TABLE 9. SOME LOW TEMPERATURE RADIATION EMISSIVITIES FOR COMMERCIAL SURFACES[a]

Surface	Emissivity ϵ
Metal Surfaces	
Aluminum:	
Clean plate	0.04–0.06
Strongly oxidized	0.11–0.19
Brass:	
Polished	0.096
Dull finish	0.22
Strongly oxidized	0.6
Copper:	
Polished	0.023
Clean	0.07
Strongly oxidized	0.8
Cast Iron, rough, strongly oxidized	0.95
Sheet Iron	0.87
Steel Plate, rough	0.94—0.97
Wrought Iron, dull oxidized	0.94
Zinc, Galvanized Sheet Iron:	
Fairly bright	0.228
Gray, oxidized	0.276
Painted Surfaces	
Aluminum paint	0.3–0.7
Other pigment paints	0.8–0.95
Miscellaneous	
Asbestos board	0.96
Asbestos paper	0.93—0.945
Carbon, candle soot	0.952
Glass	0.80
Glazed brick	0.75
Lamp black and soot	0.90—0.97
Linoleum	0.85
Marble, light gray polished	0.931
Paper	0.90—0.95
Plaster	0.95
Porcelain, glazed	0.924
Roofing paper	0.91
Rough brick	0.80—0.93
Rough concrete	0.95
Rough mortar	0.91
Smooth concrete	0.90
Stone	0.87
Water	0.95
Wood unpolished	0.89

[a]These emissivities should not be employed for solar radiation calculations. For this type of calculation reference should be made to Heat Transmission, by W. H. McAdams and The Calculation of Heat Transmission, by Margaret Fishenden and Owen A. Saunders.

The resultant resistance of R_c and R_r acting in parallel (see Fig. 4) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.312} + \frac{1}{0.76} = 4.51 \text{ Btu per hour per degree Fahrenheit.}$$

$$R_4 = 0.222 \text{ hr degree Fahrenheit per Btu.}$$

The individual resistances for a 1 ft length of pipe applying to the illustrative problem depicted in Fig. 4 have now been calculated and are summarized as follows:

- R_1 convection from the pipe wall to the cold water $= 2.8 \times 10^{-3}$ hr degree Fahrenheit per Btu.
- R_2 conduction through the pipe wall $= 8.5 \times 10^{-4}$ hr degree Fahrenheit per Btu.
- R_3 conduction through the cork insulation $= 3.9$ hr degree Fahrenheit per Btu.
- R_4 parallel convection and radiation from the surroundings $= 0.22$ hr per degree Fahrenheit per Btu.

Then

- R_T the overall resistance surroundings to cold water $= R_1 + R_2 + R_3 + R_4 = 4.1$ hr degree Fahrenheit per Btu.

Note that the controlling resistances are R_3 and R_4. That is, the neglect of R_1 and R_2 would not significantly influence the total resistance, R_T.

On the basis of this resistance calculation the heat transfer from the surroundings to the cold water may be evaluated as:

$$\frac{q_{rc}}{N} = \frac{\Delta t}{R_T} = \frac{120 - 34}{4.1} = 21 \text{ Btu per hour per foot.}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1 ft pipe length:

$$q_{rc} = 21 \text{ Btu per hour.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 6 as:

- Δt air to insulation surface $= R_4\, q_{rc} = 0.22 \times 21 = 4.6$ F.
- Δt through the insulation $= R_3\, q_{rc} = 3.9 \times 21 = 82$ F.
- Δt through the pipe wall $= R_2\, q_{rc} = 8.5 \times 10^{-4} \times 21 = 0.02$ F.
- Δt pipe wall to cold water $= R_1\, q_{rc} = 2.8 \times 10^{-3} \times 21 = 0.06$ F.

The solution was obtained on the assumption that the air temperature and the outside temperature differed by 20 F. In order to obtain a slightly better estimate of the rate of heat transfer the numerical solution should be repeated using the temperatures calculated from the previous listed temperature differences.

The foregoing problem serves to illustrate a general method of solving steady-state heat transfer problems. There are many problems which cannot be approximated by steady-state solutions. For instance, the problem of pipe line insulation in transient service; the behavior of automatically controlled thermoflow circuits; or the periodic absorption of solar energy by roof and wall structures during the day and nocturnal radiation to the *cold* sky at night. The transient heat transfer problem differs from the steady-state in that energy storage rates need to be