

which radiant exchange takes place between the surfaces of solids, as schematically shown in Fig. 3. Gaseous and luminous radiation are not considered in this discussion. Equation 3 states that the net radiation per unit transfer area of surface 1, q_r/A Btu per (hour) (square foot), which sees surface 2 through a non-absorbing medium, is proportional to the difference of the fourth powers of the absolute surface temperatures ($T_1^4 - T_2^4$). The proportionality factor ($\sigma F_A F_E$) may be conveniently separated into three parts:

$$q_r = \sigma A_1 F_A F_E (T_1^4 - T_2^4) \quad (3)$$

σ = the Stefan-Boltzmann radiation constant = 1730×10^{-18} Btu per (hour) (square foot) (Fahrenheit degree absolute temperature to the fourth power).
 F_A = the configuration 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 ac-

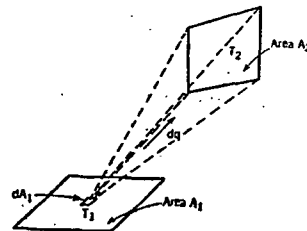


FIG. 3. RADIATION BETWEEN SURFACES

counts for the absorption and emission characteristics of the surfaces for the radiation which exists. Individual emissivities (ϵ) should be taken from Table 3 and applied, for either radiation or absorption, as follows:

- For a small body in a large enclosure, use the emissivity of the small body only: $F_E = \epsilon_1$.
- For large parallel planes, long concentric cylinders or large enclosed bodies, use both emissivities in the equation:

$$F_E = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$$

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 4³ for cold surfaces as low as -39°F to warmer surfaces as high as 139°F . The emissivities of a number of surfaces ordinarily encountered in engineering practice are shown in Table 3. For radiation at higher temperatures and various emissivities, see Table 5.

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:

$$q_{rc} = h_{rc} A (t_1 - t_2) \quad (4)$$

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 Chapter 9, (Table 1 and Fig. 3). Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

When dealing with the effect of operating temperatures upon the com-

TABLE 3. RADIATION FACTORS OR EMISSIVITIES, ϵ .^{*}
For the determination of factor F_E in Equation 3

CLASS	SURFACES	FRACTION OF BLACK-BODY RADIATION		ABSORPTIVITY FOR SOLAR RADIATION
		At 50-100 F	At 1000 F	
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, white-wash	0.85 to 0.95	0.60 to 0.75	0.3 to 0.5
6	Window glass	0.90 to 0.95	-----	Transparent ^a
7	Bright aluminum paint; gilt or bronze paint	0.4 to 0.6	-----	0.3 to 0.5
8	Dull brass, copper, or aluminum; galvanized steel; polished iron	0.2 to 0.3	0.3 to 0.5	0.4 to 0.65
9	Polished brass, copper, monel metal	0.02 to 0.05	0.05 to 0.15	0.3 to 0.5
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. ^aReflects about 8 per cent.

combined heat transfer of a given piece of equipment (as for instance a steam radiator), another form of equation is frequently used:

$$q_{rc} = B A (t_1 - t_2)^n \quad (5)$$

Values of n in this equation usually range from 1.3 to 1.5 (see Chapter 22). The chief advantage of this equation is the convenience of representing heat transfer performance on logarithmic coordinates, and the factor B should be regarded as a simple constant of proportionality.

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