HEAT EXCHANGE EQUIPMENT

Heat transfer plays a vital role in heating, refrigerating, and air-conditioning as can be seen by the many sections in chapters of *Principles of Heating, Ventilating, and Air Conditioning*, Ninth Edition, dealing with equipment whose main function is the exchange of heat, thus providing either heating or cooling. The ASHRAE Handbook volumes have a number of chapters devoted to heat transfer and heat transfer applications: in the *2021 Fundamentals* volume, Chapter 4, Heat Transfer, and Chapter 5, Two-Phase Flow; and in the 2020 *Systems and Equipment* volume, Chapter 23, Air-Cooling and Dehumidifying Coils, Chapter 27, Air-Heating Coils, and Chapter 47, Heat Exchangers. This material briefly reviews the fundamentals of *applied* heat transfer and illustrates the basic approach to heat exchanger design and analysis.

1 Modes of Heat Transfer

Heat transfer or **heat** (the "transfer" is redundant) can be defined as the transfer of energy from one region or one body to another due to a ***temperature difference***. Heat transfer is as universal as gravity since differences in temperature exist all over the universe. Unlike gravity, however, heat transfer is governed not by a single relationship but by a combination of various independent laws of physics. Heat transfer is generally divided into three distinct modes: *conduction*, *convection*, and *radiation*. Strictly speaking, only conduction and radiation are both separate and purely heat transfer processes, since convection also involves mass transfer and includes conduction.

1.1 Conduction

Conduction is the mode of heat transfer whereby energy is transported between parts of an opaque, stationary medium or between two media in direct physical contact. In gases, conduction is due to the elastic collision of molecules; in liquids and electrically non-conducting solids, it is due primarily to longitudinal oscillations of the lattice structure. In metals, thermal conduction takes place in the same manner as electrical conduction; that is, with the movement of the free electrons.

The theory of heat transfer by conduction was first proposed by Jean B. Fourier in a noted work, published in 1822 in Paris, titled *Theorie analytique de la chaleur*. Fourier's law gives the heat transfer rate past any plane by the following:

$$q = -kA\frac{\partial T}{\partial n} \tag{1}$$

where k is the thermal conductivity of the material, A is the area normal to the flow of heat, T is the temperature, and n is the distance in the direction of heat flow. The partial derivative $\partial T / \partial n$ is the temperature gradient in the direction of the heat flow. The minus sign indicates that heat flows of its own accord only in the direction of decreasing temperature (from hot to cold), in accordance with the Second Law of Thermodynamics. The thermal conductivity k is the specific property of matter that indicates a material's ability to transfer heat, expressed as energy transferred per unit time per unit area per unit temperature gradient. Table 1 provides a few order-of-magnitude values of thermal conductivity.

If heat flows in more than one direction, or there are temperature variations in more than one direction, Equation (1) cannot be directly integrated but is only a start in the development of the three-dimensional *general conduction equation*. However, for the following two simple, but important, cases to HVAC applications shown in Figure 1, direct integration is possible with these results:

Table 1   Order of Magnitude of Thermal Conductivity at Room Temperature

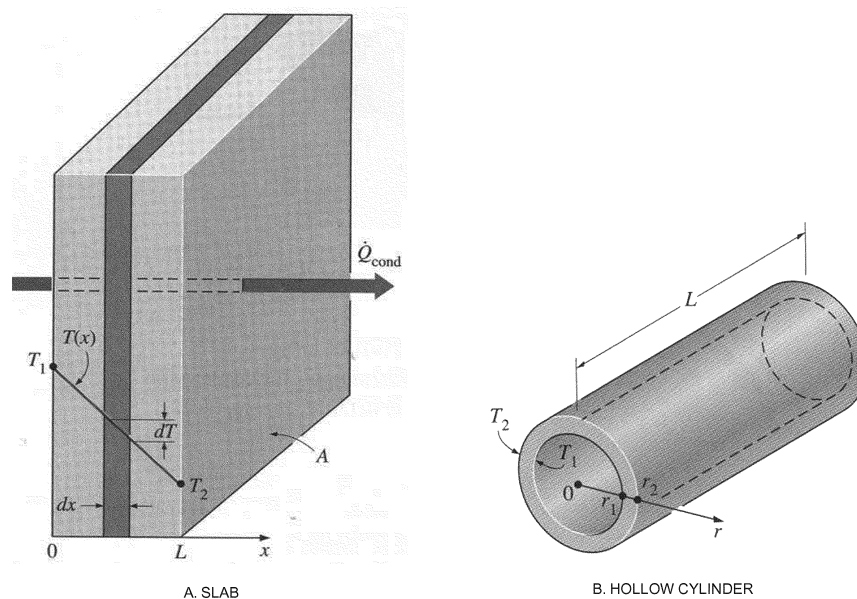| Material	k, W/(m·K)	K, Btu/h·ft·°F
Copper	400	220
Aluminum	200	110
Mild Steel	64	40
Stainless Steel	15	10
Concrete	1.4	0.8
Glass	1.1	0.6
Water	0.6	0.4
Wood	0.10	0.06
Polyvinyl chloride (PVC)	0.10	0.06
Fiberglass (medium density)	0.04	0.02
Air	0.04	0.02

Fig. 1   *Fourier's Law Applied to Two Simple Cases: (a) Slab and (b) Hollow Cylinder*

Case 1. The slab or plane wall under steady-state conditions:

$$q = kA\frac{(T_1 - T_2)}{L} = \frac{T_1 - T_2}{L/(kA)} \tag{2}$$

where L is the wall thickness, T_1 is the temperature at $x = 0$, and T_2 is the temperature at $x = L$. The quantity $L/(kA)$ can be considered the "thermal resistance" to the flow of heat.

Case 2. The hollow cylinder (tubes, pipes, etc.) under steady-state conditions:

$$q = \frac{2\pi kL(T_1 - T_2)}{\ln(r_2/r_1)} = \frac{T_1 - T_2}{[\ln(r_2/r_1)/(2\pi kL)]} \tag{3}$$

where L is the length of the cylinder, T_1 is the temperature at the inner radius r_1, and T_2 is temperature at the outer radius r_2. Here, the quantity $\ln(r_2/r_1)/(2\pi kL)$ is considered the "thermal resistance."

1.2 Convection

Convection is the mode of heat transfer whereby energy is transported by the combined action of conduction, energy storage, and mixing motion. Convection occurs between a solid surface and a moving fluid. When the fluid movement is produced by other than the heat transfer process itself (such as by a fan or pump), the convection is termed f*orced* convection. When the only motion is due to the heat transfer and the fact that warmer fluids are less dense and will naturally tend to rise, the convection is termed *free* or *natural* convection. A combination of free and forced convection may occur and is termed *mixed* convection.

The quite complex phenomenon of convection was analyzed by Sir Isaac Newton in 1701 resulting in the often called "Newton's Law of Cooling,"

$$q = hA(T_s - T_f) = \frac{(T_s - T_f)}{(1/hA)} \tag{4}$$

where h is the convective heat transfer coefficient (also called the "film coefficient"), A is the surface area in contact with the fluid, T_s (T_w sometimes used) is the surface or wall temperature, and T_f (T_∞ often used) is the fluid temperature outside the boundary layer. For convection, the quantity $1/hA$ is taken as the thermal resistance. Unfortunately, Equation (4) is actually no more nor less than the defining equation for the convection coefficient, h, and should be written:

$$h = q/[A(T_s - T_f)]$$

The equations for the governing laws for convection actually consist of five partial differential equations, namely:

- Conservation of Mass Equation
- Conservation of Energy Equation
- Momentum Equations (three, one per direction).

The h value could be obtained from solving these equations for $q,$ which is then substituted in Equation (4). Due to the mathematical complexity of the problem, most available values of the convection coefficient have been determined experimentally with empirical correlations provided for future reference and use in similar situations.

In almost every case, the fluid properties found in the correlations for predicting h depend significantly on temperature. In the case of density (and kinematic viscosity), there is also a pressure effect for gases. The temperature dependence means that there may be a significant variation in the quantities through the region of fluid near the surface (through the *boundary layer*). The accuracy of the predictive correlations depends upon the temperature(s) used for evaluating these thermodynamic and thermophysical properties. For convection occurring on an exterior surface, the *film temperature*, which is the average of the surface temperature and the undisturbed fluid temperature, is normally required. For internal flow, the *bulk mean fluid temperature* (also called the *average bulk temperature*), which is generally the average of the mean fluid inlet temperature and the mean fluid outlet temperature, is used as the temperature at which to evaluate the fluid properties. The bulk temperature is also called the "mixing cup" temperature as it represents an energy weighted average temperature. For viscous fluids, the correlation may also include the viscosity evaluated at the surface temperature.

Many engineering applications involve convection transport in noncircular tubes. At least to a first approximation, many of the circular tube results may be applied by using an effective diameter, also termed the hydraulic diameter, as the characteristic length and is defined as

$$D_e = D_h = 4A_c/P$$

where A_c and P are the flow cross-sectional area and the wetted perimeter, respectively. It is this diameter that is used in the calculation of Re_D and the Nu_D.

Table 2 provides the typical range of convective heat transfer coefficients for several common processes.

Table 2   Approximate Ranges of Convective Heat Transfer Coefficients

Flow and Fluid	h_c, W/(m²·K)	h, Btu/h·ft²·°F
Free convection, air	10	2
Free convection, water	50	10
Forced convection, air	100	20
Forced convection, water	500–10,000	100
Condensing vapor	5000–50,000	1000–10,000
Boiling liquid	1000–100,000	200–20,000

1.3 Radiation

Radiation is the mode of heat transfer wherein energy is emitted by one surface (converted from internal energy), transmitted as electromagnetic waves, and then absorbed by a receiving surface. All bodies emit radiant heat continually, with the intensity depending upon the temperature and the nature of the surface. Radiant heat is emitted by a body in the form of finite patches, or quanta, of energy. Their motion in space is similar to the propagation of light and is approximated as traveling in a straight line (slight curvature is neglected). There are many types of electromagnetic radiation, including radio waves, x-rays, gamma rays, as well as light and thermal radiation. The thermal radiation region is from about 0.1 to 100 microns, whereas the visible light portion of the spectrum is very narrow, extending from about 0.40 to 0.65 microns.

Conduction and convection heat transfer rates are driven primarily by temperature gradients and somewhat by temperature due to temperature-dependent properties; however, radiative heat transfer rates are driven by the fourth power of the absolute temperature and increase rapidly with temperature. Unlike conduction and convection, no medium is required to transmit electromagnetic energy and thermal radiation is assumed to pass undiminished through a vacuum and transparent gases. Although the rate of emission of energy is independent of the surroundings, the net heat transfer rate by radiation depends on the temperatures and spatial relationships of all surfaces involved.

The starting point for analyzing radiation heat transfer is with the answers to the three questions:

1. How much radiation is emitted (sent out) by any body?
2. Where does it go?
3. What happens when it gets there?

Depending upon the application, there are actually three answers to Question 1:

1. How much radiation is emitted at a particular wavelength?
2. How much radiation is emitted over all wavelengths?
3. How much radiation is emitted between any two wavelengths?

Radiation Emitted. The rate at which thermal radiation is emitted by an ideal surface (perfect emitter) is dependent on its absolute temperature and the wavelength. Such a surface is also a perfect absorber (i.e., it absorbs all incident radiant energy) and is called a ***blackbody***. Planck in 1901 showed that the spectral distribution of energy radiated by a blackbody at an absolute temperature T is given by:

$$E_{b\lambda} = \frac{C_1 \lambda^{-5}}{e^{C_2/\lambda T} - 1} \tag{5}$$

where

λ = wavelength, μm
T = temperature, K
C_1 = 3.743×10^8 W·μ/m² (1.187×10^8 Btu·μm⁴/h·ft²)
C_2 = 1.4387×10^4 μm/K (2.5896×10^4 μm·°R)

The symbol $E_{b\lambda}$ is used to denote the emitted flux per wavelength (monochromatic emissive power) and is defined as the energy emitted per unit surface area at wavelength lambda per unit wavelength interval

around lambda. Equation (5), called Planck's distribution law, or just ***Planck's law***, is the basic equation for thermal radiation.

The total radiation emitted over all wavelengths for a blackbody may be obtained by the direct integration of Planck's law:

$$E_b = \int_0^\infty E_{b\lambda}\, d\lambda = \sigma T^4 \tag{6}$$

The constant σ has the value of 5.67×10^{-8} W/m$^2\cdot$K^4 (0.1714×10^{-8} Btu/h$\cdot$ft$^2\cdot^\circ$R^4). This expression was deduced by Stefan in 1879 from experimental data. Boltzmann, in 1884, using classical thermodynamics, derived the expression and placed it on firm theoretical ground. The equation is called the ***Stefan-Boltzmann law***.

The radiation emitted between any two wavelengths can also be obtain by integrating Planck's law with the two wavelengths as upper and lower limits of the integral. However, the integration is not easy but fortunately has been accomplished and recorded as ***blackbody radiation functions*** in form(s) readily used, such as provided in Table 3.

The blackbody radiation function, $f_{0\text{-}\lambda}$, represents the fraction of radiation emitted from a blackbody at temperature T in the wavelength range from $\lambda = 0$ to λ. The values of f are listed in Table 3 as a function of λT, where λ is in μm and T is in K. The fraction of radiant energy emitted by a blackbody at temperature T over a finite wavelength band from λ_1 to λ is determined from

$$f_{\lambda_1 - \lambda_2}(T) = f_{\lambda_2}(T) - f_{\lambda_1}(T) \tag{7}$$

Radiation Between Any Two Surfaces. Since thermal radiation is taken to travel in straight lines, the determination of "how much goes where" becomes a matter of geometry. Hence the factor used to quantita-

Table 3   Blackbody Radiation Function, $f_{0-\lambda}$

| λT, μm$\cdot$K	f_λ	λT, μm$\cdot$K	f_λ	λT, μm$\cdot$K	f_λ
200	0.000000	4200	0.516014	8500	0.874608
400	0.000000	4400	0.548796	9000	0.890029
600	0.000000	4600	0.579280	9500	0.903085
800	0.000016	4800	0.607559	10,000	0.914199
1000	0.000321	5000	0.633747	10,500	0.923710
1200	0.002134	5200	0.658970	11,000	0.931890
1400	0.007790	5400	0.680360	11,500	0.939959
1600	0.019718	5600	0.701046	12,000	0.945098
1800	0.039341	5800	0.720158	13,000	0.955139
2000	0.066728	6000	0.737818	14,000	0.962898
2200	0.100888	6200	0.754140	15,000	0.969981
2400	0.140256	6400	0.769234	16,000	0.973814
2600	0.183120	6600	0.783199	18,000	0.980860
2800	0.227897	6800	0.796129	20,000	0.985602
3000	0.273232	7000	0.808109	25,000	0.992215
3200	0.318102	7200	0.819217	30,000	0.995340
3400	0.361735	7400	0.829527	40,000	0.997967
3600	0.403607	7600	0.839102	50,000	0.998953
3800	0.443382	7800	0.848005	75,000	0.999713
4000	0.480877	8000	0.856288	100,000	0.999905
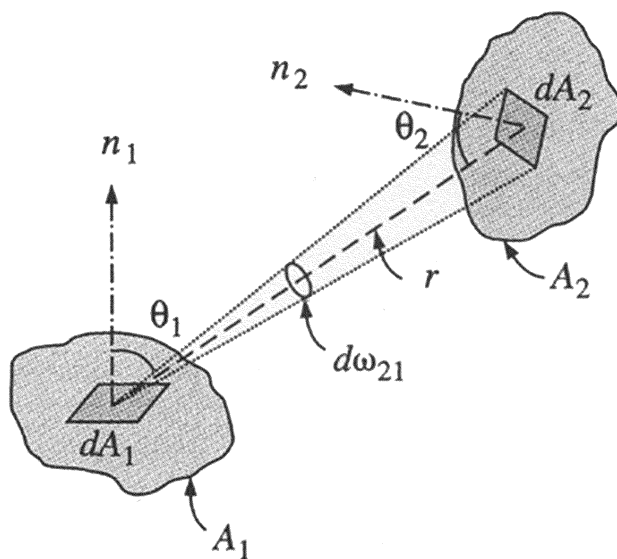
Fig. 2   View Factor Nomenclature

tively describe the fraction of the radiation leaving a surface and going to another surface is called the **geometric factor, angle factor, configuration factor,** or **view factor**. *View factor* will be used here.

The view factor F_{ij} by definition is the ***fraction*** of the ***total radiation*** leaving surface i that ***directly*** falls upon surface j. Figure 2 shows the geometry for determining the view factor between two surfaces, 1 and 2. The resulting equation for the view factor is

$$F_{12} = F_{A_1 \to A_2}$$
$$= \frac{1}{A_1} \int_{A_2} \int_{A_1} \frac{\cos \theta_1 \cos \theta_2}{\pi r^2} dA_1 dA_2 \tag{8}$$

where dA_1 and dA_2 are elemental areas of the two surfaces, r is the distance between dA_1 and dA_2, and θ_1 and θ_2 are the angles between the respective normals to dA_1 and dA_2 and the connecting line r. The solution of this equation in closed form is difficult, if not impossible, for all geometries. Numerical, graphical, and mechanical techniques have all provided alternative methods, and numerical values of the view factor for many geometries encountered in engineering may be found in the literature. It must be emphasized that the expression for the view factor is based on the assumption that the directional distribution of radiation leaving a surface is diffuse and uniformly distributed.

Two special properties play a very important role in obtaining numerical values for the complete set of view factors between the surfaces exchanging radiation. If there are n surfaces forming an enclosure, then

$$\sum_{j=1}^{n} F_{ij} = 1 \tag{9}$$

The other is the reciprocal relationship: $F_{ij}A_i = F_{ji}A_j$. In both cases, it is important to note that F_{jj} is not necessarily 0 since a concave surface may irradiate ("see") itself.
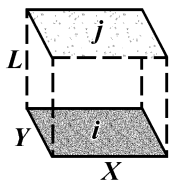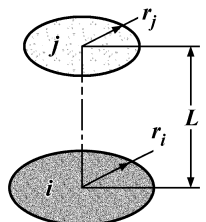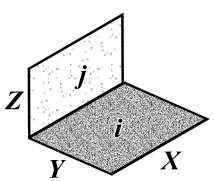
Numerical values of view factors for three common geometries are provided in Table 4 and Figure 3.

Radiation Falling on Surface. When radiant energy falls on a surface, portions may be absorbed in, reflected from, or transmitted through the material as shown in Figure 4.

Therefore, based on conservation of energy,

$$\alpha + \rho + \tau = 1 \tag{10}$$

Table 4   View Factors for Three-Dimensional Geometries

Geometry	Relation

Aligned parallel rectangles

$$\bar{X} = X/L, \qquad \bar{Y} = Y/L$$

$$F_{ij} = \frac{2}{\pi \bar{X} \bar{Y}} \left\{ \ln\left[\frac{(1+\bar{X}^2)(1+\bar{Y}^2)}{1+\bar{X}^2+\bar{Y}^2} \right]^{1/2} \right.$$

$$+ \bar{X}(1+\bar{Y}^2)^{1/2} \tan \frac{\bar{X}}{(1+\bar{Y}^2)^{1/2}}$$

$$+ \bar{Y}(1+\bar{X}^2)^{1/2} \tan^{-1} \frac{\bar{Y}}{(1+\bar{X}^2)^{1/2}}$$

$$\left. - \bar{X}\tan^{-1}\bar{X} - \bar{Y}\tan^{-1}\bar{Y} \right\}$$

Coaxial parallel disks

$$R_i = R_i/L, \qquad R_j = r_j/L$$

$$S = 1 + \frac{1+R_j^2}{R_i^2}$$

$$F_{ij} = \frac{1}{2} \left\{ S - [S^2 - 4(r_j/r_i)^2]^{1/2} \right\}$$

Perpendicular rectangles with a common edge

$$H = Z/X, \qquad W = (Y/X)$$

$$F_{ij} = \frac{1}{\pi W} \left(W \tan^{-1}\frac{1}{W} + H \tan^{-1}\frac{1}{H} \right.$$

$$- (H^2 + W^2)^{1/2} \tan^{-1} \frac{1}{(H^2+W^2)^{1/2}}$$

$$+ \frac{1}{4} \ln \left\{ \frac{(1+W^2)(1+H^2)}{1+W^2+H^2} \right.$$

$$\times \left[\frac{W^2(1+W^2+H^2)}{(1+W^2)(W^2+H^2)} \right]^{W^2}$$

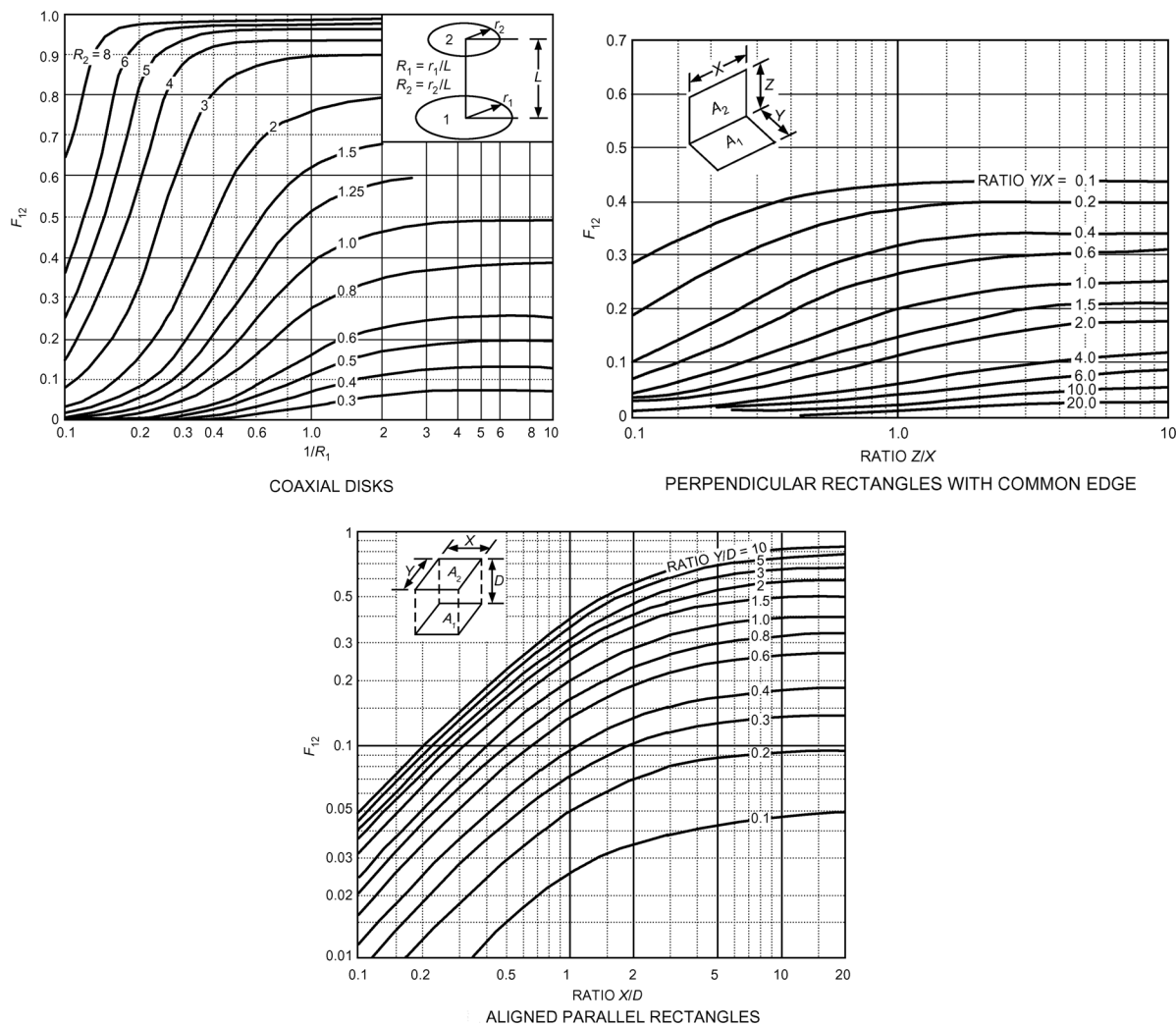$$\left. \left. \times \left[\frac{H^2(1+H^2+W^2)}{(1+H^2)(H^2+W^2)} \right]^{H^2} \right\} \right)$$

Fig. 3 View Factor Graphs for Common Geometries

where

α = fraction of incident radiation absorbed (absorptivity or absorptance)

ρ = fraction of incident radiation reflected (reflectivity or reflectance)

τ = fraction of incident radiation transmitted (transmissivity or transmittance)

For the many materials encountered in HVAC practice (other than fenestrations) that are opaque in the infrared region, $\tau = 0$, and thus $\rho = 1 - \alpha$.

Actual Surfaces (Nonblack Bodies). Materials and surfaces of engineering interest show marked divergences from the Stephan-Boltzmann and Planck laws. Actual surfaces emit and absorb less readily and are called nonblack. The emittance (or emissivity) ε of the actual surface is defined as the ratio of the radiation emitted by the surface to the radiation emitted by a blackbody at the same temperature. Emissivity is a function of the material, the condition of its surface, and its temperature. In general, the emissivity of a surface may vary with wavelengths. To overcome this complexity, gray surface behavior (ε = constant over all wavelengths) is often assumed. Many engineering surfaces actually do approximate this condition, at least in some regions of the spectrum. However, one must be especially careful at high temperature. The emissive power of a nonblack surface, at temperature T, is given by

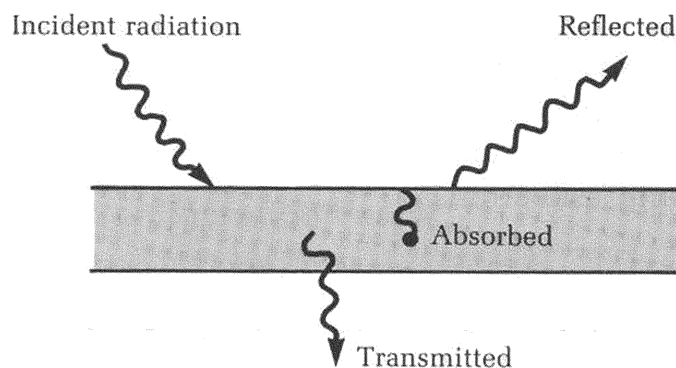$$E = \varepsilon E_b = \varepsilon \sigma T^4 \tag{11}$$

Fig. 4   Radiation Incident on Surface

where ε is the total hemispherical emittance (or emissivity) and is a strong function of the condition and temperature of the actual surface. Table 5 provides approximate emittance values of some common materials and surface finishes at room and solar temperatures. In general, both ε and α of a surface depend on the temperature and the wavelength of the radiation. ***Kirchhoff's law*** of radiation states that the emittance and the absorptance of a surface at a given temperature and wavelength are equal. In many practical applications, the surface temperature and the temperature of the source of incident radiation (major exception – solar radiation) are of the same order of magnitude, and the average absorptance of a surface is taken to be equal to its average emittance ($\alpha = \varepsilon$).

1.4 Net Radiant Energy Loss from a Surface

When radiation heat transfer is involved in the energy balance on a surface, the net heat gain (or loss) by radiation from the surface is the quantity of interest. The general problem of determining the radiation exchange in an enclosure consisting of *n* surfaces, which may see one another by no others, requires the solution of n linear algebraic equations to account for the possibly infinite number of reflections of radiation from the participating surfaces. The current method of determining this quantity is called the ***radiosity method.*** It begins with the definitions of the two terms, radiosity and irradiation, and is applicable subject to the following conditions:

- Each surface is opaque, gray, isothermal, and uniformly irradiated.
- The emission and the reflections from each surface are diffuse.

1.5 Radiosity

The *radiosity J* is the total radiation leaving a surface, per unit area, per unit time. It includes both the emitted and the reflected amounts and can be expressed for surface *i* as

$$J_i = E_i + \rho G_i = \varepsilon_i \sigma T_i^4 + (1 - \varepsilon_i) G_i \tag{12}$$

where G_i is the radiation falling upon surface *i*.

1.6 Irradiation

The *irradiation G* is the total radiation falling on a surface, per unit area, per unit time. The radiation incident on the *i*th surface is

$$A_i G_i = J_1 A_1 F_{1i} + J_2 A_2 F_{2i} + J_3 A_3 F_{3i} + \cdots$$

or

$$G_i = \sum_{j=1}^{n} J_j F_{ij}$$

(using the reciprocal rule).

Table 5   Emittance Values of Common Materials

Material and Surface Condition	Total Hemispherical Emittance	Solar Absorptance	Material and Surface Condition	Total Hemispherical Emittance	Solar Absorptance
Aluminum			Marble		
Foil	0.05	0.15	Polished	0.89	
Alloy, as received	0.04	0.37	Smooth	0.56	
Weather alloy	0.20	0.54	Paints		
Asphalt (Roofing/Pavement)	0.88		Black		
Brick	0.90	0.63	Flat	0.97	0.98
Concrete, rough	0.91	0.60	Gloss	0.90	
Copper			White		
Electroplated	0.03	0.47	Acrylic resin	0.90	0.26
Oxidized plate	0.76		Gloss	0.85	
Frost, rime	0.99		Skin	0.95	
Glass (smooth)	0.91		Soil	0.94	
Gravel	0.30		Snow (fresh)	0.82	0.13
Ice (smooth)	0.97		Stainless Steel		
Iron			Polished	0.60	0.37
Wrought, polished	0.29		Dull	0.2	
Wrought, dull	0.91		Vegetation	0.94	
			Water	0.90	0.98
			Wood (smooth)	0.84	

The radiosity of surface i is $J_i = \varepsilon_i E_{bi} + (1 - G_i)$; substituting for G_i gives

$$J_i = \varepsilon_i E_{bi} + (1 - \varepsilon_i)\sum_{}^{n} J_j F_{ik};$$

$$i = 1, 2, \ldots, n$$

(13)

a system of n linear equations in the n unknowns J_i.

Upon solving the simultaneous equations and obtaining the values for the Js, the net radiant heat loss from each surface is obtained from

$$q_i = [J_i - G_i]A_i$$

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

(14)

For the special case when only two surfaces are involved, the net loss can be written as

$$q_i = \frac{E_{b1} - E_{b2}}{\dfrac{1 - \varepsilon_1}{\varepsilon_1 A_1} + \dfrac{1}{F_{1\text{-}2}A_1} + \dfrac{1 - \varepsilon_2}{\varepsilon_2 A_2}}$$

$$= \frac{\sigma(T_1^4 - T_2^4)}{\dfrac{1 - \varepsilon_1}{\varepsilon_1 A_1} + \dfrac{1}{F_{1\text{-}2}A_1} + \dfrac{1 - \varepsilon_2}{\varepsilon_2 A_2}}$$

(15)

1.7 Radiation and Combined Heat Transfer Coefficients

A special case that occurs frequently involves radiation exchange between a small surface at T_s and a much larger, isothermal surface that completely surrounds the smaller one. The *surroundings* could, for example, be the walls of a room whose temperature T_{sur} differs from that of an enclosed surface $(T_{sur} \neq T_s)$. For such a condition, the *net* rate of radiation heat transfer *from* the surface, per unit area of the surface, is

$$q = \varepsilon A \sigma (T_s^4 - T_{sur}^4) \tag{16}$$

The T^4 dependence of radiant heat transfer complicates engineering calculations. When T_1 and T_2 are not too different, it is convenient to linearize Equation (16) by factoring the term $(\sigma T_1^4 - \sigma T_2^4)$ to obtain

$$q_{12} = \varepsilon_1 A_1 (T_1^2 + T_2^2)(T_1 + T_2)(T_1 - T_2)$$
$$\cong \varepsilon_1 A_1 \sigma (4 T_m^3)(T_1 - T_2) \tag{17}$$

for $T_1 \cong T_2$, where T_m is the mean of T_1 and T_2. This result can be written more concisely as

$$q \cong A_1 h_r (T_1 - T_2)$$

where $h_r = 4\varepsilon_1 \sigma T_m^3$ is called the **radiation heat transfer coefficient**, in Btu/h·ft^2·°F (W/m^2·K).

Heat transfer from surfaces is usually a combination of convection and radiation. It is assumed that these modes are additive, and therefore a combined surface coefficient can be used to estimate the heat flow to/from a surface:

$$h_o = h_c + h_r$$

where

h_o = overall surface coefficient, Btu/h·ft^2·°F (W/m^2·K)
h_c = convection coefficient, Btu/h·ft^2·°F (W/m^2·K)
h_r = radiation coefficient, Btu/h·ft^2·°F (W/m^2·K)

Assuming the radiant environment is equal to the temperature of the ambient air, the heat loss/gain at the surface can be calculated as

$$q = h_o A (T_{surf} - T_{amb}) \tag{18}$$

Radiation is usually significant relative to conduction or natural convection, but negligible relative to forced convection. Thus, radiation in forced-convection applications is usually disregarded, especially when the surfaces involved have low emissivities and low to moderate temperatures.

2 Heat Exchangers

A heat exchanger is a device that permits the transfer of heat from a warm fluid to a cooler fluid through an intermediate surface without mixing of the two fluids. The correct sizing and selection of heat exchangers is probably the most important single factor in designing an efficient and economical building HVAC&R system. Whether the heat exchanger is selected as an off-the-shelf item or designed especially for the application, the following factors are normally considered:

- Thermal performance
- Cost
- Pressure drop
- Space requirements
- Serviceability

The main types of heat exchangers found in HVAC&R systems are the: finned-tube (coil), shell-and-tube, and plate. Sketches of each are given in Figure 5.
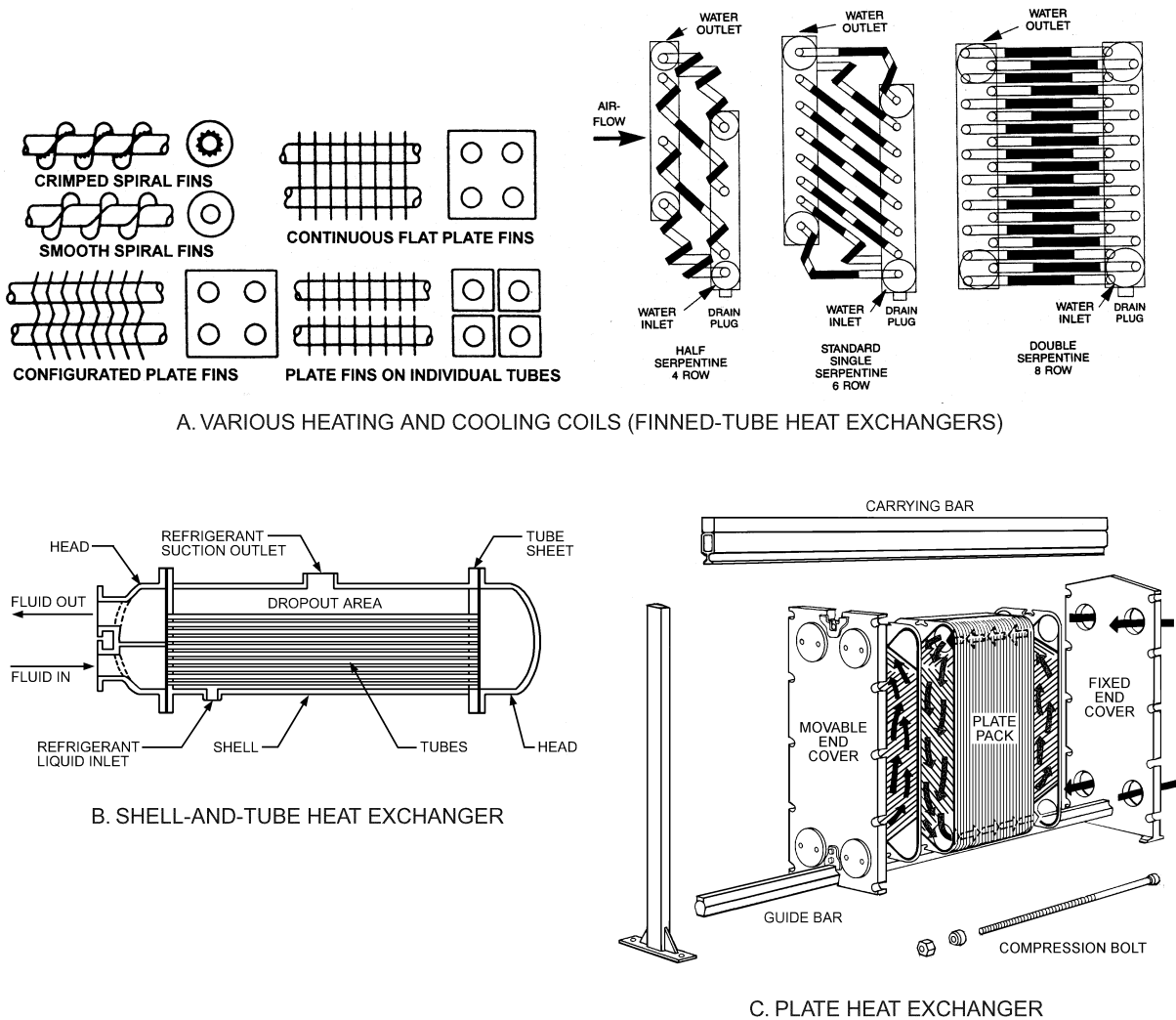
A. VARIOUS HEATING AND COOLING COILS (FINNED-TUBE HEAT EXCHANGERS)

B. SHELL-AND-TUBE HEAT EXCHANGER

C. PLATE HEAT EXCHANGER

Fig. 5 *Sketches of the Three Types of Heat Exchangers: (A) Finned-Tube, (B) Shell-and-Tube, and (C) Plate*
(Part A: Figure 1, Chapter 23; Part B: Figure 2, Chapter 42; and Part C: Figure 15, Chapter 48,
2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

2.1 Plate Fin (Extended-Surface) Coils

Most coils in HVAC systems consist of tubes with fins attached to their outer surface. Air flows over the outside of the tubes and refrigerant, steam, or water flows inside the tubes. The purpose of the fins is to increase the surface area on the air side where the convection coefficient is usually much lower than on the refrigerant, steam, or water side.

The *face area* of the coil is the cross-sectional area of the air stream at the entrance of the coil and is obtained from the *length* (sometimes called the *width*) of the coil multiplied by the *height* of the coil. The *face velocity* of the air is the volume flow rate of the air divided by the face area. The *surface area* of the coil is the heat transfer surface area in contact with the air. The *number of rows* of tubes and the *depth* of the coil are measured in the direction of the airflow.

In comparison to bare tube coils of the same capacity, finned coils are much more compact, less weight, and usually less expensive. The secondary surface area of a finned coil may be 10 to 40 times that of the bare tubes. The primary surface is that of the tubes or pipes. The secondary surface (fins) consists of thin metal plates or a spiral ribbon uniformly spaced or wound along the length of the primary surface and in intimate contact with it. The bond between fin and tube is a significant parameter in the thermal performance of the coil. The bonding is usually accomplished by expanding the tubes (often copper) into the tube holes in the

plate fins (often sheets of aluminum). The tube holes are often punched with a formed fin collar which both provides contact area as well as a means of spacing the fins uniformly along the length of the tubes. Figure 5A illustrates several finned-tube heat exchangers (coils).

More information on cooling coils and heating coils can be found in the Air-Processing Equipment online supplemental material accompanying *Principles of Heating, Ventilating, and Air Conditioning*, Ninth Edition.

2.2 Shell-and-Tube Heat Exchangers

The most common type of heat exchanger in industrial applications is probably the shell-and-tube heat exchanger. This type also finds extensive use in HVAC&R applications involving water cooled units and water and brine chillers. Figure 5B provides a sketch of the shell-and-tube heat exchanger.

Shell-and-tube heat exchangers can handle from a single tube to a large number of tubes packed in a shell with their axes parallel to that of the shell. Heat transfer takes place as one fluid flows inside the tubes while the other fluid moves outside the tubes through the shell. Baffles are usually placed in the shell to force the shell-side fluid to maintain uniform and good contact with the outside of the tubes. Shell-and-tube heat exchangers are generally classified according to the number of tube and shell passes involved.

2.3 Plate Heat Exchangers

The plate-and-frame (or just plate) type of heat exchanger continues to find additional use in HVAC applications, including the water side economizer. Plate heat exchangers consist of metal plate pairs arranged to provide separate flow paths (channels) for the two fluids. Heat transfer occurs across the plate walls. The hot and cold fluids flow in alternate passages, and thus each cold fluid stream is surrounded by two hot fluid streams, resulting in very effective heat transfer. Figure 5C provides a pictorial view of the plate exchanger and illustrates the flow paths.

The exchangers have multiple channels in series that are mounted on a frame and clamped or welded together. The rectangular plates have an opening or port at each corner. When assembled, the plates are sealed such that the ports provide manifolds to distribute the fluids through the separate flow paths. The clamped type of plate heat exchanger can be easily enlarged to meet higher heat transfer rates by simply mounting more plates. Plate exchangers are particularly well suited for liquid-to-liquid heat exchange applications, but also find use as condensers and evaporators.

3 Basic Heat Exchanger Design Equation

By applying the fundamentals of heat transfer to heat exchangers whose purpose it is to transfer heat from one fluid to another, the overall heat flow from one fluid across a barrier to a second fluid is often expressed as:

$$q = UA \, \Delta T \tag{19}$$

where
q = rate of heat transfer, Btu/h or kW
A = surface area of material separating the two fluids, ft^2 or m^2
U = overall coefficient of heat transfer, Btu/h·ft^2·°F or W/m^2·K
ΔT = *mean* temperature difference between the hot and cold fluids, °F or K

Rearranging this equation yields the basic design equation for a heat exchanger as

$$A = \frac{q}{U \, \Delta T_m} \tag{20}$$

where A is the total heat transfer area required in the exchanger. Thus, it will now be necessary to first calculate (estimate) q, U, and ΔT, as discussed in the following sections.

4 Estimation of Heat Load

The usual first step in designing a heat exchanger is to use the First Law of Thermodynamics to make an energy balance for (a) estimating the heat load (duty) of the heat exchanger, and probably (b) the required

flow rate of one of the fluids. Since the heat exchanger is normally assumed to be overall adiabatic (only energy exchange takes place within the heat exchanger, from the hot fluid to the cold fluid), the heat load is calculated in the general case from

$$q = m_h(h_{h,\,in} - h_{h,\,out}) = m_c(h_{c,\,out} - h_{c,\,in}) \tag{21}$$

where m_h and m_c are the mass flow rates of the hot and cold fluids and $h_{h,in}$, $h_{h,out}$, $h_{c,in}$, and $h_{c,out}$ are the respective enthalpies. When there is no change in phase, the enthalpy change can be replaced as follows:

$$h_{out} - h_{in} = c_p(T_{out} - T_{in})$$
$$\text{or} \quad h_{in} - h_{out} = c_p(T_{in} - T_{out}) \tag{22}$$

where c_p is the specific heat of the particular fluid.

Upon specifying the function of the particular heat exchanger(e.g., cool a known amount of hot fluid from one temperature to another), Equation (20) can be used to calculate the required heat transfer rate from the hot fluid (also the rate to the cold fluid) as well as to determine either the amount of the other fluid (if both inlet and outlet temperatures are known) or determine its outlet temperature (if its flow rate and inlet temperature are known).

5 Mean Temperature Difference

The temperature of the fluids flowing through a heat exchanger generally varies from location to location as heat is transferred from the hotter to the colder fluid. There is no single temperature difference serving as the driving force for the heat transfer and thus the rate of heat transfer with this varying temperature difference must be obtained by integrating

$$dq = U \, dA \, \Delta T$$

over the heat transfer area A along the length of the heat exchanger. The result, for either concurrent (parallel) or countercurrent (counterflow) flow conditions yields

$$q = UA(\Delta T_a - \Delta T_b)/\ln(\Delta T_a/\Delta T_b)$$

where a and b refer to the two ends of the heat exchanger.

The concept of an appropriate mean temperature difference, a single temperature difference which results in the same heat flow value, is useful and widely used in engineering practice:

$$q = UA \, \Delta T_{mean}$$

For parallel or counterflow conditions, this mean temperature difference is therefore

$$\Delta T_{mean} = (\Delta T_a - \Delta T_b)/\ln(\Delta T_a/\Delta T_b) \tag{23}$$

and is called the log mean temperature difference or LMTD.

For stream conditions other than the ideal counterflow (such as the common cross flow), a correction factor F is applied to the LMTD obtained as if the flow had been pure counterflow. Examples of these corrections factors are provided in Figure 6 (Bowman et al. 1940).

The resulting equation for the heat transfer rate becomes

$$q = UAF \, \Delta T_{m,\,cf} \tag{24}$$

For cases where at least one fluid temperature remains constant (e.g., evaporation or condensation), the correction factor is unity regardless of the flow pattern.
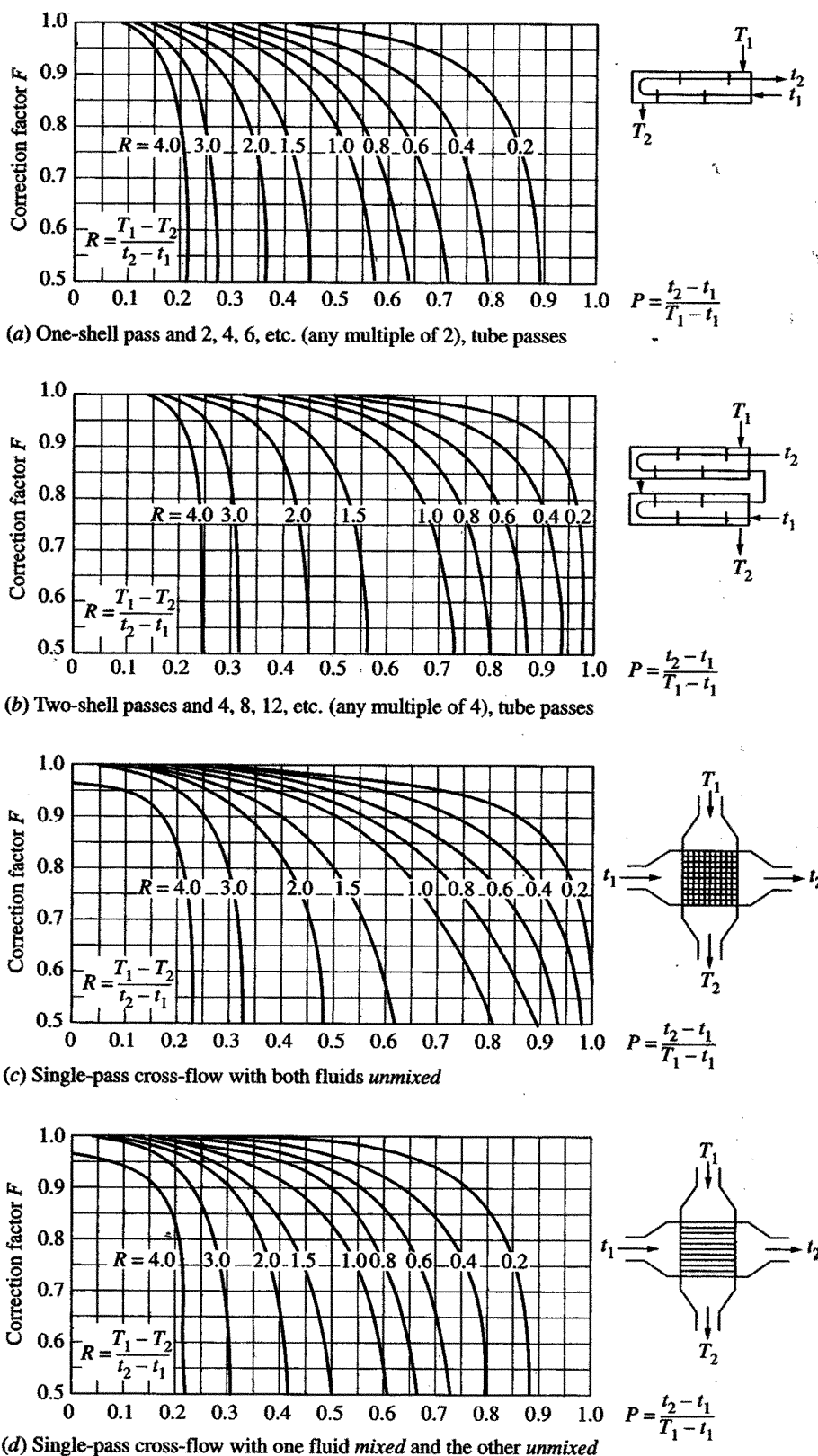
$R = \dfrac{T_1 - T_2}{t_2 - t_1}$

$P = \dfrac{t_2 - t_1}{T_1 - t_1}$

(a) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes

$R = \dfrac{T_1 - T_2}{t_2 - t_1}$

$P = \dfrac{t_2 - t_1}{T_1 - t_1}$

(b) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes

$R = \dfrac{T_1 - T_2}{t_2 - t_1}$

$P = \dfrac{t_2 - t_1}{T_1 - t_1}$

(c) Single-pass cross-flow with both fluids *unmixed*

$R = \dfrac{T_1 - T_2}{t_2 - t_1}$

$P = \dfrac{t_2 - t_1}{T_1 - t_1}$

(d) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

Fig. 6   LMTD Correction Factors for Several Flow Configurations
(Bowman et al. 1940)

6 Estimation of the Overall Heat Transfer Coefficient *U*

The governing equations for the design of heat exchangers are as follows:

$$Q = U_o A_o F \Delta T_{m, cf} = U_i A_i F \Delta T_{m, cf} \tag{25}$$

where

Q	=	amount of heat transfer for the coil to do
U	=	overall heat transfer coefficient
F	=	temperature difference correction
ΔT	=	log mean temperature difference
A_i, A_o	=	internal or outside areas

$$U = 1 / \Sigma R \tag{26}$$

where the *R*s are resistances to the flow defined by

$A/H_i A_i$	=	internal convective resistance
$A R_{fi}/A_i$	=	internal fouling in the pipes
$A \ln(d_o/d_i)/(2\pi kl)$	=	conductive resistance through wall of piping
$A R_{fo}/A_o$	=	outside fouling resistance
$A R_c/A_o$	=	contact resistance between tube and fin
$A/[h_o(A_{\text{unfin}} + \phi A_{\text{fin}})]$	=	outside convective resistance

$$\Delta T_{m, cf} = (\Delta T_a - \Delta T_b)/\ln(\Delta T_a - \Delta T_b) \tag{27}$$

These equations become

$$Q = \frac{F \Delta T_{m, cf}}{\dfrac{1}{h_i A_i} + \dfrac{R_{fi}}{A_i} + \dfrac{\ln(d_o/d_i)}{2\pi kL} + \dfrac{R_{fo}}{A_o} + \dfrac{R_c}{A_o} + \dfrac{1}{h_o(A_u + \phi A_f)}} \tag{28}$$

A complicated heat transfer phenomenon is considerably simplified by the assumption of boundary layers or films between the barrier wall and the fluids that offer resistance to heat flow. This mechanism is represented by the following equation, which assumes a constant overall heat transfer coefficient in the entire heat exchanger. This is not unreasonable when the temperature change in each fluid is small and therefore there is little change of physical properties between the inlet and the outlet.

$$U = \frac{1}{\dfrac{1}{h_o} + \dfrac{1}{R_{fo}} + \dfrac{d_w}{k_w}\left(\dfrac{A_o}{A_{\text{mean}}}\right) + \dfrac{1}{R_{fi}}\left(\dfrac{A_o}{A_i}\right) + \dfrac{1}{h_i}\left(\dfrac{A_o}{A_i}\right)} \tag{29}$$

where

U	=	overall heat transfer coefficient, Btu/h·ft^2·°F (W/m^2·K)
h_o	=	film coefficient of fluid outside tube, Btu/h·ft^2·°F (W/m^2·K)
h_i	=	film coefficient of fluid inside tube, Btu/h·ft^2·°F (W/m^2·K)
R_{fo}	=	fouling coefficient outside of tube, Btu/h·ft^2·°F (W/m^2·K)
R_{fi}	=	fouling coefficient inside of tube, Btu/h·ft^2·°F (W/m^2·K)
d_w	=	thickness of tube wall, ft (m)
k_w	=	thermal conductivity of tube, Btu·ft/h·ft^2·°F (W/m·K)
A_o/A_i	=	ratio of outside to inside tube surface
A_{mean}	=	average tube area per unit length, ft^2/ft (m^2/m)
A_o	=	outside tube area per unit length, ft^2/ft (m^2/m)

The accuracy of this relationship is limited by the reliability of the correlations for calculating the individual film coefficients, and by the arbitrary selection of fouling coefficients.

7 Extended Surfaces, Fin Efficiency, and Fin-Tube Contact Resistance

When the tube is finned on the air side to enhance heat transfer, the total heat transfer surface on the finned side becomes

$$A_s \;=\; A_{\text{total}} \;=\; A_{\text{fin}} + A_{\text{unfinned}}$$

where A_{fin} is the surface area of the fins and $A_{unfinned}$ is the area of the unfinned portion of the tube surface. However, when determining the heat transfer rate using the overall coefficient U, the appropriate total surface area to use is the effective area which means de-rating the area of the fins due to the internal thermal resistance within the fins. Thus, the appropriate surface area is

$$A \;=\; A_{\text{unfinned}} + \phi A_{\text{fin}} \tag{30}$$

The rectangular-plate fin of uniform thickness is commonly used in finned coils for heating or cooling air. It is not possible to obtain an exact mathematical solution for the efficiency of such a fin. It can be shown that an adequate approximation is to assume that the fin area served by each tube is equivalent in performance to a flat circular-plate fin of equal area. Figure 7 shows the method for determining the equivalent outer radius for this method. The corresponding efficiency for the flat plate fin can then be obtained from Figure 8.

As detailed in Chapter 4 of the 2021 *ASHRAE Handbook—Fundamentals*, the approximate fin efficiencies can also be calculated as provided in Figure 9.

The most common means of bonding the fins to the tubes on common heating and cooling coils is by mechanical expansion. Results of Sheffield et al. (1985) have shown that, for properly expanded tubes, there is a relatively narrow range of values of thermal resistance due to the fact that the tubes and fins are only in contact over a relatively small area due to surface asperities, non-roundness, and other factors. As reported by Sheffield et al., a reasonable value for the conductance between mechanically expanded copper tubes and aluminum fins is 3750 Btu/h·ft^2·°F (21 293 W/m^2·K). The thermal contact resistance (TCR) is the inverse, or 0.000267 h·ft^2·°F/Btu (0.000047 m^2·K/W).

8 Fouling Factors

After a period of operation, the heat transfer surfaces of a heat exchanger may become coated with various deposits from the fluids or may become corroded as a result of interaction between the fluids and the surface material. This coating represents an additional resistance to the flow of heat and results in decreased heat transfer performance. The effect is accounted for by a fouling factor, or fouling resistance, R_f, which is then to be added to the other resistances in the thermal path between the two fluids.

The most common type of fouling is the precipitation of solids from the fluid onto the heat transfer surface. Other types of fouling include chemical fouling (e.g., corrosion) and biological fouling from algae growth.
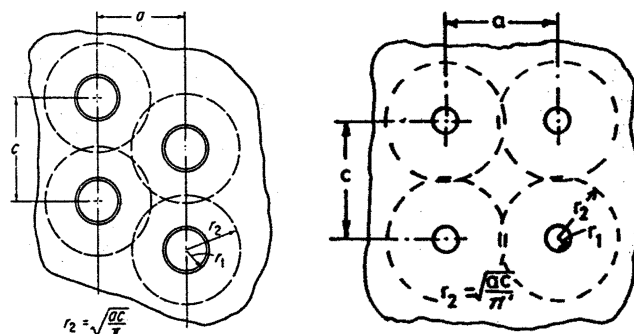

Fig. 7   Approximate Method for Obtaining Efficiency of Common Flat Plate Fin
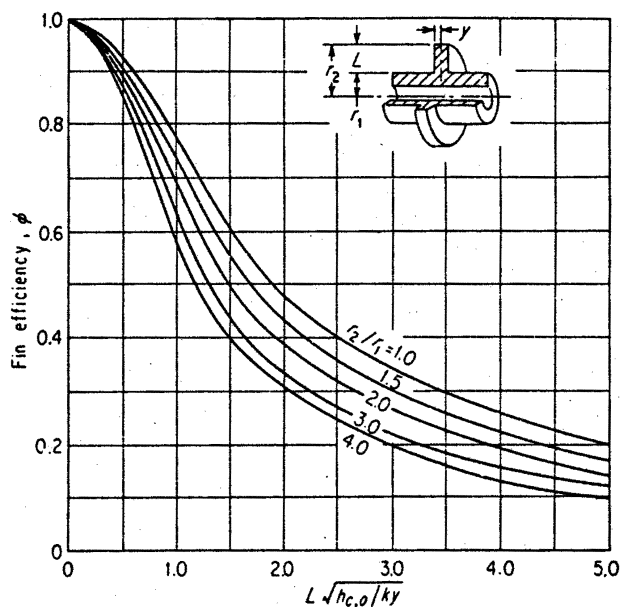
Fig. 8   Efficiency of Circular-Plate Fin of Uniform Thickness
(Gardner 1945)

$$\phi_d = [\tanh(mr\psi)]/(mr\psi)$$

where
$$m = [2h_{od}/(k_f y_f)]^{0.5}$$
$$\psi = (R/r - 1)[1 + 0.35 \ln(R/r)]$$

For in-line tube configuration,
$$R/r = 1.28\alpha(\beta - 0.2)^{0.5}$$

For triangular tube configuration,
$$R/r = 1.27\alpha(\beta - 0.3)^{0.5}$$

$$\beta = L/M \qquad \text{and} \qquad \alpha = M/r$$
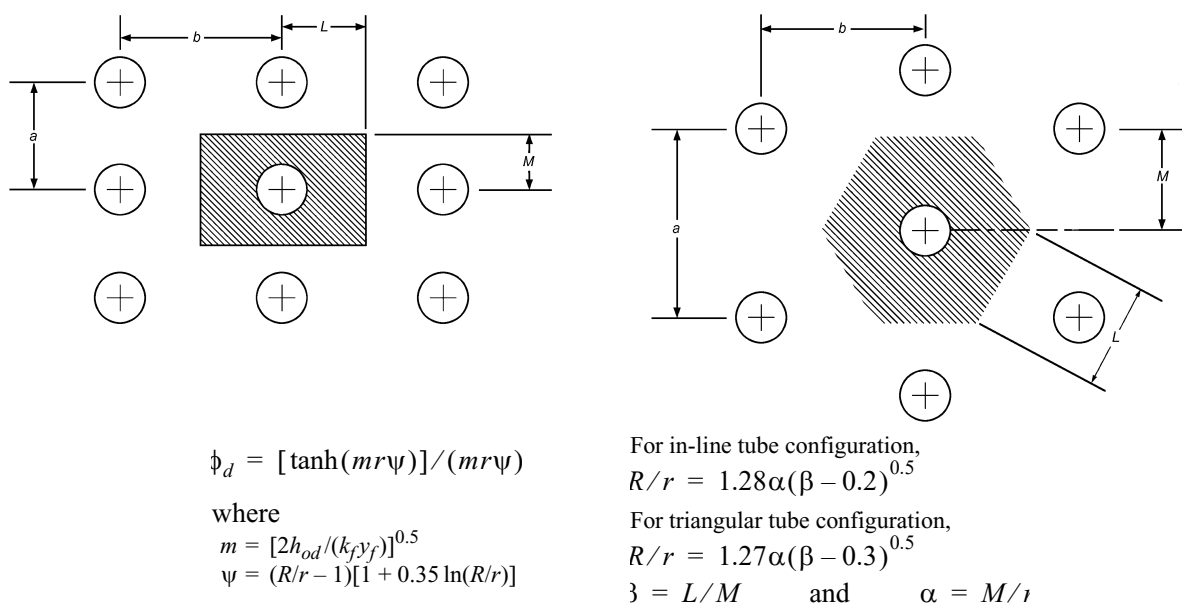
Fig. 9   Approximate Equations for Plate Fin Efficiency

The fouling factor depends upon the tube material, the nature of the fluid, and the fluid velocity. Fluid velocities less than about 3 ft/s (0.9 m/s) tend toward excess fouling. A few example values of fouling factors are provided in Table 6 and Table 7. Considerable uncertainty exists in these values, and they should be used cautiously. More comprehensive tables are available from TEMA (Tubular Exchanger Manufacturers Association) (tema.org).

There is little published data on the rate of fouling for heat exchangers in typical air conditioning and refrigeration service. For many years, the basic reference has been the *TEMA Standard* (2019). The air conditioning industry has for decades commonly used an assumed fouling level of 0.0005 h·ft^2·°F/Btu (0.00009 m^2·K/W) in both condensers and coolers. Occasionally, where a condenser was to use river water, engineers

Table 6   Fouling Factors for Water, $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot K/W$)

Type of Water	Water Velocity, ft/s (m/s)	
	3 (1) and less	Over 3 (1)
Cooling tower and spray pond		
Treated makeup	0.001 (0.00018)	0.001 (0.00018)
Untreated	0.003 (0.0005)	0.003 (0.0005)
River water (average)	0.002 (0.00035)	0.001 (0.00018)
Hard	0.003 (0.0005)	0.002 (0.00035)
Distilled or closed cycle condensate	0.001 (0.00018)	0.001 (0.00018)

Table 7   Fouling Factors for Various Fluids

Type of Fluid	Fouling Factor	
	$h \cdot ft^2 \cdot F/Btu$	$m^2 \cdot K/W$
Gases and vapors		
Steam (non-oil bearing)	0.0005	0.00009
Refrigerant vapors (oil bearing)	0.002	0.00035
Refrigerant vapors (pure)	0	0
Compressed air	0.002	0.00035
Industrial organic heat transfer media	0.001	0.00018
Liquids		
Refrigerant liquids	0.001	0.00018
Industrial organic heat transfer media	0.001	0.00018

would specify as much as 0.0020 $h \cdot ft^2 \cdot °F/Btu$ (000035 $m^2 \cdot K/W$) fouling. Based on more recent studies by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI, formerly ARI), it appears reasonable to specify a fouling factor of 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m^2 \cdot K/W$) for

- closed-loop liquid chillers and
- condensers served by well-maintained cooling towers.

9 Convective Heat Transfer Coefficients h_i and h_o

The determination of accurate values for the inside and outside convective heat transfer coefficients is critical to the accurate evaluation of heat exchanger performance. Unfortunately, even today, the available correlations for predicting these coefficients often leave much to be desired, particularly if a phase change is occurring.

The correlations presented in the following subsections are included herein primarily as (a) examples, and (b) to provide sample working relations for use with both the example heat exchanger design problems and the homework problems at the end of this material. The reader is referred to the current technical literature in heat transfer (e.g., *International Journal of Heat and Mass Transfer*, *Journal of Heat Transfer*, *International Journal of HVAC & Refrigeration Research*) for improved correlations.

9.1 Single-Phase Internal Flow in Tubes

The simplified relation of McAdams (1940) is widely used for turbulent single-phase flow in tubes and pipes:

$$\text{Nu} = 0.023(\text{Re})^{0.8}(\text{Pr})^{1/3} \tag{31}$$

The relation is relatively simple but gives maximum errors of $\pm25\%$ in the range of $0.67 < \text{Pr} < 100$. A more accurate correlation, which is also applicable for rough ducts, has been developed by Petukhov and coworkers at the Moscow Institute for High Temperature:

$$\text{Nu} = (\text{Re Pr}/X)(f/8)(\mu_b/\mu_w)^n \tag{32}$$

where

$$X = 1.07 + 12.7(\text{Pr}^{2/3} - 1)(f/8)^{1/2}$$

and

$$n = \begin{cases} 0.11 & \text{heating} \ (T_w > T_b) \\ 0.25 & \text{cooling} \ (T_w < T_b) \\ 0 & \text{gases} \end{cases}$$

This correlation is applicable for fully developed turbulent flow in the range

$$10^4 < \text{Re} < 5 \times 10^6$$

$$2 < \text{Pr} < 140 \quad \text{with 5 to 6\% error}$$

$$0.5 < \text{Pr} < 2000 \quad \text{with 10\% error}$$

$$0.08 < \mu_w/\mu_b < 40$$

All properties, except μ_w, are evaluated at the bulk temperature. For smooth tubes and pipes, the friction factor is evaluated by

$$f = (1.82 \log \text{Re} - 1.64)^{-2} \tag{33}$$

9.2 Forced Convection Boiling in Tubes

Correlations for forced convection have been developed for boiling refrigerants in horizontal tubes. All are restricted to test conditions for particular refrigerants, and one should be careful in applying them to conditions outside the test range.

Pierre (1955) introduced the load factor, $K_f = J \Delta x \, h_{fg}/L$, which effectively combines the Boiling and Martinelli numbers. In the load factor expression, J is joules equivalent of heat (778 ft·lb$_f$/Btu [1 J/J]) and Δx is the change in quality that occurred during the evaporation process. Pierre correlated R-12 and R-22 for a wide range of operating conditions with separate correlations for complete and incomplete evaporation. These correlations for the Nusselt number (Nu) with two-phase (tp) flow are as follows:

$$\text{Nu}_{tp} = 0.009(\text{Re}^2 K_f)^{0.5}$$
$$\text{for } 10^9 < \text{Re}^2 K_f < 0.7 \times 10^{12} \tag{34}$$

and exit vapor quality < 90% (incomplete evaporation)

$$\text{Nu}_{tp} = 0.0082(\text{Re}^2 K_f)^{0.5}$$
$$\text{for } 10^9 < \text{Re}^2 K_f < 0.7 \times 10^{12} \tag{35}$$

and up to 11°F (6.1°C) superheat (complete evaporation)

9.3 Forced Convection Condensation in Tubes

Condensers used for refrigeration and air-conditioning systems often involve vapor condensation inside horizontal tubes. Unfortunately, conditions within the tube are complicated and depend strongly on the velocity of the vapor flowing through the tube.

If this velocity is small, the condensate flow is from the upper portion of the tube to the bottom, from whence it flows in a longitudinal direction with the vapor. For low vapor velocities such that

$$\text{Re}_{v,i} = \left(\frac{\rho_v m_{m,v} D}{\mu_v}\right)_i < 35,000$$

where i refers to the tube inlet. An expression of the form

$$\bar{h}_D = 0.555 \left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h'_{fg}}{\mu_l(T_{\text{sat}} - T_s)D}\right]^{1/4} \tag{36}$$

is recommended where, for this case, the modified latent heat is

$$h'_{fg} \equiv h_{fg} + \frac{3}{8}c_{p,l}(T_{\text{sat}} - T_s) \tag{37}$$

At higher vapor velocities, the two-phase flow regime becomes annular, and the following correlation is preferred.

$$hD/k_l = 0.026\,\text{Pr}_l^{1/3}[\text{Re}_l + \text{Re}_v(p_l/p_s)^{1/2}]^{0.8} \tag{38}$$

where

$$\text{Re}_l = (4M_l/\pi D\mu_l) \qquad \text{Re}_v = (4M_v/\pi D\mu_v)$$

Here, M_l and M_v are, respectively, the mass flow rates of liquid and vapor. This expression is valid for $\text{Re} > 20,000$.

9.4 Condensation on Horizontal Tubes

One of the earliest investigations into laminar film condensation on horizontal tubes was carried out by Nusselt (1916). By applying a force and energy balance to the condensate film, Nusselt arrived at the following equation for condensation from a single horizontal tube:

$$h = 0.729[g\rho_l(\rho_l - \rho_v)k_l^3 h_{fg}/D_o\mu_l(T_{\text{sat}} - T_w)]^{1/4} \tag{39}$$

For a vertical tier of N horizontal tubes, the average convection coefficient (over the N tubes) may be expressed as

$$h = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h_{fg}}{N\mu_l(T_{\text{sat}} - T_s)D}\right]$$

That is, $h_N = hN^{-1/4}$, where h is the heat transfer for the first (upper) tube. Such an arrangement is often used in condenser design. The reduction in h with increasing N is due to an increase in the film thickness for each successive tube.

9.5 Boiling from Horizontal Tubes

The correlations for heat transfer under fully developed nucleate boiling conditions have been divided into two main groups: those based upon direct curve fitting of experimental data banks, called strictly empirical, and those based upon a physical model, but ultimately curve fitted by experimental results, called semi-empirical. The procedure is based on a straightforward reasoning, according to which, nucleate boiling heat transfer correlations, even those of the second group, can be reduced to a product of powers of the transport properties. These properties can be written in terms of reduced primary thermodynamic properties, such as pressure and temperature p_r and T_r, as in the Law of Corresponding States. Thus, in principle, all the heat

transfer correlations could be reduced to a product of powers of p_r and T_r, presenting a single and common form depending on numerical coefficient and exponents that can be obtained by fitting experimental data.

This correlation uses molecular weight, reduced pressure and surface roughness as the correlation parameters and can be written as

$$\left(h = 95 \cdot q^{0.67} M^{-0.5} \left(\frac{p}{p_c}\right)^{0.12 - 0.21 \log_{10} R_p} \right) \times \left(-\log_{10}\frac{p}{p_c}\right)^{-0.55} \tag{40}$$

where M is the molecular weight and R_p is the roughness of the surface.

It is generally assumed that commercial-finish copper tubes have a surface roughness of 0.4 μm.

Rohsenow's Correlation. The first and still most widely used correlation for heat transfer in nucleate pool boiling was proposed by Rohsenow (Rohsenow and Hartnett 1973) using experimental data on pool boiling from many different fluids as a guide, Rohsenow obtained

$$h = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} \left(\frac{c_{p,l}}{C_{sf} h_{fg} \mathrm{Pr}_l^n}\right)^3 \Delta T_x^2 \tag{41}$$

where

μ_l	=	viscosity of the liquid, lb/ft^2·s (kg/m·s)
h_{fg}	=	enthalpy of vaporization, Btu/lb (J/kg)
g	=	gravitational acceleration, ft^2/s (m^2/s)
ρ_l	=	density of liquid, lb/ft^3 (kg/m^3)
ρ_v	=	density of vapor, lb/ft^3 (kg/m^3)
σ	=	surface tension of liquid-vapor interface, lb$_f$/ft (N/m)
c_{pl}	=	specific heat of the liquid, Btu/lb·°F (J/kg·°C)
ΔT_x	=	$\Delta T_x = T_s - T_{sat}$
T_s	=	surface temperature, °F (°C)
T_{sat}	=	saturation temperature of the fluid, °F (°C)
C_{sf}	=	experimental constant that depends on surface-fluid combination
Pr_l	=	Prandtl number of the liquid
n	=	experimental constant that depends on the fluid = 1.0 for water, 1.7 for other fluids

The Rohsenow method correlates data for all types of nucleate-boiling processes, including pool boiling of saturated or subcooled liquids. Unfortunately, Rohsenow's Correlation can be used **only if** the C_{sf} value is known and the common values are given in Table 8. Fortunately, values for several common refrigerants have recently been reported, as shown in Table 9.

9.6 Airflow Across Finned Tubes

The outside heat transfer coefficient is a very crucial parameter which has to be estimated accurately in the design and performance simulation of finned-tube heat exchangers. It is often the controlling factor in the estimation of the overall heat transfer coefficient for the exchanger when air is the external heat transfer fluid. Most correlations have been obtained experimentally and are (a) valid only for specific surfaces and (b) proprietary for most configured fin surfaces.

Table 10 provides dimensional data for two plate fin-and-tube arrangements consisting of aluminum fins bonded to copper tubes (Schedule 18). Figure 10 gives a schematic of each surface and presents the corresponding heat transfer correlation for the external surface. The friction factor for the outside surface is also shown. The principal dimensionless groups governing these correlations are the Stanton, Prandtl, and Reynolds numbers:

$$\mathrm{St} = h/Gc_p \qquad \mathrm{Pr} = c_p \mu/k \qquad \mathrm{Re} = GD_h/\mu$$

where G is the mass velocity defined as

$$G = m/A_{min}$$

where m is the total mass flow rate of the fluid and A_{min} is the minimum free-flow cross-sectional area in the coil. The hydraulic diameter ($D_h = 4r_h$) is specified on each figure.

Table 8 Values of C_{sf} for Various Fluid-Surface Combinations
(Incropera 2007)

Fluid-Surface Combination	C_{sf}	n
Water-copper		
Scored	0.0068	1.0
Polished	0.0130	1.0
Water-stainless steel		
Chemically etched	0.0130	1.0
Mechanically polished	0.0130	1.0
Ground and polished	0.0060	1.0
Water-brass	0.0060	1.0
Water-nickel	0.006	1.0
Water-platinum	0.0130	1.0
n-Pentane-copper		
Polished	0.0154	1.7
Lapped	0.0049	1.7
Benzene-chromium	0.0101	1.7
Ethyl alcohol-chromium	0.0027	1.7

Table 9 C_{sf}s for Refrigerants

Refrigerant	Value of C_{sf}
R-12	0.008339
R-22	0.007947
R-123	0.006706
R-134a	0.006232
R-407C	0.007269
R-410A	0.008294

Table 10 Dimensional Data for Two Finned-Tube Surfaces

(a) Surface 8.0-3/8T	(b) Surface 7.75-5/8T
Tube outside diameter = 0.402 in.	Tube outside diameter = 0.676 in.
Fin pitch = 8 per in.	Fin pitch = 7.75 per in.
Fin thickness = 0.013 in.	Fin thickness = 0.016 in.
Hydraulic diameter = 0.001192 ft	Hydraulic diameter = 0.0114 ft
Free-flow area/face area = 0.534	Free-flow area/face area = 0.481
Fin area/total external area = 0.913	Fin area/total external area = 0.913
Total external area/inside tube area = 12.5	Total external area/inside tube area = 20.5
Total external area/outside tube area = 10.3	Total external area/outside tube area = 17.6
Total outside area/face area = 12.9	Total outside area/face area = 24.6
Heat transfer area/total volume = 179 ft^2/ft^3	Heat transfer area/total volume = 169 ft^2/ft^3

(A) SURFACE 8.0-3/8*T*   (B) SURFACE 7.75-5/8*t*
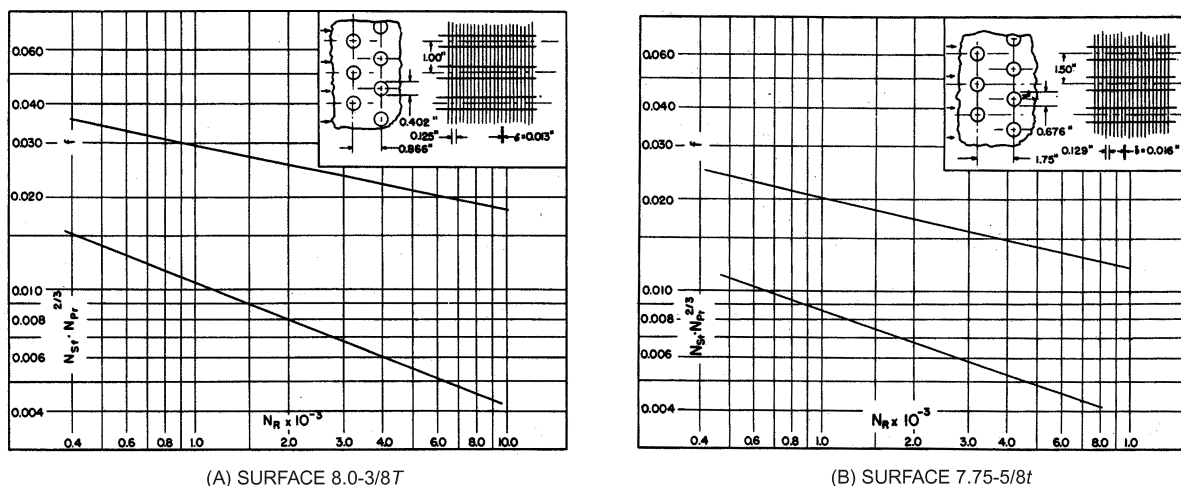
Fig. 10   Dimensional Data for Two Finned-Tube Surfaces

10 Calculation of Heat Exchanger Surface Area and Overall Size

Design methods for shell-and-tube and finned-tube heat exchangers are outlined in Tables 11 and 12, respectively.

11 Fluids and Their Thermophysical Properties

Proper evaluation of the necessary thermodynamic and thermophysical properties of the working fluids in heat exchangers is most important. Properties need to be evaluated at the correct temperature (and pressure, if gas or vapor) which may be the average bulk fluid temperature (average between in and out), the saturation temperature, the "film" temperature (average between fluid and wall), and/or the surface temperature. Unfortunately, there is no single "correct" temperature to use but varies primarily with the correlation selected for predicting the convective heat transfer coefficient.

Chapters 29, 30, and 33 of the 2021 *ASHRAE Handbook—Fundamentals* are excellent sources of fluid properties data. In addition, most heat transfer textbooks include some tables, and there are rather extensive reference handbooks either on heat transfer or thermophysical properties. And, of course, the Web is another valuable source of such data.

12 Example Finned-Tube Heat Exchanger Design

Task: An aluminum tube with $k = 1290$ Btu·in./h·ft^2·°F, ID = 1.8 in., and OD = 2 in. has circular aluminum fins $\delta = 0.04$ in. thick with an outer diameter of $D_{fin} = 3.9$ in. There are $N' = 76$ fins per foot of tube length. Steam condenses inside the tube at $t_i = 392$°F with a large heat transfer coefficient on the inner tube surface. Air at $t_\infty = 77$°F is heated by the steam. The heat transfer coefficient outside the tube is 7 Btu/h·ft^2·°F. Find the rate of heat transfer per foot of tube length.

Solution: From Figure 8 efficiency curve, the efficiency of these circular fins is

$$L = (D_{fin} - \text{OD})/2 = (3.9 - 2)/2 = 0.95 \text{ in.}$$

$$f_2/f_1 = \frac{3.9/2}{2/2} = 1.95 \text{ in.}$$

$$L\sqrt{\frac{h}{ky}} = 0.95 \text{ in.} \sqrt{\frac{7 \text{ Btu/h·ft}^2\cdot°F}{(1290 \text{ Btu·in/h·ft}^2\cdot°F)(0.02 \text{ in.})}}$$

$$\phi = 0.89$$

Table 11   Design Procedure for Shell-and-Tube Heat Exchanger

1. FIND Q OF THE CONDENSER
2. CHOOSE TUBE SIZE
3. CALCULATE MEAN TEMPERATURE
4. CALCULATE REYNOLDS NUMBER
5. USING   Q=M*CP*DELT CALCULATE MASS FLOW RATE FOR WATER
6. USING M=RHO*AREA*VELOCITY*N TUBES CALCULATE NUMBER OF TUBES
7. CALCULATE LOG MEAN TEMPERATURE DIFFERENCE USING COUNTERFLOW EXAMPLE
8. USING   Q=U*AREA*TEMP LOG MEAN CALCULATE VALUE OF   U*L
9. CALCULATE CONVECTIVE COEFFICIENT (H) FOR INSIDE TUBES
10. CALCULATE CONVECTIVE COEFFICIENT (H) FOR OUTSIDE TUBES
11. ASSUME A SURFACE TEMPERATURE FOR OUTER SURFACE OF TUBE, FIND PROPERTIES AT THE MEAN TEMPERATURE
12. LOOK UP FOULING FACTORS FOR WATER AND REFRIGERANT
13. ASSUME COPPER TUBING FOR CONDUCTIVITY (K)
14. CALCULATE NUSSELT NUMBER AND OVERALL RESISTANCE (U)
15. VERIFY ASSUMED TEMPERATURE
16. REPEAT PROCESS 10-15 IF NECESSARY
17. WHEN FINAL H AND U ARE FOUND THEN L MAY BE FOUND FROM EQUATION IN LINE 8

Table 12   Design Procedure for Finned-Tube Heat Exchanger

1. DO CALCULATIONS WITH BOTH SURFACES
2. ASSUME TUBE OF GAGE 16 FOR BOTH GIVING DIFFERENCE OF .13 INCHES BETWEEN Do AND Di
3. USING M=RHO*AREA*VELOCITY*N TUBES CALCULATE NUMBER OF TUBES
4. USING   HEIGHT=N(TUBES)*SPACING   CALCULATE HEIGHT
5. USING M=RHO*AREA*VELOCITY OF FACE   CALCULATE AREA OF FACE
6. USING A(FACE)=HEIGHT*WIDTH
7. CALCULATE REYNOLDS NUMBER INSIDE TUBES
8. USING GIVEN VALUE FOR DENSITY OF AIR USE TM=70 F
9. CALCULATE NUSSELT NUMBER, FIND H
10. CALCULATE REYNOLDS NUMBER FOR AIR USING HYDRAULIC DIAMETER
11. USING FIGURE FOR SURFACE FIND   StPr2/3, FIND H FROM WITHIN St NUMBER
12. FIND CONTACT RESISTANCE ASSUMING Al FINS AND Cu TUBES
13. ASSUME FINS ARE ANNULAR (CIRCULAR) AND HAVE A RADIUS EQUAL TO THE SPACING BETWEEN THE TUBES
14. CALCULATE ALPHA AND BETA FOR ANNULAR FIN SHAPE
15. CALCULATE K_{an} FOR ANNULAR FINS, FIND ETA
16. CALCULATE U VALUE

The fin area for $L = 1$ ft is

$$A_s \ = \ N'L \times 2\pi(D_{fin}^2 - \mathrm{OD}^2)/4 \ = \ 1338 \ \mathrm{in}^2 \ = \ 9.29 \ \mathrm{ft}^2$$

The unfinned area for $L = 1$ ft is

$$A_p \ = \ \pi \times \mathrm{OD} \times L(1 - N'\delta)$$

$$= \ \pi(2/12) \ \mathrm{ft} \times 1 \ \mathrm{ft}(1 - 76 \times 0.04/12)$$

$$= 0.39 \ \mathrm{ft}^2$$

and the total area $A = A_s + A_p = 9.68 \ \mathrm{ft}^2$. Surface efficiency is

$$\phi_s \ = \ \frac{\phi A_f + A_s}{A} \ = \ 0.894$$

and resistance of the finned surface is

$$R_s = \frac{1}{\phi_s hA} = 0.0165 \text{ h·°F/Btu}$$

Tube wall resistance is

$$R_{\text{wall}} = \frac{\ln(\text{OD}/\text{ID})}{2\pi L k_{\text{tube}}} = \frac{\ln(2/1.8)}{2\pi(1 \text{ ft})(1290/12) \text{ Btu·in/h·ft·°F}}$$

$$= 1.56 \times 10^{-4} \text{ h·°F/Btu}$$

The rate of heat transfer is then

$$q = \frac{t_i - t_\infty}{R_s + R_{\text{wall}}} = 18{,}912 \text{ Btu/h}$$

13 References

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14 Problems

Problem 1 A hot-water coil is to be sized (designed) to heat 5000 cfm of air from 70°F to 130°F with the coil face velocity based on duct space selected as 500 fpm. The inlet and outlet water temperatures are 190°F and 170°F, respectively. Determine: coil height, width, number of rows, depth.

Problem 2 A supplementary cooling coil is being added to the building's HVAC system to provide additional cooling for a computer room. The cooling coil will do sensible cooling only, taking 1500 scfm of pre-conditioned air from 70°F DB and 60°F WB to 55°F DB. A 50/50 antifreeze (ethylene glycol) solution at 50°F will be supplied to the coil. Prepare a preliminary design for the coil using Surface 8.0-3/8T with copper tubes having a final wall thickness of 0.036 in.

Problem 3 Design a steam coil to heat 8500 scfm of outdoor air from 0°F to 45°F with a face velocity of 600 fpm. Low-pressure saturated steam at 5 psig is used. Surface 7.75-5/8T is to examined first.

Problem 4 Design both the heating coil and the shell-and-tube water heat exchanger for the heating system shown in the following sketch.
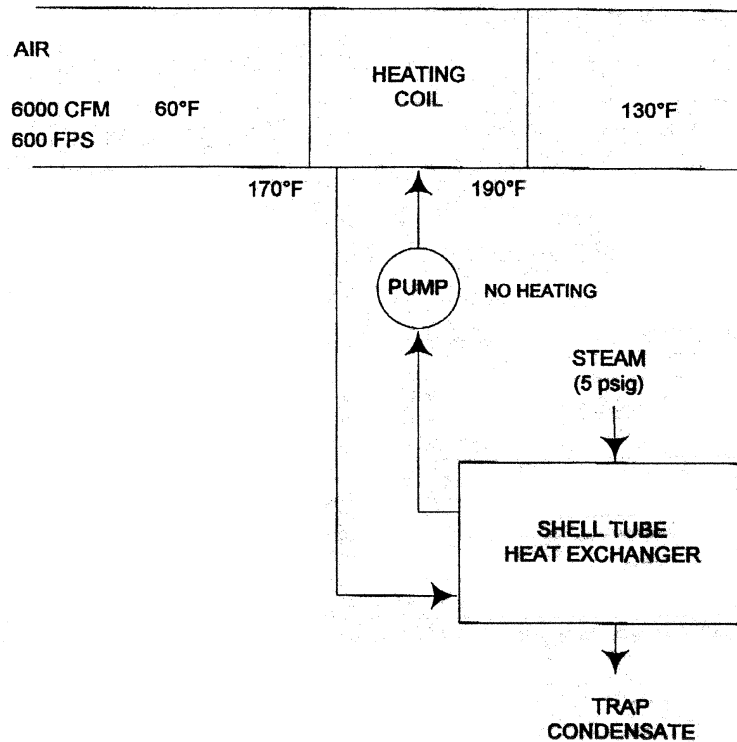
Problem 5 Design the evaporator/condenser for a cascade low-temperature refrigeration system using R-410 in the high-temperature loop and R-22 in the low-temperature loop. The shell-and-tube heat exchanger will use standard size copper tubes with a steel pipe as a shell. R-22 at the rate of 0.130 kg/s is to be condensed from saturated vapor to saturated liquid at a pressure of 0.91 MPa as it flows through the tubes. R-410A surrounds the tubes and evaporates under pool boiling conditions at a pressure of 1.1 MPa. The exterior of the heat exchanger shell is to be well insulated. Space limits the length of the exchanger to 2 m.

Problem 6 A shell-and-tube heat exchanger (as shown below) is to cool 1 L/s of water from 15°C to 5°C using R-22 evaporating at 50 kPa on the outside of the tubes. Tubes are to be of copper with a 1.41 cm ID and 1.59 cm OD. Maximum water velocity in the tubes is to be 2 m/s. Design the heat exchanger including specification of its duty (thermal rating) in kW, the design U-factor, the number of tubes per pass, and the length of the exchanger. Boiling performance may be obtained from the ASHRAE data provided in the following graphs.