Open-Ended Problems: A Future Chemical Engineering Education Approach. J. Patrick Abulencia and Louis Theodore.

6

Heat Transfer

This chapter is concerned with heat transfer. As with all the chapters in Part II, there are several sections: overview, several technical topics, illustrative open-ended problems, and open-ended problems. The purpose of the first section is to introduce the reader to the subject of heat transfer. As one might suppose, a comprehensive treatment is not provided although numerous references are included. The second section contains three open-ended problems; the authors' solutions (there may be other solutions) and also provided. The third (and final) section contains 43 problems; *no* solutions are provided here. The reader should also note that four open-ended heat transfer projects can be found in Chapter 27, Part III.

6.1 Overview

This overview section is concerned—as can be noted from its title—with heat transfer. As one might suppose, it was not possible to address all topics directly or indirectly related to heat transfer. However, additional details may be obtained from either the references provided at the end of this Overview section and/or at the end of the chapter.

Note: Those readers already familiar with the details associated with this subject may choose to bypass this Overview.

A difference in temperature between two bodies in close proximity or between two parts of the same body results in heat flow from the higher temperature to the lower temperature. There are three different (and classic) mechanisms by which this heat transfer can occur:

1. conduction,
2. convection, and
3. radiation.

When the heat transfer is the result of molecular motion (e.g., the vibrational energy of molecules in a solid being passed along from molecule to molecule), the mechanism of transfer is *conduction*. When the heat transfer results from macroscopic motion, such as currents in a fluid, the mechanism is that of *convection*. When heat is transferred by electromagnetic waves, the mechanism is defined as *radiation*. In some industrial applications, more than one mechanism; i.e., a combination of mechanisms is usually involved in the transmission of heat. However, since each mechanism is governed by its own set of physical laws, it is beneficial to discuss them independently of each other.

The five topics addressed in the Section include:

1. Conduction
2. Convection
3. Radiation
4. Condensation, Boiling, Refrigeration, and Cryogenics
5. Heat Exchangers

The reader should note that the bulk of the material for this chapter was draw from I. Farag and J. Reynolds, *Heat Transfer*, Theodore Tutorials, East Williston, NY, originally published by the USEPA/APTI, RTP, NC, 1996 [1].

6.2 Conduction

The rate of heat flow by conduction is given by Fourier's law [2].

$$q = -kA\frac{dT}{dx} \tag{6.1}$$

where q = heat flow rate (Btu/h)
x = direction of heat flow (ft)
k = thermal conductivity (Btu/hft°F)
A = heat transfer area, a plane perpendicular to the x direction (ft^2)
T = temperature (°F)

The negative sign reflects the fact that heat flow is from a high to low temperature, and therefore the sign of the derivative is opposite that of the direction of the heat flow.

Equation 6.1 may be written in the form of the general transfer rate equation:

$$\text{Transfer rate} = \frac{\text{Driving force}}{\text{Resistance}} \tag{6.2}$$

Since q in Equation 6.1 is the heat flow rate and ΔT is the driving force, the L/kA (L is the length in the direction of flow) term may be considered to be the resistance to heat flow. This approach is useful when heat is flowing by conduction in sequence through different materials.

Consider, for example, a flat incinerator wall made up of three different layers: an insulating layer, a; steel plate, b; and an outside insulating layer, c. The total resistance to heat flow through the incinerator wall is the sum of the three individual resistances; i.e.,

$$R = R_a + R_b + R_c \tag{6.3}$$

At steady state, the rate of heat flow through the wall consisting of the three layers is therefore, given by

$$q = \frac{T_1 - T_2}{(L_a / k_a A_a) + (L_b / k_b A_b) + (L_c / k_c A_c)} \tag{6.4}$$

where k_a, k_b, k_c = thermal conductivity of each section (Btu/h·ft·°F)
A_a, A_b, A_c = area of heat transfer of each section $(\text{ft})^2$; these are equal for areas of constant cross section
L_a, L_b, L_c = thickness of each layer (ft)
T_1 = temperature at inside surface of insulating wall a (°F)
T_2 = temperature at outside surface of insulating wall c (°F)

In the above example, the heat is flowing through a slab of constant cross section. In many cases of industrial importance however, this is not the case. For example, in heat flow through the walls of a cylindrical pipe in a heat exchanger or a rotary kiln, the heat transfer area increases with distance displaced from the center of the cylinder. The heat flow in this case is given by

$$q = \frac{kA_{lm}\Delta T}{L} \tag{6.5}$$

The term, A_{lm} in Equation 6.5 represents the average heat transfer area, or more accurately, the log-mean average heat transfer area. This log-mean average can be calculated by

$$A_{lm} = \frac{A_2 - A_1}{\ln(A_2 / A_1)} \tag{6.6}$$

where A_2 = outer surface area of cylinder (ft^2)
A_1 = inner surface area of cylinder (ft^2)

6.3 Convection

The flow of heat from a hot fluid to a cooler fluid through a solid wall is a situation regularly encountered in engineering equipment; examples of such equipment are heat exchangers, condensers, evaporators, boilers, and economizers. The heat absorbed by the cool fluid or given up by the hot fluid may be *sensible* heat, causing a temperature change in the fluid or it may be *latent* heat, cause a phase change such as vaporization or condensation (see also Chapter 3). The rate of heat transfer between the two streams, assuming *no* heat loss to the surroundings, may be calculated by the enthalpy (h) change for either fluid:

$$q = \Delta H = \dot{m}_h\left(h_{h1} - h_{h2}\right) = \dot{m}_c(h_{c2} - h_{c1}) \tag{6.7}$$

Equation 6.7 is applicable to the heat exchange between two fluids whether a phase change is involved or not.

In order to design a piece of heat transfer equipment, it is not sufficient to only know the heat transfer rate calculated by the enthalpy balances described above. The rate at which heat can travel from the hot fluid, through the tube walls, and into the cold fluid, must also be considered in the calculation of the contact area. The slower this rate is, for given hot and cold fluid flow rates, the more contact area is required.

The rate of heat transfer through a unit of contact area is referred to as the *heat flux* or *heat flux density* and, at any point along the tube length, is given by

$$\frac{dq}{dA} = U\left(T_h - T_c\right) \tag{6.8}$$

where dq/dA = local heat flux density (Btu/h·ft^2)
U = local overall heat transfer coefficient (Btu/h·ft^2·°F)

The use of this overall heat transfer coefficient (U) is a simple, yet powerful, concept. In most applications, it combines both conduction and convection effects, although transfer by radiation can also be included. Methods for calculation the overall heat transfer coefficient are available in the literature [3,4]. In actual practice it is not uncommon for vendors to provide a numerical value for U.

In order to apply Equation 6.8 to the *entire* heat exchanger, the equation must be *integrated*. This cannot be accomplished unless the geometry of the exchanger is first defined. For simplicity, one of the simplest exchangers will be assumed here – the double-pipe heat exchanger (see later section). This device consists of two concentric pipes. The outer surface of the outer pipe is normally well insulated so that no heat exchange with the surroundings may be assumed. One of the fluids flows through the center pipe while the other flows through the annular channel between the pipes. The fluid flows may be either *co-current*, when the two fluids flow in the same direction, or *countercurrent*, where the flows are in opposite directions. However, countercurrent arrangement is more efficient and is more commonly used.

For a heat exchanger, integration of Equation (6.8) along the exchanger length yields (after applying several simplifying assumptions),

$$q = UA\Delta T \tag{6.9}$$

This has come to be defined by some, including the authors of this text, as the *heat transfer equation*. The aforementioned simplifying assumptions are that U and ΔT do not vary along the length. Since this is not actually the case, both U and ΔT must be regarded as averages of some type. A more careful examination of Equation 6.8, assuming that only U is constant, would ultimately indicate that the appropriate average for ΔT is the log-mean average, i.e.,

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \tag{6.10}$$

where ΔT_1 and ΔT_2 are the temperature differences between the two fluids at the ends of the exchanger. The area term (A) in Equation 6.9 is the cylindrical contact area between the fluids. However, since a pipe of finite thickness separates the fluids, the cylindrical area (A) must first be defined. Any one of the infinite number of areas between and including the inside and outside surface areas of the pipe may be arbitrarily chosen for this purpose. The usual approach in practice to use either the inside (A_i) our outside (A_0) surface area; the outside area is the more commonly used of the two. Since the value of the overall heat transfer coefficient depends on the area chosen, it should be subscripted to correspond to the area on which it is based. Equation 6.9 based on the outside surface area now becomes

$$q = U_0 A_0 \Delta T_{lm} \tag{6.11}$$

Comparing Equation 6.11 to the general rate equation (6.2), it can be seen that $(U_0, A_0)^{-1}$ may be regarded as the resistance to heat transfer between the two fluids, i.e.,

$$R_o = \frac{1}{U_0 A_0} \tag{6.12}$$

The *total* resistance to heat transfer across the wall therefore may be divided into three contributions: the *inside film*, the *tube wall*, and the *outside film*. This is restated mathematically in Equations 6.13 and 6.14.

$$R_t = R_i + R_w + R_0 \tag{6.13}$$

or

$$\frac{1}{U_0 A_0} = \frac{1}{h_i A_i} + \frac{x}{k A_{lm}} + \frac{1}{h_0 A_0} \tag{6.14}$$

where h_i = inside film coefficient (Btu/h·ft²·°F)
h_o = outside film coefficient (Btu/h·ft²·°F)

These film coefficients are almost always determined experimentally. Many empirical correlations can be found in the literature for a variety of fluids and exchanger geometries. Typical values of film coefficients are available in the literature [3–6].

After a period of service, thin films of foreign materials such as dirt, scale or products of corrosion build up on the tube wall surfaces. As shown in Equations 6.15 and 6.16 these films R_{fi} and R_{fo} introduce additional resistances to heat flow, and reduce the overall heat transfer coefficient:

$$R_t = R_i + R_\mathrm{h} + R_w + R_\mathrm{fo} + R_\mathrm{o} \tag{6.15}$$

For this new condition,

$$\frac{1}{U_0 A_0} = \frac{1}{h_i A_i} + \frac{1}{h_{di} A_i} + \frac{x}{k A_{lm}} + \frac{1}{h_{do} A_o} + \frac{1}{h_0 A_0} \tag{6.16}$$

where h_{di}=inside fouling factor (Btu/h·ft²·°F)
h_{do}=outside fouling factor (Btu/h·ft²·°F)
Typical values of fouling factors are provided in the literature [3–6].

6.4 Radiation

Radiation becomes important as a heat transfer mechanism only when the temperature of the source or system is very high. The driving force for conduction and convection is the temperature difference between the source and the receptor; the actual temperatures have only minor influence since it is the difference in temperatures that count. For these two mechanisms, it usually does not matter whether the temperatures are 100°F and 50°F or 500°F and 450°F. Radiation on the other hand is strongly influenced by the temperature level; as the temperature level increases, the effectiveness of radiation as a heat transfer mechanism increases rapidly. It follows that, at very low temperatures, conduction and convection are the major contributors to the total heat transfer; however, at very high temperatures, radiation is the controlling or predominant factor. At temperatures in between, the fraction contributed by radiation depends on such factors as the convection film coefficient and the nature of the radiating surface [3]. To cite two extreme examples, for large pipes losing heat by natural convection, the temperature at which radiation accounts for roughly one half of the total heat transmission is around room temperature; for fine wires, this temperature is significantly above that.

A perfect or ideal radiator, referred to as a *black body*, emits energy at a rate proportional to the *fourth* power of the absolute temperature T of the body. When two bodies exchange heat by radiation, the net heat exchange is then proportional to the difference between each T^4. This is shown in Equation 6.17, which is based on the classic Stefan-Boltzmann law of thermal radiation:

$$q = \sigma FA\left(T_h^4 - T_c^4\right) \tag{6.17}$$

where σ = 0.1714 x10^{-8} Btu/hr·ft^2·R^4(Stefan-Boltzmann constant)
A = area of either surface (chosen arbitrarily)(ft^2)
F = view factor (dimensionless)
T_h = absolute temperature of the hotter body (°R)
T_c = absolute temperature of the colder body (°R)

The view factor (F) above depends on the geometry of the system; i.e., the surface geometries of the two bodies plus the spatial relationship between them, and on the surface chosen for A. Values of F are available in the literature for many geometries [3–5]. It is important to note that Equation 6.17 applies only to black bodies and is valid only for thermal radiation.

Other types of surfaces besides black bodies are less capable of radiating energy, although the T^4 law is generally obeyed. The ratio of energy radiating from one of the *gray* (non-black) bodies to that radiating from the black body under the same conditions is defined as the *emissivity* (ε). For *gray* bodies, Equation (6.17) becomes

$$q = \sigma FF_{\varepsilon}A\left(T_h^4 - T_c^4\right) \tag{6.18}$$

where F_{ε} = emissivity function, dependent on the emissivity of each body .

Relationships for F_{ε} for various geometries are available in the literature [3–5].

6.5 Condensation, Boiling, Refrigeration, and Cryogenics

Phase-change processes involve changes (sometimes significantly) in the density, viscosity, heat capacity, and thermal conductivity of the fluid in question. The heat transfer process and the applicable heat transfer coefficients for boiling and condensation is generally more involved and complicated than that for a single-phase process. It is therefore not surprising that more real-world applications involving boiling and condensation require the use of empirical correlations; many of these are available in the literature [3–6].

One of the main cost considerations when dealing with refrigeration and cryogenics is the cost of building and powering the unit. This is a costly element in the overall economics, so it is important to efficiently transfer heat in the refrigeration and cryogenic processes. Since the cost of equipment can be expensive, there are a number of factors that should be considered when choosing the applicable equipment. Cryogenics plays a major role in the chemical processing industry. Its importance lies in such processes as the recover of valuable feedstocks from natural gas streams, upgrading the heat content of fuel gas, purifying many process and waste streams, producing ethylene, as well as other chemical processes.

6.6 Heat Exchangers

Heat exchangers are defined as equipment that affect the transfer of thermal energy (in the form of heat) from one fluid to another. The simplest exchangers involve the direct mixing of hot and cold fluids. Most industrial exchangers are those in which the fluids are separated by a wall. The latter type, referred to by some as a *recuperator*, can range from a simple plane wall between two flowing fluids to more complex configurations involving multiple passes, fins, or baffles. Conductive and convective heat transfer principles (see earlier subsections) are required to describe and design these units; radiation effects are generally neglected. Kern [3] and Theodore [4] provide an extensive predictive and design calculations, most of which are based on Equation 6.9 – the *heat transfer equation*.

The presentation in this section keys on the description of the various heat exchanger equipment (and their classification), not on the log-mean temperature difference driving force, ΔT_{tm}, the overall heat transfer coefficient, U, etc., developed earlier in the section on convection. However, the design and predictive equation is the *heat transfer equation* provided in Equation 6.9. Three heat exchangers are briefly discussed below.

The *double-pipe* unit consists of two concentric pipes. Each of the two fluids—hot and cold—flow either through the inside of the inner pipe or through the annulus formed between the outside of the inner pipe and the inside of the outer pipe. Generally, it is more economical (from a heat efficiency and design perspective) for the hot fluid to flow through the inner pipe and the cold fluid through the annulus, thereby reducing heat losses to the surroundings. In order to ensure sufficient contacting time, pipes longer than approximately 20 ft are extended by connecting them to *return bends*. The length of pipe is generally kept to a maximum of 20 ft because the weight of the piping may cause the pipes to sag. Sagging may allow the inner pipe to touch the outer pipe, distorting the annulus flow region (as well as the velocity profile/distribution) and disturbing proper operation. When two pipes are connected in a "U" configuration that entails a return bend, the bend is referred to as a *hairpin*. In some instances, several hairpins may be connected in series. Additional information and calculation details are provided by Kern [3] and Theodore [4].

Shell-and-tube (also referred to as tube and bundle) heat exchangers provide a large heat transfer area economically and practically. The tubes are placed in a bundle and the ends of the tubes are mounted in tube sheets. The tube bundle is enclosed in a cylindrical shell, through which the second fluid flows. Most shell-and-tube exchangers used in practice are of welded construction. The shells are built from a piece of pipe with flanged ends and necessary branch connections. The shells consist of seamless pipe up to 24 inches in diameter; they are made of bent and welded steel plates if above 24 inches. Channel sections are usually of built-up construction, with welding-neck forged-steel flanges, rolled-steel barrels and welded-in pass partitions. Shell covers are either welded directly to the shell, or are built-up constructions of flanged and dished heads and welding-neck forged-steel flanges. The tube sheets are usually nonferrous castings in which the holes for inserting the tubes have been drilled and reamed before assembly. Baffles are usually employed to both control the flow of the fluid outside the tubes and provide turbulence.

One method of increasing the heat transfer rate is to increase the surface area of the heat exchanger. This can be accomplished by mounting metal *fins* on a tube in such a way that there is good metallic contact between the base of the fin and the wall of the tube. With this contact, the temperature throughout the fins will approximate that of the temperature of the heating (or cooling) medium and the high thermal conductivity of most metals used in practice reduces the resistance to heat transfer by conduction in the fins. Consequently, the surface will be increased without more tubes [7].

Extended surfaces, or fins, are classified into longitudinal fins, transverse fins, and spine fins. *Longitudinal fins* (also termed straight fins) are attached continuously along the length of the surface. *Transverse or circumferential fins* are positioned approximately perpendicular to the pipe or tube axis and are usually used in the cooling of gases. *Annular fins* are examples of continuous transverse fins. *Spine of peg fins* employ cones or cylinders, which extend from the heat transfer surface, and are used for either longitudinal flow or cross flow.

Other heat exchangers including: evaporators, waste heat boilers, condensers, and quenchers, details of which are provided by Theodore [4]. Material on cooling towers is also available in the literature [8].

A detailed and expanded treatment of heat transfer is available in the following three references [9–11]:

1. D. Green and R. Perry (editors), *Perry's Chemical Engineers' Handbook,* 8th edition, McGraw-Hill, New York City, NY, 2008 [9].
2. L. Theodore, *Chemical Engineering: The Essential Reference*, McGraw-Hill, New York City, NY, 2014 [10].
3. L. Theodore, *Heat Transfer for the Practicing Engineer*, John Wiley & Sons, Hoboken, NJ, 2011 [11].

6.7 Illustrative Open-Ended Problems

This and the last section provide open-ended problems. However, solutions *are* provided for the three problems in this section in order for the reader to hopefully obtain a better understanding of these problems which differ from the traditional problems/illustrative examples. The first problem is relatively straightforward while the third (and last problem) is somewhat more difficult and/or complex. Note that solutions are not provided for the 43 open-ended problems in the next section. Additional case study projects are located in Chapter 27, Part III.

Problem 1: A double-pipe heat exchanger is designed to heat a discharge stream to a required temperature of 105°F. However, the exchanger is currently only partially heating the stream. You have been asked to briefly describe what steps can be taken to get the heat exchanger back to design specifications.

Solution: Depending on the type of heat exchanger, the approaches to bring the heat exchanger back to design specification are different. However, most of them include the following:

1. cleaning of heat exchanger;
2. installation or removal of insulation;
3. correction of leakage;

4. upgrading or replacement of gaskets;
5. increasing the mass flow rate of the cooling fluid; and,
6. change the cooling medium (e.g., have colder water pass through the heat exchanger).

Problem 2: A plant has three streams to be heated (see Table 6.1) and three streams to be cooled (see Table 6.2). Cooling water (90°F supply, 155°F return) and steam (saturated at 250 psia) are available. Calculate the heating and cooling duties and indicate what utility (or utilities) should be employed. Comment on the type of utility that should be employed.

Solution: The sensible heating duties (in units of Btu/h) can now be computed and compared.

$$\text{Heating: } 7{,}475{,}000 + 6{,}612{,}000 + 9{,}984{,}000 = 24{,}071{,}000 \text{ Btu/h}$$
$$\text{Cooling: } 12{,}600{,}000 + 4{,}160{,}000 + 3{,}150{,}000 = 19{,}910{,}000 \text{ Btu/h}$$
$$\text{Heating} - \text{Cooling} = 24{,}071{,}000 - 19{,}910{,}000 = 4{,}161{,}000 \text{ Btu/h}$$

At a minimum, 4,161,000 Btu/h will have to be supplied by steam or another hot medium. The type of heating medium will be dictated by availability, location economics, safety considerations, etc. This problem will be revisited in Part III, Chapter 27 – *Heat Transfer Term Projects*.

Table 6.1 Streams to be Heated in Problem 2

Stream	Flowrate, lb/h	c_p, Btu/lb·F	T_{in}, °F	T_{out}, °F
1	50,000	0.65	70	300
2	60,000	0.58	120	310
3	80,000	0.78	90	250

Table 6.2 Streams to be Cooled in Problem 2

Stream	Flowrate, lb/h	c_p, Btu/lb·F	T_{in}, °F	T_{out}, °F
4	60,000	0.70	420	120
5	40,000	0.52	300	100
6	35,000	0.60	240	90

Table 6.3 Heat Exchanger Duty Requirements for Problem 2.

Stream	**Duty, Btu/h**
1	7,475,000
2	6,612,000
3	9,984,000
4	12,600,000
5	4,160,000
6	3,150,000

The heat exchanger requirements and the physical layout of the exchangers receive treatment in a later chapter.

Problem 3: It would normally seem that the thicker the insulation on a pipe, the less the heat loss; i.e., increasing the insulation should reduce the heat loss to the surroundings. But, this is not always the case. There is a "critical insulation thickness" above which the system will experience a greater heat loss due to an increase in insulation. This situation arises for "small" diameter pipes when the increase in area increases more rapidly than the resistance of the thicker insulation. Develop an equation describing this phenomena. Provide a graphical analysis of your results.

Comment: Refer to the literature for additional details [3–4].

Solution: Consider the system shown in Figure 6.1. One notes that the area terms for the heat transfer equations in rectangular coordinates are no longer the same in cylindrical coordinates for pipes (e.g., for the inside surface, the heat transfer area is given by $2\pi r_i L$). Applying the general heat transfer equation to a pipe/cylinder system leads to:

$$g = \frac{T_i - T_o}{\dfrac{1}{2\pi r_i L}\left(\dfrac{1}{h_i}\right) + \dfrac{\Delta x_w}{k_w 2\pi L r_{\text{lm},w}} + \dfrac{\Delta x_i}{k_i 2\pi L r_{\text{lm},i}} + \dfrac{1}{2\pi r L}\left(\dfrac{1}{h_0}\right)}$$

$$= \frac{2\pi L(T_i - T_o)}{\dfrac{1}{r_i h_i} + \dfrac{\ln(r_o / r_i)}{k_w} + \dfrac{\ln(r / r_o)}{k_i} + \dfrac{1}{r h_0}} = \frac{2\pi L\left(T_i - T_o\right)}{f(r)} \tag{6.19}$$

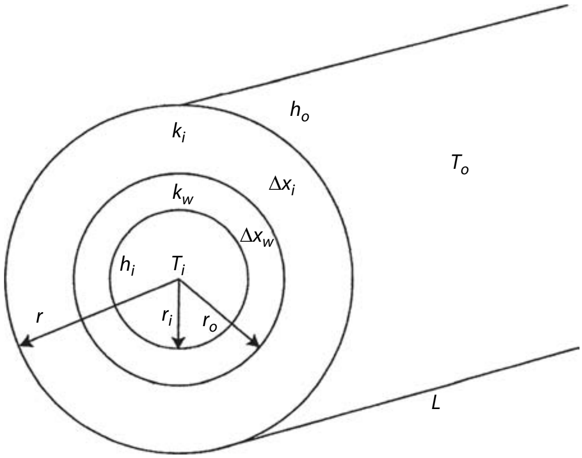

Figure 6.1 Critical insulation thickness for a Pipe; problem 3.

where

$$f(r) = \frac{1}{r_i h_i} + \frac{\ln(r_o / r_i)}{k_w} + \frac{\ln(r / r_o)}{k_i} + \frac{1}{r h_0} \tag{6.20}$$

Assuming that q goes through a maximum or minimum as r is varied, l'Hôpital's rule can be applied to Equation (6.20):

$$\frac{dq}{dr} = 2\pi L(T_i - T_o)\left[\frac{-\dfrac{df(r)}{dr}}{f(r)^2}\right] = 0 \tag{6.21}$$

with

$$-\frac{df(r)}{dr} = -\frac{1}{rk_i} + \frac{1}{r^2 h_0}$$

For $dq/dr = 0$, one may therefore write

$$\frac{df}{dr} = -\frac{df(r)}{dr} = \frac{1}{rk_i} + \frac{1}{r^2 h_0} = 0 \tag{6.22}$$

For this maximum/minimum condition, set $r = r_c$ and solve for r_c.

$$r_c = \frac{k_i}{h_o} \tag{6.23}$$

The second derivative of *dq*/*dr* of Equation 6.21 provides information as to whether *g* experiences a maximum or minimum at r_c.

$$\frac{d}{dr}\left(\frac{dq_r}{dr}\right)+\frac{1}{r^2k_i}-\frac{1}{r^3h_o}=\frac{h_o^2}{k^3}-\frac{2h_o^2}{k^3}=\frac{h_o^2}{k^3}(1-2) \tag{6.24}$$

Clearly, the second derivative is a negative number, is therefore a *maximum* $r = r_c$. The term q then decreases monotonously as r is increased beyond r_c. However, one should exercise care in interpreting the implications of the above development. This results applies *only* if r_o is *less* than r_c; i.e., it generally applies to "small" diameter pipes/tubes. Thus, r_c represents the outer radius (not the thickness) of the insulation that will maximize the heat loss and at which point any further increase in insulation thickness will result in an increase in heat loss.

A graphical plot of the resistance R versus r is provided in Figure 6.2. (The curve is inverted for the plot of q or r). One notes that the maximum heat loss from a pipe occurs when the critical radium equals the ratio of the thermal conductivity of the insulation to the surface coefficient of heat transfer. This ratio has the dimension of length (e.g., ft). The equation for r_i can be rewritten in terms of a dimensionless number defined as the Biot number *Bi*.

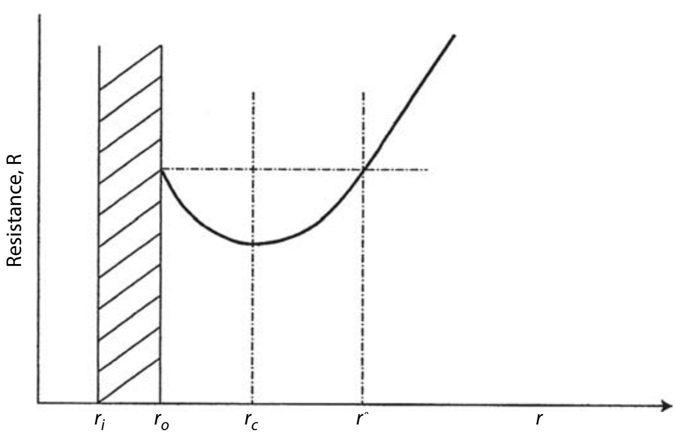

Figure 6.2 Resistance associated with the critical insulation thickness for a bare surface.

$$\frac{k_i}{h_o r_i} = \left(Bi\right)^{-1} \tag{6.25}$$

To reduce q below that for a bare wall ($r = r_0$), r must be greater than r_c; i.e., $r>r_c$. The radius at which this occurs is denoted as r^*. The term r^* may be obtained by solving the equation

$$q_{\text{bare}} = q_{l,r^*} \tag{6.26}$$

so that

$$\frac{2\pi L(T_i - T_o)}{\frac{1}{r_i h_i} + \frac{\ln(r_o / r_i)}{k_w} + \frac{1}{r_\text{o} h_\text{o}}} = \frac{2\pi L(T_i - T_o)}{\frac{1}{r_i h_i} + \frac{\ln(r_o / r_i)}{k_w} + \frac{\ln(r^* / r_o)}{k_i} + \frac{1}{r^* h_0}} \tag{6.27}$$

One may now solve for r^* using a suitable trial-and-error procedure. See Figure 6.2 once again. This topic will be revisited in Chapter 16, Illustrative Open-ended Problem #3.

6.8 Open-Ended Problems

This last Section of the chapter contains open-ended problems as they relate to heat transfer. No detailed and/or specific solution is provided; that task is left to the reader, noting that each problem has either a unique solution or a number of solutions or (in some cases) no solution at all. These are characteristics of open-ended problems described earlier.

There are comments associated with some, but not all, of the problems. The comments are included to assist the reader while attempting to solve the problems. However, it is recommended that the solution to each problem should initially be attempted without the assistance of the comments.

There are 43 open-ended problems in this section. As stated above, if difficulty is encountered in solving any particular problem, the reader should next refer to the comment, if any is provided with the problem. The reader should also note that the more difficult problems are generally located at or near the end of the section.

1. Describe the early history associated with heat transfer.
2. Discuss the recent advances in heat transfer technology.

3. Select a refereed, published article on heat transfer from the literature and provide a review.
4. Provide some normal everyday domestic applications involving the general topic of heat transfer.
5. Develop an original problem that would be suitable as an illustrative example in a book on heat transfer.
6. Prepare a list of the various books which have been written on heat transfer. Select the three best and justify your answer. Also select the three weakest books and, once again, justify your answer.
7. Develop a new method to experimentally determine the heat conductivity of a
   - solid;
   - liquid;
   - gas; and
   - slurry
8. Define tube/pipe *pitch* and *clearance* in layman terms.
9. Discuss the role *bypass* plays in the operation and performance of a heat exchanger.
10. Discuss the role flow arrangements in a heat exchanger can play in increasing heat transfer/recovery.
11. Discuss the effect that noncircular ducts have on the general heat transfer equations and the various heat transfer coefficient correlations.
12. Discuss the effect of conduits with varying cross-sectional areas have on the general heat transfer equation.
13. Consider the impact on the heat transfer performance of a heat exchanger as a function of the heat capacity of various cooling mediums flowing through the tubes of a shell-and-tube unit. Qualitatively discuss this effect as the heat capacity of the medium increases. Also, qualitatively discuss the effect of other physical properties on the exchanger's performance.
14. Most home heating systems employing either natural gas or oil are (relative to industrial systems) inefficient. Outline various procedures that can be applied to increase the thermal efficiency of these units. Include economic considerations in the analysis.
15. Describe the similarities and differences between the atmospheric *lapse rate* and natural convection heat transfer [10].
16. Global warming has become a reality and sea levels around the world have reportedly increased approximately 12

inches. As an authority on heat transfer, the President of the United States has invited you to participate in an emergency meeting whose agenda is concerned with providing the technical community with recommendations to reduce and/or eliminate rising sea levels. Suggest possible solutions.

17. Obtain order of magnitude values of industrial film coefficients for various boiling liquids, various condensing vapors, dropwise condensation for various vapors, film-type condensation for various vapors, boiling liquids, heating of various liquids and vapors, cooling of various liquids and vapors, and superheated steam.
    Comment: This will require a rather extensive investigation.
18. Discuss the differences and problems that can arise when condensing mixed vapors (as opposed to pure vapors).
19. Develop improved operation, maintenance and inspection (OM&I) procedures for heat exchangers.
20. Is there sufficient justification to employ the use of the log-mean temperature difference (LMTD) as the actual/true driving force in a heat exchanger.
21. Develop equations describing the "F" factor employed in shell-and-tube heat exchanger calculations for the following geometries:
    1. One shell pass: 2 or more tube passes
    2. Two shell pass: 4 or more tube passes
    3. Four shell pass: 8 or more tube passes
    4. Six shell pass: 12 or more tube passes
    5. Cross flow with both fluids unmixed
    6. Cross flow with one fluid unmixed

    Comment: Refer to the literature [4].
22. Consider a finned heat exchanger. Qualitatively discuss the effect that the thermal conductivity of the fins have on the heat transfer rate of the exchanger. Also discuss the effect of the heat capacity of the fin material.
23. Ganapathy, in his article titled "Size or Check Waste Heat Borders Quickly," *Hydrocarbon Processing*, New York City, NY, 169-170, September, 1984, provides a chart to design and predict the performance of a boiler. Convert that chart to equation form [12].
24. One option available to a plant manager when a tube within a heat exchanger fails is to simply plug the inlet of the tube. Develop an equation to describe the impact on the heat

transfer performance of the exchanger as a function of both the number of tubes within the exchanger and the number of plugged tubes.

25. Attempt to develop a general all-purpose equation that can be employed to describe the overall heat transfer coefficient for most of the heat exchangers in use today. Clearly define and quantify the terms/ factors that affect the coefficient.
26. Describe the hazards, risks, and safety associated with cryogenic operation and detail procedures that can be implemented to reduce those problems.
    Comment: Refer to L. Theodore and R. Dupont's *Environmental Health and Hazard Risk Assessment: Principles and Calculations text*, CRC Press/Taylor & Francis Group, Boca Raton, FL, 2012 [13].
27. Describe the hazards, risks, and safety associated with boiling operations and detail procedures that can be implemented to reduce those problems.
    Comment: Refer to L. Theodore and R. Dupont's *Environmental Health and Hazard Risk Assessment: Principles and Calculations text*, CRC Press/Taylor & Francis Group, Boca Raton, FL, 2012 [13].
28. Develop a generalized equation to describe the radiation view factor as a function of the geometry for aligned parallel rectangles.
    Comment: Refer to the literature [4].
29. Develop a generalized equation to describe the radiation view factor as a function of the geometry for perpendicular rectangles with a common edge.
    Comment: Refer to the literature [4].
30. Develop a generalized equation to describe the radiation view factor as a function of the geometry for coaxial parallel disks.
    Comment: Refer to the literature [4]
31. Outline Wilson's method for evaluating the *inside* film coefficient for a double pipe heat exchanger unit.
    Comment: Refer to E. Wilson, *Trans. AMSE*, 37, 47, ASME, New York City, NY, 1915 [14] This problem will be revisited in Part III, Chapter 27 – *Heat Transfer Term Projects.*
32. Devise a new method of reducing heat loss from a hot surface.
    Comment: Is insulation the only option?

33. Explain why entropy calculations should be included in some heat exchanger design calculations.
    Comment: Refer to L. Theodore et al, *Thermodynamics for the Practicing Engineer*, John Wiley & Sons, Hoboken, NJ, 2010 [15]
34. Design a new type of heat exchanger.
35. Design a new fin.
36. Develop an optimum design procedure that could be employed with the various classes of heat exchangers.
    Comment: This will require a rather extensive investigation.
37. Develop a comprehensive procedure for designing and predicting the performance of a cooling tower.
38. You have been hired by Theodore Consultants to estimate the energy requirements to maintain a small 10 ft high, 20 ft x 20 ft laboratory facility located along the Equator. Outline how to calculate the energy requirements if the temperature in the lab is to be maintained at 72°F. Clearly specify all information required in performing the analysis e.g., the properties of walls, ceiling and floor.
39. Refer to Problem 3 in the previous Section. Develop an expression for the optimum thickness of insulation from an economic perspective to cover a flat surface if the annual cost per unit area of insulation is directly proportional to the thickness. Neglect the air film (resistance) coefficient.
40. Refer to the previous problem. Resolve the problem but include the air film resistance. The air-film coefficient may be assumed constant and independent of insulation thicknesses.
41. CORENZA Partners designed a shell-and-tube heat exchanger to operate with a maximum discharge temperature of a hot stream to be cooled to 90°F. Once the unit was installed and running, the exchanger operated with a discharge temperature of 105°F. Rather than purchase a new exchanger, what options are available to bring the unit into compliance with the specified design temperature? Include how your answers would be affected if the unit is a shell-and-tube exchanger.
42. Environmentalists are now concerned that Earth's surface (not the atmosphere) will increase in temperature this century. Comment on the validity of this concern by estimating any temperature increases due to thermal activity within the Earth's surface.

43. Lou Theodore, a supposed authority on heat transfer and energy conservation, has indicated that there is a near limited amount of energy in the water covering the Earth's surface. Outline how this energy could be extracted from the water and discuss some of the economic considerations.

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