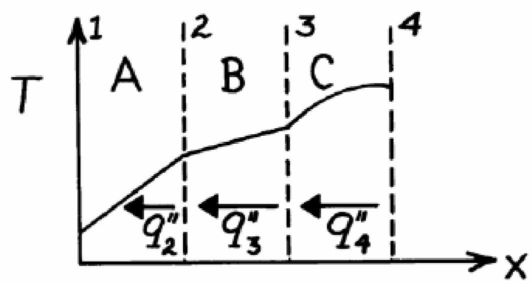
COMMENTS: The dominant contribution to the total thermal resistance is made by the insulation. Hence uncertainties in knowledge of h or k_{A1} have a negligible effect on the accuracy of the k_I measurement.

PROBLEM 3.71

KNOWN: Temperature distribution in a composite wall.

FIND: (a) Relative magnitudes of interfacial heat fluxes, (b) Relative magnitudes of thermal conductivities, and (c) Heat flux as a function of distance x.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional conduction, (3) Constant properties.

ANALYSIS: (a) For the prescribed conditions (one-dimensional, steady-state, constant k), the parabolic temperature distribution in C implies the existence of heat generation. Hence, since dT/dx *increases* with *decreasing* x, the heat flux in C *increases* with *decreasing* x. Hence,

$$q_3'' > q_4''$$

However, the linear temperature distributions in A and B indicate no generation, in which case

$$q_2'' = q_3''$$

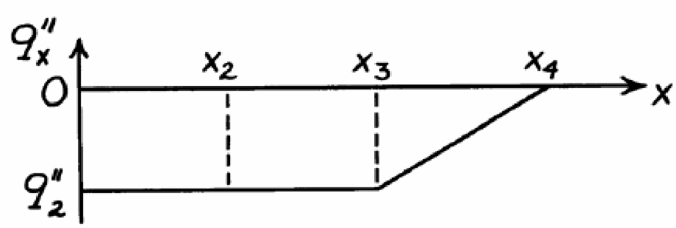
(b) Since conservation of energy requires that $q''_{3,B} = q''_{3,C}$ and $dT/dx)_B < dT/dx)_C$, it follows from Fourier's law that

$$k_B > k_C.$$

Similarly, since $q''_{2,A} = q''_{2,B}$ and $dT/dx)_A > dT/dx)_B$, it follows that

$$k_A < k_B.$$

(c) It follows that the flux distribution appears as shown below.



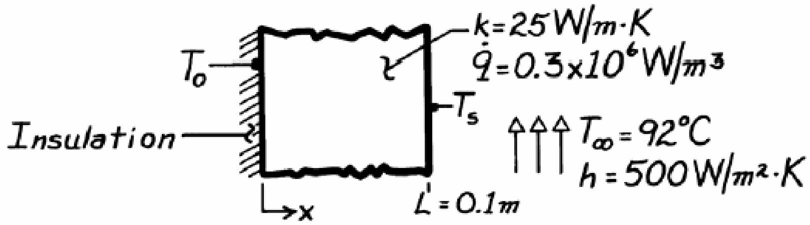
COMMENTS: Note that, with $dT/dx)_{4,C} = 0$, the interface at 4 is adiabatic.

PROBLEM 3.72

KNOWN: Plane wall with internal heat generation which is insulated at the inner surface and subjected to a convection process at the outer surface.

FIND: Maximum temperature in the wall.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional conduction with uniform volumetric heat generation, (3) Inner surface is adiabatic.

ANALYSIS: From Eq. 3.42, the temperature at the inner surface is given by Eq. 3.43 and is the maximum temperature within the wall,

$$T_o = \dot{q}L^2 / 2k + T_s.$$

The outer surface temperature follows from Eq. 3.46,

$$T_s = T_\infty + \dot{q}L/h$$

$$T_s = 92^\circ C + 0.3\times10^6 \frac{W}{m^3}\times 0.1m/500W/m^2\cdot K = 92^\circ C + 60^\circ C = 152^\circ C.$$

It follows that

$$T_o = 0.3\times10^6\, W/m^3\times(0.1m)^2 / 2\times 25W/m\cdot K + 152^\circ C$$

$$T_o = 60^\circ C + 152^\circ C = 212^\circ C. \qquad <$$

COMMENTS: The heat flux leaving the wall can be determined from knowledge of h, T_s and T_∞ using Newton's law of cooling.

$$q''_{conv} = h(T_s - T_\infty) = 500W/m^2\cdot K(152-92)^\circ C = 30kW/m^2.$$

This same result can be determined from an energy balance on the entire wall, which has the form

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

where

$$\dot{E}_g = \dot{q}AL \qquad \text{and} \qquad \dot{E}_{out} = q''_{conv}\cdot A.$$

Hence,

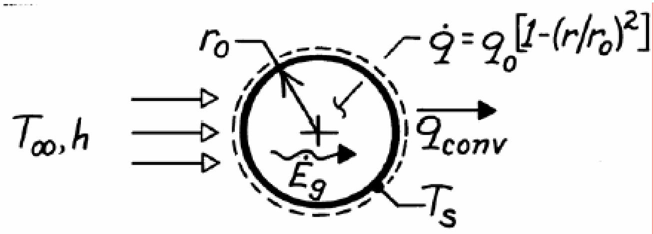
$$q''_{conv} = \dot{q}L = 0.3\times10^6 W/m^3\times 0.1m = 30kW/m^2.$$

PROBLEM 3.94

KNOWN: Radial distribution of heat dissipation of a spherical container of radioactive wastes. Surface convection conditions.

FIND: Radial temperature distribution.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional conduction, (3) Constant properties, (4) Negligible temperature drop across container wall.

ANALYSIS: The appropriate form of the heat equation is

$$\frac{1}{r^2}\frac{d}{dr}\left(r^2\frac{dT}{dr}\right) = -\frac{\dot{q}}{k} = -\frac{\dot{q}_o}{k}\left[1-\left(\frac{r}{r_o}\right)^2\right].$$

Hence $$r^2\frac{dT}{dr} = -\frac{\dot{q}_o}{k}\left(\frac{r^3}{3}-\frac{r^5}{5r_o^2}\right)+C_1$$

$$T = -\frac{\dot{q}_o}{k}\left(\frac{r^2}{6}-\frac{r^4}{20r_o^2}\right)-\frac{C_1}{r}+C_2.$$

From the boundary conditions,

$$dT/dr\,|_{r=0} = 0 \quad \text{and} \quad -k\,dT/dr\,|_{r=r_o} = h\left[T(r_o)-T_\infty\right]$$

it follows that $C_1 = 0$ and

$$\dot{q}_o\left(\frac{r_o}{3}-\frac{r_o}{5}\right) = h\left[-\frac{\dot{q}_o}{k}\left(\frac{r_o^2}{6}-\frac{r_o^2}{20}\right)+C_2-T_\infty\right]$$

$$C_2 = \frac{2r_o\dot{q}_o}{15h}+\frac{7\dot{q}_o r_o^2}{60k}+T_\infty.$$

Hence $$T(r) = T_\infty+\frac{2r_o\dot{q}_o}{15h}+\frac{\dot{q}r_o^2}{k}\left[\frac{7}{60}-\frac{1}{6}\left(\frac{r}{r_o}\right)^2+\frac{1}{20}\left(\frac{r}{r_o}\right)^4\right].$$ <

COMMENTS: Applying the above result at r_o yields

$$T_s = T(r_o) = T_\infty + (2r_o\dot{q}_o/15h).$$

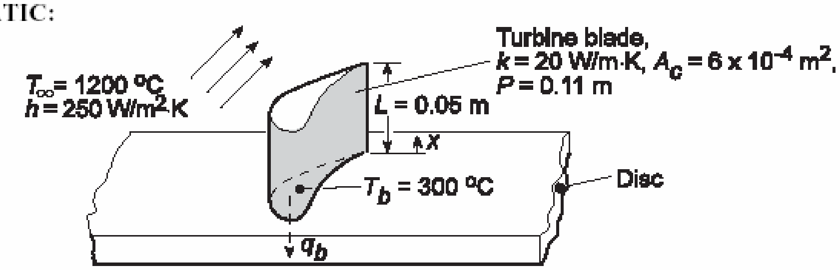
The same result may be obtained by applying an energy balance to a control surface about the container, where $\dot{E}_g = q_{conv}$. The maximum temperature exists at r = 0.

PROBLEM 3.116

KNOWN: Dimensions and thermal conductivity of a gas turbine blade. Temperature and convection coefficient of gas stream. Temperature of blade base and maximum allowable blade temperature.

FIND: (a) Whether blade operating conditions are acceptable, (b) Heat transfer to blade coolant.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional, steady-state conduction in blade, (2) Constant k, (3) Adiabatic blade tip, (4) Negligible radiation.

ANALYSIS: Conditions in the blade are determined by Case B of Table 3.4.

(a) With the maximum temperature existing at x = L, Eq. 3.75 yields

$$\frac{T(L)-T_\infty}{T_b-T_\infty}=\frac{1}{\cosh mL}$$

$$m=\left(hP/kA_c\right)^{1/2}=\left(250\,W/m^2\cdot K\times 0.11m/20\,W/m\cdot K\times 6\times 10^{-4}m^2\right)^{1/2}$$

$$m = 47.87\ m^{-1} \quad \text{and} \quad mL = 47.87\ m^{-1}\times 0.05\ m = 2.39$$

From Table B.1, cosh mL = 5.51. Hence,

$$T(L)=1200^\circ C+(300-1200)^\circ C/5.51=1037^\circ C \qquad <$$

and the operating conditions are acceptable.

(b) With $M=\left(hPkA_c\right)^{1/2}\Theta_b=\left(250\,W/m^2\cdot K\times 0.11m\times 20\,W/m\cdot K\times 6\times 10^{-4}m^2\right)^{1/2}\left(-900^\circ C\right)=-517\,W$,

Eq. 3.76 and Table B.1 yield

$$q_f = M\tanh mL = -517\,W\left(0.983\right) = -508\,W$$

Hence, $q_b=-q_f=508\,W$ $\qquad <$

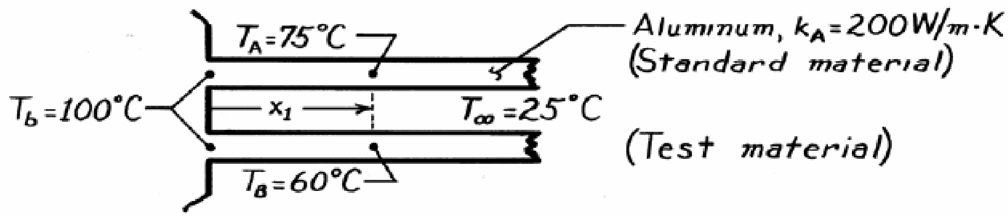
COMMENTS: Radiation losses from the blade surface and convection from the tip will contribute to reducing the blade temperatures.

PROBLEM 3.129

KNOWN: Base temperature, ambient fluid conditions, and temperatures at a prescribed distance from the base for two long rods, with one of known thermal conductivity.

FIND: Thermal conductivity of other rod.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state, (2) One-dimensional conduction along rods, (3) Constant properties, (4) Negligible radiation, (5) Negligible contact resistance at base, (6) Infinitely long rods, (7) Rods are identical except for their thermal conductivity.

ANALYSIS: With the assumption of infinitely long rods, the temperature distribution is

$$\frac{\theta}{\theta_b} = \frac{T - T_\infty}{T_b - T_\infty} = e^{-mx}$$

or

$$\ln \frac{T - T_\infty}{T_b - T_\infty} = -mx = \left[\frac{hP}{kA}\right]^{1/2} x$$

Hence, for the two rods,

$$\frac{\ln\left[\dfrac{T_A - T_\infty}{T_b - T_\infty}\right]}{\ln\left[\dfrac{T_B - T_\infty}{T_b - T_\infty}\right]} = \left[\frac{k_B}{k_A}\right]^{1/2}$$

$$k_B^{1/2} = k_A^{1/2} \frac{\ln\left[\dfrac{T_A - T_\infty}{T_b - T_\infty}\right]}{\ln\left[\dfrac{T_B - T_\infty}{T_b - T_\infty}\right]} = (200)^{1/2} \frac{\ln \dfrac{75-25}{100-25}}{\ln \dfrac{60-25}{100-25}} = 7.524$$

$$k_B = 56.6 \text{ W/m}\cdot\text{K}.$$ <

COMMENTS: Providing conditions for the two rods may be maintained nearly identical, the above method provides a convenient means of measuring the thermal conductivity of solids.

PROBLEM 3.134b

For copper (300K) : $k \cong 400$ W/m.K (Table 1.1)

For each fin :

$$m = \left(\frac{hP}{kA_c}\right)^{1/2} = \left[\frac{h\,\pi D}{k(\pi D^2/4)}\right]^{1/2} = \left(\frac{4h}{kD}\right)^{1/2} = \left(\frac{4\times 1000}{400\times 1.5\times 10^{-3}}\right)^{1/2} = 81.7\ m^{-1}$$

$$L_c = L + \frac{D}{4} = 15.75\times 10^{-3} \Rightarrow mL_c = 1.287$$

$$\eta_f = \frac{\tanh(mL_c)}{mL_c} = 0.667$$

$$q_{max} = h(\pi D L_c)(T_b - T_\infty) = 4.08\ W \Rightarrow q_f = \eta_f\, q_{max} = 2.7\ W$$

For all 16 fins: $q_{finned} = N q_f = 16\times 2.7 = 43.2\ W$

For unfinned surface: $q_{unfinned} = hA_{unfinned}(T_b - T_\infty) = h\left[A_{chip} - N\frac{\pi D^2}{4}\right](T_b - T_\infty)$

$$= (1000)\left[(12.7\times 10^{-3})^2 - (16)\frac{\pi(1.5\times 10^{-3})^2}{4}\right](75-20)$$

$$= 7.3\ W$$

Total heat transfer: $q_t = q_{finned} + q_{unfinned} = 43.2 + 7.3 = 50.5\ W$

Note:- Contact resistance and heat loss from the bottom of the board were small and therefore neglected in above calculations. If they are both accounted for, $q_t = 50.9$ W.

- Alternatively, one can calculate η_o and solve the problem that way

$$\eta_o = 1 - \frac{NA_f}{A_t}(1-\eta_f) = 1 - \frac{(16)(7.4\times 10^{-5})}{1.32\times 10^{-3}}(1-0.667) = 0.7$$

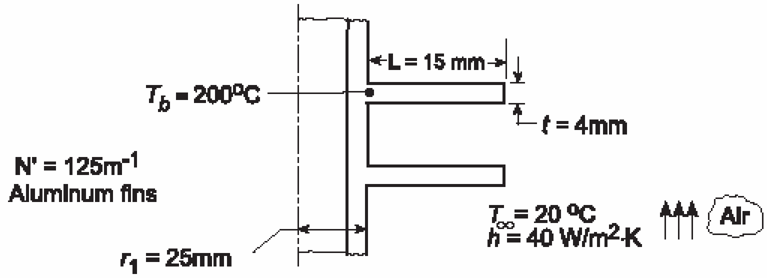
$$q_t = \eta_o h A_t (T_b - T_\infty) = 50.8\ W \quad \text{(same as above)}$$

PROBLEM 3.145

KNOWN: Dimensions and base temperature of aluminum fins of rectangular profile. Ambient air conditions.

FIND: (a) Fin efficiency and effectiveness, (b) Rate of heat transfer per unit length of tube.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional radial conduction in fins, (3) Constant properties, (4) Negligible radiation, (5) Negligible base contact resistance, (6) Uniform convection coefficient.

PROPERTIES: *Table A-1*, Aluminum, pure (T ≈ 400 K): k = 240 W/m·K.

ANALYSIS: (a) The fin parameters for use with Figure 3.19 are

$$r_{2c} = r_2 + t/2 = 40\text{ mm} + 2\text{ mm} = 0.042\text{ m} \qquad L_c = L + t/2 = 15\text{ mm} + 2\text{ mm} = 0.017\text{ m}$$

$$r_{2c}/r_1 = 0.042\text{ m}/0.025\text{ m} = 1.68 \qquad A_p = L_c t = 0.017\text{ m} \times 0.004\text{ m} = 6.8\times10^{-5}\text{ m}^2$$

$$L_c^{3/2}\left(h/kA_p\right)^{1/2} = \left(0.017\text{ m}\right)^{3/2}\left[40\text{ W/m}^2\cdot\text{K}/240\text{ W/m}\cdot\text{K}\times6.8\times10^{-5}\text{ m}^2\right]^{1/2} = 0.11$$

The fin efficiency is $\eta_f \approx 0.97$. From Eq. 3.86,

$$q_f = \eta_f q_{max} = \eta_f h A_{f(ann)}\theta_b = 2\pi\eta_f h\left[r_{2c}^2 - r_1^2\right]\theta_b$$

$$q_f = 2\pi\times0.97\times40\text{ W/m}^2\cdot\text{K}\left[\left(0.042\right)^2-\left(0.025\right)^2\right]\text{m}^2\times180^\circ\text{C} = 50\text{ W}$$ <

From Eq. 3.81, the fin effectiveness is

$$\varepsilon_f = \frac{q_f}{hA_{c,b}\theta_b} = \frac{50\text{ W}}{40\text{ W/m}^2\cdot\text{K}\;2\pi\left(0.025\text{ m}\right)\left(0.004\text{ m}\right)180^\circ\text{C}} = 11.05$$ <

(b) The rate of heat transfer per unit length is

$$q' = N'q_f + h\left(1-N't\right)\left(2\pi r_1\right)\theta_b$$

$$q' = 125\times50\text{ W/m} + 40\text{ W/m}^2\cdot\text{K}\left(1-125\times0.004\right)\left(2\pi\times0.025\text{ m}\right)\times180^\circ\text{C}$$

$$q' = \left(6250+565\right)\text{W/m} = 6.82\text{ kW/m}$$ <

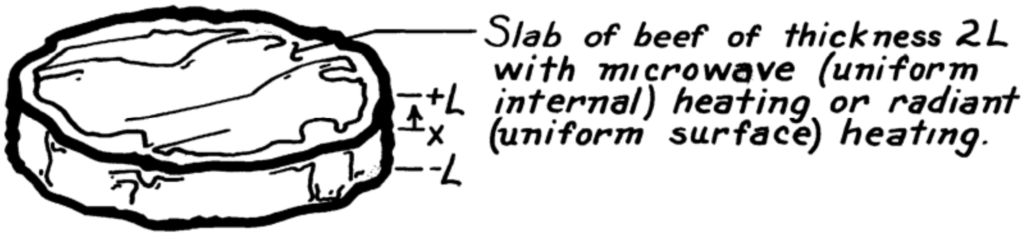
COMMENTS: Note the dominant contribution made by the fins to the total heat transfer.

PROBLEM 5.3

KNOWN: Microwave and radiant heating conditions for a slab of beef.

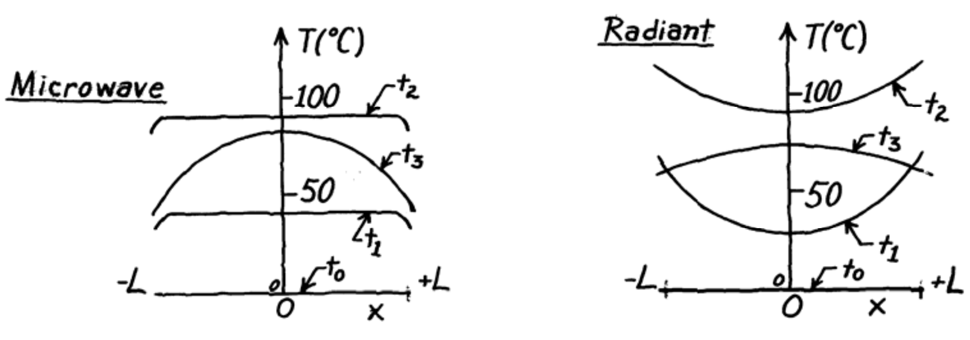
FIND: Sketch temperature distributions at specific times during heating and cooling.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional conduction in x, (2) Uniform internal heat generation for microwave, (3) Uniform surface heating for radiant oven, (4) Heat loss from surface of meat to surroundings is negligible during the heating process, (5) Symmetry about midplane.

ANALYSIS:



COMMENTS: (1) With uniform generation and negligible surface heat loss, the temperature distribution remains nearly uniform during *microwave heating.* During the subsequent surface cooling, the maximum temperature is at the midplane.

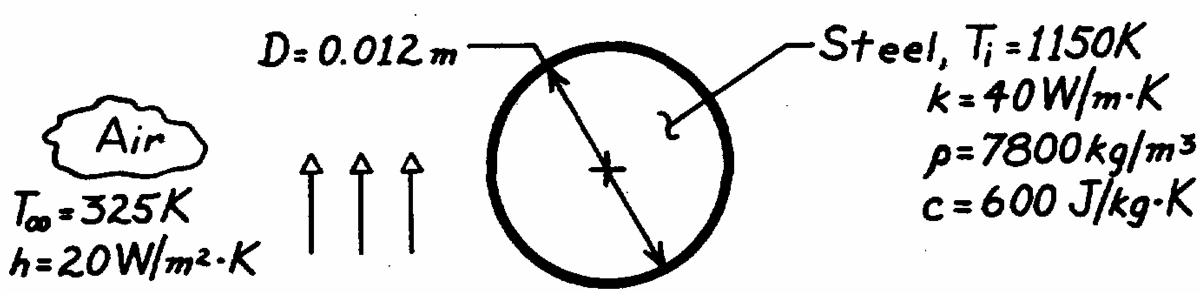
(2) The interior of the meat is heated by conduction from the hotter surfaces during *radiant heating,* and the lowest temperature is at the midplane. The situation is reversed shortly after cooling begins, and the maximum temperature is at the midplane.

PROBLEM 5.5

KNOWN: Diameter and initial temperature of steel balls cooling in air.

FIND: Time required to cool to a prescribed temperature.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible radiation effects, (2) Constant properties.

ANALYSIS: Applying Eq. 5.10 to a sphere ($L_c = r_o/3$),

$$Bi = \frac{hL_c}{k} = \frac{h(r_o/3)}{k} = \frac{20\ W/m^2 \cdot K\ (0.002m)}{40\ W/m \cdot K} = 0.001.$$

Hence, the temperature of the steel remains approximately uniform during the cooling process, and the lumped capacitance method may be used. From Eqs. 5.4 and 5.5,

$$t = \frac{\rho V c_p}{hA_s} \ln\frac{T_i - T_\infty}{T - T_\infty} = \frac{\rho\left(\pi D^3/6\right)c_p}{h\pi\ D^2} \ln\frac{T_i - T_\infty}{T - T_\infty}$$

$$t = \frac{7800kg/m^3\,(0.012m)\,600J/kg \cdot K}{6 \times 20\ W/m^2 \cdot K} \ln\frac{1150 - 325}{400 - 325}$$

$$t = 1122\ s\ = 0.312h$$ <

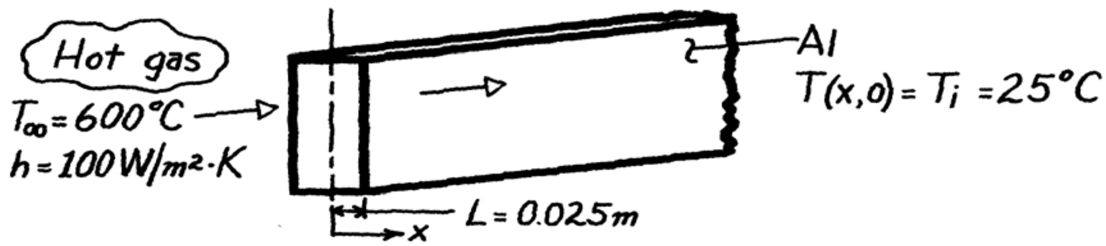
COMMENTS: Due to the large value of T_i, radiation effects are likely to be significant during the early portion of the transient. The effect is to shorten the cooling time.

PROBLEM 5.11

KNOWN: Configuration, initial temperature and charging conditions of a thermal energy storage unit.

FIND: Time required to achieve 75% of maximum possible energy storage. Temperature of storage medium at this time.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional conduction, (2) Constant properties, (3) Negligible radiation exchange with surroundings.

PROPERTIES: *Table A-1*, Aluminum, pure $\left(\bar{T} \approx 600\text{K} = 327^\circ\text{C}\right)$: k = 231 W/m·K, c = 1033 J/kg·K, ρ = 2702 kg/m^3.

ANALYSIS: Recognizing the characteristic length is the half thickness, find

$$\text{Bi} = \frac{hL}{k} = \frac{100\ \text{W/m}^2 \cdot \text{K} \times 0.025\text{m}}{231\ \text{W/m} \cdot \text{K}} = 0.011.$$

Hence, the lumped capacitance method may be used. From Eq. 5.8,

$$Q = (\rho V c)\theta_i \left[1 - \exp(-t/\tau_t)\right] = -\Delta E_{st} \tag{1}$$

$$-\Delta E_{st,max} = (\rho V c)\theta_i. \tag{2}$$

Dividing Eq. (1) by (2),

$$\Delta E_{st} / \Delta E_{st,max} = 1 - \exp(-t/\tau_{th}) = 0.75.$$

Solving for $\tau_{th} = \dfrac{\rho V c}{h A_s} = \dfrac{\rho L c}{h} = \dfrac{2702\ \text{kg/m}^3 \times 0.025\text{m} \times 1033\ \text{J/kg} \cdot \text{K}}{100\ \text{W/m}^2 \cdot \text{K}} = 698\text{s}.$

Hence, the required time is

$$-\exp(-t/698\text{s}) = -0.25 \qquad \text{or} \qquad t = 968\text{s}.$$ <

From Eq. 5.6,

$$\frac{T - T_\infty}{T_i - T_\infty} = \exp(-t/\tau_{th})$$

$$T = T_\infty + (T_i - T_\infty)\exp(-t/\tau_{th}) = 600^\circ\text{C} - \left(575^\circ\text{C}\right)\exp(-968/698)$$

$$T = 456^\circ\text{C}.$$ <

COMMENTS: For the prescribed temperatures, the property temperature dependence is significant and some error is incurred by assuming constant properties. However, selecting properties at 600K was reasonable for this estimate.

PROBLEM 5.43

a) If the surface temperature is suddenly reduced, this is analogous to $h \to \infty$ and $T_\infty = T_s$. Therefore, $Bi \to \infty$ or $Bi^{-1} = 0$.

$$\theta_o^* = \frac{T_o - T_\infty}{T_i - T_s} = 0.5 \;,\; Bi^{-1} = 0 \Rightarrow \underset{\text{(Heisler charts)}}{Fo \cong 0.38} \Rightarrow t = \frac{Fo L^2}{\alpha} \cong 63 \text{ s}.$$

b) Since we cannot use Heisler charts for the temperature gradient, use the one-term approximation.

$$\theta^* = C_1 e^{-\zeta_1^2 Fo} \cos(\zeta_1 x^*) \Rightarrow \frac{\partial \theta^*}{\partial x^*} = C_1 e^{-\zeta_1^2 Fo} (-\zeta_1) \sin(\zeta_1 x^*)$$

$$Bi = \infty \Rightarrow \zeta_1 = 1.5708 \;,\; C_1 = 1.2733 \text{ (Table 5.1)} \Rightarrow \left.\frac{\partial \theta^*}{\partial x^*}\right|_{x^*=1} = -0.78$$

$$\frac{\partial\left(\frac{T - T_\infty}{T_i - T_\infty}\right)}{\partial\left(\frac{x}{L}\right)} = \frac{L}{T_i - T_\infty} \frac{\partial T}{\partial x} \Rightarrow \left.\frac{\partial T}{\partial x}\right|_{x=L} = \frac{T_i - T_\infty}{L} \left.\frac{\partial \theta^*}{\partial x^*}\right|_{x^*=1} = \left(\frac{300}{0.01}\right)(-0.78) = -23{,}560 \frac{°C}{m}$$

PROBLEM 5.57

$$Bi = \frac{h(r_o/3)}{k} = \frac{(5000)(10\times10^{-3}/3)}{50} = 0.33 > 0.1 \Rightarrow \text{Heisler charts}$$

Again, swith to $Bi^{-1} = \frac{k}{r_o h} = 1.0$, $\frac{r}{r_o} = 1$ (surface) $\Rightarrow \frac{\theta}{\theta_o} = \frac{T - T_\infty}{T_o - T_\infty} = 0.6$

$$\Rightarrow \frac{1000 - 1300}{T_o - 1300} = 0.6 \Rightarrow T_o = 800\ K \Rightarrow \theta_o^* = \frac{T_o - T_\infty}{T_i - T_\infty} = \frac{800 - 1300}{300 - 1300} = 0.5$$

$$Bi^{-1} = 1.0,\ \theta_o^* = 0.5 \Rightarrow Fo \cong 0.4 \Rightarrow t = \frac{Fo.r_o^2}{\alpha} = \frac{Fo.r_o^2}{(k/\rho c_p)}$$

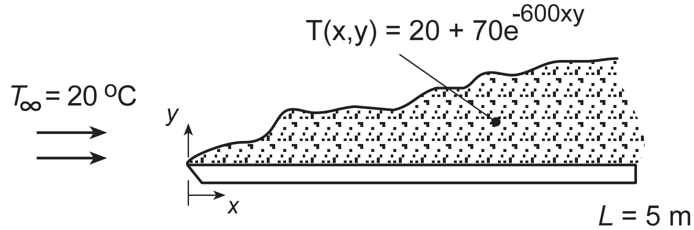
$$t = \frac{(0.4)(10\times10^{-3})^2}{[50/(7800)(500)]} = 3.1\ s$$

PROBLEM 6.8

KNOWN: Temperature distribution in boundary layer for air flow over a flat plate.

FIND: Variation of local convection coefficient along the plate and value of average coefficient.

SCHEMATIC:



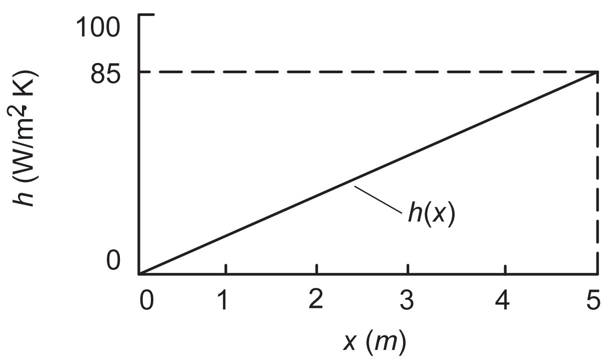
ANALYSIS: From Eq. 6.5,

$$h = -\frac{k\,\partial T/\partial y\big|_{y=0}}{\left(T_s - T_\infty\right)} = +\frac{k\left(70 \times 600x\right)}{\left(T_s - T_\infty\right)}$$

where $T_s = T(x,0) = 90°C$. Evaluating k at the arithmetic mean of the freestream and surface temperatures, $\bar{T} = (20 + 90)°C/2 = 55°C = 328\ K$, Table A.4 yields k = 0.0284 W/m·K. Hence, with $T_s - T_\infty = 70°C = 70\ K$,

$$h = \frac{0.0284\,W/m\cdot K\left(42,000x\right)K/m}{70\,K} = 17x\left(W/m^2\cdot K\right)$$ <

and the convection coefficient increases linearly with x.



The average coefficient over the range 0 ≤ x ≤ 5 m is

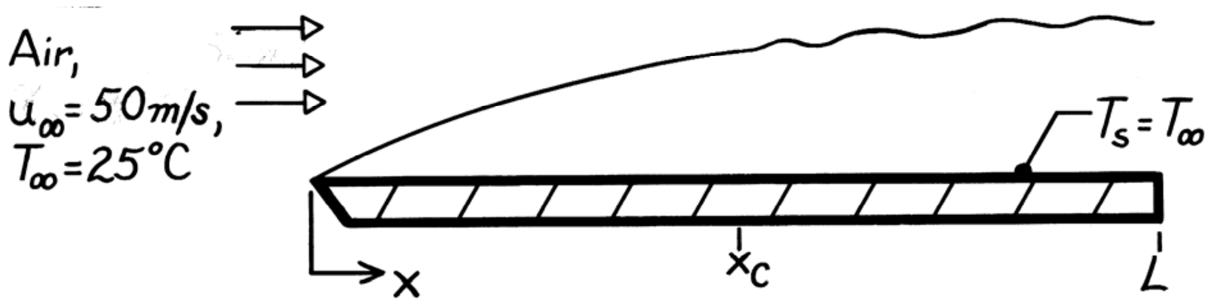
$$\bar{h} = \frac{1}{L}\int_0^L h\,dx = \frac{17}{5}\int_0^5 x\,dx = \frac{17}{5}\left.\frac{x^2}{2}\right|_0^5 = 42.5\,W/m^2\cdot K$$ <

PROBLEM 6.15

KNOWN: Air speed and temperature in a wind tunnel.

FIND: (a) Minimum plate length to achieve a Reynolds number of 10^8, (b) Distance from leading edge at which transition would occur.

SCHEMATIC:



ASSUMPTIONS: (1) Isothermal conditions, $T_s = T_\infty$.

PROPERTIES: *Table A-4*, Air (25°C = 298K): $\nu = 15.71 \times 10^{-6} m^2/s$.

ANALYSIS: (a) The Reynolds number is

$$Re_x = \frac{\rho u_\infty x}{\mu} = \frac{u_\infty x}{\nu}.$$

To achieve a Reynolds number of 1×10^8, the minimum plate length is then

$$L_{min} = \frac{Re_x \nu}{u_\infty} = \frac{1\times10^8 \left(15.71\times10^{-6} m^2/s\right)}{50 \text{ m/s}}$$

$$L_{min} = 31.4 \text{ m.}$$ <

(b) For a transition Reynolds number of 5×10^5

$$x_c = \frac{Re_{x,c} \nu}{u_\infty} = \frac{5\times10^5 \left(15.71\times10^{-6} m^2/s\right)}{50 \text{ m/s}}$$

$$x_c = 0.157 \text{ m.}$$ <

COMMENTS: Note that

$$\frac{x_c}{L} = \frac{Re_{x,c}}{Re_L}$$

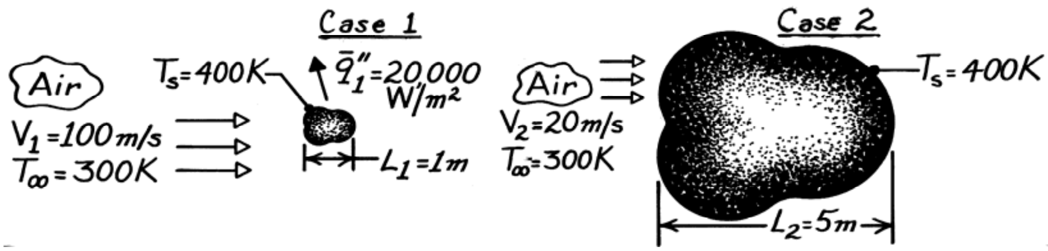
This expression may be used to quickly establish the location of transition from knowledge of $Re_{x,c}$ and Re_L.

PROBLEM 6.18

KNOWN: Characteristic length, surface temperature and average heat flux for an object placed in an airstream of prescribed temperature and velocity.

FIND: Average convection coefficient if characteristic length of object is increased by a factor of five and air velocity is decreased by a factor of five.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Constant properties.

ANALYSIS: For a particular geometry,

$$\overline{Nu}_L = f\left(Re_L, Pr\right).$$

The Reynolds numbers for each case are

Case 1: $$Re_{L,1} = \frac{V_1 L_1}{\nu_1} = \frac{(100\text{m/s})1\text{m}}{\nu_1} = \frac{100\ \text{m}^2/\text{s}}{\nu_1}$$

Case 2: $$Re_{L,2} = \frac{V_2 L_2}{\nu_2} = \frac{(20\text{m/s})5\text{m}}{\nu_2} = \frac{100\ \text{m}^2/\text{s}}{\nu_2}.$$

Hence, with $\nu_1 = \nu_2$, $Re_{L,1} = Re_{L,2}$. Since $Pr_1 = Pr_2$, it follows that

$$\overline{Nu}_{L,2} = \overline{Nu}_{L,1}.$$

Hence,

$$\bar{h}_2 L_2 / k_2 = \bar{h}_1 L_1 / k_1$$
$$\bar{h}_2 = \bar{h}_1 \frac{L_1}{L_2} = 0.2\ \bar{h}_1.$$

For *Case 1*, using the rate equation, the convection coefficient is

$$q_1 = \bar{h}_1 A_1 \left(T_s - T_\infty\right)_1$$
$$\bar{h}_1 = \frac{\left(q_1 / A_1\right)}{\left(T_s - T_\infty\right)_1} = \frac{q_1''}{\left(T_s - T_\infty\right)_1} = \frac{20,000\ \text{W/m}^2}{\left(400-300\right)\text{K}} = 200\ \text{W/m}^2 \cdot \text{K}.$$

Hence, it follows that for *Case 2*

$$\bar{h}_2 = 0.2 \times 200\ \text{W/m}^2 \cdot \text{K} = 40\ \text{W/m}^2 \cdot \text{K}.$$ <

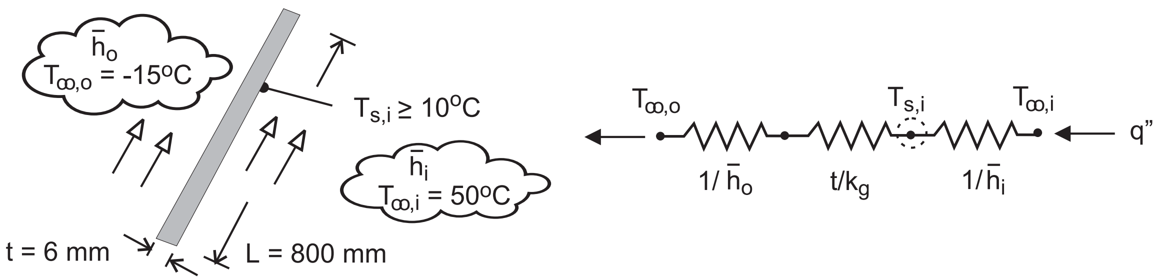
COMMENTS: If $Re_{L,2}$ were *not* equal to $Re_{L,1}$, it would be necessary to know the specific form of $f(Re_L, Pr)$ before $\bar{h}_2$ could be determined.

PROBLEM 6.32

KNOWN: Ambient, interior and dewpoint temperatures. Vehicle speed and dimensions of windshield. Heat transfer correlation for external flow.

FIND: Minimum value of convection coefficient needed to prevent condensation on interior surface of windshield.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state, (2) One-dimensional heat transfer, (3) Constant properties.

PROPERTIES: *Table A-3*, glass: k_g = 1.4 W/m·K. Prescribed, air: k = 0.023 W/m·K, ν = 12.5 × 10^{-6} m^2/s, Pr = 0.70.

ANALYSIS: From the prescribed thermal circuit, conservation of energy yields

$$\frac{T_{\infty,i} - T_{s,i}}{1/\bar{h}_i} = \frac{T_{s,i} - T_{\infty,o}}{t/k_g + 1/\bar{h}_o}$$

where $\bar{h}_o$ may be obtained from the correlation

$$\overline{Nu}_L = \frac{\bar{h}_o L}{k} = 0.030\,Re_L^{0.8}\,Pr^{1/3}$$

With V = (70 mph × 1585 m/mile)/3600 s/h = 30.8 m/s, Re_D = (30.8 m/s × 0.800 m)/12.5 × 10^{-6} m^2/s = 1.97 × 10^6 and

$$\bar{h}_o = \frac{0.023\,W/m\cdot K}{0.800\ m} 0.030\left(1.97\times10^6\right)^{0.8}(0.70)^{1/3} = 83.1\,W/m^2\cdot K$$

From the energy balance, with $T_{s,i} = T_{dp}$ = 10°C

$$\bar{h}_i = \frac{\left(T_{s,i} - T_{\infty,o}\right)}{\left(T_{\infty,i} - T_{s,i}\right)}\left(\frac{t}{k_g} + \frac{1}{\bar{h}_o}\right)^{-1}$$

$$\bar{h}_i = \frac{(10+15)°C}{(50-10)°C}\left(\frac{0.006\ m}{1.4\,W/m\cdot K} + \frac{1}{83.1\,W/m^2\cdot K}\right)^{-1}$$

$$\bar{h}_i = 38.3\,W/m^2\cdot K$$ <

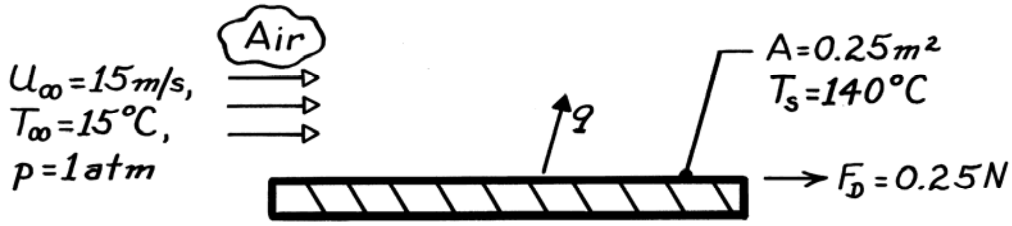
COMMENTS: The output of the fan in the automobile's heater/defroster system must maintain a velocity for flow over the inner surface that is large enough to provide the foregoing value of $\bar{h}_i$. In addition, the output of the heater must be sufficient to maintain the prescribed value of $T_{\infty,i}$ at this velocity.

PROBLEM 6.35

KNOWN: Air flow conditions and drag force associated with a heater of prescribed surface temperature and area.

FIND: Required heater power.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Reynolds analogy is applicable, (3) Bottom surface is adiabatic.

PROPERTIES: *Table A-4*, Air (T_f = 350K, 1atm): ρ = 0.995 kg/m^3, c_p = 1009 J/kg·K, Pr = 0.700.

ANALYSIS: The average shear stress and friction coefficient are

$$\overline{\tau}_s = \frac{F_D}{A} = \frac{0.25\text{ N}}{0.25\text{ m}^2} = 1\text{ N/m}^2$$

$$\overline{C}_f = \frac{\overline{\tau}_s}{\rho\, u_\infty^2/2} = \frac{1\text{ N/m}^2}{0.995\text{ kg/m}^3(15\text{m/s})^2/2} = 8.93\times10^{-3}.$$

From the Reynolds analogy,

$$\overline{St} = \frac{\overline{h}}{\rho\, u_\infty c_p} = \frac{\overline{C}_f}{2}\text{Pr}^{-2/3}.$$

Solving for $\overline{h}$ and substituting numerical values, find

$$\overline{h} = 0.995\text{ kg/m}^3(15\text{m/s})\ 1009\text{ J/kg}\cdot\text{K}\left(8.93\times10^{-3}/2\right)\ (0.7)^{-2/3}$$

$$\overline{h} = 85\text{ W/m}^2\cdot\text{K}.$$

Hence, the heat rate is

$$q = \overline{h}\,A\,(T_s - T_\infty) = 85\text{W/m}^2\cdot\text{K}\left(0.25\text{m}^2\right)\ (140-15)^\circ\text{C}$$

$$q = 2.66\text{ kW}.$$ <

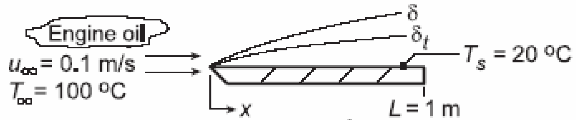
COMMENTS: Due to bottom heat losses, which have been assumed negligible, the actual power requirement would exceed 2.66 kW.

PROBLEM 7.2

KNOWN: Temperature and velocity of engine oil. Temperature and length of flat plate.

FIND: (a) Velocity and thermal boundary layer thickness at trailing edge, (b) Heat flux and surface shear stress at trailing edge, (c) Total drag force and heat transfer per unit plate width, and (d) Plot the boundary layer thickness and local values of the shear stress, convection coefficient, and heat flux as a function of x for $0 \le x \le 1$ m.

SCHEMATIC:



ASSUMPTIONS: (1) Critical Reynolds number is 5×10^5, (2) Flow over top and bottom surfaces.

PROPERTIES: *Table A.5*, Engine Oil (T_f = 333 K): $\rho = 864$ kg/m^3, $\nu = 86.1 \times 10^{-6}$ m^2/s, k = 0.140 W/m·K, Pr = 1081.

ANALYSIS: (a) Calculate the Reynolds number to determine nature of the flow,

$$Re_L = \frac{u_\infty L}{\nu} = \frac{0.1\,m/s \times 1\,m}{86.1\times10^{-6}\,m^2/s} = 1161$$

Hence the flow is laminar at x = L. From Eqs. 7.19 and 7.24,

$$\delta = 5L\,Re_L^{-1/2} = 5(1\,m)(1161)^{-1/2} = 0.147\,m \qquad <$$

$$\delta_t = \delta\,Pr^{-1/3} = 0.147\,m(1081)^{-1/3} = 0.0143\,m \qquad <$$

(b) The local convection coefficient, Eq. 7.23, and heat flux at x = L are

$$h_L = \frac{k}{L}0.332\,Re_L^{1/2}\,Pr^{1/3} = \frac{0.140\,W/m\cdot K}{1\,m}0.332(1161)^{1/2}(1081)^{1/3} = 16.25\,W/m^2\cdot K$$

$$q''_x = h_L(T_s - T_\infty) = 16.25\,W/m^2\cdot K(20-100)^\circ C = -1300\,W/m^2 \qquad <$$

Also, the local shear stress is, from Eq. 7.20,

$$\tau_{s,L} = \frac{\rho u_\infty^2}{2}0.664\,Re_L^{-1/2} = \frac{864\,kg/m^3}{2}(0.1\,m/s)^2\,0.664(1161)^{-1/2}$$

$$\tau_{s,L} = 0.0842\,kg/m\cdot s^2 = 0.0842\,N/m^2 \qquad <$$

(c) With the drag force per unit width given by $D' = 2L\bar{\tau}_{s,L}$ where the factor of 2 is included to account for both sides of the plate, it follows from Eq. 7.29 that

$$D' = 2L\left(\rho u_\infty^2/2\right)1.328\,Re_L^{-1/2} = (1\,m)864\,kg/m^3\,(0.1\,m/s)^2\,1.328(1161)^{-1/2} = 0.337\,N/m \qquad <$$

For laminar flow, the average value $\bar{h}_L$ over the distance 0 to L is twice the local value, h_L,

$$\bar{h}_L = 2h_L = 32.5\,W/m^2\cdot K$$

The total heat transfer rate per unit width of the plate is

$$q' = 2L\bar{h}_L(T_s - T_\infty) = 2(1\,m)32.5\,W/m^2\cdot K(20-100)^\circ C = -5200\,W/m \qquad <$$

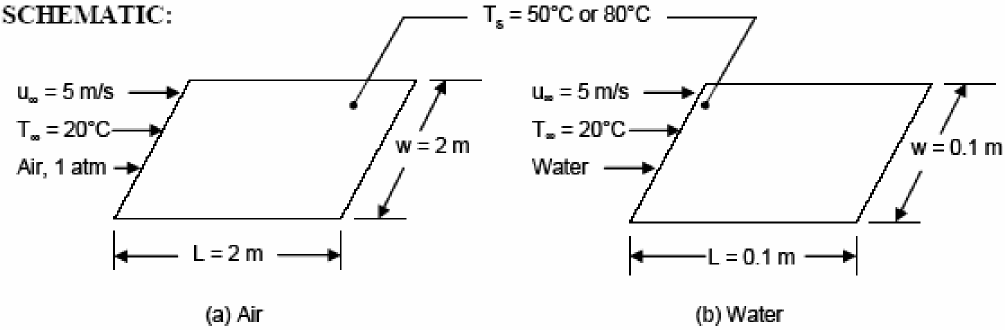
Continued...

PROBLEM 7.11

KNOWN: Dimensions and surface temperatures of a flat plate. Velocity and temperature of air and water flow parallel to the plate.

FIND: (a) Average convective heat transfer coefficient, convective heat transfer rate, and drag force when L = 2 m, w = 2 m. (b) Average convective heat transfer coefficient, convective heat transfer rate, and drag force when L = 0.1 m, w = 0.1 m.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Boundary layer assumptions are valid, (3) Constant properties, (4) Transition Reynolds number is 5×10^5.

PROPERTIES: Using *IHT*, Air (p = 1 atm, T_f = 35°C = 308 K): Pr = 0.706, k = 26.9×10^{-3} W/m·K, $\nu = 1.669 \times 10^{-5}$ m²/s, ρ = 1.135 kg/m³. Air (p = 1 atm, T_f = 50°C = 323 K): Pr = 0.704, k = 28.0×10^{-3} W/m·K, $\nu = 1.82 \times 10^{-5}$ m²/s, ρ = 1.085 kg/m³. Water (T_f = 308 K): Pr = 4.85, k = 0.625 W/m·K, $\nu = 7.291 \times 10^{-7}$ m²/s, ρ = 994 kg/m³. Water (T_f = 323 K): Pr = 3.56, k = 0.643 W/m·K, $\nu = 5.543 \times 10^{-7}$ m²/s, ρ = 988 kg/m³.

ANALYSIS:

(a) We begin by calculating the Reynolds numbers for the two different surface temperatures:

$$Re_{L1} = \frac{u_\infty L}{\nu_1} = \frac{5 \text{ m/s} \times 2 \text{ m}}{1.669 \times 10^{-5} \text{ m}^2/\text{s}} = 5.99 \times 10^5$$

$$Re_{L2} = \frac{u_\infty L}{\nu_2} = \frac{5 \text{ m/s} \times 2 \text{ m}}{1.82 \times 10^{-5} \text{ m}^2/\text{s}} = 5.49 \times 10^5$$

Therefore, in both cases the flow is turbulent at the end of the plate and the conditions in the boundary layer are "mixed."

The average drag coefficient can be calculated from Equation 7.40. For the first case,

$$\overline{C}_{f,L1} = 0.074\, Re_{L1}^{-1/5} - 1742\, Re_{L1}^{-1}$$

$$= 0.074(5.99 \times 10^5)^{-1/5} - 1742(5.99 \times 10^5)^{-1} = 2.27 \times 10^{-3}$$

Then

$$F_{D1} = \overline{C}_{f,L1} \frac{1}{2} \rho u_\infty^2 A_s$$

$$= 2.27 \times 10^{-3} \times \frac{1}{2} \times 1.135 \text{ kg/m}^3 \times (5 \text{ m/s})^2 \times 8 \text{ m}^2 = 0.257 \text{ N}$$

$$= 0.257 \text{ N}$$ <

Continued….

PROBLEM 7.11 (Cont.)

The average Nusselt number is calculated from Equation 7.38, with A = 871 for a transition Reynolds number of 5×10^5.

$$\overline{Nu}_{L1} = (0.037\,Re_L^{4/5} - 871)\,Pr^{1/3}$$
$$= \left[0.037(5.99 \times 10^5)^{4/5} - 871\right](0.706)^{1/3} = 604.$$

Then

$$\overline{h}_{L1} = \overline{Nu}_{L1} k/L = 604 \times 26.9 \times 10^{-3}\ \text{W/m}\cdot\text{K}/2\ \text{m} = 8.13\ \text{W/m}^2\cdot\text{K} \qquad <$$

and

$$q_1 = \overline{h}_{L1} A_s (T_s - T_\infty) = 8.13\ \text{W/m}^2\cdot\text{K} \times 8\ \text{m}^2 \times (50°\text{C} - 20°\text{C}) = 1950\ \text{W} \qquad <$$

Similarly for $T_s = 80°C$ we find

$F_{D2} = 0.227$ N, $\overline{h}_{L2} = 7.16$ W/m^2·K, $q_2 = 3440$ W

<

(b) Repeating the calculations for water

$$Re_{L1} = \frac{u_\infty L}{\nu} = \frac{5\ \text{m/s} \times 0.1\ \text{m}}{7.291 \times 10^{-7}\ \text{m}^2/\text{s}} = 6.86 \times 10^5$$

$$Re_{L2} = 9.02 \times 10^5$$

The flow is turbulent at the end of the plate in both cases.

$$\overline{C}_{f,L1} = 0.074(6.86 \times 10^5)^{-1/5} - 1742(6.86 \times 10^5)^{-1} = 2.49 \times 10^{-3}$$

$$F_{D1} = 2.49 \times 10^{-3} \times 1/2 \times 994\ \text{kg/m}^3 \times (5\ \text{m/s})^2 \times 0.02\ \text{m}^2 = 0.620\ \text{N} \qquad <$$

$$\overline{Nu}_L = \left[0.037(6.86 \times 10^5)^{4/5} - 871\right](4.85)^{1/3} = 1450$$

$$\overline{h}_{L1} = 1450 \times 0.625\ \text{W/m}\cdot\text{K}/0.1\ \text{m} = 9050\ \text{W/m}^2\cdot\text{K} \qquad <$$

$$q_1 = 9050\ \text{W/m}^2\cdot\text{K} \times 0.02\ \text{m}^2 \times (50°\text{C} - 20°\text{C}) = 5430\ \text{W} \qquad <$$

For the higher surface temperature,

$F_{D2} = 0.700$ N, $\overline{h}_{L2} = 12{,}600$ W/m^2·K, $q_2 = 15{,}100$ W <

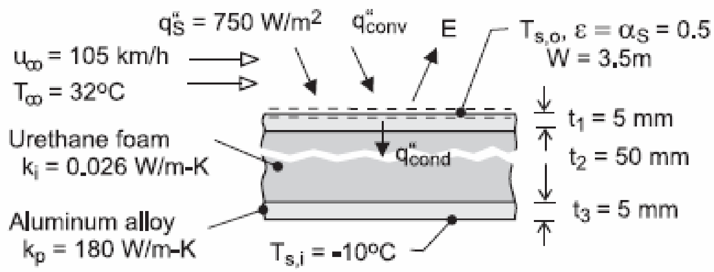
COMMENTS: (1) For air, kinematic viscosity increases with increasing temperature. This decreases the Reynolds number which causes the transition to turbulence to move downstream, thereby decreasing the drag force and average heat transfer coefficient. The heat transfer rate increases for the higher surface temperature, however, because of the greater temperature difference between the surface and air. (2) For water, kinematic viscosity decreases with increasing temperature, causing the opposite trends as for air. The heat transfer rate increases dramatically for the higher surface temperature because of the increases in both the heat transfer coefficient and temperature difference. (3) Even though the water flows over a plate that is 400 times smaller, the drag force and heat transfer rate are larger than for air because of the smaller viscosity and greater density, thermal conductivity, and Prandtl number. The discrepancy is particularly great for the hear transfer rate. (4) The problem highlights the importance of carefully accounting for the temperature dependence of thermal properties.

PROBLEM 7.20

KNOWN: Material properties, inner surface temperature and dimensions of roof of refrigerated truck compartment. Truck speed and ambient temperature. Solar irradiation.

FIND: (a) Outer surface temperature of roof and rate of heat transfer to compartment, (b) Effect of changing radiative properties of outer surface, (c) Effect of eliminating insulation.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible irradiation from the sky, (2) Turbulent flow over entire outer surface, (3) Average convection coefficient may be used to estimate average surface temperature, (4) Constant properties.

PROPERTIES: *Table A-4*, air (p = 1 atm, $T_f \approx 300$K): $\nu = 15.89 \times 10^{-6}$ m^2/s, k = 0.0263 W/m·K, Pr = 0.707.

ANALYSIS: (a) From an energy balance for the outer surface,

$$\alpha_S G_S + q''_{conv} - E = q''_{cond} = \frac{T_{s,o} - T_{s,i}}{R''_{tot}}$$

$$\alpha_S G_S + \bar{h}\left(T_\infty - T_{s,o}\right) - \varepsilon\sigma T_{s,o}^4 = \frac{T_{s,o} - T_{s,i}}{2R''_p + R''_i}$$

where $R''_p = \left(t_1 / k_p\right) = 2.78 \times 10^{-5}\,\text{m}^2 \cdot \text{K/W}$, $R''_i = \left(t_2 / k_i\right) = 1.923\,\text{m}^2 \cdot \text{K/W}$, and with $Re_L = u_\infty L/\nu$ $= 29.2\,\text{m/s} \times 10\text{m} / 15.89 \times 10^{-6}\,\text{m}^2/\text{s} = 1.84 \times 10^7$,

$$\bar{h} = \frac{k}{L} 0.037\, Re_L^{4/5}\, Pr^{1/3} = \frac{0.0263\,\text{W/m}\cdot\text{K}}{10\text{m}} 0.037\left(1.84 \times 10^7\right)^{4/5} \left(0.707\right)^{1/3} = 56.2\,\text{W/m}^2 \cdot \text{K}$$

Hence,

$$0.5\left(750\,\text{W/m}^2 \cdot \text{K}\right) + 56.2\,\text{W/m}^2 \cdot \text{K}\left(305 - T_{s,o}\right) - 0.5 \times 5.67 \times 10^{-8}\,\text{W/m}^2 \cdot \text{K}^4\, T_{s,o}^4 = \frac{T_{s,o} - 263\text{K}}{\left(5.56 \times 10^{-5} + 1.923\right)\text{m}^2 \cdot \text{K/W}}$$

Solving, we obtain

$$T_{s,o} = 306.8\text{K} = 33.8°\text{C} \qquad <$$

Hence, the heat load is

$$q = \left(W \cdot L\right) q''_{cnd} = \left(3.5\text{m} \times 10\text{m}\right) \frac{(33.8 + 10)°\text{C}}{1.923\,\text{m}^2 \cdot \text{K/W}} = 797\,\text{W} \qquad <$$

(b) With the special surface finish $\left(\alpha_S = 0.15,\ \varepsilon = 0.8\right)$,

Continued …..

PROBLEM 7.20 (Cont.)

$$T_{s,o} = 300.1\text{K} = 27.1°\text{C}$$ <

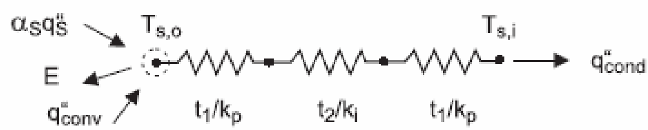
$$q = 675.3\text{W}$$ <

(c) Without the insulation ($t_2 = 0$) and with $\alpha_S = \varepsilon = 0.5$,

$$T_{s,o} = 263.1\text{K} = -9.9°\text{C}$$ <

$$q = 90,630\text{W}$$ <

COMMENTS: (1) Use of the special surface finish reduces the solar input, while increasing radiation emission from the surface. The cumulative effect is to reduce the heat load by 15%. (2) The thermal resistance of the aluminum panels is negligible, and without the insulation, the heat load is *enormous*.

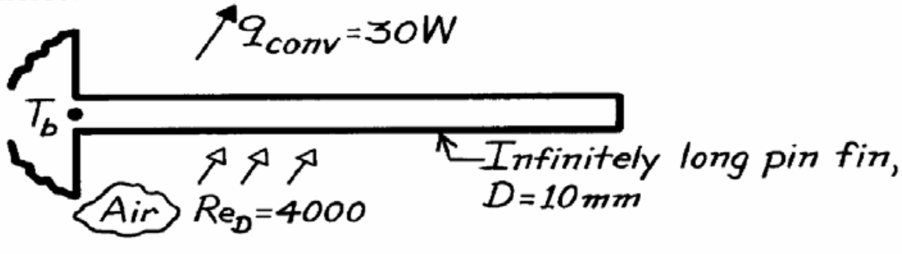


PROBLEM 7.45

KNOWN: Pin fin of 10 mm diameter dissipates 30 W by forced convection in cross-flow of air with $Re_D = 4000$.

FIND: Fin heat rate if diameter is doubled while all conditions remain the same.

SCHEMATIC:



ASSUMPTIONS: (1) Pin behaves as infinitely long fin, (2) Conditions of flow, as well as base and air temperatures, remain the same for both situations, (3) Negligible radiation heat transfer.

ANALYSIS: For an infinitely long pin fin, the fin heat rate is

$$q_f = q_{conv} = \left(\overline{h}PkA_c\right)^{1/2}\theta_b$$

where $P = \pi D$ and $A_c = \pi D^2/4$. Hence,

$$q_{conv} \sim \left(\overline{h}\cdot D\cdot D^2\right)^{1/2}.$$

For forced convection cross-flow over a cylinder, an appropriate correlation for estimating the dependence of $\overline{h}$ on the diameter is

$$\overline{Nu}_D = \frac{\overline{h}D}{k} = CRe_D^m\ Pr^{1/3} = C\left(\frac{VD}{\nu}\right)^m Pr^{1/3}.$$

From Table 7.2 for $Re_D = 4000$, find m = 0.466 and

$$\overline{h}\sim D^{-1}(D)^{0.466} = D^{-0.534}.$$

It follows that

$$q_{conv} \sim \left(D^{-0.534}\cdot D\cdot D^2\right)^{1/2} = D^{1.23}.$$

Hence, with $q_1 \rightarrow D_1$ (10 mm) and $q_2 \rightarrow D_2$ (20 mm), find

$$q_2 = q_1\left(\frac{D_2}{D_1}\right)^{1.23} = 30\text{ W}\left(\frac{20}{10}\right)^{1.23} = 70.4\text{ W}.$$ <

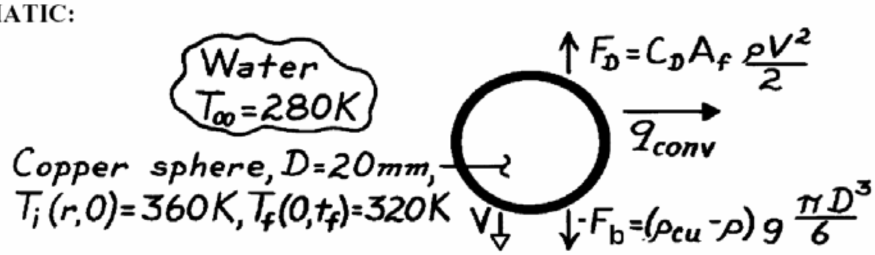
COMMENTS: The effect of doubling the diameter, with all other conditions remaining the same, is to increase the fin heat rate by a factor of 2.35. The effect is nearly linear, with enhancements due to the increase in surface and cross-sectional areas ($D^{1.5}$) exceeding the attenuation due to a decrease in the heat transfer coefficient ($D^{-0.267}$). Note that, with increasing Reynolds number, the exponent m increases and there is greater heat transfer enhancement due to increasing the diameter.

PROBLEM 7.73

KNOWN: Diameter and initial and final temperatures of copper spheres quenched in a water bath.

FIND: (a) Terminal velocity in the bath, (b) Tank height.

SCHEMATIC:



ASSUMPTIONS: (1) Sphere descends at terminal velocity, (2) Uniform, but time varying surface, temperature.

PROPERTIES: *Table A-1*, Copper (350K): $\rho = 8933$ kg/m^3, k = 398 W/m·K, $c_p = 387$ J/kg·K; *Table A-6*, Water ($T_\infty = 280$ K): $\rho = 1000$ kg/m^3, $\mu = 1422 \times 10^{-6}$ N·s/m^2, k = 0.582 W/m·K, Pr = 10.26; ($T_s \approx 340$ K): $\mu_s = 420 \times 10^{-5}$ N·s/m^2.

ANALYSIS: A force balance gives $C_D\left(\pi D^2/4\right)\rho V^2/2 = \left(\rho_{cu} - \rho\right) g\, \pi D^3/6$,

$$C_D\, V^2 = \frac{4D}{3}\frac{\rho_{cu} - \rho}{\rho} g = \frac{4 \times 0.02\text{ m}}{3} \cdot \frac{8933 - 1000}{1000} 9.8\text{ m/s}^2 = 2.07\text{ m}^2/\text{s}^2.$$

An iterative solution is needed, where C_D is obtained from Figure 7.8 with $Re_D = VD/\nu = 0.02$ m V/1.42 × 10^{-6} m^2/s = 14,085 V (m/s). Convergence is achieved with

$$V \approx 2.1\text{ m/s} \qquad <$$

for which Re_D = 29,580 and $C_D \approx 0.46$. Using the Whitaker expression

$$\overline{Nu}_D = 2 + \left(0.4 \times 29{,}850^{1/2} + 0.06 \times 29{,}850^{2/3}\right)(10.26)^{0.4}(1422/420)^{1/4} = 439$$

$$\overline{h} = \overline{Nu}_D\ k/D = 439 \times 0.582\text{ W/m}\cdot\text{K}/0.02\text{ m} = 12{,}775\text{ W/m}^2\cdot\text{K}.$$

To determine applicability of lumped capacitance method, find $Bi = \overline{h}\left(r_o/3\right)/k_{cu} = 12{,}775$ W/m$^2\cdot$K $(0.01\text{ m}/3)/398\text{W/m}\cdot\text{K} = 0.11$. Applicability is marginal. Using Eq. 5.50c, $\theta_o^* = C_1 \exp(-\xi_1^2 Fo)$ and from Table 5.1 at $B_i = \overline{h}\, r_o/k = 0.32$, $C_1 = 1.0937$, $\xi_1 = 0.9472$. Substituting into the preceding equation yields

$0.5 = 1.0937 \exp(-0.94722\ Fo)$ from which

$Fo = 0.87 = \alpha t_f / r_o^2$

With $\alpha_{cu} = k/\rho c_p$ = 398 W/m·K/(8933 kg/m^3) (387 J/kg·K) = 1.15 × 10^{-4} m^2/s, find

$$t_f = 0.87\ (0.01\text{ m})^2/1.15 \times 10^{-4}\text{m}^2/\text{s} = 0.76\text{ s}.$$

Required tank height is

$$H = t_f \cdot V = 0.76\text{ s} \times 2.1\text{ m/s} = 1.6\text{ m}. \qquad <$$

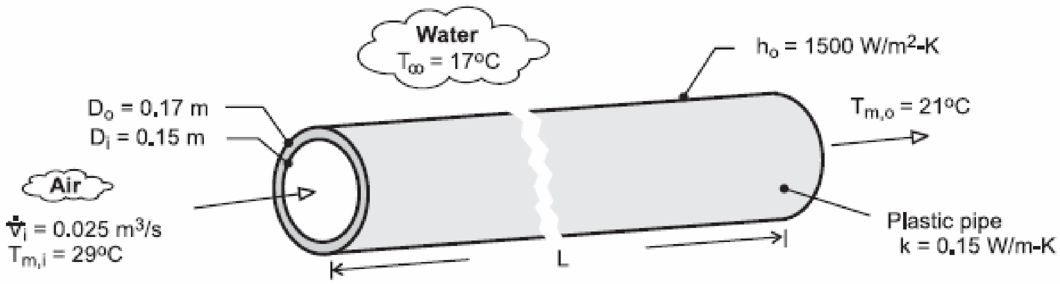
COMMENTS: Note that the terminal velocity is not reached immediately. Reduced V implies reduced $\overline{h}$ and increased t_f. The Fourier number, Fo, is greater than 0.2. Hence, use of Eq. 5.50c is justified.

PROBLEM 8.31

KNOWN: Thermal conductivity and inner and outer diameters of plastic pipe. Volumetric flow rate and inlet and outlet temperatures of air flow through pipe. Convection coefficient and temperature of water.

FIND: Pipe length and fan power requirement.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state, (2) Negligible heat transfer from air in vertical legs of pipe, (3) Ideal gas with negligible viscous dissipation and pressure variation, (4) Smooth interior surface, (5) Constant properties.

PROPERTIES: *Table A-4*, Air ($T_{m,i}$ = 29°C): $\rho_i = 1.155\,kg/m^3$. Air $(\overline{T}_m = 25°C)$: $c_p = 1007$ J/kg·K, $\mu = 183.6 \times 10^{-7}$ N·s/m^2, $k_a = 0.0261$ W/m·K, Pr = 0.707.

ANALYSIS: From Eq. (8.45a)

$$\frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}} = \exp\left(-\frac{\overline{U}A_s}{\dot{m}\,c_p}\right)$$

where, from Eq. (3.32),
$$\left(\overline{U}A_s\right)^{-1} = R_{tot} = \frac{1}{\overline{h}_i \pi D_i L} + \frac{\ln\left(D_o/D_i\right)}{2\pi L k} + \frac{1}{h_o \pi D_o L}$$

With $\dot{m} = \rho_i \dot{\forall}_i = 0.0289\,kg/s$ and $Re_D = 4\dot{m}/\pi D_i \mu = 13{,}350$, flow in the pipe is turbulent. Assuming fully developed flow throughout the pipe, and from Eq. (8.60),

$$\overline{h}_i = \frac{k_a}{D_i} 0.023\,Re_D^{4/5}\,Pr^{0.3} = \frac{0.0261\,W/m\cdot K \times 0.023}{0.15m}\left(13,350\right)^{4/5}\left(0.707\right)^{0.3} = 7.20\,W/m^2\cdot K$$

$$\left(\overline{U}A_s\right)^{-1} = \frac{1}{L}\left(\frac{1}{7.21\,W/m^2\cdot K \times \pi \times 0.15m} + \frac{\ln\left(0.17/0.15\right)}{2\pi \times 0.15\,W/m\cdot K} + \frac{1}{1500\,W/m^2\cdot K \times \pi \times 0.17m}\right)$$

$$\overline{U}A_s = \frac{L}{\left(0.294 + 0.133 + 0.001\right)} = 2.335\,L\ \ W/K$$

$$\frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}} = \frac{17-21}{17-29} = 0.333 = \exp\left(-\frac{2.335\,L}{0.0289\,kg/s \times 1007\,J/kg\cdot K}\right) = \exp\left(-0.0802L\right)$$

$$L = -\frac{\ln\left(0.333\right)}{0.0802} = 13.7m$$ <

From Eqs. (8.22a) and (8.22b) and with $u_{m,i} = \dot{\forall}_i / \left(\pi D_i^2/4\right) = 1.415\,m/s$, the fan power is

$$P = \left(\Delta p\right)\dot{\forall} \approx f\,\frac{\rho_i u_{m,i}^2}{2D_i} L\,\dot{\forall}_i = 0.0294\frac{1.155\,kg/m^3\left(1.415\,m/s\right)^2}{2\left(0.15m\right)} 13.7m \times 0.025\,m^3/s = 0.078\,W$$ <

where $\qquad f = 0.316\,Re_D^{-1/4} = 0.0294$ from Eq. (8.20a).

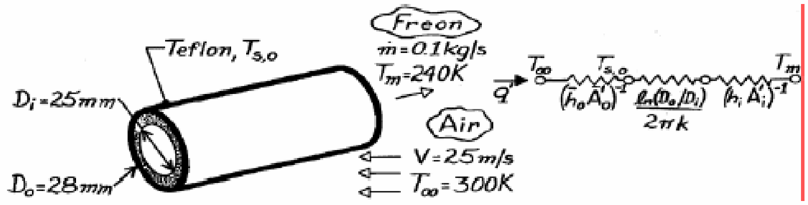
COMMENTS: (1) With L/Di = 91, the assumption of fully developed flow throughout the pipe is justified. (2) The fan power requirement is small, and the process is economical. (3) The resistance to heat transfer associated with convection at the outer surface is negligible.

PROBLEM 8.50

KNOWN: Flow rate and temperature of Refrigerant-134a passing through a Teflon tube of prescribed inner and outer diameter. Velocity and temperature of air in cross flow over tube.

FIND: Heat transfer per unit tube length.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional radial conduction, (3) Constant properties, (4) Fully developed flow.

PROPERTIES: *Table A-4*, Air (T = 300K, 1 atm): $\nu = 15.89 \times 10^{-6}$ m^2/s, k = 0.0263 W/m·K, Pr = 0.707; *Table A-5*, R-134a (T = 240K): $\mu = 4.202 \times 10^{-4}$ N·s/m^2, k = 0.1073 W/m·K, Pr = 5.0; *Table A-3*, Teflon (T ≈ 300K): k = 0.35 W/m·K.

ANALYSIS: Considering the thermal circuit shown above, the heat rate is

$$q' = \frac{T_\infty - T_m}{\left(1/\bar{h}_o \pi D_o\right) + \left[\ell n\left(D_o / D_i\right)/2\pi k\right] + \left(1/h_i \pi D_i\right)}.$$

$$Re_{D,i} = \frac{4\dot{m}}{\pi D_i \mu} = \frac{0.4 \text{ kg/s}}{\pi(0.025\text{m})\ 4.202\times10^{-4} \text{ N}\cdot\text{s/m}^2} = 12{,}120$$

and the flow is turbulent. Hence, from the Dittus-Boelter correlation

$$h_i = \frac{k}{D_i} 0.023\, Re_{D,i}^{4/5} Pr^{0.4} = \frac{0.1073 \text{ W/m}\cdot\text{K}}{0.025 \text{ m}} 0.023(12{,}120)^{4/5}(5)^{0.4} = 347 \text{ W/m}^2\cdot\text{K}.$$

With $\quad Re_{D,o} = \dfrac{VD_o}{\nu} = \dfrac{(25 \text{ m/s})0.028 \text{ m}}{15.89\times10^{-6} \text{ m}^2/\text{s}} = 4.405\times10^4$

it follows from Eq. 7.53 and Table 7.4 that

$$\bar{h}_o = \frac{k}{D} 0.26\, Re_{D,o}^{0.6} Pr^{0.37} = \frac{0.0263 \text{ W/m}\cdot\text{K}}{0.028 \text{ m}} 0.26\left(4.405\times10^4\right)^{0.6}(0.707)^{0.37} = 131 \text{ W/m}^2\cdot\text{K}.$$

Hence

$$q' = \frac{T_\infty - T_m}{\left(131 \text{ W/m}^2\cdot\text{K}\pi 0.028 \text{ m}\right)^{-1} + \ell n(28/25)/2\pi(0.350 \text{ W/m}\cdot\text{K}) + \left(347 \text{ W/m}^2\cdot\text{K}\pi 0.025 \text{ m}\right)^{-1}}$$

$$q' = \frac{(300-240)\text{K}}{(0.087+0.052+0.037)\ \text{K}\cdot\text{m/W}} = 343 \text{ W/m}. \qquad <$$

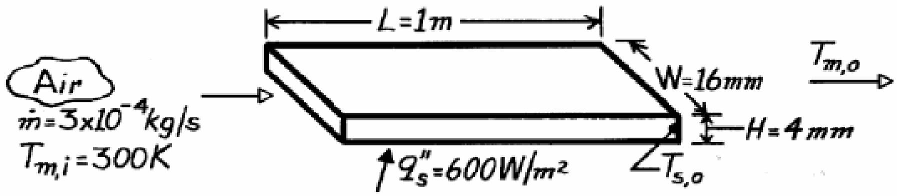
COMMENTS: The three thermal resistances are comparable. Note that $T_{s,o} = T_\infty - q'/h_o\pi D_o$ = 300K - 343 W/m/131 W/m^2·K π 0.028 m = 270 K.

PROBLEM 8.74

KNOWN: Flow rate and inlet temperature of air passing through a rectangular duct of prescribed dimensions and surface heat flux.

FIND: Air and duct surface temperatures at outlet.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Uniform surface heat flux, (3) Constant properties, (4) Atmospheric pressure, (3) Fully developed conditions at duct exit, (6) Ideal gas with negligible viscous dissipation and pressure variation.

PROPERTIES: *Table A-4*, Air $\left(\overline{T}_m \approx 300\text{K, 1 atm}\right)$: $c_p = 1007$ J/kg·K, $\mu = 184.6 \times 10^{-7}$ N·s/m^2, $k = 0.0263$ W/m·K, Pr = 0.707.

ANALYSIS: For this uniform heat flux condition, the heat rate is

$$q = q_s''\ A_s = q_s''\left[2(L\times W)+2(L\times H)\right]$$

$$q = 600\ \text{W/m}^2\left[2(1\text{m}\times 0.016\text{m})+2(1\text{m}\times 0.004\text{m})\right] = 24\ \text{W}.$$

From an overall energy balance

$$T_{m,o}=T_{m,i}+\frac{q}{\dot{m}\,c_p} = 300\text{K}+\frac{24\ \text{W}}{3\times 10^{-4}\ \text{kg/s}\times 1007\ \text{J/kg}\cdot\text{K}} = 379\ \text{K}. \qquad <$$

The surface temperature at the outlet may be determined from Newton's law of cooling, where

$$T_{s,o} = T_{m,o} + q''/h.$$

From Eqs. 8.66 and 8.1

$$D_h = \frac{4\ A_c}{P} = \frac{4(0.016\text{m}\times 0.004\text{m})}{2(0.016\text{m}+0.004\text{m})} = 0.0064\ \text{m}$$

$$Re_D = \frac{\rho\, u_m\, D_h}{\mu} = \frac{\dot{m}\, D_h}{A_c \mu} = \frac{3\times 10^{-4}\text{kg/s}(0.0064\text{m})}{64\times 10^{-6}\text{m}^2\left(184.6\times 10^{-7}\ \text{N}\cdot\text{s/m}^2\right)} = 1625.$$

Hence the flow is laminar, and from Table 8.1

$$h = \frac{k}{D_h}5.33 = \frac{0.0263\ \text{W/m}\cdot\text{K}}{0.0064\ \text{m}}5.33 = 22\ \text{W/m}^2\cdot\text{K}$$

$$T_{s,o} = 379\ \text{K} + \frac{600\ \text{W/m}^2}{22\ \text{W/m}^2\cdot\text{K}} = 406\ \text{K}. \qquad <$$

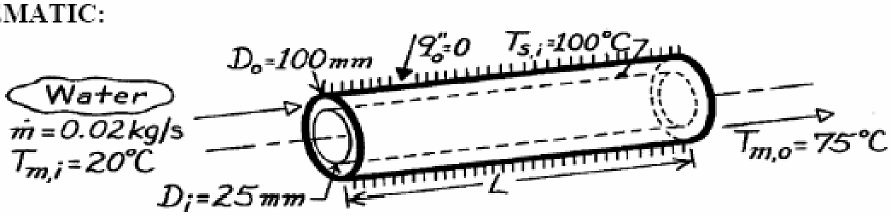
COMMENTS: The calculations should be reperformed with properties evaluated at $\overline{T}_m = 340$ K. The change in $T_{m,o}$ would be negligible, and $T_{s,o}$ would decrease slightly.

PROBLEM 8.93

KNOWN: Surface thermal conditions and diameters associated with a concentric tube annulus. Water flow rate and inlet temperature.

FIND: (a) Length required to achieve desired outlet temperature, (b) Heat flux from inner tube at outlet.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Fully developed conditions throughout, (3) Adiabatic outer surface, (4) Uniform temperature at inner surface, (5) Constant properties, (6) Water is incompressible liquid with negligible viscous dissipation.

PROPERTIES: *Table A-6*, Water $(\overline{T}_m = 320K)$: $c_p = 4180$ J/kg·K, $\mu = 577 \times 10^{-6}$ N·s/m^2, k = 0.640 W/m·K, Pr = 3.77.

ANALYSIS: (a) From Eq. 8.41a,

$$L = -\frac{\dot{m}\, c_p}{P\overline{h}} \ell n \frac{\Delta T_o}{\Delta T_i} = -\frac{\dot{m}\, c_p}{\pi\, D_i \overline{h}} \ell n \frac{T_s - T_{m,o}}{T_s - T_{m,i}}.$$

With $$Re_D = \frac{\rho\, u_m D_h}{\mu} = \frac{\dot{m}\,(D_o - D_i)}{(\pi/4)\,\left(D_o^2 - D_i^2\right)\mu} = \frac{4\,\dot{m}}{\pi(D_o + D_i)\,\mu}$$

$$Re_D = \frac{4 \times 0.02 \text{ kg/s}}{\pi(0.125\text{m})577 \times 10^{-6} \text{N}\cdot\text{s/m}^2} = 353$$

the flow is laminar. Hence, from Eq. 8.69 and Table 8.2,

$$\overline{h} = h_i = \frac{k}{D_h} Nu_i = \frac{0.64 \text{ W/m}\cdot\text{K}}{(0.100 - 0.025)\text{ m}} 7.37 = 63 \text{ W/m}^2\cdot\text{K}$$

and $$L = -\frac{0.02 \text{ kg/s } (4180 \text{ J/kg}\cdot\text{K})}{\pi(0.025\text{m})63 \text{ W/m}^2\cdot\text{K}} \ell n \frac{(100-75)^\circ \text{C}}{(100-20)^\circ \text{C}} = 19.7 \text{ m.}$$ <

(b) From Eq. 8.68

$$q''_i(L) = h_i\left(T_{s,i} - T_{m,o}\right) = 63 \frac{\text{W}}{\text{m}^2.\text{K}}(100-75)^\circ \text{C} = 1575 \text{ W/m}^2.$$ <

COMMENTS: The total heat rate to the water is

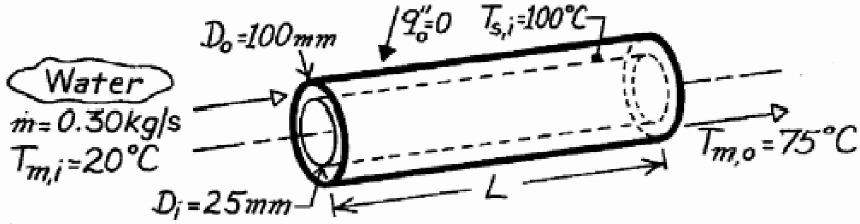
$$q = \dot{m}\, c_p\, \left(T_{m,o} - T_{m,i}\right) = 0.02 \text{ kg/s} \times 4180 \text{ J/kg}\cdot\text{K}\left(55^\circ\text{C}\right) = 4598 \text{ W.}$$

PROBLEM 8.94

KNOWN: Surface thermal conditions and diameters associated with a concentric tube annulus. Water flow rate and inlet temperature.

FIND: Length required to achieve desired outlet temperature.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Fully developed conditions throughout, (3) Adiabatic outer surface, (4) Uniform temperature at inner surface, (5) Constant properties, (6) Incompressible liquid with negligible viscous dissipation.

PROPERTIES: *Table A-6*, Water $\left(\overline{T}_m = 320K\right)$: $c_p = 4180$ J/kg·K, $\mu = 577 \times 10^{-6}$ N·s/m^2, k = 0.640 W/m·K, Pr = 3.77.

ANALYSIS: From Eq. 8.42a,

$$L = -\frac{\dot{m}\,c_p}{P\,\overline{h}}\ell n\frac{\Delta T_o}{\Delta T_i} = -\frac{\dot{m}\,c_p}{\pi\,D_i\overline{h}}\ell n\frac{T_s - T_{m,o}}{T_s - T_{m,i}}.$$

With

$$Re_D = \frac{\rho\,u_m D_h}{\mu} = \frac{\dot{m}\,(D_o - D_i)}{(\pi/4)\,\left(D_o^2 - D_i^2\right)\mu} = \frac{4\,\dot{m}}{\pi\,(D_o + D_i)\,\mu}$$

$$Re_D = \frac{4 \times 0.30\text{ kg/s}}{\pi\,(0.125\text{m})\,577 \times 10^{-6}\text{N}\cdot\text{s/m}^2} = 5296$$

and the flow is turbulent. Hence, from Eq. 8.60,

$$\overline{h} = \frac{k}{D_h}Nu_D = 0.023\frac{k}{D_h}Re_D^{4/5}\,Pr^{0.4}$$

$$\overline{h} = 0.023\frac{0.640\text{ W/m}\cdot\text{K}}{0.075\text{ m}}(5296)^{4/5}(3.77)^{0.4} = 318\text{ W/m}^2\cdot\text{K}$$

and hence the required length is

$$L = -\frac{0.30\text{ kg/s}\,(4180\text{ J/kg}\cdot\text{K})}{\pi\,(0.025\text{m})\;318\text{ W/m}^2\cdot\text{K}}\ell n\frac{(100-75)^\circ\text{C}}{(100-20)^\circ\text{C}} = 58.4\text{ m.}$$ <

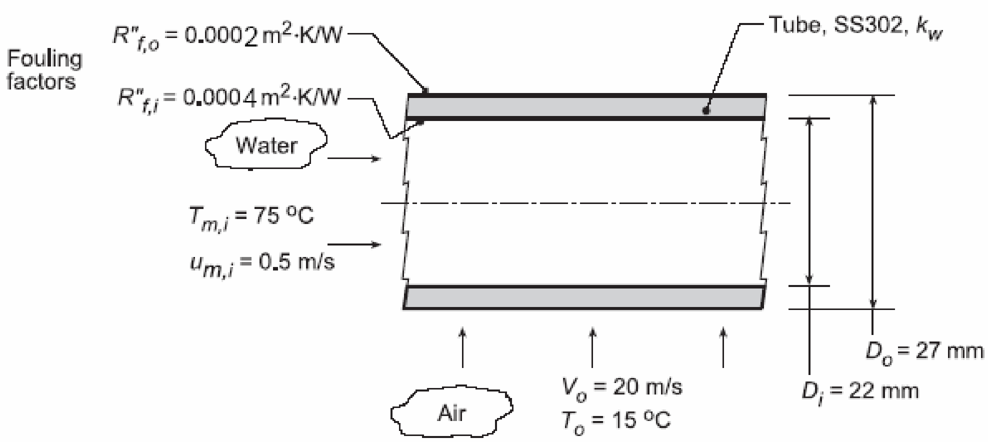
COMMENTS: (1) Increasing $\dot{m}$ by a factor of 15 increases Re_D accordingly, and the flow is turbulent. However, $\overline{h}$ increases by a factor of only 5, from the result of Problem 8.99, in which case the tube length must be a factor of 3 larger than that of Problem 8.99. (2) The Gnielinski correlation would be more accurate than the Dittus-Boelter correlation for the low (but turbulent) conditions suggested by the value of the Reynolds number.

PROBLEM 11.2

KNOWN: Type-302 stainless tube with prescribed inner and outer diameters used in a cross-flow heat exchanger. Prescribed fouling factors and internal water flow conditions.

FIND: (a) Overall coefficient based upon the outer surface, U_o, with air at T_o =15°C and velocity V_o = 20 m/s in cross-flow; compare thermal resistances due to convection, tube wall conduction and fouling; (b) Overall coefficient, U_o, with water (rather than air) at T_o = 15°C and velocity V_o = 1 m/s in cross-flow; compare thermal resistances due to convection, tube wall conduction and fouling; (c) For the water-air conditions of part (a), compute and plot U_o as a function of the air cross-flow velocity for $5 \le V_o \le 30$ m/s for water mean velocities of $u_{m,i}$ = 0.2, 0.5 and 1.0 m/s; and (d) For the water-water conditions of part (b), compute and plot U_o as a function of the water mean velocity for $0.5 \le u_{m,i} \le 2.5$ m/s for air cross-flow velocities of V_o = 1, 3 and 8 m/s.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Fully developed internal flow,

PROPERTIES: *Table A.1*, Stainless steel, AISI 302 (300 K): k_w = 15.1 W/m·K; *Table A.6*, Water ($\overline{T}_{m,i}$ = 348 K): ρ_i = 974.8 kg/m³, $\mu_i = 3.746 \times 10^{-4}$ N·s/m², k_i = 0.668 W/m·K, Pr_i = 2.354; *Table A.4*, Air (assume $\overline{T}_{f,o}$ = 315K, 1 atm): k_o = 0.02737 W/m·K, $\nu_o = 17.35 \times 10^{-6}$ m²/s, Pr_o = 0.705.

ANALYSIS: (a) For the water-air condition, the overall coefficient, Eq. 11.1, based upon the outer area can be expressed as the sum of the thermal resistances due to convection (cv), tube wall conduction (w) and fouling (f):

$$1/U_o A_o = R_{tot} = R_{cv,i} + R_{f,i} + R_w + R_{f,o} + R_{cv,o}$$

$$R_{cv,i} = 1/\overline{h}_i A_i \qquad R_{cv,o} = 1/\overline{h}_o A_o$$

$$R_{f,i} = R''_{f,i}/A_i \qquad R_{f,o} = R''_{f,o}/A_o$$

and from Eq. 3.28,

$$R_w = \ln(D_o/D_i)/(2\pi L k_w)$$

The convection coefficients can be estimated from appropriate correlations.

Continued...

Estimating $\overline{h}_i$: For internal flow, characterize the flow evaluating thermophysical properties at $T_{m,i}$ with

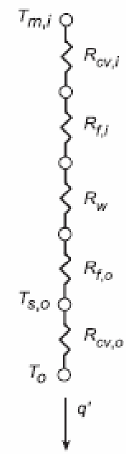
$$Re_{D,i} = \frac{u_{m,i} D_i}{\nu_i} = \frac{0.5\,m/s \times 0.022m}{3.746\times10^{-4}\,N\cdot s/m^2 / 974.8\,kg/m^3} = 28,625$$

For the turbulent flow, use the Dittus-Boelter correlation, Eq. 8.60,

$$Nu_{D,i} = 0.023\,Re_{D,i}^{0.8}\,Pr_i^{0.3}$$

$$Nu_{D,i} = 0.023(28,625)^{0.8}(2.354)^{0.3} = 109.3$$

$$\overline{h}_i = Nu_{D,i}\,k_i/D_i = 109.3\times0.668\,W/m^2\cdot K/0.022m = 3313\,W/m^2\cdot K$$



Estimating $\overline{h}_o$: For external flow, characterize the flow with

$$Re_{D,o} = \frac{V_o D_o}{\nu_o} = \frac{20\,m/s\times0.027m}{17.35\times10^{-6}\,m^2/s} = 31,124$$

evaluating thermophysical properties at $T_{f,o} = (T_{s,o} + T_o)/2$ when the surface temperature is determined from the thermal circuit analysis result,

$$\left(T_{m,i} - T_o\right)/R_{tot} = \left(T_{s,o} - T_o\right)/R_{cv,o}$$

Assume $T_{f,o}$ = 315 K, and check later. Using the Churchill-Bernstein correlation, Eq. 7.54, find

$$\overline{Nu}_{D,o} = 0.3 + \frac{0.62\,Re_{D,o}^{1/2}\,Pr_o^{1/3}}{\left[1+(0.4/Pr_o)^{2/3}\right]^{1/4}}\left[1+\left(\frac{Re_{D,o}}{282,000}\right)^{5/8}\right]^{4/5}$$

$$\overline{Nu}_{D,o} = 0.3 + \frac{0.62(31,124)^{1/2}(0.705)^{1/3}}{\left[1+(0.4/0.705)^{2/3}\right]^{1/4}}\left[1+\left(\frac{31,124}{282,000}\right)^{5/8}\right]^{4/5}$$

$$\overline{Nu}_{D,o} = 102.6$$

$$\overline{h}_o = \overline{Nu}_{D,o}\,k_o/D_o = 102.6\times0.02737\,W/m\cdot K/0.027m = 104.0\,W/m\cdot K$$

Using the above values for $\overline{h}_i$, and $\overline{h}_o$, and other prescribed values, the thermal resistances and overall coefficient can be evaluated and are tabulated below.

$R_{cv,i}$ (K/W)	$R_{f,i}$ (K/W)	R_w (K/W)	$R_{f,o}$ (K/W)	$R_{cv,o}$ (K/W)	U_o ($W/m^2\cdot K$)	R_{tot} (K/W)
0.00436	0.00578	0.00216	0.00236	0.1134	92.1	0.128

The major thermal resistance is due to outside (air) convection, accounting for 89% of the total resistance. The other thermal resistances are of similar magnitude, nearly 50 times smaller than $R_{cv,o}$.

(b) For the water-water condition, the method of analysis follows that of part (a). For the internal flow, the estimated convection coefficient is the same as part (a). For an assumed outer film coefficient, $\overline{T}_{f,o}$ = 292 K, the convection correlation for the outer water flow condition V_o = 1 m/s and T_o = 15°C,

Continued...

PROBLEM 11.18

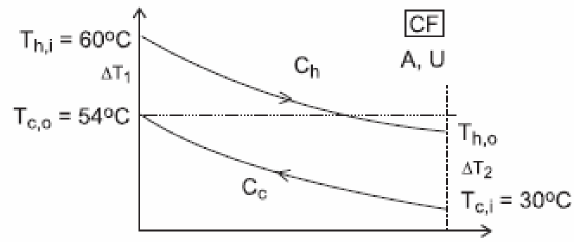
KNOWN: Concentric tube heat exchanger with area of 50 m^2 with operating conditions as shown on the schematic.

FIND: (a) Outlet temperature of the hot fluid; (b) Whether the exchanger is operating in counterflow or parallel flow; or can't tell from information provided; (c) Overall heat transfer coefficient; (d) Effectiveness of the exchanger; and (e) Effectiveness of the exchanger if its length is made very long

SCHEMATIC:

Operating conditions
Concentric tube HXer, A = 50 m^2

	Hot fluid (h)	Cold fluid (c)
Capacity rate, kW/K	6	3
Inlet temperature, °C	60	30
Outlet temperature, °C	–	54



ASSUMPTIONS: (1) Negligible heat loss to surroundings, (2) Constant properties.

ANALYSIS: From overall energy balances on the hot and cold fluids, find the hot fluid outlet temperature

$$q = C_c\left(T_{c,o} - T_{c,i}\right) = C_h\left(T_{h,i} - T_{h,o}\right) \tag{1}$$

$$3000\ \text{W/K}\ (54-30)\text{K} = 6000\left(60 - T_{h,o}\right) \qquad T_{h,o} = 48°\text{C} \qquad <$$

(b) HXer must be operating in counterflow (CF) since $T_{h,o} < T_{c,o}$. See schematic for temperature distribution.

(c) From the rate equation with A = 50 m^2, with Eq. (1) for q,

$$q = C_c\left(T_{c,o} - T_{c,i}\right) = UA\Delta T_{\ell m} \tag{2}$$

$$\Delta T_{\ell m} = \frac{\Delta T_1 - \Delta T_2}{\ell m\left(\Delta T_1 / \Delta T_2\right)} = \frac{(60-54)\text{K} - (48-30)\text{K}}{\ell n(6/18)} = 10.9°\text{C} \tag{3}$$

$$3000\ \text{W/K}(54-30)\text{K} = U \times 50\ \text{m}^2 \times 10.9\,\text{K}$$

$$U = 132\ \text{W/m}^2\cdot\text{K} \qquad <$$

(d) The effectiveness, from Eq. 11.19, with the cold fluid as the minimum fluid, $C_c = C_{min}$,

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_c\left(T_{c,o} - T_{c,i}\right)}{C_{min}\left(T_{h,i} - T_{c,i}\right)} = \frac{(54-30)\text{K}}{(60-30)\text{K}} = 0.8 \qquad <$$

(e) For a very long CF HXer, the outlet of the minimum fluid, $C_{min} = C_c$, will approach $T_{h,i}$. That is,

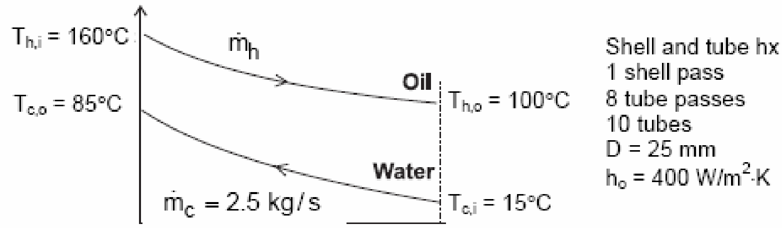
$$q \to C_{min}\left(T_{c,o} - T_{c,i}\right) \to q_{max} \qquad \varepsilon = 1 \qquad <$$

PROBLEM 11.22

KNOWN: Inlet and outlet temperatures for a shell-and-tube heat exchanger with 10 tubes making eight passes. Heat transfer coefficient for oil flowing in shell. Mass flow rate of water in tubes. Tube diameter.

FIND: Oil flow rate required to achieve specified outlet temperature. Tube length required to achieve specified water heating.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat loss to the surroundings, (2) Constant properties, (3) Negligible tube wall thermal resistance and fouling effects, (4) Fully developed water flow in tubes.

PROPERTIES: *Table A.5*, unused engine oil: ($\overline{T}_h$ = 130°C): c_p = 2350 J/kg·K. *Table A.6*, water ($\overline{T}_c$ = 50°C): c_p = 4181 J/kg·K, $\mu = 548 \times 10^{-6}$ N·s/m^2, k = 0.643 W/m·K, Pr = 3.56.

ANALYSIS: From the overall energy balance, Eq. 11.7b, the heat transfer required of the exchanger is

$$q = \dot{m}_c c_{p,c}(T_{c,o} - T_{c,i}) = 2.5 \text{ kg/s} \times 4181 \text{ J/kg}\cdot\text{K}(85-15)°\text{C} = 7.317 \times 10^5 \text{ W}$$

Hence from Eq. 11.6b,

$$\dot{m}_h = \frac{q}{c_{p,h}(T_{h,i} - T_{h,o})} = \frac{7.317 \times 10^5 \text{ W}}{2350 \text{ J/kg}\cdot\text{K}(160-100)°\text{C}} = 5.19 \text{ kg/s} \qquad <$$

The required tube length may be obtained using the ε-NTU method. We first calculate the heat capacity rates, $C_h = \dot{m}_h c_{p,h}$ = 12,195 W/K, $C_c = \dot{m}_c c_{p,c}$ =10,453 W/K. Thus, $C_{min} = C_c$, and $C_r = C_{min}/C_{max}$ = 0.857. Then from Eq. 11.21,

$$\varepsilon = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} = \frac{(85-15)°\text{C}}{(160-15)°\text{C}} = 0.483$$

Using Eqs. 11.30b,c for one shell pass and an even number of tube passes, we find

Continued…

PROBLEM 11.22 (Cont.)

$$E = \frac{2/\varepsilon - (1+C_r)}{(1+C_r^2)^{1/2}} = \frac{2/0.483 - (1+0.857)}{(1+0.857^2)^{1/2}} = 1.74$$

$$NTU = -(1+C_r^2)^{-1/2}\ln\left(\frac{E-1}{E+1}\right) = -(1+0.857^2)^{-1/2}\ln\left(\frac{1.74-1}{1.74+1}\right) = 0.997$$

Thus UA = NTU×C_{min} = 10,420 W/K. To find the required tube length, we must know the heat transfer coefficients for the water flow. We calculate the Reynolds number, with $\dot{m}_1 = \dot{m}_c / N$ = 0.25 kg/s defined as the water flow rate per tube, Eq. 8.6 yields

$$Re_D = \frac{4\dot{m}_1}{\pi D \mu_c} = \frac{4 \times 0.25\ \text{kg/s}}{\pi(0.025\ \text{m})548 \times 10^{-6}\ \text{N}\cdot\text{s/m}^2} = 23,234$$

Hence the flow is turbulent, and from Eq. 8.60,

$$Nu_D = 0.023\, Re_D^{4/5}\, Pr^{0.4} = 0.023(23,234)^{4/5}(3.56)^{0.4} = 119$$

and

$$h_c = \frac{k_c}{D} Nu_D = \frac{0.643\ \text{W/m}\cdot\text{K}}{0.025\ \text{m}} 119 = 3060\ \text{W/m}^2\cdot\text{K}$$

Hence U = $[1/h_c + 1/h_h]^{-1}$ = 354 W/m^2·K and we can find the required tube length from

$$L = \frac{UA}{UN\pi D} = \frac{10,420\ \text{W/K}}{354\ \text{W/m}^2\cdot\text{K} \times 10 \times \pi \times 0.025\ \text{m}} = 37.5\ \text{m}$$ <

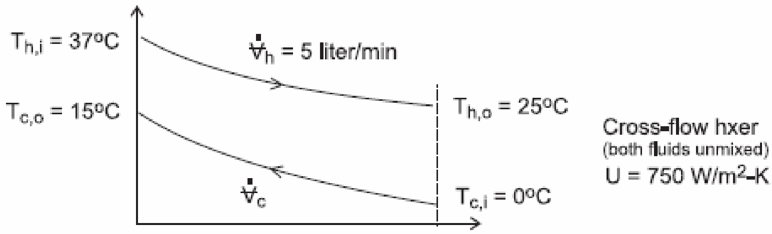
COMMENTS: (1) With L/D = 1516, the assumption of fully developed conditions throughout the tube is justified. (2) With eight passes, the shell length is approximately L/8 = 4.7 m.

PROBLEM 11.34

KNOWN: Cross-flow heat exchanger (both fluids unmixed) cools blood to induce body hypothermia using ice-water as the coolant.

FIND: (a) Heat transfer rate from the blood, (b) Water flow rate, $\dot{\forall}_c$ (liter/min), (c) Surface area of the exchanger, and (d) Calculate and plot the blood and water outlet temperatures as a function of the water flow rate for the range, $2 \le \dot{\forall} \le 4$ liter/min, assuming all other parameters remain unchanged.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Negligible losses to the surroundings, (3) Overall heat transfer coefficient remains constant with water flow rate changes, and (4) Constant properties.

PROPERTIES: *Table A-6*, Water $\left(\overline{T}_c = 280\text{K}\right)$, $\rho = 1000\ \text{kg/m}^3$, $c = 4198\ \text{J/kg}\cdot\text{K}$. Blood (given): $\rho = 1050\ \text{kg/m}^3$, $c = 3740\ \text{J/kg}\cdot\text{K}$.

ANALYSIS: (a) The heat transfer rate from the blood is calculated from an energy balance on the hot fluid,

$$\dot{m}_h = \rho_h \dot{\forall}_h = 1050\ \text{kg/m}^3 \times \left(5\ \text{liter/min} \times 1\ \text{min}/60\ \text{s}\right) \times 10^{-3}\,\text{m}^3/\text{liter} = 0.0875\ \text{kg/s}$$

$$q = \dot{m}_h c_h \left(T_{h,i} - T_{h,o}\right) = 0.0875\ \text{kg/s} \times 3740\ \text{J/kg}\cdot\text{K}\left(37 - 25\right)\text{K} = 3927\ \text{W} \qquad < (1)$$

(b) From an energy balance on the cold fluid, find the coolant water flow rate,

$$q = \dot{m}_c\, c_c \left(T_{c,o} - T_{c,i}\right) \qquad (2)$$

$$3927\ \text{W} = \dot{m}_c \times 4198\ \text{J/kg}\cdot\text{K}\left(15 - 0\right)\text{K} \qquad \dot{m}_c = 0.0624\ \text{kg/s}$$

$$\dot{\forall}_c = \dot{m}_c/\rho_c = 0.0624\ \text{kg/s}/1000\ \text{kg/m}^3 \times 10^3\text{liter/m}^3 \times 60\ \text{s/min} = 3.74\ \text{liter/min} \qquad <$$

(c) The surface area can be determined using the effectiveness-NTU method. The capacity rates for the exchanger are

$$C_h = \dot{m}_h c_h = 327\ \text{W/K} \qquad C_c = \dot{m}_c\, c_c = 262\ \text{W/K} \qquad C_{min} = C_c \qquad (3, 4, 5)$$

From Eq. 11.18 and 11.19, the maximum heat rate and effectiveness are

Continued …..

$$q_{max} = C_{min}\left(T_{h,i} - T_{c,i}\right) = 262\ W/K\ (37-0)K = 9694\ W \quad (6)$$

$$\varepsilon = q / q_{max} = 3927/9694 = 0.405 \quad (7)$$

For the cross flow exchanger, with both fluids unmixed, substitute numerical values into Eq. 11.32 to find the number of transfer units, NTU, where $C_r = C_{min} / C_{max}$.

$$\varepsilon = 1 - \exp\left[(1/C_r)NTU^{0.22}\left\{\exp\left[-C_r NTU^{0.78}\right]-1\right\}\right] \quad (8)$$

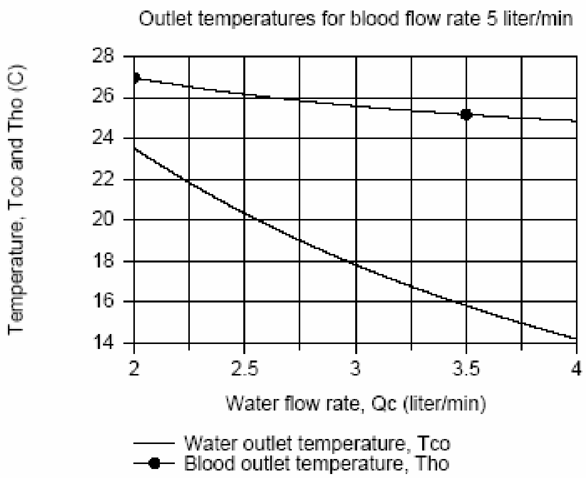
$$NTU = 0.691$$

From Eq. 11.24, find the surface area, A.

$$NTU = UA / C_{min}$$

$$A = 0.691 \times 262\ W/K / 750\ W/m^2 \cdot K = 0.241\ m^2$$ <

(d) Using the foregoing equations in the *IHT* workspace, the blood and water outlet temperatures, $T_{h,o}$ and $T_{c,o}$, respectively, are calculated and plotted as a function of the water flow rate, all other parameters remaining unchanged.



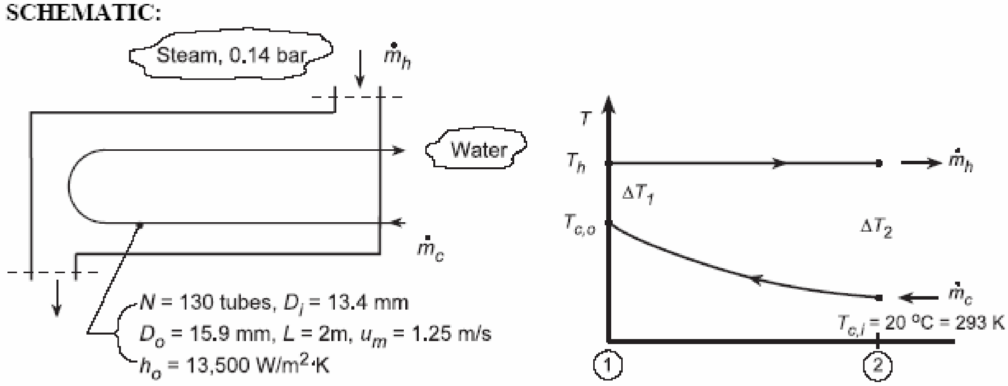
From the graph, note that with increasing water flow rate, both the blood and water outlet temperatures decrease. However, the effect of the water flow rate is greater on the water outlet temperature. This is an advantage for this application, since it is desirable to have the blood outlet temperature relatively insensitive to changes in the water flow rate. That is, if there are pressure changes on the water supply line or a slight mis-setting of the water flow rate controller, the outlet blood temperature will not change markedly.

PROBLEM 11.35

KNOWN: Steam at 0.14 bar condensing in a shell and tube HXer (one shell, two tube passes consisting of 130 brass tubes of length 2 m, D_i = 13.4 mm, D_o = 15.9 mm). Cooling water enters at 20°C with a mean velocity 1.25 m/s. Heat transfer convection coefficient for condensation on outer tube surface is h_o = 13,500 W/m^2·K.

FIND: (a) Overall heat transfer coefficient, U, for the HXer, outlet temperature of cooling water, $T_{c,o}$, and condensation rate of the steam $\dot{m}_h$; and (b) Compute and plot $T_{c,o}$ and $\dot{m}_h$ as a function of the water flow rate $10 \le \dot{m}_c \le 30$ kg/s with all other conditions remaining the same, but accounting for changes in U.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat loss to surroundings, (2) Constant properties, (3) Fully developed water flow in tubes.

PROPERTIES: *Table A-6,* Steam (0.14 bar): $T_{sat} = T_h$ = 327 K, h_{fg} = 2373 kJ/kg, c_p = 1898 J/kg·K; *Table A-6,* Water (Assume $T_{c,o} \approx 44$°C or $\overline{T}_c \approx 305$ K): $v_f = 1.005 \times 10^{-3}$ m^3/kg, c_p = 4178 J/kg·K, $\mu_f = 769 \times 10^{-6}$ N·s/m^2, k_f = 0.620 W/m·K, Pr_f = 5.2; *Table A-1,* Brass - 70/30 (Evaluate at $\overline{T} = (T_h + \overline{T}_c)/2 = 316$ K): k = 114 W/m·K.

ANALYSIS: (a) The overall heat transfer coefficient based upon the outside tube area follows from Eq. 11.5,

$$U_o = \left[\frac{1}{h_o} + \frac{r_o}{k}\ell n\frac{r_o}{r_i} + \left(\frac{r_o}{r_i}\right)\frac{1}{h_i}\right]^{-1}. \qquad (1)$$

The value for h_i can be estimated from an appropriate internal flow correlation. First determine the nature of the flow within the tubes. From Eq. 8.1,

$$Re_{D_i} = \rho u_m \frac{D_i}{\mu} = \frac{\left(1.005\times10^{-3}\,m^3/kg\right)^{-1}\times1.25\,m/s\times13.4\times10^{-3}\,m}{769\times10^{-6}\,N\cdot s/m^2} = 21,673\,.$$

The water flow is turbulent and fully developed (L/D_i = 2 m /13.4 × 10^{-3} m = 150 > 10). The Dittus-Boelter correlation with n = 0.4 is appropriate,

$$Nu_D = h_i D_i/k_f = 0.023\,Re_D^{0.8}\,Pr_f^{0.4} = 0.023\times(21,673)^{0.8}(5.2)^{0.4} = 130.9$$

Continued...

$$h_i = \frac{k_f}{D_i} Nu_D = \frac{0.620 \, W/m \cdot K}{13.4 \times 10^{-3} \, m} \times 130.9 = 6057 \, W/m^2 \cdot K.$$

Substituting numerical values into Eq. (1), the overall heat transfer coefficient is

$$U_o = \left[\frac{1}{13,500 \, W/m^2 \cdot K} + \frac{\left(15.9 \times 10^{-3} \, m\right)/2}{114 \, W/m \cdot K} \ell n \frac{15.9}{13.4} + \frac{15.9}{13.4} \times \frac{1}{6057 \, W/m^2 \cdot K} \right]^{-1}$$

$$U_o = \left[7.407 \times 10^{-5} + 1.193 \times 10^{-5} + 19.590 \times 10^{-5} \right]^{-1} W/m^2 \cdot K = 3549 \, W/m^2 \cdot K. \quad <$$

To find the outlet temperature of the water, we'll employ the $\varepsilon - NTU$ method. From an energy balance on the cold fluid,

$$T_{c,o} = T_{c,i} + q/C_c \quad (3)$$

where the heat rate can be expressed as

$$q = \varepsilon q_{max} \qquad q_{max} = C_{min} \left(T_{h,i} - T_{h,o} \right). \quad (4,5)$$

The minimum capacity rate is that of the cold water since $C_h \to \infty$. Evaluating, find

$$C_{min} = C_c = \left(\dot{m} c_p \right)_c = 22.8 \, kg/s \times 4178 \, J/kg \cdot K = 95,270 \, W/K.$$

where

$$\dot{m}_c = \left(\rho A u_m \right) N = 995.0 \, kg/m^3 \times \pi/4 \left(0.0134 m \right)^2 \times 1.25 \, m/s \times 130 = 22.8 \, kg/s$$

To determine ε, use Fig. 11.12 (one shell and any multiple of tube passes) with $C_r = 0$ and

$$NTU = \frac{U_o A_o}{C_{min}} = \frac{3549 \, W/m^2 \cdot K \left(\pi 0.0159 m \times 2m \times 130 \times 2 \right)}{95,270 \, W/K} = 0.968$$

where 130 and 2 represent the number of tubes and passes, respectively, to find $\varepsilon \approx 0.62$. Combining Eqs. (4) and (5) into Eq. (3), find

$$T_{c,o} = T_{c,i} + \varepsilon C_{min} \left(T_{h,i} - T_{c,i} \right) / C_c = 20^\circ C + 0.62 \left(327 - 293 \right) K = 41.1^\circ C. \quad <$$

The condensation rate of the steam is given by

$$\dot{m}_h = q/h_{fg} \quad (6)$$

where the heat rate can be determined from Eq. (3) with $T_{c,o}$,

$$\dot{m}_h = C_c \left(T_{c,o} - T_{c,i} \right) / h_{fg} = 95,270 \, W/K \left(41.1 - 20.0 \right) K / 2373 \times 10^3 \, J/kg \cdot K = 0.85 \, kg/s. \quad <$$

(b) Using the *IHT Heat Exchanger Tool, All Exchangers, $C_r = 0$,* and the *Properties Tool* for *Water,* a model was developed and the cold outlet temperature and condensation rate were computed and plotted.

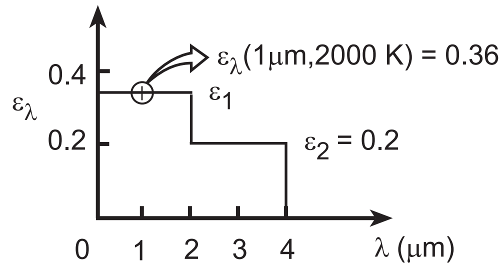
Continued...

PROBLEM 12.32

KNOWN: Metallic surface with prescribed spectral, directional emissivity at 2000 K and 1 μm (see Example 12.6) and additional measurements of the spectral, hemispherical emissivity.

FIND: (a) Total hemispherical emissivity, ε, and the emissive power, E, at 2000 K, (b) Effect of temperature on the emissivity.

SCHEMATIC:



ANALYSIS: (a) The total, hemispherical emissivity, ε, may be determined from knowledge of the spectral, hemispherical emissivity, ε_λ, using Eq. 12.36.

$$\varepsilon(T) = \int_0^\infty \varepsilon_\lambda(\lambda)E_{\lambda,b}(\lambda,T)\,d\lambda \big/ E_b(T) = \varepsilon_1\int_0^{2\mu m} \frac{E_{\lambda,b}(\lambda,T)d\lambda}{E_b(T)} + \varepsilon_2\int_{2\mu m}^{4\mu m} \frac{E_{\lambda,b}(\lambda,T)d\lambda}{E_b(T)}$$

or from Eqs. 12.36 and 12.28,

$$\varepsilon(T) = \varepsilon_1 F_{(0\to\lambda_1)} + \varepsilon_2\left[F_{(0\to\lambda_2)} - F_{(0\to\lambda_1)}\right]$$

From Table 12.1,

$$\lambda_1 = 2\,\mu m, \quad T = 2000\text{ K}: \quad \lambda_1 T = 4000\,\mu m\cdot K, \quad F_{(0\to\lambda_1)} = 0.481$$

$$\lambda_2 = 4\,\mu m, \quad T = 2000\text{ K}: \quad \lambda_2 T = 8000\,\mu m\cdot K, \quad F_{(0\to\lambda_2)} = 0.856$$

Hence,

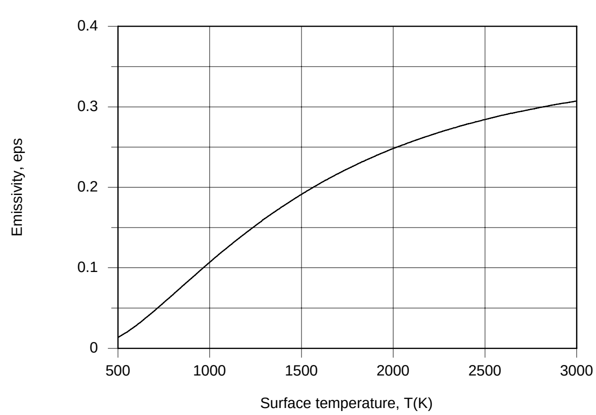
$$\varepsilon(T) = 0.36\times 0.481 + 0.20(0.856 - 0.481) = 0.25$$ <

The total emissive power at 2000 K is

$$E(2000\text{ K}) = \varepsilon\,(2000\text{ K})\cdot E_b\,(2000\text{ K})$$

$$E(2000\text{ K}) = 0.25\times 5.67\times 10^{-8}\,W/m^2\cdot K^4 \times (2000\text{ K})^4 = 2.27\times 10^5\,W/m^2\,.$$ <

(b) Using the *Radiation* Toolpad of IHT, the following result was generated.



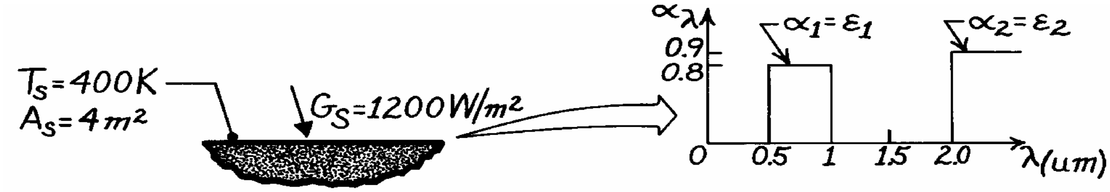
Continued...

PROBLEM 12.46

KNOWN: Area, temperature, irradiation and spectral absorptivity of a surface.

FIND: Absorbed irradiation, emissive power, radiosity and net radiation transfer from the surface.

SCHEMATIC:



ASSUMPTIONS: (1) Opaque, diffuse surface behavior, (2) Spectral distribution of solar radiation corresponds to emission from a blackbody at 5800 K.

ANALYSIS: The absorptivity to solar irradiation is

$$\alpha_S = \frac{\int_0^\infty \alpha_\lambda G_\lambda d\lambda}{G} = \frac{\int_0^\infty \alpha_\lambda E_{\lambda b}(5800\ K)d\lambda}{E_b} = \alpha_1 F_{(0.5\to 1\mu m)} + \alpha_2 F_{(2\to\infty)}.$$

From Table 12.1,	$\lambda T = 2900\ \mu m\cdot K$:	$F_{(0\to 0.5\ \mu m)} = 0.250$
	$\lambda T = 5800\ \mu m\cdot K$:	$F_{(0\to 1\ \mu m)} = 0.720$
	$\lambda T = 11,600\ \mu m\cdot K$:	$F_{(0\to 2\ \mu m)} = 0.941$

$$\alpha_S = 0.8(0.720 - 0.250) + 0.9(1 - 0.941) = 0.429.$$

Hence, $$G_{abs} = \alpha_S G_S = 0.429\left(1200\ W/m^2\right) = 515\ W/m^2.$$ <

The emissivity is

$$\varepsilon = \int_0^\infty \varepsilon_\lambda E_{\lambda b}(400\ K)d\lambda / E_b = \varepsilon_1 F_{(0.5\to 1\mu m)} + \varepsilon_2 F_{(2\to\infty)}.$$

From Table 12.1,	$\lambda T = 200\ \mu m\cdot K$:	$F_{(0\to 0.5\ \mu m)} = 0$
	$\lambda T = 400\ \mu m\cdot K$:	$F_{(0\to 1\ \mu m)} = 0$
	$\lambda T = 800\ \mu m\cdot K$	$F_{(0\to 2\ \mu m)} = 0.$

Hence, $\varepsilon = \varepsilon_2 = 0.9$,

$$E = \varepsilon\sigma T_s^4 = 0.9\times 5.67\times 10^{-8}\ W/m^2\cdot K^4\,(400\ K)^4 = 1306\ W/m^2.$$ <

The radiosity is

$$J = E + \rho_S G_S = E + (1-\alpha_S)G_S = [1306 + 0.571\times 1200]W/m^2 = 1991\ W/m^2.$$ <

The net radiation transfer from the surface is

$$q_{net} = (E - \alpha_S G_S)A_s = (1306 - 515)W/m^2 \times 4\ m^2 = 3164\ W.$$ <

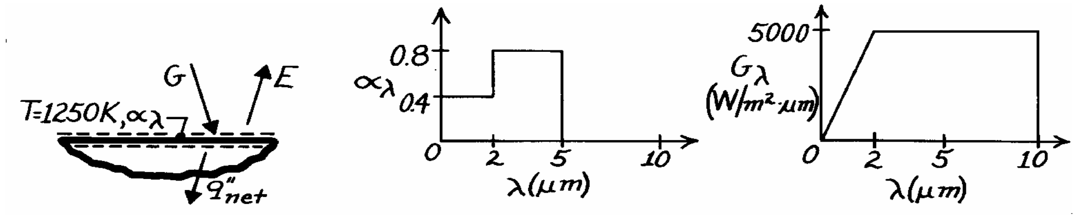
COMMENTS: Unless 3164 W are supplied to the surface by other means (for example, by convection), the surface temperature will decrease with time.

PROBLEM 12.51

KNOWN: Spectral distribution of surface absorptivity and irradiation. Surface temperature.

FIND: (a) Total absorptivity, (b) Emissive power, (c) Nature of surface temperature change.

SCHEMATIC:



ASSUMPTIONS: (1) Opaque, diffuse surface behavior, (2) Convection effects are negligible.

ANALYSIS: (a) From Eqs. 12.43 and 12.44, the absorptivity is defined as

$$\alpha \equiv G_{abs} / G = \int_0^\infty \alpha_\lambda \, G_\lambda \, d\lambda / \int_0^\infty G_\lambda \, d\lambda.$$

The absorbed irradiation is,

$$G_{abs} = 0.4\left(5000\,W/m^2 \cdot \mu m \times 2\,\mu m\right)/2 + 0.8 \times 5000\,W/m^2 \cdot \mu m\left(5-2\right)\mu m + 0 = 14,000 \;\; W/m^2.$$

The irradiation is,

$$G = \left(2\,\mu m \times 5000 \;\; W/m^2 \cdot \mu m\right)/2 + \left(10-2\right)\mu m \times 5000 \;\; W/m^2 \cdot \mu m = 45,000 \;\; W/m^2.$$

Hence, $\alpha = 14,000 \;\; W/m^2 / 45,000 \;\; W/m^2 = 0.311.$ <

(b) From Eq. 12.36, the emissivity is

$$\varepsilon = \int_0^\infty \varepsilon_\lambda \, E_{\lambda,b} \, d\lambda / E_b = 0.4\int_0^2 E_{\lambda,b} \, d\lambda / E_b + 0.8\int_2^5 E_{\lambda,b} \, d\lambda / E_b$$

From Table 12.1, $\lambda T = 2\ \mu m \times 1250K = 2500K$, $F_{(0-2)} = 0.162$

$\lambda T = 5\ \mu m \times 1250K = 6250K$, $F_{(0-5)} = 0.757$.

Hence, $\varepsilon = 0.4 \times 0.162 + 0.8\left(0.757 - 0.162\right) = 0.54.$

$$E = \varepsilon E_b = \varepsilon \sigma T^4 = 0.54 \times 5.67 \times 10^{-8}\,W/m^2 \cdot K^4\left(1250K\right)^4 = 74,751\,W/m^2.$$ <

(c) From an energy balance on the surface, the net heat flux *to* the surface is

$$q''_{net} = \alpha G - E = \left(14,000 - 74,751\right)W/m^2 = -60,751\,W/m^2.$$

Hence the temperature of the surface is *decreasing.* <

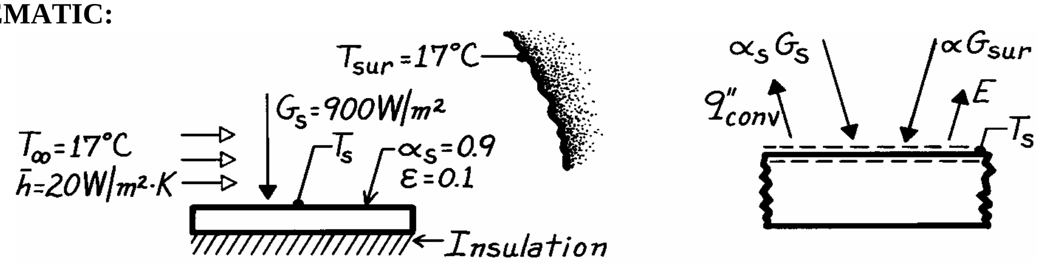
COMMENTS: Note that $\alpha \neq \varepsilon$. Hence the surface is not gray for the prescribed conditions.

PROBLEM 12.109

KNOWN: Plate exposed to solar flux with prescribed solar absorptivity and emissivity; convection and surrounding conditions also prescribed.

FIND: Steady-state temperature of the plate.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Plate is small compared to surroundings, (3) Backside of plate is perfectly insulated, (4) Diffuse behavior.

ANALYSIS: Perform a surface energy balance on the top surface of the plate.

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

$$\alpha_S G_S + \alpha G_{sur} - q''_{conv} - \varepsilon E_b(T_s) = 0$$

Note that the effect of the surroundings is to provide an irradiation, G_{sur}, on the plate; since the spectral distribution of G_{sur} and $E_{\lambda,b}(T_s)$ are nearly the same, accordiing to Kirchoff's law, $\alpha = \varepsilon$. Recognizing that $G_{sur} = \sigma T_{sur}^4$ and using Newton's law of cooling, the energy balance is

$$\alpha_S G_S + \varepsilon\sigma T_{sur}^4 - \bar{h}(T_s - T_\infty) - \varepsilon \cdot \sigma T_s^4 = 0.$$

Substituting numerical values,

$$0.9 \times 900\ \mathrm{W/m^2} + 0.1 \times 5.67 \times 10^{-8}\,\mathrm{W/m^2 \cdot K} \times (17 + 273)^4\,\mathrm{K}^4$$
$$-20\mathrm{W/m^2 \cdot K}(T_s - 290)\mathrm{K} - 0.1\left(5.67 \times 10^{-8}\,\mathrm{W/m^2 \cdot K^4}\right)T_s^4 = 0$$

$$6650\ \mathrm{W/m^2} = 20\ T_s + 5.67 \times 10^{-9}\ T_s^4.$$

From a trial-and-error solution, find

$$T_s = 329.2\ \mathrm{K}.$$ <

COMMENTS: (1) When performing an analysis with both convection and radiation processes present, all temperatures must be expressed in absolute units (K).

(2) Note also that the terms $\alpha\ G_{sur} - \varepsilon\ E_b\ (T_s)$ could be expressed as a radiation exchange term, written as

$$q''_{rad} = q/A = \varepsilon\sigma\left(T_{sur}^4 - T_s^4\right).$$

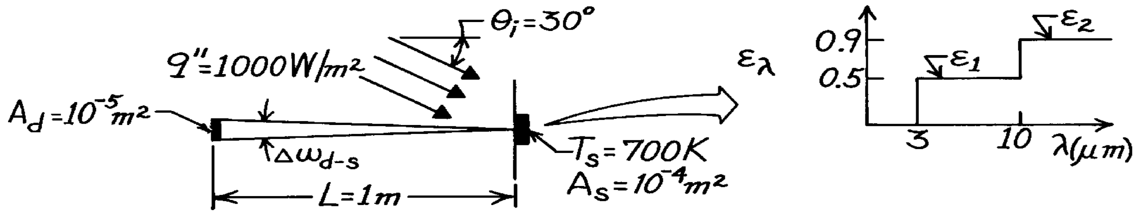
The conditions for application of this relation were met and are namely: surroundings much larger than surface, diffuse surface, and spectral distributions of irradiation and emission are similar (or the surface is gray).

PROBLEM 12.47

KNOWN: Temperature and spectral emissivity of a receiving surface. Direction and spectral distribution of incident flux. Distance and aperture of surface radiation detector.

FIND: Radiant power received by the detector.

SCHEMATIC:



ASSUMPTIONS: (1) Target surface is diffuse, (2) $A_d/L^2 << 1$.

ANALYSIS: The radiant power received by the detector depends on emission and reflection from the target.

$$q_d = I_{e+r} A_s \cos\theta_{d-s} \Delta\omega_{d-s}$$

$$q_d = \frac{\varepsilon\sigma T_s^4 + \rho G}{\pi} A_s \frac{A_d}{L^2}$$

$$\varepsilon = \frac{\int_0^\infty \varepsilon_\lambda E_{\lambda b}(700\ K)\, d\lambda}{E_b(700\ K)} = \varepsilon_1 F_{(3\to 10\,\mu m)} + \varepsilon_2 F_{(10\to\infty)}.$$

From Table 12.1, $\lambda T = 2100\ \mu m{\cdot}K$: $F_{(0\to 3\,\mu m)} = 0.0838$

$\lambda T = 7000\ \mu m{\cdot}K$: $F_{(0\to 10\,\mu m)} = 0.8081$.

The emissivity can be expected as

$$\varepsilon = 0.5(0.8081 - 0.0838) + 0.9(1 - 0.8081) = 0.535.$$

Also,

$$\rho = \frac{\int_0^\infty \rho_\lambda G_\lambda\, d\lambda}{G} = \frac{\int_0^\infty (1-\varepsilon_\lambda) q''_\lambda\, d\lambda}{q''} = 1\times F_{(0\to 3\,\mu m)} + 0.5\times F_{(3\to 6\mu m)}$$

$$\rho = 1\times 0.4 + 0.5\times 0.6 = 0.70.$$

Hence, with $G = q''\cos\theta_i = 866\ W/m^2$,

$$q_d = \frac{0.535\times 5.67\times 10^{-8}\ W/m^2\cdot K^4\,(700\ K)^4 + 0.7\times 866\ W/m^2}{\pi} 10^{-4} m^2 \frac{10^{-5} m^2}{(1m)^2}$$

$$q_d = 2.51\times 10^{-6}\ W. \qquad <$$

COMMENTS: A total radiation detector cannot discriminate between emitted and reflected radiation from a surface.

PROBLEM 13.1

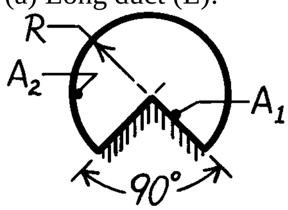
KNOWN: Various geometric shapes involving two areas A_1 and A_2.

FIND: Shape factors, F_{12} and F_{21}, for each configuration.

ASSUMPTIONS: Surfaces are diffuse.

ANALYSIS: The analysis is not to make use of tables or charts. The approach involves use of the reciprocity relation, Eq. 13.3, and summation rule, Eq. 13.4. Recognize that reciprocity applies to two surfaces; summation applies to an enclosure. Certain shape factors will be identified by inspection. Note L is the length normal to page.

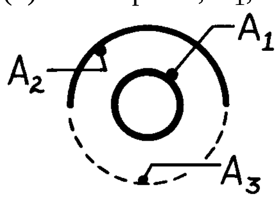
(a) Long duct (L):



By inspection, $F_{12} = 1.0$ <

By reciprocity, $F_{21} = \dfrac{A_1}{A_2} F_{12} = \dfrac{2\,RL}{(3/4)\cdot 2\pi RL} \times 1.0 = \dfrac{4}{3\pi} = 0.424$ <

(b) Small sphere, A_1, under concentric hemisphere, A_2, where $A_2 = 2A$

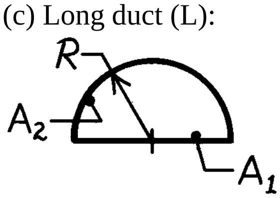


Summation rule $F_{11} + F_{12} + F_{13} = 1$

But $F_{12} = F_{13}$ by symmetry, hence $F_{12} = 0.50$ <

By reciprocity, $F_{21} = \dfrac{A_1}{A_2} F_{12} = \dfrac{A_1}{2A_1} \times 0.5 = 0.25.$ <

(c) Long duct (L):



<

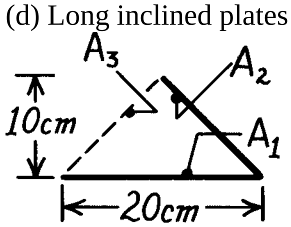
By reciprocity, $F_{21} = \dfrac{A_1}{A_2} F_{12} = \dfrac{2RL}{\pi RL} \times 1.0 = \dfrac{2}{\pi} = 0.637$ <

Summation rule, $F_{22} = 1 - F_{21} = 1 - 0.64 = 0.363.$ <

By inspection,

$F_{12} = 1.0$

(d) Long inclined plates (L):

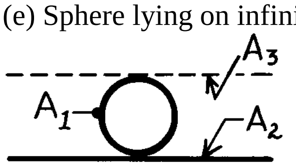


Summation rule, $F_{11} + F_{12} + F_{13} = 1$

But $F_{12} = F_{13}$ by symmetry, hence $F_{12} = 0.50$ <

By reciprocity, $F_{21} = \dfrac{A_1}{A_2} F_{12} = \dfrac{20L}{10(2)^{1/2} L} \times 0.5 = 0.707.$ <

(e) Sphere lying on infinite plane



Summation rule, $F_{11} + F_{12} + F_{13} = 1$

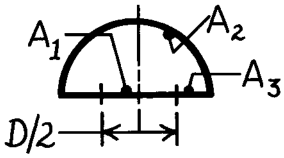
But $F_{12} = F_{13}$ by symmetry, hence $F_{12} = 0.5$ <

By reciprocity, $F_{21} = \dfrac{A_1}{A_2} F_{12} \rightarrow 0$ since $A_2 \rightarrow \infty$. <

Continued …..

PROBLEM 13.1 (Cont.)

(f) Hemisphere over a disc of diameter D/2; find also F_{22} and F_{23}.



By inspection, $F_{12} = 1.0$ <

Summation rule for surface A_3 is written as

$$F_{31} + F_{32} + F_{33} = 1. \quad \text{Hence, } F_{32} = 1.0.$$

By reciprocity, $$F_{23} = \frac{A_3}{A_2} F_{32}$$

$$F_{23} = \left\{ \left[\frac{\pi D^2}{4} - \frac{\pi (D/2)^2}{4} \right] / \frac{\pi D^2}{2} \right\} 1.0 = 0.375.$$

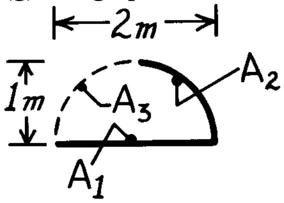
By reciprocity, $$F_{21} = \frac{A_1}{A_2} F_{12} = \left\{ \frac{\pi}{4} \left[\frac{D}{2} \right]^2 / \frac{\pi D^2}{2} \right\} \times 1.0 = 0.125.$$ <

Summation rule for A_2, $$F_{21} + F_{22} + F_{23} = 1 \text{ or}$$

$$F_{22} = 1 - F_{21} - F_{23} = 1 - 0.125 - 0.375 = 0.5.$$ <

Note that by inspection you can deduce $F_{22} = 0.5$

(g) Long open channel (L):



Summation rule for A_1

$$F_{11} + F_{12} + F_{13} = 0$$

but $F_{12} = F_{13}$ by symmetry, hence $F_{12} = 0.50$. <

By reciprocity, $$F_{21} = \frac{A_1}{A_2} F_{12} = \frac{2 \times L}{(2\pi 1)/4 \times L} = \frac{4}{\pi} \times 0.50 = 0.637.$$

COMMENTS: (1) Note that the summation rule is applied to an enclosure. To complete the enclosure, it was necessary in several cases to define a third surface which was shown by dashed lines.

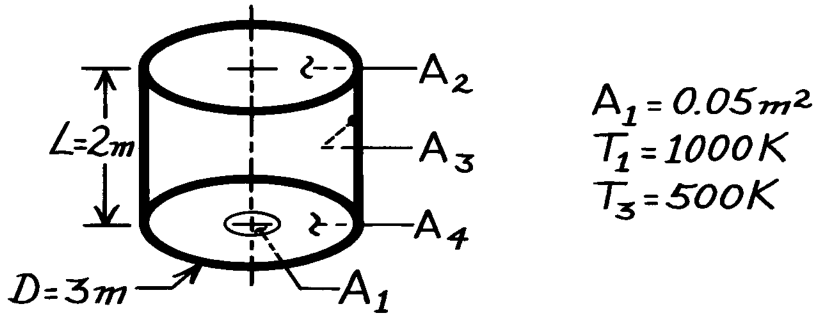
(2) Recognize that the solutions follow a systematic procedure; in many instances it is possible to deduce a shape factor by inspection.

PROBLEM 13.19

KNOWN: Arrangement of three black surfaces with prescribed geometries and surface temperatures.

FIND: (a) View factor F_{13}, (b) Net radiation heat transfer from A_1 to A_3.

SCHEMATIC:



ASSUMPTIONS: (1) Interior surfaces behave as blackbodies, (2) $A_2 >> A_1$.

ANALYSIS: (a) Define the enclosure as the interior of the cylindrical form and identify A_4. Applying the view factor summation rule, Eq. 13.4,

$$F_{11} + F_{12} + F_{13} + F_{14} = 1. \tag{1}$$

Note that $F_{11} = 0$ and $F_{14} = 0$. From Eq. 13.8,

$$F_{12} = \frac{D^2}{D^2 + 4L^2} = \frac{(3\text{m})^2}{(3\text{m})^2 + 4(2\text{m})^2} = 0.36. \tag{2}$$

From Eqs. (1) and (2),

$$F_{13} = 1 - F_{12} = 1 - 0.36 = 0.64. \qquad <$$

(b) The net heat transfer rate from A_1 to A_3 follows from Eq. 13.13,

$$q_{13} = A_1\ F_{13}\ \sigma\left(T_1^4 - T_3^4\right)$$

$$q_{13} = 0.05\text{m}^2 \times 0.64 \times 5.67 \times 10^{-8}\,\text{W/m}^2 \cdot \text{K}^4 \left(1000^4 - 500^4\right)\text{K}^4 = 1700\,\text{W}. \qquad <$$

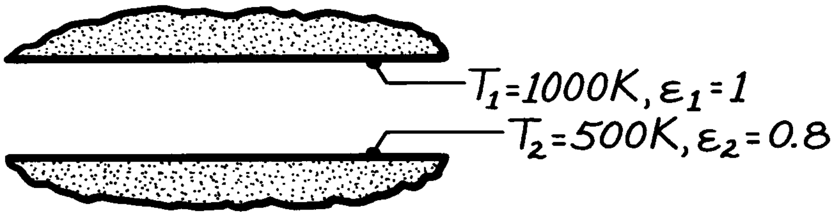
COMMENTS: Note that the summation rule, Eq. 13.4, applies to an enclosure; that is, the total region above the surface must be considered.

PROBLEM 13.41

KNOWN: Two horizontal, very large parallel plates with prescribed surface conditions and temperatures.

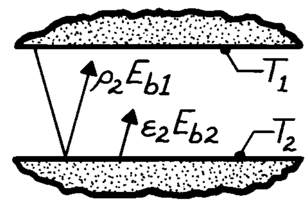
FIND: (a) Irradiation to the top plate, G_1, (b) Radiosity of the top plate, J_1, (c) Radiosity of the lower plate, J_2, (d) Net radiative exchange between the plates per unit area of the plates.

SCHEMATIC:



ASSUMPTIONS: (1) Plates are sufficiently large to form a two surface enclosure and (2) Surfaces are diffuse-gray.

ANALYSIS: (a) The irradiation to the upper plate is defined as the radiant flux incident on that surface. The irradiation to the upper plate G_1 is comprised of flux emitted by surface 2 and reflected flux emitted by surface 1.



$$G_1 = \varepsilon_2 E_{b2} + \rho_2 E_{b1} = \varepsilon_2 \sigma T_2^4 + (1-\varepsilon_2)\sigma T_1^4$$

$$G_1 = 0.8 \times 5.67 \times 10^{-8}\,\text{W/m}^2\cdot\text{K}^4\,(1000\ \text{K})^4 + (1-0.8) \times 5.67 \times 10^{-8}\,\text{W/m}^2\cdot\text{K}^4\,(500\ \text{K})^4$$

$$G_1 = 2835\,\text{W/m}^2 + 11,340\,\text{W/m}^2 = 14,175\,\text{W/m}^2. \qquad <$$

(b) The radiosity is defined as the radiant flux leaving the surface by emission and reflection. For the blackbody surface 1, it follows that

$$J_1 = E_{b1} = \sigma T_1^4 = 5.67 \times 10^{-8}\,\text{W/m}^2\cdot\text{K}^4\,(1000\ \text{K})^4 = 56,700\,\text{W/m}^2. \qquad <$$

(c) The radiosity of surface 2 is then,

$$J_2 = \varepsilon_2 E_{b2} + \rho_2 G_2.$$

Since the upper plate is a blackbody, it follows that $G_2 = E_{b1}$ and

$$J_2 = \varepsilon_2 E_{b1} + \rho_2 E_{b1} = \varepsilon_2 \sigma T_2^4 + 1(1-\varepsilon_2)\sigma T_1^4 = 14,175\,\text{W/m}^2. \qquad <$$

Note that $J_2 = G_1$. That is, the radiant flux leaving surface 2 (J_2) is incident upon surface 1 (G_1).

(d) The net radiation heat exchange per unit area can be found by three relations.

$$q_1'' = J_1 - G_1 = (56,700 - 14,175)\,\text{W/m}^2 = 42,525\,\text{W/m}^2$$

$$q_{12}'' = J_1 - J_2 = (56,700 - 14,175)\,\text{W/m}^2 = 42,525\,\text{W/m}^2 \qquad <$$

The exchange relation, Eq. 13.24, is also appropriate with $\varepsilon_1 = 1$,

$$q_1'' = -q_2'' = q_{12}''$$

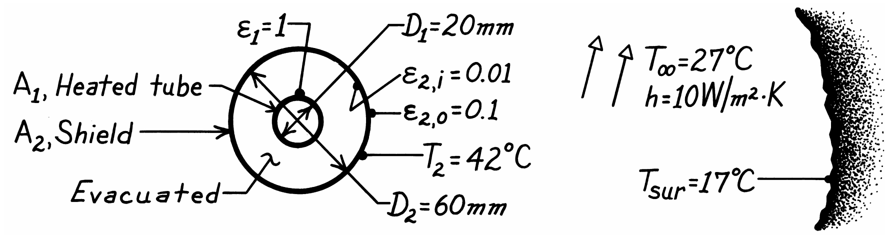
$$q_1'' = \varepsilon_2 \sigma \left(T_1^4 - T_2^4\right) = 0.8 \times 5.67 \times 10^{-8}\,\text{W/m}^2\cdot\text{K}^4\left(1000^4 - 500^4\right)\text{K}^4 = 42,525\,\text{W/m}^2.$$

PROBLEM 13.62

KNOWN: Heated tube with radiation shield whose exterior surface is exposed to convection and radiation processes.

FIND: Operating temperature for the tube under the prescribed conditions.

SCHEMATIC:

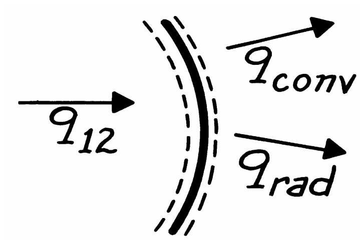


ASSUMPTIONS: (1) Steady-state conditions, (2) No convection in space between tube and shield, (3) Surroundings are large compared to the shield and are isothermal, (4) Tube and shield are infinitely long, (5) Surfaces are diffuse-gray, (6) Shield is isothermal.

ANALYSIS: Perform an energy balance on the shield.

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

$$q_{12} - q_{conv} - q_{rad} = 0$$



where q_{12} is the net radiation exchange between the tube and inner surface of the shield, which from Eq. 13.20 is,

$$-q_{12} = \frac{A_1\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_{2,i}}{\varepsilon_{2,i}}\frac{D_1}{D_2}}$$

Using appropriate rate equations for q_{conv} and q_{rad}, the energy balance is

$$\frac{A_1\sigma\left(T_1^4 - T_2^4\right)}{1 + \frac{1-\varepsilon_{2,i}}{\varepsilon_{2,i}}\frac{D_1}{D_2}} - hA_2\left(T_2 - T_\infty\right) - \varepsilon_{2,o}A_2\sigma\left(T_2^4 - T_{sur}^4\right) = 0$$

where $\varepsilon_1 = 1$. Substituting numerical values, with $A_1/A_2 = D_1/D_2$, and solving for T_1,

$$\frac{(20/60)\times 5.67\times 10^{-8}\,\mathrm{W/m^2\cdot K^4}\left(T_1^4 - 315^4\right)\mathrm{K}^4}{1 + (1-0.01/0.01)(20/60)} - 10\ \mathrm{W/m^2\cdot K}(315-300)\mathrm{K}$$

$$-0.1\times 5.67\times 10^{-8}\,\mathrm{W/m^2\cdot K^4}\left(315^4 - 290^4\right)\mathrm{K}^4 = 0$$

$$T_1 = 745\ \mathrm{K} = 472°\mathrm{C}. \qquad <$$

COMMENTS: Note that all temperatures are expressed in kelvins. This is a necessary practice when dealing with radiation and convection modes.

ME225 - Heat Transfer
A core curriculum of Mechanical Engineering

- Homework Problems & Solutions
- Date: 2010.02
- By: David Malawey

The following section of this PDF is approximately 197 pages of homework problems and solutions for the 1-semester course. The materials consist of some problems from the textbook, with solutions & notes from the professor, some problems written by the professor, and hand-drawn solutions, and possibly problems from other sources. Altogether these problems are delivered to the students for the homework of the complete course for Heat Transfer.

ANSWERS TO END-OF-CHAPTER PROBLEMS
Fundamentals of Heat and Mass Transfer (6th Edition)
Introduction to Heat Transfer (5th Edition)

F.P. Incropera
D.P. DeWitt
T.L. Bergman
A.S. Lavine

CHAPTER 1

1.1	14.5 W/m^2, 58 W
1.2	2667 W
1.3	4312 W, $4.14/d
1.4	0.10 W/m·K
1.5	8400 W
1.6	19,600 W, 120 W
1.7	54 mm
1.8	16.6 W/m^2, 35.9 W
1.9	375 mm
1.10	110.40°C, 110.24C
1.11	1.1°C
1.12	(a) 9800 W/m^2
1.13	(a) 1400 W/m^2; (b) 18,000 W/m^2
1.14	(a) 22.0 $W/m^2 \cdot K$; (b) 22.12, 0.6
1.15	4570 $W/m^2 \cdot K$, 65 $W/m^2 \cdot K$
1.16	51.8°C, 3203°C
1.17	6.3 m/s
1.18	0.35 W, 5.25 W
1.19	2.94 W
1.21	15 mW
1.22	6.3 $W/m^2 \cdot K$
1.23	102.5°C
1.25	254.7 K
1.27	0.42, 264 W
1.28	(a) 18,405 W; (b) $6450
1.30	3.5%
1.31	(a) 0.223 W; (b) 3.44 W
1.32	(a) 8.1 W; (b) 0.23 kg/h
1.33	100°C
1.34	(a) 0, 144 W, 144 W, 0; 0, 144 W, 144 W, 0; (b) 2.04×10^5 W/m^3; (c) 39.0 $W/m^2 \cdot K$
1.35	(a) 0.052°C/s; (b) 48.4°C
1.36	375 W, 1.8×10^{-4} W, 0.065 W
1.37	6380 kWh, $510 or $170
1.38	(a) 4180 s; (b) 319 K, 359 K; (c) 830 K

1.39 (a) 0.0181 m^3/s, 4.7 m/s; (b) 5.97 W
1.40 840 kW
1.41 (a) 32.5 kW/m^2, 126 kW/m^2, 17.6 K/s, 68.6 K/s
1.43 (a) 104 K/s; (b) 1251 K
1.46 (a) 1950 A
1.47 132 J/kg·K
1.48 (a) –0.084 K/s; (b) 439 K
1.49 (a) 1.41×10^{-3} kg/s
1.50 3.2 h
1.51 (a) 60.6×10^{-3} $kg/s\cdot m^2$, 121 g/m^2; (b) 32.3×10^{-3} $kg/s\cdot m^2$
1.53 49°C
1.54 (a) 7.13×10^{-3} m^3/s; (b) 70°C
1.55 (a) 86.7°C; (b) 47°C
1.56 (a) –0.044 K/s; (b) 230 W, 230 W
1.57 (a) 26°C; (b) 4.0×10^{-5} ℓ/s
1.58 (b) 80°C
1.59 12.2 $W/m^2\cdot K$, 12.2 $W/m^2\cdot °C$
1.60 (a) 600 K
1.61 375 $W/m^2\cdot K$
1.62 (a) 5500 W/m^2; (b) 87.8°C
1.63 (a) 5268 W; (b) 41°C
1.65 345°C
1.66 (a) 84°C
1.67 214 K, 20.0 mW
1.68 (a) 190.6 W
1.69 (a) 153°C
1.70 (a) 386 W/m^2; (b) 27.7°C; (c) 55%
1.71 (a) 47.0°C or 39.9°C

CHAPTER 2

2.8 (a) –280 K/m, 14.0 kW/m^2; (b) 80 K/m, –4.0 kW/m^2; (c) 110°C, –8.0 kW/m^2; (d) 60°C, 4.0 kW/m^2; (e) –20°C, –10.0 kW/m^2
2.9 (a) 2000 K/m, –200 kW/m^2; (b) –2000 K/m, 200 kW/m^2; (c) 2000 K/m, –200 kW/m^2
2.11 0, 60 K/m
2.12 100°C, 18.75 W; –40°C, 16.25 W
2.14 1010 W, $1050; 151 W, $157; 10.1 W, $10
2.15 (a) 85 μW
2.16 14.5 $(Btu/h\cdot ft^2\cdot °F)^{-1}$, 18 $(Btu/h\cdot ft^2\cdot °F)^{-1}$, 18 $(Btu/h\cdot ft^2\cdot °F)^{-1}$
2.17 (a) 15.0 W/m·K, 400 K; (b) 70.0 W/m·K, 380 K
2.18 (b) 5.9×10^{-3} W/m·K, (c) 0.74×10^{-3} °C, (d) 25.02°C
2.19 765 J/kg·K, 36.0 W/m·K
2.22 (a) 0, 0.98×10^5 W/m; (b) 56.8 K/s
2.23 (a) 2×10^5 W/m^3; (b) 0, 10,000 W/m^2
2.24 (a) 200 W/m^2, 182 W/m^2, 18 W/m^2; (b) 4.3 $W/m^2\cdot K$

2.25 (b) 2×10^5 W/m^3; (c) –2950 W/m^2, 5050 W/m^2; (d) 51 W/m^2·K, 101 W/m^2·K; (f) –2×10^5 W/m^3; (g) 20°C, 4.94×10^6 J/m^2
2.26 (a) 10^6 W/m^3; (b) 120°C, 10^4 K/m, –10^5 K/m^2; (c) 220°C, 10^4 K/m, –10^5 K/m^2; (d) 220°C, 2×10^4 K/m, –2×10^5 K/m^2
2.40 (a) 0.20 m; (b) 0; (c) –144,765 W; (d) 72,380 W, –72,380 W
2.41 (d) 133°C, 122°C, 133.1°C
2.43 (d) 52.5°C
2.45 (c) 18.0 kW/m^2, –360 K/m; (d) 8.73×10^6 J/m^2; (e) 8.73×10^6 J/m^2
2.51 (a) 1.8×10^6 W/m^3; (c) 1.8×10^5 W/m^2; (d) 7.77×10^7 J/m^2
2.53 (a) 25°C, 35°C; (b) 50°C, 30°C; (c) 86.1°C
2.54 (b) 3.18×10^8 W/m^3, 1.59×10^5 W/m^2

CHAPTER 3

3.2 (a) 7.7°C, 4.9°C
3.3 (a) 1270 W/m^2
3.4 (b) 2830 W/m2
3.5 14.1 W/m^2
3.6 (a) 996 W/m^2·K, 0.40%; (b) 14.5 W/m^2·K, 37.9%
3.7 (a) 0.553; (b) 22.1°C, 10.8°C; (c) –56.3°C
3.8 (a) 29.4 W
3.9 1.53 W/m·K
3.10 (a) 17 mm; (b) 20.7 mm; (c) 40.3 kW, 550 W
3.11 (b) 86 mm
3.12 0.79 W/m·K, 43.6 W/m·K
3.13 (b) 4.21 kW; (c) 0.6%
3.14 1.30×10^8 J
3.15 0.185 K/W
3.16 2.13
3.17 (b) 64 m, 8, 307 kW, 40%
3.18 (a) 0.0875 m, 0.963 m, 0.0008 m, 0.015 m; (b) 573.5 K, 313.1 K, 53.5 K, 313.1 K; (c) 135 W, 33.75 W, 108.4 W, 10.2 W
3.19 (b) 327°C
3.20 (a) 34,600 W/m^2
3.21 3.70×10^4 W/m^2, 55.6°C
3.22 (a) 762 W
3.24 590 W
3.25 (a) 1.97×10^6 K/W
3.27 (b) 49°C; (c) 67,200 W/m^2
3.28 (a) 0.268 W
3.29 (b) 5.76 kW
3.30 (b) 189 W
3.34 (a) 48.3×10^6 K/W; (b) 62.3°C
3.35 (a) 603 W/m
3.36 0.784 m, 0.784 m, 0.025 m
3.37 2380 W/m

3.39 (a) 12.6 W/m; (b) 7.7 W/m
3.40 (a) 214 mm, 420 W/m
3.41 (a) 251 W/m; (b) 23.5°C
3.42 (a) 4 W/m, 1.27×10^6 W/m^3; (b) 58°C; (c) 37.6°C, 34.8°C
3.43 4.5 W/m, 13 mm
3.44 (b) 239°C
3.45 (a) 47.1 W/m; (c) 3.25 h
3.47 (a) 779°C; (b) 1153°C, 779°C; (c) 0.0175 m, 318°C
3.48 (a) 3727 W/m, 163 W/m; (b) 0.26 yr
3.49 1830 W/m
3.51 5 mm
3.52 (b) 1040 W/m, 407 K, 325 K
3.53 (a) 0.01 m; (b) 770 W/m, 909 W/m; (c) 55 mm
3.54 8670 W
3.55 (a) 99.8%
3.57 0.062 W/m·K
3.59 5.34 mm
3.60 (a) 489 W, (b) 120°C
3.61 13.5 mm, 91%
3.62 601 K
3.63 181 W/m^2·K
3.64 (b) 35.5 mW, 44.9 mW
3.65 (b) 3000 W/m^2
3.67 (a) 1.69 m
3.68 77.9°C
3.69 0.157 W
3.72 212°C
3.73 (a) 4.0×10^6 W/m^3, 15.3 W/m·K; (c) 835°C, 360°C
3.74 (a) 50.2°C
3.76 (a) 180°C; (c) 50°C; (d) 180°C
3.78 (a) 530°C, 380°C; (b) 328°C, 290°C
3.79 (b) 60°C, 65°C; (c) 200 W/m^2; (d) 55°C
3.86 29 A, 3.0 m, 3.2 kW
3.88 (b) 1458 K
3.89 (a) 6410 A; (b) –15,240 W; (c) –10,990 W/m, –11,350 W/m, 4250 W/m, 3890 W/m
3.90 804 s
3.91 (a) 938 K, 931 K; (b) 3×10^8 W/m^3
3.92 (a) 71.8°C, 51.0°C; (b) 192°C
3.95 (a) 36.6°C; (b) 129.4°C; (c) 337.7°C
3.96 (a) 5.26°C, 5.14°C
3.99 63.7°C, 160 W/m
3.102 (c) –17.2 W, 23.6 W
3.103 (b) 164.3°C, 145.1°C
3.105 (b) 347°C
3.107 (b) 160°C
3.109 510 nm

3.113 (a) 305°C, (b) 272°C
3.115 (c) 62.4°C
3.116 (b) 508 W
3.117 (c) 0.333, 607°C
3.119 (a) 420%; (b) 29%
3.120 156.5°C, 128.9°C, 107.0°C
3.121 (a) Case A: 151 W/m, 0.96, 20.1, 0.50 m·K/W, 95.6°C
Case B: 144 W/m, 0.92, 19.3, 0.52 m·K/W, 96.0°C
Case D: 450 W/m, 0, 60.0, 0.17 m·K/W, 25°C
3.122 (a) Case A: 151 W/m, 0.96, 20.1, 0.50 m·K/W, 95.6°C
Case B: 144 W/m, 0.92, 19.2, 0.52 m·K/W, 96.0°C
Case D: 450 W/m, 0, 60.0, 0.167 m·K/W, 25°C
3.123 Rectangular fin: 130 W/m, 0.98, 4.5×10^{-5} m^2
Triangular fin: 117 W/m, 0.98, 2.3×10^{-5} m^2
Parabolic fin: 116 W/m, 0.96, 1.5×10^{-5} m^2
3.124 121 W
3.125 (a) 0.37 W; (b) 1.04×10^5 W
3.126 5.2 nm, 0.2 nm
3.127 17.5 W/m·K
3.128 (a) 1.31 W; (b) 1.34 W
3.129 56.6 W/m·K
3.130 (b) 5995 W, –4278 W
3.131 (a) 2.44×10^{-3} K/W
3.132 (a) 31.8 W
3.133 1.30×10^{-3} W, 8.64×10^{-3} W
3.134 (b) 50.9 W
3.136 (a) 276 W
3.138 37.0 W
3.140 (b) 2830 W/m
3.141 1315%
3.142 93.7°C
3.143 (a) 0.99, 6.0; (b) 110.8 W/m
3.144 (a) 12.8 W; (b) 2.91 kW/m
3.145 (a) 0.97, 11.1; (b) 6.82 kW/m
3.146 394 K, 383 K, 381 K
403 K, 392 K, 382 K, 381 K
3.147 39,300 W/m, 405 K, 393 K, 384 K, 382 K
3.148 (a) 1.40 W
3.149 4025 W/m
3.152 3.3×10^{-5} ℓ/s

CHAPTER 4

4.2 94.5°C
4.3 5.6 kW/m

4.7 (c) 2.70, 3.70, 0.19°C, 0.14°C
4.9 92.7°C
4.10 84 W/m
4.11 9.9 W/m
4.12 94.9°C
4.13 110 W/m
4.14 6.72L, 612 W/m
4.15 1122 W/m
4.16 12.5 W/m, 11.3 W/m
4.17 5.30 kW
4.18 72.1°C, 78.1°C, 70.0°C
4.19 (a) 10.3 W; (b) 0.21 mm
4.20 316 kW
4.21 (a) 1.62 kW/m; (b) 93°C
4.22 (a) 0.1°C
4.23 3.62 kW, 254°C, 251°C
4.24 (a) 4.46 W, 98.2°C; 3.26 W, 78.4°C; (b) 4.09 W, 92.1°C; 3.05 W, 75.1°C
4.25 (a) 1.2°C
4.26 694 W
4.27 (a) 57°C; (b) 138 W
4.28 156.3 W, 81.3°C
4.29 (a) 745 W/m; (b) 3.54 kg/s
4.30 (a) 74.6 kW/m; (b) 2.38×10^8 W/m^3, 315°C
4.31 0.70 W
4.43 6710 W/m
4.44 (a) 422 K, 363 K, 443 K; (b) 156 W/m
4.45 (a) 362.4 K, 390.2 K, 369.0 K; (b) 7500 W/m
4.46 (a) 122.0°C, 94.5°C, 95.8°C, 79.7°C; (b) 1117 W/m; (c) –1383 W/m
4.47 (a) T1 = 46.6°C, T2 = 45.7°C, T3 = 45.4°C, T4 = 49.2°C, T5 = 48.5°C, T6 = 48.0°C, T7 = 47.9°C, 10,340 W/m
4.48 (a) T1 = 160.7°C, T2 = 95.6°C, T3 = 48.7°C; (b) 743 W/m
4.49 (a) 1.473 W/m; (b) T3 = 89.2°C
4.50 (a) 205.0 W/m; (b) 156.3 W/m
4.51 (a) 118.8°C, 156.3°C, 168.8°C, 206.3°C, 162.5°C; (b) 117.4°C, 156.1°C, 168.9°C, 207.6°C, 162.5°C
4.52 (a) 272.2°C, 952 W/m; (b) 271.0°C, 834 W/m
4.53 (a) 348.6 K, 368.9 K, 374.6 K, 362.4 K, 390.2 K, 398.0 K; (b) 1.53×10^8 W/m^3
4.54 3.00 kW/m
4.55 (a) 1.57 kW/m; (b) 1.52 kW/m
4.56 456 W/m
4.58 1487 W/m
4.59 (a) 7.72 kW/m; (b) 7.61 kW/m
4.60 94.0°C
4.61 1477 W/m
4.62 (a) 100 W/m, 1
4.64 (a) 1010 W/m, 100°C, 1; (b) 805 W/m, 100°C

4.65 (b) 128 W/m
4.66 (b) 1.48×10^8 W/m^3
4.67 (b) 1135 W/m, (c) –1365 W/m
4.69 (a) 2939 W/m^2
4.70 (b) 1205 W/m
4.71 1.604 W
4.73 (c) 25.0 W/m
4.75 (a) 2.38×10^{-3} m·K/W; (b) 3.64×10^{-3} m·K/W
4.77 (a) 878 W/m, 5.69×10^{-2} m·K/W, 220 W
4.78 (a) 131 W/m; (b) 129 W/m
4.79 (a) 1683 W/m
4.80 (a) 52.6 kW/m^2
4.82 (b) 2.43×10^7 W/m^3; (c) 47.5°C, 0.45

4S.1 2.83, 7.34 kW/m
4S.2 4.26ℓ, 245 W/m
4S.3 (a) 50°C; (b) 0.53ℓ, 3975 W/m; (d) 1.70ℓ, 12,750 W/m
4S.4 2.34ℓ
4S.5 2.58ℓ, 11,600 W/m; 1.55ℓ, 6975 W/m
4S.6 (a) 7.8 kW/m; (b) 0.24 m
4S.7 4ℓ, 45.6 kW/m; 3.5ℓ, 39.9 kW/m
4S.8 3500 W/m, 4500 W/m

CHAPTER 5

5.5 1122 s
5.7 35.3 W/m^2·K
5.8 7.04 h
5.9 168 s
5.10 859 s
5.11 968 s, 456°C
5.12 984 s, 272.5°C
5.13 0.0041 m^2·K/W
5.14 (a) 84.1°C, 83.0°C
5.15 21.8 m
5.16 1.08 h, 1220 K
5.17 (a) 209 s, 105 m; (b) 1.03 m/s, 1.69 m/s
5.18 88.7°C, 8.31 s
5.19 (b) 825 s, 122°C
5.20 (b) 63.8°C
5.21 3.54 m
5.22 (d) 1.04 h, 0.826 h, 0.444 h
5.23 2.52 m, 0.022 J
5.24 (a) 1190 s, 199 s; (b) 24.1 s; (c) 21.0 s
5.25 (a) 1 ms
5.26 1.56×10^{-4} s, 2.28×10^{-5} s

5.27	80°C, 38.3 s
5.28	45.7°C, 13.0 s
5.29	(a) 25.8 s, 9.5 s, 616 s; 118.9 s, 43.8 s, 2840 s; 40 s; (b) 29.5 s, 11.0 s, 708 s
5.30	(a) 98.1°C; (b) 1.67 h
5.31	(a) 90 μW; (b) 0.71 μs, 1,71 μs; (c) 400×10^6 bits, 6.84 s
5.32	(a) 86 ms; (b) 147 ms
5.33	(a) 11.3 min; (b) 5.9 min
5.35	960 s
5.36	(a) 56.3 min
5.37	861 s
5.38	(a) 45.4°C, 43.1°C, –7305 W/m^2, -2.72×10^7 J/m; (b) 4.4 min
5.39	(a) 33,800 s
5.40	491 s
5.41	(a) 164 s, 367 s
5.42	(a) 10.9 s
5.43	(a) 63 s; (b) -2.36×10^4 °C/m
5.44	(a) 1100 s
5.45	0.613 W/m·K, 2.73×10^6 J/m^3·K
5.46	(a) 45.3 min; (b) 2.21×10^7 J/m^2
5.47	63.8 s, 51.8°C
5.48	(a) 486 K
5.49	(a) 145 s
5.50	(a) 194 s
5.51	596 K
5.52	254°C
5.53	17 min, 149 kW
5.54	579 s
5.55	(a) 0.30 W/m·K
5.57	3.4 s
5.58	140 s, 36 mm/s
5.59	(a) 42 s, 114°C; (b) 40 kW
5.60	(b) 72 s; (c) 7125 W/m^2; (d) 3364 J; (e) 428 K
5.61	(b) 2.8 h, 107 s; (c) 3.48×10^6 J, 3405 J
5.62	(a) 98.6 s, (b) 100°C
5.63	1020 s, 257.3°C
5.64	(a) 100 s
5.65	(a) 94.2 s, 0.0025; (b) 3.0 s
5.68	4.99×10^5 J/m^2
5.69	1793 s
5.71	(a) 2.81 min; (b) 56 kJ
5.73	53.5°C
5.74	(a) 0.34 mm, 2.36 mm
5.75	1.41 W/m·K
5.76	0.45 W/m·K
5.77	(a) 276°C, 315°C
5.78	(a) 310 s

5.81 365.9 K
5.82 (b) 0.43 s
5.84 51.4 kN
5.85 (a) 870 W; (b) 28.2 yr
5.86 21.8 ns
5.87 10.6 s
5.88 (a) 31.8°C, 58.°C; (b) 34.2°C, 65.9°C
5.89 4.13 yr
5.90 (a) 32°C, 22°C; (b) 34.3 W/m^2, 22°C, 27°C; (c) 27.4°C, 26.6°C
5.91 3.1 μm
5.96 (b) 230°C
5.100 24.1°C, 71.5°C
5.105 161 s, 1364°C, 2.42 m
5.106 275°C, 312°C
5.107 502.3 K, 300.1 K
5.108 (a) 119.3°C, 45.1°C
5.109 (a) 66°C, 32°C
5.113 (a) ~230 s
5.114 (a) ~14.3 s
5.115 (b) 550 s
5.116 (a) 136 s; (b) 73 s
5.118 (b) 158°C
5.120 (a) 54.8°C; (b) 54.7°C
5.123 (b) 806 K, 1.17 s
5.124 (a) 402.7 K, (b) 368.7 K, (c) 362.5 K
5.128 (a) 54 days; (b) 13 days

5S.1 1170 s, 410°C, 537°C
5S.2 96 $W/m^2 \cdot K$
5S.3 0.0073 m/s
5S.4 7.6 min
5S.5 (a) 3607 s; (b) 51°C
5S.6 199°C
5S.7 (a) 12 s; (b) –1.0°C; (c) –3.4 J
5S.8 (a) 5.1 s, 68.3°C
5S.9 1.83 h
5S.10 434 K, 320 K
5S.11 (a) 260°C
5S.12 ~2.75 h
5S.13 561 K, 604 K
5S.14 (a) 402.7 K, 370.5 K, 362.4 K

CHAPTER 6

6.2 705 $W/m^2 \cdot K$, –171.4°C/m, –17,060°C/m
6.3 –9200 W/m^2

6.4 2.0
6.5 1.33
6.7 10.9 $W/m^2 \cdot K$, 1.0
6.8 42.5 $W/m^2 \cdot K$
6.10 67.35 $W/m^2 \cdot K$
6.13 600 W/m^2, 18.9 W
6.14 (a) 20.9 m/s
6.15 (a) 31.4 m; (b) 0.157 m
6.16 7.95 m, 275 m, 0.056 m; 10.5 m, 20.9 m, 0.049 m
6.17 (b) 29.9×10^{-6} m^2/s, 5.98×10^{-6} m^2/s, 2.99×10^{-6} m^2/s; (c) 5.23 m, 1.05 m, 0.523 m
6.18 40 $W/m^2 \cdot K$
6.19 2066 W
6.20 (a) 34.3 $W/m^2 \cdot K$; (b) 59.0 $W/m^2 \cdot K$
6.21 1.11
6.23 2.0
6.24 6.69
6.26 42.5°C
6.27 (a) 47.2°C; (b) 13.2 m/s
6.29 88.6
6.31 281, 284, 180, 465 $W \cdot s^{0.85}/m^{2.7} \cdot K$
6.32 38.3 $W/m^2 \cdot K$
6.33 16.6 mm, 250 K
6.34 240 W
6.35 2.66 kW
6.36 26.0 N/m^2
6.37 0.785 N
6.38 4260 W/m^2
6.39 14.3 W
6.40 1.55×10^{-3} m/s
6.41 (a) 0.0179 m/s; (b) 0.75×10^{-5} kg/s; (c) 5.31×10^{-5} kg/s
6.42 2.76×10^{-5} $kg/s \cdot m^2$, 1.37×10^{-3} m/s
6.43 10^{-6} $kg/s \cdot m^2$
6.45 0.025 m/s
6.46 385, 6.29×10, 0.62, 0.0073; 1613, 6.29×10^4, 2.56, 0.0187; 1.06×10^7, 1577, 6.74×10^5, 76.9
6.47 (a) 120 $W/m^2 \cdot K$; (d) 0.51 m/s
6.48 (a) 4.64×10^{-4} kg/s; (b) 1247 W
6.49 (a) 0.975×10^{-3} $kmol/m^3$, 0.0258 atm; (b) 9.28×10^{-4} $kg/s \cdot m^2$
6.50 (a) 1.63×10^{-4} kg/s; (b) 282.2 K
6.51 1.50×10^{-3} kg
6.52 359 $W/m^2 \cdot K$
6.53 (a) 25.8 m/s; (b) 25.2 $W/m^2 \cdot K$, 80.8 $W/m^2 \cdot K$, 162 $W/m^2 \cdot K$
6.54 (a) 0.031, 0.80; (b) 198 $W/m^2 \cdot K$; (c) 74%
6.55 2×10^{-3} $kg/s \cdot m^2$
6.56 0.00395 m/s

6.57 (b) 9.92 W
6.58 7.78 km
6.59 (a) 7.25×10^{-6} kg/s; (b) 22.4 W, 2791 W/m^2
6.60 (a) +795 W/m^2, +131 W/m^2, –563.1 W/m^2; (c) 1227 W/m^2
6.61 784 W
6.62 (a) 0.70 kg/h; (b) 85.6°C
6.63 21.8 W
6.64 2950 W
6.65 (a) 4.7°C; (b) 0.0238 m/s; (c) 16.2°C
6.66 (a) 0.0142 bar, 0.214; (b) 0.225; (c) 0.242
6.67 0.218
6.68 (a) 22.7 W/m^2·K, 454 W; (b) 2.12×10^{-2} m/s, 3.32 kg/h; (c) 2685 W
6.69 (a) 0.172 m/s, 65.3 s; (b) 173 W/m^2·K, 77,500 W/m^2; (c) 429°C
6.70 (a) 6.71×10^{-3} m/s; (b) 8.97 W/m^2·K
6.71 (a) 0.0113 m/s; (b) 90 J
6.72 (b) –0.32 K/s

6S.2 40.83°C
6S.3 (b) 1510 W/m, 453 kg/s^2·m
6S.4 (a) 34.2 N/m^2, 0.738 N/m^2; 6840 W/m^2, 148 W/m^2; (b) 1.37×10^6 W/m^3, 2.95×10^4 W/m^3; (c) 34.0°C, 30.5°C
6S.5 (a) 0.0028, 0.0056, 13.4; (b) 3.38
6S.7 (b) 117°C
6S.8 (a) 6.66×10^7 W/m^3; (b) 1460 W; (c) 81.2°C, 303°C
6S.14 (e) 6.71×10^{-4} kg/s

CHAPTER 7

7.1 (a) 3.99 mm, 4.48 mm; 0.93 mm, 0.52 mm; 23.5 mm, 1.27 mm; 0.34 mm, 1.17 mm
7.2 (a) 0.147 m, 0.0143 m; (b) –1300 W/m^2, 0.0842 N/m^2; (c) 0.337 N/m, -5200 W/m
7.3 (a) 0.126 mm, 0.399 mm, 1.262 mm, 141 mm; (b) 6.07 N/m^2, 1.92 N/m^2, 0.61 N/m^2, 0.528 m/s, 0.167 m/s, 0.053 m/s
7.8 (a) 8.71×10^5 W/m^3; (b) 158.4°C
7.9 (a) 51.1 W, 12.2 W, 8.3 W, 255.3 W
7.10 13,600 W/m, 9780 W/m, 5530 W/m
7.11 (a) 0.257 N, 8.13 W/m^2·K, 1950 W; 0.227 N, 7.16 W/m^2·K, 3440 W; (b) 0.620 N, 9050 W/m^2·K, 5430 W; 0.700 N, 12,600 W/m^2·K, 15,100 W
7.12 (c) 4110 W/m^2·K, 4490 W/m^2·K, 5070 W/m^2·K
7.13 3.2×10^5, 1.5×10^6
7.15 (a) 17.6 W; (b) 143.6 W
7.16 (a) 5830 W/m
7.17 29°C
7.18 (a) 13.5 W, 47.6°C; (b) 13.9 W, 41.0°C
7.19 (a) 14.3 W, 125.9°C
7.20 (a) 33.8°C, 797 W; (b) 675 W; (c) 90.6 kW

7.21	1570 W
7.22	(a) 8.7 $W/m^2 \cdot K$, 14.5 $W/m^2 \cdot K$, 11.1 $W/m^2 \cdot K$
7.23	(b) ±4%
7.24	6780 W, -0.26°C/s
7.25	(b) 213°C
7.26	(b) 256°C; (c) 210°C
7.27	–0.99 K/s, –1.47 K/s, 1.91 m
7.28	(b) 0.47; (c) 0.47
7.29	46.2°C
7.30	2.73 m/s
7.31	(a) 23.4 W, 28.2°C; (b) 271 W, 72.6°C; 105 W, 200°C
7.32	(a) 27°C; (b) 77°C
7.33	(a) 64°C; (c) 6.6 m/s
7.34	42.5°C
7.35	(a) 98.3°C, 17.1 $W/m^2 \cdot K$; (b) 86.1°C
7.36	(a) ∞, 2.61 $W/m^2 \cdot K$; ∞, 2.74 $W/m^2 \cdot K$; (b) 4.22 $W/m^2 \cdot K$, 5.41 $W/m^2 \cdot K$
7.37	(a) 55 W; (b) 39 W
7.38	0.30 W, 0.81 W
7.39	0.66 W, 0.60 W
7.40	(a) 30.8 $W/m^2 \cdot K$, 56.7 $W/m^2 \cdot K$; (b) 0.633, 0.555
7.41	(a) 71.1 W/m, 20.4 kW/m, 1640 W/m
7.42	3.24 N/m, 520 W/m
7.43	(a) 603 K; (b) 200 s
7.44	(a) 562 K; (b) 155 s
7.45	70.4 W
7.46	(a) 2.20 W; (b) 37.4 mm; (c) 89.6; (d) 435%
7.47	(a) 235 $W/m^2 \cdot K$; (b) 0.87 W; (c) 1.02 W
7.48	(a) 1012°C, 1014°C
7.49	(b) 97 m/s
7.50	(b) 195 mA
7.52	(a) 27.6°C; (b) 27.6°C
7.53	(a) 45.8°C; (b) 68.3°C
7.54	(a) 0.924 W
7.55	(a) 1.41 W
7.56	(a) \$0.415/m·d; (b) \$0.363/m·d
7.57	(a) 3640 W/m
7.58	452.2 K
7.59	(b) 18.7 mm
7.61	(a) 0.20 m/s; (b) 2.80×10^4 W/m^2, 52%
7.62	(a) 12.8 s
7.63	(b) 340°C; (c) 309°C
7.65	3.26 m/s
7.66	(a) 0.011 N; (b) 3.14 W
7.67	(a) 0.489 N, 510 W; (b) 0.452×10^{-3} N, 1.59 W
7.68	10.3 W
7.69	(a) 67 s; (b) 48 s

7.70 (a) 18.8°C; (b) 672°C
7.71 (a) 2 m/s, (b) 1.11 mm, 1.25×10^{-3} m^3
7.72 (a) 737°C; (b) 274 s; (c) 760°C, 230 s
7.73 2.1 m/s, 1.6 m
7.74 1.1 m/s, 10.3 m
7.75 (a) 0.0011 s, 166.7 m/s; (b) 7.8×10^{-4} s
7.76 180 mm
7.77 (a) 0.3 m/s; (b) 1.86 mm
7.78 (a) 6.83 s; (b) 936 K
7.79 (b) 337°C
7.80 726 K, 735 K
7.81 (a) 92.2 W; (b) 1.76×10^8 W/m^3, 1422 K; (c) 1789 K
7.82 (a) 20.4; (b) 45.8°C
7.83 234 W/m^2·K, 0.0149 bar, 28.4 kW/m
7.84 58.5 kW, 5.9×10^{-3} bar
7.85 16, 684 N/m^2
7.86 –532 kW/m
7.87 (a) 7670 W, 47.6°C; (b) 195 N/m^2, 56 W
7.88 (a) 39°C; (b) 0.00993 bar, 6.26 kW
7.89 (a) 435 W/m^2·K; (b) 321 K, 2453 W
7.91 (a) 363 K; (b) 355 kW, 0.163 kg/s
7.92 1.68 W
7.93 1.45 K/s
7.94 (c) 0.179 kg/s
7.96 11,600 W, 140 W
7.97 (a) 3.56 s, 17 μm
7.98 (a) 470.5 W/m^2·K; (b) 40.1 s
7.99 5044 s
7.100 65.7 kW
7.101 (a) 941°C, 333 W; (b) 1314°C
7.102 (a) 10.5°C, 0.0134 kg/s
7.103 (a) 41.2°C, 0.0154 kg/s
7.104 (a) 0.0728 m/s; (b) 0.0028 kg/s
7.105 (a) 9.29×10^{-4} kg/s·m, 2265 W/m
7.106 5.93 kW
7.107 (a) 0.0135 kg/s·m
7.108 3.57 W
7.109 (a) 4.48×10^{-3} kg/s·m^2; (b) 11,800 W/m^2; (c) 9.95×10^{-4} kg/s·m
7.110 (a) 9.5×10^{-4} kg/s, 9.5×10^{-4} kg/s, 9.3×10^{-4} kg/s; (b) 334.7 K, 332.8 K, 327.1 K
7.112 (a) 0.123 kg/h, 44.9 W; (b) 52.3°C
7.113 (b) 2.12×10^{-4} kg/s·m^2
7.114 28.6 min
7.115 337 W/m, 483 W/m, 589 W/m
7.116 (b) 5.35×10^{-4} kg/s·m; (c) 1.275 kW/m
7.117 2.73×10^6 kg/day

7.118 2775 W, -0.13°C/s
7.119 0.0016 kg/s·m^2
7.120 (a) 4.81×10^{-4} kg/s·m^2, 1268 W/m^2
7.121 (a) 20.3 kg/h; (b) 102 days
7.122 (a) 4.76×10^{-2} m/s; (b) 3.80×10^{-2} m/s
7.123 (a) 2.31×10^{-4} kg/s·m
7.124 (a) 0.30 kg/h·m, 110 W/m; (b) 52.3°C
7.126 (a) 485 W; (b) 3135 W
7.128 (a) 9424 W/m, 0.0036 kg/s·m
7.129 45.6°C, 0.21
7.130 (b) 1.82×10^{-7} kg/s, -8.9 K/s
7.131 159 s
7.132 3.60×10^{-8} kg/s
7.133 278 K
7.134 (a) 0.080 m/s; (b) 50 ms; (c) 1.01×10^{-10} kg/s
7.135 35 body diameters/s
7.136 9.06×10^{-8} kg/s
7.137 (a) 1.12×10^{-8} kg/s; (b) 2.3 K
7.138 (a) 1.83; (b) 28.0°C, 0.00129
7.139 0.0016 kg/s·m^2

CHAPTER 8

8.1 0.041 m/s, -0.86×10^{-5} bar/m
8.2 0.0215 bar
8.3 (a) 0.289 bar, 1.42 kW; (b) 0.402 bar, 1.97 kW
8.4 (a) 53.8 bar, 146 W
8.7 367 K
8.8 0.50 m/s, 20.0°C
8.9 0.84 MPa, 0.46 MW, 0.46°C
8.10 (c) 0.46°C
8.11 (b) 44.2°C
8.15 (b) 60.8°C
8.16 (a) 471 W, 113.5°C, 60°C, 153.5°C; (b) 353 W, 90.2°C, 20°C, 150.2°C
8.17 (b) 362 K
8.18 8
8.21 (c) 6018 W
8.22 (a) 35°C, 16,000 W
8.23 (a) 28.4°C, 44.9°C; 73.3°C, 64.5°C;73.3°C, 65.1°C
8.24 90 kg/h, 1360 W
8.25 26.8 m
8.26 1281 W, 15.4 m
8.27 12,680 W/m^2, 121°C, 52.7°C
8.28 (b) 1266 s
8.29 5, -2.54°C/s

8.30	(a) 1.56 m, 28.3 days
8.31	13.7 m, 0.078 W
8.32	(a) 10.6 m
8.33	$0.411/m·d
8.34	(a) 8.87 m, (b) 52.6°C
8.35	(a) 81.7°C; (b) 9.1×10^{-4} bars, 0.19 W
8.36	(a) 63°C; (b) 66.1°C
8.37	(a) 29.9°C, -1212 W, 4.03 N/m^2
8.38	99°C, 5 atm, 1.14×10^{-3} kg/s
8.39	0.11 m
8.40	0.39 m
8.41	7840 W/m^2, 7040 W/m^2
8.42	(a) 323 K; (b) 325 K
8.43	(a) 578°C
8.44	(a) 1384 K; (b) 5.2×10^{-2} kg/s, 890 K
8.45	(a) 198°C, 71 N/m^2, 221 W; (b) 195°C, 119 N/m^2, 415 W
8.46	(a) 85.6°C, 661 W
8.47	(a) 42.2°C, 7500 W, 1.29 h; (b) 0.324 h
8.49	(a) 100 kW; (b) 2.67 kW
8.50	343 W/m
8.51	80.3 W/m
8.52	(a) 543°C, 232°C
8.53	81.4°C
8.54	(a) 37.1°C; (b) 47.4°C
8.55	(a) 98.5 m; (b) 10.6 m
8.56	(a) 35.1°C; (b) 95.8°C; (c) 96.9°C
8.57	489 W/m
8.58	(a) 10 kW; (b) 3.4 m
8.60	(a) 409 W/m^2·K; (b) 93.4 W/m^2·K; (c) 76.1 W/m^2·K, 15°C
8.61	9.3 m
8.62	15.7°C, 438 W
8.63	(b) 111°C, 9.1×10^6 W
8.65	(a) 35°C; (b) 5.6; (d) 732 W/m^2·K; (e) –66.7 W; (f) 0.81 m
8.66	(b) 244 W/m, 187°C
8.67	0.90, 308.3 K
8.68	(b) 53°C
8.69	85°C, 53 mm
8.73	33.2°C
8.74	379 K, 406 K
8.75	88°C
8.76	(a) 0.485 W; (b) 0.753 W; (c) 0.205 Pa, 1.02 Pa
8.77	(a) 53,700 W/m^2; (b) 123,400 W/m^2
8.78	2720 W, 312 K
8.79	(a) 2600 W/m
8.80	(a) 1.9 m
8.81	9330 W/m^2·K, 1960 W/m^2·K

8.82 (b) 305.3 K, 15.7 kW
8.84 30°C, 156 N/m^2
8.85 (a) 3.35 kW; (b) 69 s
8.86 59.7°C, 20°C, 75.7°C; 27.9°C, 20°C, 37.5°C
8.87 510 W/m^2
8.89 116.5°C, 5 atm, 1.38×10^{-3} kg/s
8.90 2.51 m, 388°C
8.91 (a) 704.5 K; (b) 683 K, 1087 K, 300 K
8.92 (a) ~25.1°C
8.93 19.7 m, 1575 W/m^2
8.94 58.4 m
8.95 (a) 20.6°C, 0.085 W; (b) 18.9°C, 0.128 W; (c) 21.5°C
8.96 (a) 9.77 m; (b) 159 mm; (c) 775 N/m^2, 379 N/m^2; (d) 2.25×10^{-3} kg/s
8.97 6.90×10^{-2} kg/s, 5.79 m, 67.8 mm
8.98 (a) 35.1°C; (b) 33.2°C
8.99 (a) 3190 W, -2039 W; (b) 9197 W, -8065 W
8.100 (a) 81.3°C
8.101 310 K, 418 W
8.102 (a) 307.8 K, 373 W; (b) 704 W
8.103 (a) 298.6 K, 45 W; (b) 349.9 K, 3.0 W
8.104 (a) 0.48°C; (b) 43.6°C, 27.6°C
8.105 (a) 52 mm; (b) 316 K; (c) 15.9×10^6 Pa; (d) 1.63 km, 1650 s
8.106 (b) 2.60×10^{-5} kg/s, 307.7 K
8.107 2.42×10^{-2} m/s
8.108 0.0032 m/s
8.109 0.024 m/s
8.110 3.89×10^{-2} m/s, 107 $W/m^2 \cdot K$; 1.12×10^{-2} m/s, 31 $W/m^2 \cdot K$
8.111 (a) 50.6 W; (b) 34.2°C
8.113 0.017 kg/m^3, 4.33×10^{-6} kg/s
8.114 2.1 m
8.115 (a) 3.64 m; (b) 46.7 W
8.116 (a) 8.7 mm Hg, 0.0326 kg/m^3; (b) 3.2×10^{-5} kg/s
8.117 (a) 0.0050 m/s; (b) 0.134 liter/day
8.118 (d) 0.0181 kg/m^3

CHAPTER 9

9.1 300.9×10^{-6} K^{-1}
9.2 727, 12.5, 512, 1.01×10^6
9.3 5.8, 663, 209
9.4 5.20, 23.78, 108.7
9.5 0.881
9.6 (a) 17.5 mm; (b) 0.47 m/s, 3.5 mm; (c) 4.3 $W/m^2 \cdot K$; (d) 0.60 m
9.7 12.6 mm
9.8 4.42 $W/m^2 \cdot K$, 4.51 $W/m^2 \cdot K$

9.9	(a) 34 mm; (b) 16.8 W
9.10	(a) 3.03 $W/m^2 \cdot K$; (b) 2.94 $W/m^2 \cdot K$
9.12	(b) 1154 s
9.13	11.7 W
9.14	(b) –0.099 K/s
9.15	6.5 $W/m^2 \cdot K$
9.17	153 W
9.18	(b) 223 W, $0.43/day
9.19	273.8 K, 174.8 W
9.20	274.4 K, 273.2 K, 168.8 W
9.21	(a) 4010 W; (b) 2365 s
9.22	(a) 347 $W/m^2 \cdot K$
9.23	(a) 364.2 W; (b) 46.5°C, 16.4%
9.24	–0.136 K/s
9.25	(a) 118 kW/m^3
9.26	24.8 W/m^2
9.27	74°C, 68°C
9.28	(a) 35.8°C; (b) 28.8°C; (c) 68.8°C, 49.5°C
9.29	(a) 94.3 W
9.30	(a) 74.9°C, 1225 W; (b) 71.4°C, 1262 W
9.31	(a) 2.13×10^6 W, 92%
9.33	(a) 72.7 W; (b) 483 W
9.34	86.5 W/m
9.35	(a) 19.3°C, 19.3°C, 20.1°C; 21.6 W/m, 27.0 W/m, 26.2 W/m
9.36	468 W
9.37	(a) 238 W/m
9.38	0.560 $W/m \cdot K$, 0.815
9.39	(a) 7.46 $W/m^2 \cdot K$; (b) 8.49 $W/m^2 \cdot K$, 264 K
9.40	90.2 W
9.41	56.7°C, 4.30 K/W, 3.56 K/W, 0.30 K/W, 4.2×10^{-3} K/W
9.42	45.3°C, 1007 W
9.43	(a) 2.91×10^5 W; (b) 8.53×10^4 W, 1463 K; (c) 0.13 m
9.45	(a) 250 W; (b) 181 s
9.46	(a) 42.9 min; (b) 11.0 min
9.47	(a) 4.0 $W/m^2 \cdot K$; (b) 3.2 $W/m^2 \cdot K$; (d) 45.7°C, 34.0°C
9.48	(a) 18.9 W; (b) 174°C
9.49	(a) 3.71 $W/m^2 \cdot K$, 44.5 W/m; (b) 4.50 $W/m^2 \cdot K$, 54.0 W; (c) 4.19 $W/m^2 \cdot K$, 50.3 W/m
9.50	(a) 92.7; (b) 85.2; (c) 83.2
9.51	21.6 kW
9.52	57°C
9.53	780 W/m
9.54	698 W/m
9.57	(a) 56.8°C; (b) 1335 W; (c) 7.27 h
9.58	(a) 929 W/m; (b) 2340 W/m; (c) 187 W/m
9.59	103 W/m
9.60	(a) 50.8 W/m, 0.953; (b) 56.9 W/m

9.61 79°C
9.62 64.8°C
9.63 (b) 8 $W/m^2 \cdot K$, 30.2 W/m
9.64 (a) 2.04 W; (b) 1.97 W
9.65 \$0.265/m·d
9.67 581°C, 183 s
9.69 (a) 207 $W/m^2 \cdot K$; (b) 387 $W/m^2 \cdot K$
9.70 (b) ~1.1 h
9.71 (a) 9.84 $W/m^2 \cdot K$; (b) 32.1 $W/m^2 \cdot K$; (c) 221 s; (d) 245 s
9.72 (a) 163 W/m, 41.7°C; (b) 60.8°C, 12.5°C, 52.8 W/m
9.73 (a) 45.8 m, 7
9.74 (a) 33.3 kW; (b) ~855 s, 9.07 kg
9.75 (a) 13.6 $W/m^2 \cdot K$, 4.7 $W/m^2 \cdot K$, 70.2°C, 20.5; (b) 70.7°C, 18.1
9.77 (c) 0.17 m/s, 0.00185 m/s
9.78 (a) 1.55 W; (b) 187 W; (c) 57.0 W
9.79 8.62 W
9.80 (a) 10.2 $W/m^2 \cdot K$, 1210 $W/m^2 \cdot K$
9.82 7.12 mm, 63.1 W
9.83 91.8 W
9.84 (a) 7.2 W, \$0.10
9.85 (a) 28.8 W
9.86 8.13 mm, 19, 380 W
9.87 (a) 18.0 W; (b) 19.6 mm
9.88 26.3 kW/m
9.89 7 mm
9.90 61 W
9.91 84 W/m^2
9.92 (a) 429 kg; (b) 105×10^6 J, 0.102; (c) 33.4 kg
9.93 (a) 124 W/m^2; (b) 146 W/m^2; (c) 26 W/m^2
9.94 (a) 1.57
9.95 (a) 525 W/m^2; (b) 4; (c) 101 W/m^2
9.97 (a) 9.1°C, -9.6°C, 35.7 W
9.98 (a) 74%, 35.4°C
9.99 43.9 W, 28.3 W
9.100 (a) 466 W
9.101 (a) 44.9 W/m; (b) 47.3 W/m
9.102 43.4 W/m
9.103 (a) 30.7°C; (b) 0.685 μm/s
9.104 463 W/m
9.105 0.022 kg/s
9.106 1.33
9.108 2.0 m/s
9.109 (a) 54.6 W/m; (b) 72.3 W/m; (c) 70.7 W/m; (d) 235 W/m
9.111 7.2 kW, -0.28°C/s
9.112 43.9 W, 47.9 W
9.113 6.21×10^{-5} kg/s·m

9.114 (a) 9.18 W; (b) 6.04×10^5; (c) 4.47 W/m^2·K; (d) 0.00398 m/s, 1.44 kg/day, 40.2 W; (e) 59 W

9.115 1253 MW, 246 MW, 1105 MW

9.116 77 A, 102 A

CHAPTER 10

10.2 4640 W/m^2, 8500 W/m^2, 40,000 W/m^2·K

10.3 (c) 0.032 mm

10.4 (a) 38,600 W/m^2·K; (b) 0.017

10.5 13,690 W/m^2·K

10.6 20.6°C

10.7 73 kW/m^2, 232 kW/m^2; 105 kW/m^2, 439 kW/m^2

10.8 (b) 89°C

10.9 1.34 MW/m^2, 0.512 MW/m^2, 0.241 MW/m^2, 1.26 MW/m^2

10.10 8.55 kW, 14 kg/h, 0.384, 30°C

10.11 (a) 110°C, 1.043×10^6 W/m^2; (b) 109°C

10.12 (a) 3.5 W, -19°C

10.13 559 W, 6.89×10^{-4} kg/s, 0.026

10.14 ~9, 119°C

10.15 152.6°C, 166.7°C

10.16 175 A

10.17 4.70 MW/m^2, 23.8 MW/m^2

10.20 55.4°C

10.21 0.81 MW/m^2

10.23 (a) 73.8°C, 82°C; (b) 1.13×10^5 W/m^2

10.24 (a) 0.0131; (b) 144.6°C, 182.1°C

10.26 (a) 180 W/m^2·K, 0.067; (b) 300°C

10.27 835 W

10.28 (a) 858 kW/m

10.29 1.34 MW/m^2, 2048 K

10.30 (a) 907 W/m, 1.4 kg/h·m; (b) 107.6°C, 1.4 kg/h·m

10.31 (b) 82%

10.32 0.475 MW/m

10.33 (a) 146 kW/m^2

10.34 4.16×10^4 W/m^2

10.35 (a) 0.18 s; (b) 37.4 s

10.37 68.0 kW/m

10.38 (a) 197°C, 0,672; (b) 691°C

10.39 (a) 35.6°C; (b) 54.0°C

10.40 3.03 mm, 3.65 mm, 4.93 mm, 2.14 mm, 1.43 mm

10.41 1.11×10^{-3} kg/s

10.42 6.69 mm

10.43 16.0 kW, 7.1×10^{-3} kg/s

10.44 40.4 kW, 17.8×10^{-3} kg/s

10.45 (a) 78°C
10.46 (a) 1.07 MW, 0.444 kg/s; (b) 0.98 MW, 0.407 kg/s
10.47 2.21 kW, 2.44×10^{-3} kg/s
10.48 (a) 649 kW/m; 0.272 kg/s·m
10.50 4.8 MW/m, 1.93 kg/s·m
10.51 (a) 48.6°C, 4270 W/m, 0.0018 kg/s·m
10.53 19.1 kW, 8.39×10^{-3} kg/s
10.54 4.28×10^{-3} kg/s·m
10.55 28.3 kW/m, 1.16×10^{-2} kg/s·m
10.56 (a) 144 mm
10.57 0.0080 kg/s, 0.0020 kg/s; 0.0086 kg/s, 0.0014 kg/s
10.58 (a) 459 kg/h·m
10.59 22,200 W/m^2·K, 9050 W/m^2·K
10.60 (a) 0.0118 kg/s, 7.98×10^{-3} kg/s; (b) 50.6°C, 55.4°C; (c) 6.23 m
10.61 2.78×10^{-3} kg/s
10.62 2.0 s, 3.91×10^{-5} kg
10.63 7100 W/m^2·K, 9.0×10^{-3} kg/s·m
10.64 144,900 W/m^2·K, 7.56×10^{-4} kg/s
10.65 6.29×10^{-2} kg/s·m
10.66 0.023 kg/s, 0.0014 kg/s
10.67 2.11 W, 0.0332 m^2
10.68 (a) 114.0°C; (b) 80.9°C, 2.6×10^{-4} kg/s
10.69 (a) 79.4°C, 55.7 W; (b) 16.7 mm
10.70 (a) 0.286 kg/s, 0.089 kg/s; (b) 379.7 K, 1.27 bar
10.72 (a) 1870 W, 325 K, 300 K

CHAPTER 11

11.2 (a) 92 W/m^2·K; (b) 690 W/m^2·K
11.3 (a) 1.52; (b) 0.304; (c) 0.10
11.4 249 W/m^2·K, 1140 W/m^2·K
11.5 11,800 W/m
11.6 920 W/m
11.7 (a) 2255 W/m^2·K; (b) 1800 W/m^2·K; (c) 29.4°C
11.8 (a) 98 W/m^2·K; (b) 512 W/m^2·K
11.9 168 W/m^2·K
11.10 29.5 W/m^2·K
11.11 12.6 W/m^2·K
11.13 3.09 m^2, 2.64 m^2, 2.83 m^2, 2.84 m^2
11.14 4.75 m^2
11.15 5.64×10^{-4} m^2·K/W
11.16 (a) 0.97 m; (b) 2250 W, 145°C, 338 W/m^2·K, 9.59×10^{-4} m^2·K/W
11.18 (a) 48°C; (c) 132 W/m^2·K; (d) 0.8; (e) 1
11.20 (a) 7600 W, 48.1°C; (b) 40 m
11.21 0.0029 m^2·K/W

11.22 5.19 kg/s, 37.5 m
11.23 (a) 1.58 m^2
11.24 (a) 357 K, 515 K; (b) 30.7 $W/m^2{\cdot}K$
11.25 (a) 785 W/K, 5.0 m; (b) 18.4°C
11.26 (a) 13 m^2, 679 kW, 37.5°C; (b) $178,000/yr
11.27 (a) 74%; (b) 24°C
11.28 76.4°C
11.29 (a) 8.9 mm, 3.4 m
11.30 79.8°C, 8.70×10^{-4} kg/s
11.31 (a) 399 K, 70
11.32 33.1 m^2
11.33 (a) 130 m; (b) 1985 $W/m^2{\cdot}K$; (d) 5.8%
11.34 (a) 3930 W; (b) 3.74 liter/min; (c) 0.24 m^2
11.35 (a) 3550 $W/m^2{\cdot}K$, 41.1°C, 0.85 kg/s
11.36 (a) 344.4 K
11.37 (a) 1.89 kg/s
11.38 (a) 720, 0.858 m; (b) 0.513 m
11.39 (a) 41.9 m^2, 20.7 kg/s; (b) 0.936 kg/s
11.40 (b) 0.50
11.41 36.8°C, 37.5°C
11.42 50°C, 61.7°C
11.43 (a) 3.07 m^2, 33.4°C
11.44 (a) 9.6 m
11.45 (a) 508 $W/m^2{\cdot}K$; (b) 1.95 m
11.46 (a) 55.7°C, 41.9°C; (b) 2300 $W/m^2{\cdot}K$; (d) 74.4°C, 0.55
11.47 (a) 11,200 m^2; (b) 1994 kg/s
11.48 2.83 kg/s
11.49 (a) 206; (b) 26.1°C; (c) 17.4 kg/s
11.50 (a) 686 W, 11.0 °C; (b) 10,087 W, 24.5°C; (c) $1765
11.51 (a) 129.7°C
11.52 808 $W/m^2{\cdot}K$
11.53 (a) 61 $W/m^2{\cdot}K$; (b) 0.64; (c) 46.2°C
11.54 68, 7.1 m
11.55 (a) 71 m^2
11.56 (a) 0.53×10^5 W, 0.47
11.57 34.1°C
11.58 36.6°C, 143.5°C
11.59 241 m^2
11.60 (a) 29.4%
11.61 1014 K, 800 K, 514 K, 0.70
11.62 (a) 204 kW, 57.3°C, 42.7°C; (b) 2.33 m
11.63 (a) 44.6 kW; (b) 0.65; (c) 0.55
11.64 (a) 8280 W; (b) 98.4°C, 87.2°C; (d) 5470 W, 0.66
11.66 (a) 0.0354 m or 0.261 m, 0.34 m
11.67 41 m^2
11.68 (a) 26.8°C

11.69 (a) 2×10^5 W, 19.6°C, 19.6°C
11.70 4.56 m
11.71 (a) 73.9 W/m^2·K; (b) 496 K, 369 K
11.72 (a) 104°C
11.74 (a) 0.752 m^2; (b) 0.576 m^2, 0.0855 kg/s; (c) 0.723 m^2, 317 K
11.75 (a) 0.672 kW; (b) 8°C, 61°C, 1.43 kW
11.76 (a) 335 K; (b) 8.11 m; (c) 0.666
11.77 (a) 30.3%; (b) 27.5 kg/s
11.78 (a) 86.5 kW/K
11.79 66.3 W/m^2·K
11.80 0.082 m^3, ~13, ~11, 0.54 m
11.81 56.2 W/m^2·K, 0.026 m^3
11.82 3
11.83 285 K
11.84 ~11
11.85 564 K

11S.1 160 W/m^2·K
11S.2 29.6 W/m^2·K
11S.3 4.75 m^2
11S.4 5.74×10^{-4} m^2·K/W
11S.5 (a) 1.5 m^2
11S.6 33.1 m^2
11S.7 9.6 m
11S.8 (a) 11,100 m^2; (b) 1994 kg/s
11S.9 878 W/m^2·K
11S.10 243 m^2
11S.11 4.8 m

CHAPTER 12

12.1 12.1 W/m^2, 28.0 W/m^2, 19.8 W/m^2
12.2 (a) 1.38×10^{-3} W; (b) 2.76 W/m^2
12.3 8×10^{-6} J
12.4 (a) 193 mm; (b) 940 W/m^2
12.5 (a) 60 mW; (b) 95.5 W/m^2·sr; (c) 47.8 μW
12.6 1086 W/m^2
12.7 (a) 1446 W/m^2
12.8 167.6 W/m^2
12.9 0.25
12.10 (a) 2000 W/m^2; (b) 637 W/m^2·sr; (c) 0.50
12.11 (a) 0.10 W; (b) 1273 W/m^2·sr; (d) 0.10 W; (e) 3.6×10^{-7} W, 90 mW/m^2; (f) 5.09×10^{-7} W, 127 mW/m^2; (g) 63.7 mW/m^2
12.13 (a) 0.393 m
12.14 (a) 0.32 W/m^2
12.15 789 K, 22 W; 273 K, 0.031 W; 1606 K, 37.7 W; 1750 K, 2761 W; 485 K, 0.628 W

12.16 (a) 133 W, 76 W, 232 W, 5.54 μm
12.17 7348 W/m^2
12.18 498 K
12.19 279 K
12.20 (a) 6.3×10^7 W/m^2; (b) 5774 K; (c) 0.50 μm; (d) 278 K
12.21 (a) 6.74×10^{-5} sr; (b) 2.0×10^7 $W/m^2 \cdot sr$
12.22 (a) 0.50 μm, 1.16 μm, 1.93 μm, 9.50 μm, 48.3 μm; (b) 0.125, 0.366, 0.509
12.23 (a) 0.226, 0.044; (b) 0.50 mm, 1.0 mm
12.24 (a) 0.0137, 0.376; 0.359, 0.132
12.25 (a) 9.84 kW/m^2, (b) 6.24 kW/m^2
12.26 278.4 W
12.28 (b) 0.90 K, 7.1 K
12.29 (a) 0.352; (b) –1977 K/s
12.30 (a) 0.249; (b) 0.358
12.32 (a) 0.25, 2.27×10^5 W/m^2
12.33 (a) 66.2 W/m^2; (b) 8.76 μm
12.34 0.636, 76.9 kW/m^2
12.35 1.45
12.39 (c) 995.3 K
12.40 (a) 3.3%, 5.3%; (b) 10%
12.41 0.373, -5.78 K/s, 311 s
12.42 34.7 kW/m^2
12.43 (b) 7500 W/m^2; (c) 2250 W/m^2; (d) 0.30
12.44 (a) 0.50, 0.60; (b) 3.62×10^5 W/m^2, 5.44×10^5 W/m^2; (c) 0.126 $W/m^2 \cdot \mu m$; (d) 10.3 μm
12.45 (a) 0.383; (b) 0.958, 0.240
12.46 515 W/m^2, 1306 W/m^2, 1991 W/m^2, 3164 W
12.47 2.51×10^{-6} W
12.48 1480 μW
12.49 (a) 0.774; (b) 0.1
12.50 (a) 0.45; (b) 0.56; (c) -22,750 W/m^2
12.51 (a) 0.311; (b) 74,750 W/m^2; (c) -60,750 W/m^2
12.52 348 K
12.53 (a) 0.669; (b) 0.745; (c) 111 kW/m^2; (d) 176 kW/m^2
12.54 (a) 1.76×10^{-4} W; (b) 0.384×10^{-4} W
12.55 (a) 0.00099; (b) 0.295; (c) 0.861
12.56 (a) 0.839, 0.568; (b) 0.329, 0.217
12.57 (b) 8.88 W, 78.2 W
12.59 (b) 0.599, 0.086, 0.315; (c) 1; (d) –615 W/m^2
12.60 0.85, 0.85, 3.9×10^6 W/m^2, 4.6×10^6 W/m^2, 4.6×10^6 W/m2; 0.297, 0.106, 137 W/m^2, 4.6×10^6 W/m^2, 1.38×10^6 W/m^2
12.62 (a) 0; (b) 0.5
12.63 (a) 0.574; (b) 0.145; (c) 0.681, 0.539
12.64 0, 0.3, 0.7, 0.303, 606 W/m^2, 952 $W/m^2 \cdot K$
12.65 (a) 0, 577.4 K; (b) 0.89, 0.10; (c) 5600 W/m^2, 630 W/m^2; (d) 0.89, 0.10
12.66 0.34, 0.80, 1700 W/m^2, -700 W/m^2
12.67 (a) 0.60; (b) 200 W/m^2; (c) -1200 W/m^2

12.68 (b) 1.42×10^5 W/m^2
12.69 (a) 0.720; (b) 0.756; (c) -11.8 kW/m^2
12.70 (a) 0.225, 0.388, 0.5; (b) 443 kW/m^2
12.71 7000 W/m^2, 0.94
12.72 29.2 W, 344 K
12.73 (a) 0.748; (b) 29.6°C; (c) 402 W/m^2
12.74 (a) 117°C
12.75 (a) 22.2°C
12.76 (a) 25°C
12.77 (a) 19.8 W; (b) 538.2 K; (c) 855 s
12.78 (a) 145 W/m^2; (b) 9.23 μW; (c) 8.10 μW
12.79 999 K
12.80 8.08×10^{-8} W
12.81 (a) 1.15 μW; (b) 1.47 μW
12.82 3.38×10^{-7} W
12.83 571 m/s, -15.5 m/s, 146 m, 0.256 s
12.84 (a) 134 K
12.85 (a) 1718 μV; (c) 1382 μV, 878 μV, 373 μV
12.86 (a) 23°C; (b) 44.6°C
12.88 (a) 5441 W/m^2
12.89 (a) 0.49; (b) 0.20; (c) 24.5 W
12.90 (a) 0.375; (b) 0.702; (c) 0.375
12.91 (b) 20 K/s
12.92 (a) 2.93 W/m·K
12.93 (a) 1425 K, 1.87×10^5 W/m^2; (b) 5.95×10^{-5} W
12.94 (a) 314 μW; (b) 871 K
12.95 796 K, 793 K
12.96 (a) 0.643; (b) 0.200; (c) 1646 kW/m^2; (e) 119 s
12.97 (a) 839 K; (b) 770 K; (c) 850 K
12.98 32.5 s
12.99 (a) 1.83 s; (b) 54.3 W
12.100 (a) 1.56×10^9 W/m^3; (b) 15.6 W/mm^2, 7.8 W/mm^2; (c) 772 s
12.101 (a) 126 W/m^2·K
12.102 (a) 0.799, 0.536; (b) 11.7 K/s; (c) 0.536; (d) 82 s
12.103 (a) 763 W; (b) 930 K; (c) 537 s
12.104 (a) 0.80, 0.713; (b) 233 kW/m^2, 9.75 K/s; (d) 186 s; (e) 413 s; (f) 899 s
12.105 (a) 0.51 μm, 8.33 μm; (c) 0.30, 0.59, 0.68; (d) 0.66, 0.68; (e) 73.5 kW/m^2
12.106 (a) 48.5°C
12.107 139°C, 3.47 W
12.109 329.2 K
12.110 (a) 0.8, 0.625; (b) 352 K
12.111 (a) 0.634; (b) 0.25; (c) 460 W/m^2
12.115 (b) 35.5°C
12.116 (a) 8.16×10^5 W/m^2
12.117 0.129 m, 39.1%
12.118 (a) 1.60×10^7 W, 75.8%

12.119 (b) 295.2 K; (c) 704 W
12.121 (a) 0.704; (b) 0.20; (c) 6.07 $W/m^2 \cdot K$
12.122 (a) 55.2°C; (b) 35.9°C
12.123 (a) 4.887 W; (c) 43.5°C
12.124 96.5 W/m^2
12.125 (a) 77.6°C; (b) –35.3°C
12.126 379 K, 656 K
12.127 (a) 197 K; (b) 340 K
12.128 439 K
12.129 0.95
12.130 (a) 837 W, 1026 W
12.131 (a) 30.5°C, 2766 W; (b) –7.6°C, 4660 W; (c) 80°C, 9557 W
12.132 (a) 1.286; (b) 352 K, 145 K
12.133 714 W/m^2
12.134 (a) 308 K, 278 K; (b) 211 K, 178 K
12.135 (b) $\sim 5.5 \times 10^6$ s
12.136 (a) 0 or ∞, 294 K; (b) 13.57 μm, 205 K, 310.4 K
12.137 (a) 57.9°C, 180 W; (b) 71.7°C; (c) –30.4°C
12.138 4.7°C, 16.2°C
12.140 (a) 43.1°C; (b) 29.2°C
12.141 25°C, 3.1×10^{-4} kg/s
12.142 (a) 0.0436 kg/m^3, 0.0144 kg/m^3, 6.76×10^{-6} kg/s, 21.2 W; (b) 503 W/m^2, 3105 W/m^2, 627 W/m^2

CHAPTER 13

13.1 (a) 1.0, 0.424; (b) 0.50, 0.25; (c) 0.637, 0.363; (d) 0.50, 0.707; (e) 0.5, 0; (f) 1.0, 0.125; (g) 0.50, 0.637
13.2 (c) ~0.62
13.4 (b) 0.0767, 0.553; 0.0343, 0.800
13.6 (a) 0.781; (b) 0.110
13.9 (a) 0.09
13.10 (b) 0.01
13.11 (a) 0.038; (b) 0.23
13.12 0.41
13.13 0.163
13.15 27.7 $\mu W/m^2$
13.16 (a) 354 mW/m^2; (b) 1695 mW/m^2
13.17 36,900 W
13.18 0.0492 $kg/s \cdot m$
13.19 (a) 0.64; (b) 1700 W
13.20 1.69×10^5 W/m
13.21 456 K
13.22 (a) 13.4 W, 6825 W/m^2
13.23 (a) 579.4 K; (b) 583 K
13.24 (a) 413 K; (b) 1312 W

13.25 544 K, 828 K
13.26 (a) 255 W; (b) 970 K, 837.5 K
13.28 1.58×10^4 W/m^2
13.29 0.162 W
13.30 11.87 kW
13.31 (c) 17 kW/m^2; (d) 635 μW
13.32 63.8 W/m^2
13.33 (a) 1.19 W; (b) 1.48 μW
13.34 (a) 69 mW; (b) 934.5 W/m^2; (c) 1085 W/m^2; (d) 1085 W/m^2
13.35 (a) 507 K
13.36 (a) 308 K
13.37 774 K
13.38 (a) 48.8 K/s; (b) 15 s; (c) 14.6 s, 12.0 s, 6.8 s
13.39 1680 W/m, 2517 W/m, 916 K
13.40 (a) 109.7 W/m, 110.1 W/m; (b) 98.5 W/m, 103.8 W/m
13.41 (a) 14.2 kW/m^2, (b) 56.7 kW/m^2; (c) 14.2 kW/m^2; (d) 42.5 kW/m^2
13.42 (a) 1.58 W; (b) 0.986
13.43 46.2 kW/m^2, 0.814
13.44 (a) 0.963
13.46 (a) 30.13 W; (b) 30.65 W; (c) 31.41 W; (d) 55.8 W/kg, 79.2 W/kg, 75.1 W/kg
13.47 (a) 15.2 W/m
13.48 66.4°C, 71.4°C
13.49 30.2 W/m
13.50 69°C
13.51 (a) 45.6 kW/m^2; (b) 2.1 kW/m^2
13.52 1.14×10^{-4} kg/s
13.53 (a) 1995 W; (b) 191 W; (c) 983 W, 1.5%
13.54 548 K, 474 K
13.56 0.138
13.58 (a) 338 K; (b) 25.3 W/m^2
13.59 90 mW
13.60 9
13.61 0.50 W/m, -49%
13.62 472°C
13.63 1225 K, 1167 K
13.64 (a) 896 K, 986 K; (b) 950 K, 990 K
13.65 (a) 288 W; (b) –1692 W; (c) 207 W; (d) –1133 W
13.66 423 K
13.67 10.1 kW/m
13.68 3295 W
13.69 (a) 12.6 W; (b) 0.873; (c) 590 K
13.70 (a) 169 kW/m; (b) 1320 K
13.71 (a) 1.83
13.72 (a) 8520 W/m; (b) 732 K
13.73 12.6 kW/m^2
13.74 (a) 9870 W/m; (b) 853 K

13.75 (a) 25.3 kW; (b) 18.2 kW
13.76 (a) 1228 K; (b) 1117 K
13.77 (b) 43.8 kW, 764 K
13.78 266 W/m
13.79 1046 W
13.80 (a) 37 W/m; (b) 9.2%
13.81 (b) 1.32 W
13.82 (a) 1842 W; (b) 1840 W
13.83 –538 W, –603 W, 1141 W
13.85 611 K
13.86 –200 W, 5037 W, –4799 W, 0, 0
13.87 583 K
13.88 317 W
13.89 (a) 52.7 kW/m^2, 2.89 kW
13.90 (a) 41.3 kW; (b) 6.55 kW
13.91 (a) –1153 W; (b) 0.57 K/s; (c) 715 K
13.92 (b) 842.5 K
13.93 (a) 14.0°C, 8.6°C
13.94 (a) 54 kW/m; (b) 1346 K, 25.7 kW/m
13.95 (a) 373 K, 0.76; (b) 388.4 K, 1.84; (c) 361.6 K, 0.40
13.96 4600 W, 64 W/m^2·K
13.97 (a) 29.5 W/m, 483 K; (b) 441 W/m, 352 K
13.98 (a) 35 W/m^2·K; (b) 484 K
13.99 (a) 792.2 K, 792 K; (b) 783.2 K, 783 K
13.100 (a) 526 K
13.101 (a) 577 K; (b) 156 W/m; (c) 3120 W
13.103 (a) 289.4 W/m^2; (b) 376.2 W/m^2; (c) 328.7 W/m^2
13.104 (b) 366 K, 1920 W/m^2; 598 K, 10,850 W/m^2
13.105 5700 K/W
13.106 19.2 W
13.107 (a) 159 W/m^2, 7.3 W/m^2
13.108 (a) 145 W/m^2; (b) 306 W/m^2; (c) 165 W/m^2, 156 W/m^2
13.109 (a) 8.8°C, –7.4°C, 91.3 W
13.110 669 W/m^2, 199 W/m^2
13.111 (a) 131 W/m
13.112 59.5°C, 89 W/m
13.113 (a) 466 W, 1088 W
13.114 (a) 54.6 W; (b) 413 K
13.115 (a) 30 W/m^2
13.116 (a) 8075 W/m; (b) 796 K, 10 kW/m
13.117 (a) 19.89 W/m^2; (b) 100.6 W/m^2; (c) 122.1 W/m^2
13.118 0.023 kg/s
13.119 (a) 3.45 m; (b) 0.88 m
13.120 (b) 1295 K, 498 K; (c) 8 kW/m
13.121 (b) 1213 W/m; (c) 576 W/m; (d) 502 K
13.122 (a) 178 W/m^2·K, 543 K, 0.098 kg/s; (b) 0.140 kg/s

13.123 1040 K
13.124 (a) 0.0197; (b) 825 K, 108 kW
13.125 15.1 kW
13.126 21.9 kW/m
13.127 98 kW/m
13.128 135 kW/m^2
13.129 380 K, 89.6 kW/m^2
13.130 0.515 kg/s
13.131 0.004 kg/s, 4.5 m/s, 367 K
13.132 (a) 530 K; (b) 0.00613 m/s
13.133 (a) 0.00341 kg/s·m; (b) 889 K, 811 K

CHAPTER 14

14.1 0.233, 0.767
14.2 0.04 $kmol/m^3$, 1.78 kg/m^3, 0.5, 0.61; 0.04 $kmol/m^3$, 1.13 kg/m^3, 0.5, 0.39
14.3 0.0837, 0.1219 kg/m^3, 0.0406 $kmol/m^3$, 35.8 kg/kmol, 0.0146 kg
14.4 (b) 0.31, 0.27, 0.42; 0.35, 0.40, 0.25
14.6 0.36×10^{-4} m^2/s, 0.52×10^{-4} m^2/s
14.8 3.14×10^{-15} kmol/s
14.10 (a) 1.087×10^{-8} kmol/s; (b) 1.107×10^{-8} kmol/s
14.13 0.257 $kg/m^2 \cdot h$
14.17 (a) 14.7×10^{-12} kg/s; (b) -3.5×10^{-7} bar/s
14.18 1.5×10^{-8} $kmol/m^2 \cdot s$
14.19 (a) 0.0189, 0.0120, 1, 1; (b) 0.205, 0.226, 5.53×10^{-6}, 9.83×10^{-6}
14.20 (a) 1.31×10^{-9} $kmol/m^2 \cdot s$; (b) 0.0807 $kmol/m^3$, 0.0404 $kmol/m^3$
14.21 4.05×10^{-9} kg/s
14.22 1.9×10^{-22} kg/s·m
14.23 4×10^{-15} kg/s
14.24 1.39×10^{-6} kmol/s·m
14.25 (a) 1.91×10^{-3} $kmol/m^3 \cdot bar$; (b) 0.32×10^{-16} kmol/s; (c) 0.32×10^{-16} kmol/s; (d) 0
14.26 0.0034
14.27 0.022 kg/h
14.28 (a) 0.10; (b) 5.61×10^{-5} kg/s
14.29 6.66×10^{-9} kmol/s
14.31 0.0008, 7.48×10^{-8} $kmol/s \cdot m^2$, 2.24×10^{-7} $kmol/s \cdot m^2$
14.33 (b) 0.02 $kmol/m^3$; (d) 9.60×10^{-5} $kmol/m^2 \cdot s$
14.34 (c) 3×10^{-5} $kmol/m^3$
14.35 (c) 5.11×10^{-6} $kmol/m^3$, 2.18×10^{-15} kmol/s
14.37 (b) 5.38×10^{-11} $kmol/s \cdot m^2$
14.38 (b) 9.08×10^{-7} $kmol/m^3$
14.39 333 s
14.40 0.02 s
14.41 8.42 h
14.42 (b) 0.0314

14.43 (a) 2071 days; (b) 198.3 kg/m^3; (c) 198.3 kg/m^3
14.44 (b) 1.85 $kmol/m^3$
14.46 (a) 32 kg/m^3; (c) 42.9 h
14.47 (a) 32 kg/m^3; (c) 42.9 h; (d) 218 h
14.48 3.3×10^{-3} $kmol/m^3 \cdot bar$, 8.5×10^{-13} m^2/s
14.49 3940 s
14.51 (a) 0.0778 $kmol/m^3$, 1.24×10^{-4}; (b) 0.0117 $kmol/m^3$, 1.9×10^{-5}; 0.0346 $kmol/m^3$, 5.5×10^{-5}
14.53 (a) 2.5 μm; (b) 1.7 h; (c) 3.9 h