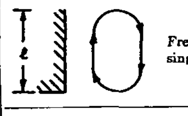
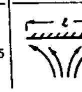
Table 2 .... Approximate Unit Conductances for Thermal Convection for Several Flow Systems (Concluded)

Case	System		Heat Transfer Equation[a] and Its Limits of Application	
			Free Convection[c]	
11 Ref. 5		Free convection past a heated horizontal cylinder.	*Equation for Air* $h_c = 0.271\left(\frac{P}{P_a}\right)^{0.5}\left(\frac{\Delta t}{D}\right)^{0.25}$	*where* $10^3 < N_{Gr} < 10^9$
12 Ref. 6		Free convection past a single vertical surface.	*Equation for Air* $h_c = 0.354\left(\frac{P}{P_a}\right)^{0.50}\left(\frac{\Delta t}{l}\right)^{0.25}$	*where* $10^3 < N_{Gr} < 10^7$
			$h_c = 0.420\left(\frac{P}{P_a}\right)^{0.50}\left(\frac{\Delta t}{l}\right)^{0.25}$	*where* $10^8 < N_{Gr} < 10^{10}$
13 Ref. 5		Free convection past a heated horizontal surface (face up).	*Equation for Air* $h_c = 0.478\left(\frac{P}{P_a}\right)^{0.50}\left(\frac{\Delta t}{l}\right)^{0.25}$	*where* $10^3 < N_{Gr} < 10^9$
14 Ref. 5		Free convection past a heated horizontal surface (face down).	*Equation for Air* $h_c = 0.239\left(\frac{P}{P_a}\right)^{0.50}\left(\frac{\Delta t}{l}\right)^{0.25}$	*where* $10^3 < N_{Gr} < 10^7$

Nomenclature and Dimensions for Table 2

c_p = heat capacity at constant pressure, Btu per (pound) (Fahrenheit degree).
D = cylinder diameter, feet.
f = subscript denoting film.
g = body force per unit mass, feet per hour per hour. (For static system on earth, $g = 32.2 \times 3600^2$ feet per hour per hour.)
G = 3600 $u_m\rho$ = mass flow per unit cross-sectional area normal to flow, pounds per (hour) (square foot of flow cross-section).
N_{Gr} = Grashof modulus, dimensionless ($N_{Gr} = D^3\rho^2\beta\Delta t g/\mu^2$).
h_c = average unit thermal convective conductance from the leading edge of surface to the position x, Btu per (hour) (square foot) (Fahrenheit degree).
h_{cx} = local unit thermal convective conductance, at the position x from the leading edge of surface, Btu per (hour) (square foot) (Fahrenheit degree).
k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot thickness).
l = a dimension of the system, feet.
m = a subscript denoting *mean*.
P = pressure, atmospheres.
P_a = pressure (atmospheric), atmospheres.
t = temperature, Fahrenheit.
T = temperature, Fahrenheit, absolute.
u = fluid velocity, feet per second.
V = volume, cubic feet.
x = a dimension of the system, feet.
β = coefficient of cubical expansion ($\beta = \frac{1}{V}\left(\frac{dV}{dT}\right)_p$); for perfect gases $\beta = 1/T$.
Δt = difference between wall and fluid temperatures, Fahrenheit degrees.
μ = fluid viscosity, pounds per (hour) (foot).
ρ = density, pounds per cubic foot.
∞ = infinity, referring the quantity to a point not directly affected by the phenomenon in question.

[a] Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{surface} + t_{fluid})$ divided by 2.
[b] These expressions are suitable approximations to longitudinal flow in other than right circular cylinders, provided the hydraulic diameter is employed as the conduit dimension parameter. For non-circular cross-sections, the hydraulic diameter is equal to four times the cross-sectional area divided by the wetted perimeter.
[c] For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates, increases towards 0.33. The above equations employing an exponent equal to 0.25 are applicable in the intermediate range indicated.

F_A = the geometrical factor which is dimensionless and ≤ 1. This factor accounts for the shape and relative position of the two surfaces. The value of $F_A = 1$ may be used in the cases of large parallel planes, long concentric cylinders or smaller bodies in large enclosures.
F_E = the emissivity factor which is also dimensionless and ≤ 1. This factor accounts for the absorption and emission characteristics of the surfaces for the radiation which exists. Emissivities or absorptivities (ϵ) for many common surfaces, are given in Table 3. The value of F_E for large parallel planes, long concentric cylinders, or large enclosed bodies is $1 \div (1/\epsilon_1 + 1/\epsilon_2 - 1)$.

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 4[7] for cold surfaces as low as −39 F to warmer surfaces as high as 139 F. Some net radiation exchange solutions for several common radiation systems are given in Table 5.

There are several methods by which the geometrical

factor F_A can be determined. One method involves the use of a mechanical geometrical integrator (Reference 8). Photographic and other methods are given in References 9 and 10.

Equivalent Conductance for Radiation

Although Equation 5 is a suitable equation for describing radiant exchange, it is not convenient for computations where other modes of energy transfer are operative. For such cases, it is convenient to define an *equivalent conductance for radiation* by the equation:

$$q_r = h_r A (t_1 - t_2) \tag{6}$$

The conductance h_r thus defined is a function of the shape-emissivity factor, as well as the temperatures of the radiator and receiver. Fig. 7 shows a plot of the equivalent conductance for two *black bodies* (i.e., with emissivities equal to unity) which exchange energy only with one another.

Combined Convection and Radiation

It should be noted that the previous equations and tables give the heat transfer by convection and by radiation computed separately. In many practical cases it is desirable to treat *convection and radiation as a single combined process*, using a first-power equation:

$$q_{rc} = h_{rc} A (t_1 - t_2) \tag{7}$$

where q_{rc} is the total heat flow due to radiation and convection, in Btu per hour. Values of h_{rc}, the surface or film conductance for combined radiation and convection, are given in Table 3 and Fig. 4 of Chapter 9. Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

HEAT-FLOW RESISTANCE

In most of the steady-state heat transfer problems encountered in air conditioning applications, more than one of the heat transfer mechanisms are effective, and the thermal current flows through several resistances in series or in parallel. In using the resistance concept, the calculations involved are analogous to the application of Ohm's Law in electricity, viz., the heat flow or thermal current is directly proportional to the thermal potential or temperature difference, and inversely proportional to the thermal resistance:

$$q_{rc} = \frac{t_1 - t_2}{R} \tag{8}$$

Following the electrical analogy, when there is a thermal current flowing through several *resistances in series*, the resistances are additive:

$$R_T = R_1 + R_2 + R_3 + \cdots + R_n \tag{9}$$

Similarly, conductance is the reciprocal of resistance, and for heat flow through several *resistances in parallel*, the conductances are additive:

$$C_T = \frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \cdots + \frac{1}{R_n} \tag{10}$$

Practical Heat Transfer Problems

The use of these relations for resistance and conductance makes possible the solution of many practical heat transfer problems. As discussed in Chapters 9, 23, and 32, the practical analyses of heat transfer in building walls, in fin-tube coils, and in pipe coverings, are usually computed by this method. The same resistance analysis may be applied to complicated steady-state *conduction* problems. Table 6 gives the resistances in six common cases of steady-state conduction.

A complete analysis by the resistance method is well illustrated by considering the heat transfer from the air outside to the cold water inside of an insulated pipe. The temperature gradients and the nature of the resistance analysis are indicated by the two sketches of Fig. 8.

Since air is sensibly transparent to radiation, there will be some heat transfer by both radiation and convection to the outer insulation surface. The mechanisms act in parallel on the air side. The total transfer by radiation and convection then passes through the insulating layer and the pipe wall by thermal conduction, and thence by convection and radiation into main cold water streams. (Radiation is not significant on the water side as liquids are sensibly opaque to radiation, although water transmits energy in the visible region.) The contact resistance between the insulation and the pipe wall is presumed to be equal to zero.

Referring to Fig. 8, the heat transferred for a given length N of pipe, q_{rc}, Btu per hour, may be thought of as flowing through the parallel resistances R_r and R_c, associated with the insulation surface radiation and convection transfer. Then the flow is through the resistance offered to thermal conduction by the insulation, R_3, through the pipe wall resistance, R_2, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_t, (hour) (Fahrenheit degrees) per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \tag{11}$$

where the resultant parallel resistance R_4 is obtained from

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c} \tag{12}$$

Provided the individual resistances may be evaluated, the total resistance can be obtained from this relation. Then the heat transfer for the length of pipe (N, ft) can be established by the relation

$$q_{rc}\ (\text{Btu per hour}) = \frac{(t_a - t_f)}{R_T} \tag{13}$$

For a unit length of the pipe the heat transfer rate is

$$\frac{q_{rc}}{N}\ \text{Btu per (hour) (foot)} = \frac{(t_a - t_f)}{R_T N} \tag{14}$$

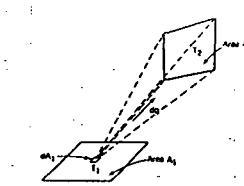


Fig. 3 .... Radiation Between Surfaces