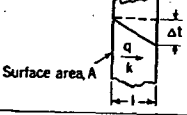
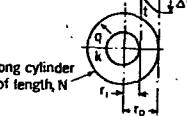
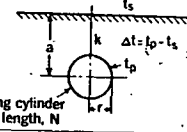
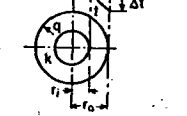
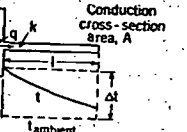
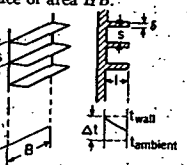


TABLE 8. SOLUTIONS FOR SOME STEADY-STATE THERMAL CONDUCTION PROBLEMS^{ab}

No.	SYSTEM	Expressions for the resistance R entering into the equation: $q = \Delta t/R$ (Btu per hour)
1.	Flat wall or curved wall if curvature is small (wall thickness less than 0.1 of inside diameter). 	$R = \frac{l}{kA}$
2.	Radial flow through a right circular cylinder. 	$R = \frac{\log_e \frac{r_2}{r_1}}{2\pi k N}$ (See footnote c).
3.	The buried cylinder. 	$R \approx \frac{\log_e \frac{2a}{r}}{2\pi k N}$; $R = \frac{\cosh^{-1} \frac{a}{r}}{2\pi k N}$ for $\frac{a}{r} \geq 3$ (See footnote c).
4.	Radial flow in a hollow sphere. 	$R = \frac{\frac{1}{r_1} - \frac{1}{r_2}}{4\pi k}$
5.	The straight fin or rod heated at one end. 	$R = \frac{m}{h_0 p \tanh ml}$ (see footnotes d and e). For $ml > 2.3$, $\tanh ml \approx 1$ $m = \sqrt{h_0 p / kA}$ A = conduction cross-section area. p = perimeter of cross-section A . h_0 = unit conductance to the surroundings from the fin surface. k = thermal conductivity fin material. Δt = wall temperature—ambient temperature
6.	Finned surface of area HB . 	$R = \frac{(s + \delta)}{h_0 \left(\frac{2}{m} \tanh ml + s \right) HB}$ $m = \sqrt{\frac{h_0 p}{kA}} = \sqrt{\frac{2 h_0}{k\delta}}$ Δt defined as in Case 5 above.

^aThe dimensions to be employed in these solutions are: length of dimension p, l, r = feet; units of k = Btu per hour per square foot per degree Fahrenheit for one foot thickness; units of h, B = Btu per hour per square foot per degree Fahrenheit; units of area, A = square feet.
^bThe thermal conductivity, k , in these solutions should be taken at the average material temperature (see Table 5).
^c $\log_e x = 2.303 \log_{10} x$.
^dThis expression can also be employed as an approximation for tapered fins or of angular fins by employing average magnitudes of A and p .
^e $\tanh$ is the hyperbolic tangent.

where the resultant parallel resistance R_4 is obtained from:

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

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

$$q_{rc} \text{ (Btu per hour)} = \frac{t_o - t_i}{R_T} \quad (10)$$

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

$$\frac{q_{rc}}{N} \text{ (Btu per hour foot)} = \frac{(t_o - t_i)}{R_T N} \quad (11)$$

The temperature drop, Δt , through an individual resistance may then be calculated from the relation:

$$\Delta t = R q_{rc}$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable integration of the rate Equations 1, 2 and 3 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \quad (12)$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R . Table 8 lists such solutions for six different conduction systems. Table 2 in Chapter 4 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k , to be employed in the expressions of Table 8.

The solution applicable to the problem depicted in Fig. 4, for the calculation of R_2 and R_3 , is case 2 in Table 8. Thus for a 1 ft length of 2 in. nominal size pipe (I. D. = 2.067 in., O. D. = 2.375 in.) insulated with 1 in. of cork:

$$R_2 = \frac{\log_e \frac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ hr degree Fahrenheit per Btu.}$$

$$R_3 = \frac{\log_e \frac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ hr degree Fahrenheit per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1 , and from the air to the surface of the insulating material, R_c , are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. The unit conductances for thermal convection, h , Btu per hour per square foot per degree Fahrenheit, have been determined by test for many flow systems. These data may be employed to predict the conductances for similar flow systems. Table 5 summarizes some empirical equations expressing such test results: