Table 3 . . . . Radiation Factors or Emissivities, ϵ*

For the determination of factor F_E in Equation 3

Class	Surfaces	Fraction of Black-Body Radiation At 50–100 F	Fraction of Black-Body Radiation At 1000 F	Absorptivity for Solar Radiation
1	A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99	0.97 to 0.99	0.97 to 0.99
2	Black non-metallic surfaces such as asphalt, carbon, slate, paint, paper	0.90 to 0.98	0.90 to 0.98	0.85 to 0.98
3	Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.85 to 0.95	0.75 to 0.90	0.65 to 0.80
4	Yellow and buff brick and stone, firebrick, fire clay	0.85 to 0.95	0.70 to 0.85	0.50 to 0.70
5	White or light-cream brick, tile, paint or paper, plaster, whitewash	0.85 to 0.95	0.60 to 0.75	0.30 to 0.50
6	Window glass	0.90 to 0.95	....	Transparent[a]
7	Bright aluminum paint; gilt or bronze paint	0.40 to 0.60	....	0.30 to 0.50
8	Dull brass, copper, or aluminum; galvanized steel; polished iron	0.20 to 0.30	0.30 to 0.50	0.40 to 0.65
9	Polished brass, copper, monel metal	0.02 to 0.05	0.05 to 0.15	0.30 to 0.50
10	Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0.04	0.05 to 0.10	0.10 to 0.40

* Emissivities of other materials may be found in Reference 4. [a] Reflects about 8 percent.

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

$$\Delta t = R q_{rs} \tag{15}$$

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 manipulation of the rate Equations 1, 2, and 5 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \tag{16}$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 6 lists such solutions for six different conduction systems. Table 4 in Chapter 9 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 6, after dividing k by 12.

The solution applicable to the problem depicted in Fig. 8, for the calculation of R_2 and R_3, is Case 2 in Table 6. 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 material having a conductivity of 0.025:

$$R_2 = \frac{\log_e \dfrac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ (hr) (F deg) per Btu.}$$

$$R_3 = \frac{\log_e \dfrac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ (hr) (F deg) per Btu.}$$

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_4, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. These resistances are also directly dependent upon the temperature distribution, and for this reason it is necessary first to guess, on the basis of the problem statement, a temperature distribution upon which to base the initial calculations. Since the values of h_c for heat transfer between water and pipe walls are relatively high in this temperature range, it is logical to assume only a small temperature difference between the temperature of the fluid body and the temperature of the pipe wall. For the purpose of an initial guess, this temperature difference will be assumed to be 2 deg. On the other hand, h_c for heat transfer from air to a body is relatively small, and a higher temperature difference would be expected between these masses. The value initially assumed here will be 20 deg. In summary the temperature distribution in the system is assumed as follows:

Fluid temperature = 34 F.
Inner pipe wall temperature = 36 F.
Outer insulation surface temperature = 100 F.
Ambient air temperature = 120 F.

With these assumptions and the problem statement, it is now possible to calculate values for the convective resistances. If it is found in the ultimate solution of the problem that the temperature distribution is different from that assumed, it will then be necessary to repeat the solution procedure.

If reference now be made to Table 2, it is found that Case 3 of this table is a system similar to that encountered in the

Table 4 . . . . Heat Transmission by Radiation for Black-Body Conditions[a]

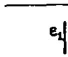
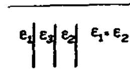
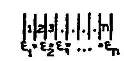
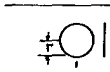
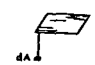
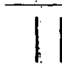
Expressed in Btu per (square foot) (hour)

Temp F Deg	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60.5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67.7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1

	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	87.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

[a] *Example:* Radiation from walls of room at 32 F to surface at −25 F for effective emissivity of 0.95 = (102 − 62.3) 0.95 = 37.7 Btu per (square foot) (hour).

Table 5 . . . Net Radiation Solutions

System		Solution	Remarks
	Two infinite parallel planes.	$\dfrac{q_r}{A} = \dfrac{1}{\dfrac{1}{\epsilon_1} + \dfrac{1}{\epsilon_2} - 1}\sigma(T_1^4 - T_2^4)$	Considering interreflections. (Reference 5)
	One radiation shield between two infinite parallel planes.	$\dfrac{q_r}{A} = \left(\dfrac{1}{2}\right)\dfrac{1}{\dfrac{1}{\epsilon_1} + \dfrac{1}{\epsilon_2} - 1}\sigma(T_1^4 - T_2^4)$	Considering interreflections. (Reference 5)
	n radiation shields between two infinite parallel planes.	$\dfrac{q_r}{A} = \dfrac{1}{n+1}\left(\dfrac{q_r}{A}\right)_0$ where $\left(\dfrac{q_r}{A}\right)_0$ is the net radiation exchange without the shields.	Considering interreflections. (Reference 5)
	Two concentric spheres or two infinitely long cylinders.	$\dfrac{q_r}{A_1} = \dfrac{1}{\dfrac{1}{\epsilon_1} + \dfrac{A_1}{A_2}\left(\dfrac{1}{\epsilon_2} - 1\right)}\sigma(T_1^4 - T_2^4)$	Considering interreflections and diffuse surfaces. (Reference 5)
	Two areas dA_1 and dA_2	$\dfrac{dq_r}{dA_1} = \epsilon_1\epsilon_2\sigma F_A(T_1^4 - T_2^4)$	Surface diffuse, neglecting interreflection. (Reference 5)
	Tube of infinite length parallel to an infinite wall.	$\dfrac{q_r}{2\pi r N} = \left(\dfrac{1}{2}\right)\epsilon_1\epsilon_2\sigma(T_1^4 - T_2^4)$ where N is the length of cylinder from which q_r is exchanged.	Neglecting interreflections. (Reference 5)
	Surfaces are perfect radiators. Surface element dA and rectangle above and parallel to it, with one corner of rectangle contained in normal to dA.	See Fig. 4	(Reference 4)
	Surfaces are perfect radiators. Adjacent rectangles in perpendicular planes.	See Fig. 5	(Reference 4)
	Surfaces are perfect radiators. Opposed parallel rectangles and discs of equal size.	See Fig. 6	(Reference 4)