Case 11 of Table 2 fits the conditions of the problem if only free convection heating of the pipe is assumed. The equation in this case is as follows:

$$h_c = 0.271\left(\frac{P}{P_o}\right)^{0.5}\left(\frac{\Delta t}{D}\right)^{0.25} \quad (16)$$

where

$$\Delta t = 20\text{ F}$$
$$D = 0.364\text{ ft}$$
$$P = P_0 = \text{one atmosphere}$$

therefore,

$$h_c = 0.271\left(\frac{20}{0.364}\right)^{0.25}$$

$$h_c = 0.737 \text{ Btu per (hr) (sq ft) (F deg).}$$

Using the surface area of the insulation, the value of the resistance per unit length is determined.

$$A = \pi\left(\frac{4.375}{12}\right) \times 1 = 1.14 \text{ sq ft}$$

$$R_c = \frac{1}{h_c A} = \frac{1}{0.737 \times 1.14}$$

$$R_c = 1.19 \text{ (hr) (F deg) per Btu.}$$

This result may not be deemed conservative inasmuch as the expression is for still air. If, however, the air is not still, but flows at approximately 5 mph or 7 fps, the heat transfer equation for forced convection would apply. This equation is Case 5 of Table 2.

$$h_{c(\text{average})} = 0.211(T_f)^{0.43}\frac{(u_\infty \rho)^{0.8}}{D^{0.4}} \quad (17)$$

$$T_f = \frac{100 + 120}{2} + 460 = 570 \text{ Rankine (Fahrenheit absolute)}$$

$$u_\infty = 7 \text{ fps}$$

$$\rho = 0.076\left(\frac{520}{570}\right) = 0.0694 \text{ lb per cu ft}$$

$$D = 0.364 \text{ ft}$$

$$h_{c(\text{average})} = \frac{0.211(570)^{0.43}(7 \times 0.0694)^{0.8}}{(0.364)^{0.4}}$$

$$h_{c(\text{average})} = 2.73 \text{ Btu per (hr) (sq ft) (F deg)}$$

and

$$R_c \text{ (Forced Convection)} = \frac{1}{h_c A} = \frac{1}{2.73 \times 1.14}$$

$$R_c = 0.321 \text{ (hr) (F deg) per Btu.}$$

The radiation resistance, R_r, which acts in parallel with the resistance just calculated, can be computed with the aid of Fig. 7. The pipe wall, assumed at 100 F *sees* the surroundings at 120 F. If these two temperatures are used with Fig. 7, a value for $\frac{h_r}{F_A F_E}$ is determined directly.

$$\frac{h_r}{F_A F_E} = 1.4 \text{ Btu per (hr) (F deg) (sq ft).}$$

The angle factor, F_A, is unity, and for an estimated surface emissivity of 0.95 (see Table 3), F_E = 0.95. Therefore,

$$h_r = 1.4\, F_A F_E = 1.4 \times 1 \times 0.95$$

$$h_r = 1.33 \text{ Btu per (hr) (F deg) (sq ft)}$$

and the radiation resistance, R_r, is then the following:

$$R_r = \frac{1}{h_r A} = \frac{1}{1.33 \times 1.14}$$

$$R_r = 0.659 \text{ (hr) (F deg) per Btu.}$$

The resultant resistance of R_c and R_r acting in parallel (see Fig. 8) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.321} + \frac{1}{0.659} = 4.54 \text{ Btu per (hr) (F deg)}$$

$$R_4 = 0.216 \text{ (hr) (F deg) per Btu.}$$

The overall resistance, R_T, surroundings to cold water, is the sum of $R_1 + R_2 + R_3 + R_4$ = 4.12 (hr) (F deg) per Btu for 1-ft length of pipe. Note that the controlling resistances are R_3 and R_4, and that neglect of both 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:

$$\frac{q_{rc}}{N} = \frac{t_o - t_f}{R_T} = \frac{120 - 34}{4.12} = 20.8 \text{ Btu per (hr) (ft)}$$

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

Since the calculation is based on a 1-ft pipe length,

$$q_{rc} = 20.8 \text{ Btu per hr.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 14 as:

$$\Delta t = qR$$

$t_o - t_{s3}$ (air to insulation surface) $= qR_4 = 20.8 \times 0.216 = 4.49$ F deg

$t_{s3} - t_{s2}$ (through the insulation) $= qR_3 = 20.8 \times 3.9 = 81.2$ F deg

$t_{s2} - t_{s1}$ (through the pipe wall) $= qR_2 = 20.8 \times 8.5 \times 10^{-4} = 0.018$ F deg

$t_{s1} - t_f$ (pipe wall to cold water) $= qR_1 = 20.8 \times 3.73 \times 10^{-3} = 0.078$ F deg

This solution was obtained on the temperature distribution assumptions initially made. It is apparent that a better solution could be obtained if the whole problem were reiterated using the temperature distribution just calculated.