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Revision Memo

12/01	Page 8	Equation (15) has been modified to include q_f.
	Page 21	Description for Program T2504 has been deleted.
	Page 22	Pipe size definition is clarified.

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SCOPE

The techniques for calculating the steady-state heat loss (or gain) from insulated or uninsulated equipment are covered in this section. Equations are presented for evaluating the process-side and environmental-side heat transfer coefficients and thermal resistances.

REFERENCES

DESIGN PRACTICES (BESIDES OTHER SECTIONS OF SECTION XVI)

Section VIII	Furnaces
Section IX	Heat Exchange Equipment
Section XIV	Fluid Flow

OTHER REFERENCES

1. Tenner, A. R., and Golan, L. P., *Graphic Procedure for Calculating Heat Transfer Coefficients and Flow Regimes for Single and Two-Phase Flow in Tubes*, ERE Report No. EE.6ER.71 (3/24/71).
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5. McAdams, W. H., *Heat Transmission,* McGraw-Hill, New York (1954).
6. White, R. R., and Churchill, S. W., *Experimental Foundations of Chemical Engineering,* A.I.Ch.E. Journal, 5, No. 3, 354 - 360 (Sept., 1959).
7. General Electric Heat Transfer Design Data Book.

PROCESS-SIDE HEAT TRANSFER COEFFICIENT

The process-side heat transfer resistance is a direct function of the process heat transfer coefficient, h_f. This coefficient depends on the thermal properties of the fluid, the flow regime (laminar or turbulent), the type of flow (free or forced convection), and the number of phases present. Only free convection (for process-side heat transfer) will be covered in this section. The heat transfer correlations for forced-convection cases can be found in **Section XIX**, *Heat Exchange Equipment*, **Section VIII**, *Furnaces*, and ERE Report No. EE.6ER.71, *Graphic Procedure for Calculating Heat Transfer Coefficients and Flow Regimes and Two-Phase Flow in Tubes.*

NON-FLOW HEAT TRANSFER COEFFICIENT TO VERTICAL AND HORIZONTAL SURFACES

The heat transfer coefficient between a hydrocarbon and a liquid vessel wall for non-flow cases is:

$$h_f = \frac{16.8\,(T_f - T_i)^{0.25}}{(\mu_f)^{0.4}} \qquad \text{(Customary)} \qquad \text{Eq. (1)}$$

$$h_f = \frac{6.97\,(T_f - T_i)^{0.25}}{(\mu_f)^{0.4}} \qquad \text{(Metric)} \qquad \text{Eq. (1)M}$$

where:
- h_f = Heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)
- T_f = Temperature of the process fluid, °F (°C)
- T_i = Temperature of the surface in contact with the fluid,°F (°C)
- μ_f = The absolute viscosity of the process fluid, cP (Pa•s), evaluated at the film temperature, $(T_f + T_i) / 2$, °F (°C)

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PROCESS-SIDE HEAT TRANSFER COEFFICIENT (Cont)

Note that in a cone roof storage tank, the area above the liquid level is treated as a “dry" wall as compared to the “wet" wall involving the liquid and the vessel wall.

If the process fluid is not hydrocarbon liquid, the general heat transfer correlation presented as **Eq. (3)** and **Table 1** may be used for calculating the heat transfer coefficient, provided that the thermal properties for the fluid are used.

TANK HEATER COEFFICIENTS

For tank heaters (steam, finned) the following film coefficient may be used for calculating the rate of heat transfer to heavy fuel oil or asphalt:

$$U_o = \frac{30\,(T_c - T_f)^{0.14}}{(\mu_f)^{0.4}} \qquad \text{(Customary)} \qquad \text{Eq. (2)}$$

$$U_o = \frac{11.7\,(T_c - T_f)^{0.14}}{(\mu_f)^{0.4}} \qquad \text{(Metric)} \qquad \text{Eq. (2)M}$$

where: U_o = Overall heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)
T_c = Temperature of the steam, °F (°C)
Other terms are as defined for **Eq. (1)**

ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT

The heat transfer coefficient on the exterior of a body is difficult to predict exactly. Conditions in practice are usually different from the conditions by which the heat transfer correlation has been derived. However, a number of the correlations have been found to yield reliable results. The recommended procedures for predicting the environment-side heat transfer coefficient are given below.

CONVECTION COEFFICIENT

Convection to a Stagnant Fluid - Heat transfer to a stagnant fluid is by natural convection. The amount of heat transferred depends upon the fluid properties, the temperature difference between the surface and the fluid, and the orientation of the surface. The natural convection heat transfer coefficient can be expressed as:

$$h_{nc} = C\left(\frac{k_f}{L}\right)(Gr\ Pr)^n \qquad \text{Eq. (3)}$$

where: h_{nc} = Natural convection heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)
C = See **Table 1**
k_f = Thermal conductivity of the fluid, Btu/h ft^2 °F/ft (W/m°C), evaluated at the film temperature
L = Height of vertical surface, smallest linear dimension of horizontal plate or diameter of horizontal cylinder, ft (m)
n = See **Table 1**
Gr Pr, the product of Grashof and Prandtl numbers is:

$$Gr\ Pr = \frac{g\,\beta\,\rho^2\,C_p\,L^3\,\Delta T}{2.42\,\mu_f\,k_f} \qquad \text{(Customary)} \qquad \text{Eq. (4)}$$

$$Gr\ Pr = \frac{10^3\,g\,\beta\,\rho^2\,C_p\,L^3\,\Delta T}{\mu_f\,k_f} \qquad \text{(Metric)} \qquad \text{Eq. (4)M}$$

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ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT (Cont)

where: g = Gravitational constant, 4.17 x 108 ft/h^2 (9.81 m/s^2)
β = Volumetric thermal expansion coefficient, °F^{-1} (°C^{-1})
ρ = Density of fluid, lb/ft^3 (kg/m^3), evaluated at the film temperature
C_p = Specific heat capacity of fluid, Btu/lb °F (kJ/kg °C), evaluated at the film temperature
ΔT = Temperature difference between surface and fluid, °F (°C)
$T_i - T_f$ or $T_f - T_i$
μ_f = Absolute viscosity of the fluid, cP (Pa•s), evaluated at the film temperature

The fluid properties are evaluated at the film temperature, $(T_f + T_i) / 2$, °F (°C)

The values of C and n are given in **Table 1,** below:

TABLE 1
CONSTANTS C AND n FOR NATURAL CONVECTION HEAT TRANSFER COEFFICIENT

RANGE OF (Gr Pr)	>>	10^3 TO 10^9		> 10^9	
VALUE OF CONSTANTS IN EQ. (3)	>>	C	n	C	n
Vertical planes, vertical and horizontal cylinders		0.59	1/4	0.13	1/3
Horizontal plates, hot surface facing up or cold surface facing down		0.71	1/4	0.17	1/3
Horizontal plates, hot surface facing down or cold surface facing up		0.35	1/4	0.08	1/3

The natural convective heat transfer coefficient for air has been plotted on **Figures 1** and **2**. These figures may be used over the whole atmospheric temperature range (-50°F to 100°F or -46°C to 38°C) with fair accuracy. The natural convective heat transfer coefficient for water has been plotted on **Figure 3**. This figure has been reprinted from the General Electric Heat Transfer Design Data Book. The value of h_{nc} determined from **Figure 3** must be multiplied by the correction factor which can be found on the insert.

Convection to a Moving Fluid - Heat transfer to a moving fluid, forced convection, depends on the fluid properties, the fluid velocity with respect to the surface, and the size of the surface.

The heat transfer coefficient for air flowing across a cylinder is:

$$h_{fc} = 1.93 \left(\frac{k_a}{\sqrt{\nu_a}}\right) \sqrt{\frac{V}{D}} + 0.001875 \left(\frac{k_a}{\nu_a}\right) V \qquad \text{(Customary)} \qquad \text{Eq. (5)}$$

$$h_{fc} = \left(1.46 \times 10^4\right) \left(\frac{k_a}{\sqrt{\nu_a}}\right) \sqrt{\frac{V}{D}} + \left(1.26 \times 10^3\right) \left(\frac{k_a}{\nu_a}\right) V \qquad \text{(Metric)} \qquad \text{Eq. (5)M}$$

where: h_{fc} = Forced convection heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)
k_a = Air thermal conductivity, Btu/h ft^2 °F/ft (W/m°C), at $\frac{T_a + T_s}{2}$, °F (°C)
ν_a = Air kinematic viscosity, ft^2/s (mm^2/s), at $\frac{T_a + T_s}{2}$, °F (°C)
V = Wind speed, mph (m/s)
D = Outside diameter of cylinder, in. (mm)
T_a = Air temperature, °F (°C)
T_s = Temperature of surface in contact with air, °F (°C)

Although **Eq. (5)** has been established for fluid flow across a cylinder, it is recommended that this equation be used also for flat plates.

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ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT (Cont)

The value of the forced convection heat transfer coefficient can also be determined from **Figure 4,** where the heat transfer coefficient has been calculated on the basis of an air film temperature of 50°F (10°C).

For water flowing over a cylinder, the equation for forced convection heat transfer is:

$$h_{fc} = 12 \left(\frac{k_f}{D}\right) \left[0.35 + 0.56\ (Re)^{0.5}\right] (Pr)^{0.31} \qquad \text{(Customary)} \qquad \text{Eq. (6)}$$

$$h_{fc} = (10^3) \left(\frac{k_f}{D}\right) \left[0.35 + 0.56(Re)^{0.5}\right] (Pr)^{0.31} \qquad \text{(Metric)} \qquad \text{Eq. (6)M}$$

where: Re = Reynolds number, dimensionless, defined by **Eq. (7)**
Pr = Prandtl number, dimensionless, defined by **Eq. (8)**
Other terms are as previously defined.

$$Re = \frac{51.2\ \rho\ D\ v}{\mu_f} \qquad \text{(Customary)} \qquad \text{Eq. (7)}$$

$$Re = \frac{\left(10^{-3}\right) \rho\ D\ v}{\mu_f} \qquad \text{(Metric)} \qquad \text{Eq. (7)M}$$

$$Pr = \frac{2.42\ C_p\ \mu_f}{k_f} \qquad \text{(Customary)} \qquad \text{Eq. (8)}$$

$$Pr = \frac{(10^3)\ C_p\ \mu_f}{k_f} \qquad \text{(Metric)} \qquad \text{Eq. (8)M}$$

where: v = Water velocity, ft/s (m/s)
Other terms are as previously defined.

Combined Natural and Forced Convection - A number of situations involve convection heat transfer which is neither forced nor free in nature. These circumstances arise when a fluid passes over a heated surface at a rather low velocity. Coupled with the "forced" flow velocity is a convective velocity which is generated by the buoyancy forces resulting from a reduction in the fluid density near the heated surface.

A forced convection situation involving a fluid velocity of 100 ft/s (30 m/s), for example, would overshadow most free convection effects. A forced flow situation at very low velocities [~ 1 ft/s (0.3 m/s)] might be influenced by free convection currents. As a rule of thumb, when $Gr / (Re)^2 > 1.0$, then free convection is of primary importance.

If the natural and forced convection heat transfer coefficients are the same order of magnitude, then for vertical surfaces and horizontal pipes they may be combined as:

$$h_c = \sqrt{(h_{nc})^2 + (h_{fc})^2} \qquad \text{Eq. (9)}$$

where: h_c = Combined forced and natural convection heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)
h_{nc} = Natural convection heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)
h_{fc} = Forced convection heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)

For flat horizontal surfaces, the larger of either the natural or forced convection coefficient should be used.

RADIATION COEFFICIENT

Radiation to Surroundings - The energy leaving a surface by radiation is a function of:

- The temperature of the surface, T_s.
- The emissivity of the surface, ε.
- The temperature "seen" by the surface or the temperature of surroundings, T_{sur}.

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ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT (Cont)

The temperature seen by the surface may be the ground surface temperature, the outside surface temperature of other equipment, the temperature of the sky, or any combination of these. Usually, a surface "sees" the ground, other equipment, and the sky simultaneously.

The calculation of the percentage of energy radiated to each temperature is complicated; therefore, the designer should assume an average surroundings temperature for use in calculations. The air temperature is usually suitable for use as the average surroundings temperature, unless many high-temperature sources such as fires or flares are near, in which case the Heat Transfer Section of EED should be consulted for analysis.

The heat transferred by radiation from a surface totally enclosed by the surroundings at T_{sur} is:

$$q''_r = \varepsilon\,\sigma\left[(T_s + 460)^4 - (T_{sur} + 460)^4\right] \quad \text{(Customary)} \quad \text{Eq. (10)}$$

$$q''_r = \varepsilon\,\sigma\left[(T_s + 273)^4 - (T_{sur} + 273)^4\right] \quad \text{(Metric)} \quad \text{Eq. (10)M}$$

where: σ = 0.1714 x 10^{-8} Btu/h ft^2 °R^4 (5.669 x 10^{-8} W/m^2 °K^4)
q''_r = Radiant heat transfer rate, Btu/h ft^2 (W/m^2)
ε = Emissivity of surface at T_s, dimensionless
T_s = Temperature of surface, °F (°C)
T_{sur} = Temperature of surroundings, °F (°C)

Eq. (10) may be simplified to:

$$q''_r = h_r\,(T_s - T_{sur}) \quad \text{Eq. (11)}$$

$$\text{if } h_r = \varepsilon\,\sigma\left[(T_s + 460)^2 + (T_{sur} + 460)^2\right]\left[(T_s + 460) + (T_{sur} + 460)\right] \quad \text{(Customary)} \quad \text{Eq. (12)}$$

$$\text{if } h_r = \varepsilon\,\sigma\left[(T_s + 273)^2 + (T_{sur} + 273)^2\right]\left[(T_s + 273) + (T_{sur} + 273)\right] \quad \text{(Metric)} \quad \text{Eq. (12)M}$$

The value of the radiant heat transfer coefficient, h_r, can be calculated by **Eq. (12)** or read from **Figure 5** (approximate emissivity values of the most common surfaces are listed on **Figure 5**).

Effect of Solar Radiation - The solar energy absorbed by a surface can have a significant effect on the equilibrium temperature of that surface. The following simplified procedure may be used to estimate the magnitude of the energy gain and equilibrium surface temperature which can be expected as a result of exposure to solar radiation.

The rate at which solar energy impinges on a surface placed normal to the sun's rays at the outer fringes of the atmosphere is 442 Btu/h ft^2 (1395 W/m^2). This is the solar constant G_o. The rate at which solar energy reaches the earth is substantially less than this value, because the radiation is absorbed and scattered by the atmosphere. The attenuation of the solar radiant intensity by the earth's atmosphere depends on the length of the path through which the radiation must travel. This is related to the position of the sun.

The radiant energy G_n, incident upon a surface at ground level, placed normal to the rays of the sun, can be estimated from the equation:

$$G_n = G_o\,\tau_a^m = 442\,\tau_a^m \quad \text{(Customary)} \quad \text{Eq. (13)}$$

$$G_n = G_o\,\tau_a^m = 1395\,\tau_a^m \quad \text{(Metric)} \quad \text{Eq. (13)M}$$

where: G_o = Solar constant, 442 Btu/h ft^2 (1395 W/m^2)
G_n = Radiant energy incident on a surface at ground level normal to the sun's rays, Btu/h ft^2 (W/m^2)
τ_a = Transmission coefficient for unit air mass (0.81 to 0.62), dimensionless
m = Ratio of the actual path length to the shortest possible path

The value of τ_a is slightly less in the summer than in winter, because the atmosphere contains more water vapor during the summer. It also varies with the condition of the sky, ranging from 0.81 on a clear day to 0.62 on a cloudy one. A mean value of 0.7 is generally considered acceptable for most purposes.

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ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT (Cont)

The value of m depends on the angle between the zenith and the direction of the sun. The values of m for various north latitude locations at noon, local solar time (maximum solar radiation), are given in **Table 2**, below. If the receiving surface is not normal to the direction of the sun's rays (the usual case), then the incident radiation will be reduced by the cosine of the angle between the direction of the sun and a line normal to the surface.

$$G_i = G_n \cos i \qquad \text{Eq. (14)}$$

For a horizontal surface, values of cos i (at noon) are also given in **Table 2.**

TABLE 2
SOLAR RADIATION EFFECT

NORTH LATITUDE	m	cos i
25°	1.004	0.996
30°	1.015	0.985
35°	1.035	0.966
40°	1.064	0.940
45°	1.103	0.906
50°	1.155	0.866

Use of the values for τ_a, m, and cos i will permit one to calculate the maximum incident radiation on a body. If an analysis of different times during the day or of different inclinations of the surface are required, consult **Reference 4** (Kreith).

The fraction of the incident radiation absorbed by a surface depends on the solar absorptivity coefficient for the body, G_{abs}, which equals $\alpha\, G_i$. Consult **Table 3** for the solar absorptivity coefficients for a number of common substances. The equilibrium temperature of a surface exposed to solar radiation can be calculated from **Eq. (15)**:

$$G_{abs} + q_a + q_c + q_k - q_r + q_f = 0 \qquad \text{Eq. (15)}$$

where: G_{abs} = Solar radiation absorbed, Btu/h ft^2 (W/m^2)
q_a = Atmospheric radiation absorbed, Btu/h ft^2 (W/m^2). On a clear day, $q_a = 0.1\, G_{abs}$; on a cloudy day $q_a = 0.5\, G_{abs}$
q_c = Net heat absorbed by convection, Btu/h ft^2 (W/m^2) (negative if heat is given off)
q_k = Net conduction (if any) to the body, Btu/h ft^2 (W/m^2)
q_r = Radiation emitted, Btu/h ft^2 (W/m^2)
q_f = Net heat absorbed by convection from process side, Btu/h ft^2 (W/m^2) negative if process fluid is cooler than surface.

Table 3 also contains equilibrium temperatures, T_s, that have been calculated for various horizontal surfaces, assuming a value of G_{abs} = 300 Btu/h ft^2 (946 W/m^2), still air at 100°F (38°C) and a surface radiating to the surroundings at 100°F (38°C).

TABLE 3
SOLAR ABSORPTIVITY COEFFICIENTS

TYPE OF SURFACE	SOLAR ABSORPTIVITY COEFFICIENT	EMISSIVITY, ε	EQUILIBRIUM TEMPERATURE	
			°F	°C
White paint (one mil thick, 25 µm thick)	0.15	0.94	123	51
Aluminum paint (2 coats)	0.35	0.27 to 0.67	170 to 154	77 to 68
Galvanized steel (new; body tarnished)	0.65; 0.89	0.17; 0.53	220; 221	104; 105
Stainless steel jacket	0.8(?)	0.3(?)	228(?)	109(?)
Aluminum jacket	0.2 to 0.4(?)	0.1 to 0.2(?)	152 to 181(?)	67 to 83(?)
Black body	1.0	1.0	205	96

Note: (?) denotes uncertain value.

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ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT (Cont)

The above information indicates that white paint is by far the best choice when additional heat input due to solar radiation is undesirable. This is due to the combination of low solar absorptivity and high long-wave length emissivity. It should be pointed out that the emissivity and absorptivity values are not very well known, even for commonly used materials, as can be seen from the number of (?) marks in **Table 3.**

Effect of Radiant Cooling at Night - Maximum radiant heat loss will occur on a clear night. To predict this value, one should assume a value of -50°F (-46°C) for T_{sur} in **Eq. (11)**.

COMBINED RADIATION AND CONVECTION COEFFICIENT

Often the radiation and convection coefficients are the same order of magnitude, and it is convenient to combine these factors into a single overall heat transfer coefficient at an average environment temperature. This overall heat transfer coefficient, h_e, is the sum of all the heat transfer coefficients. The average environment temperature, T_{eav}, is the weighted average of the temperature of each sink involved, the "weights" being the heat transfer coefficients corresponding to the temperatures involved.

The effect of solar radiation, which is expressed as a heat flux value, G_{abs}, must be included as follows:

$$h_e = \sum_j h_j \qquad \text{Eq. (16)}$$

$$T_{eav} = \frac{\sum_j h_j T_j + G_{abs}}{h_e} \qquad \text{Eq. (17)}$$

where: $\sum_j h_j$ = the summation of the atmospheric-side heat transfer coefficients, and

$\sum_j h_j \bullet T_j$ = the summation of the products of each atmosphere side heat transfer coefficient and its sink temperature (normally $h_c \bullet T_a$ and $h_r \bullet T_{sur}$).

An example is presented in Sample Problem No. 1.

CONDUCTION TO THE GROUND

The heat transfer through the bottom of a storage tank to the ground can be calculated only when the thermal conductivity, k_g, of the ground is known and is constant with depth or can be assigned an average constant value. If the ground conductivity for a particular location has not been measured, it can be estimated as a function of the dry soil density and the moisture content from **Figure 6** for sandy soils, and from **Figure 7** for silty and clay soils.

Energy Loss Through the Bottom of a Storage Tank - The energy loss through the bottom of a storage tank can be expressed as:

$$q_{fl} = 2 D_b (T_s - T_g)(k_g / 12) \qquad \text{(Customary)} \qquad \text{Eq. (18)}$$

$$q_{fl} = 2 D_b (T_s - T_g)(k_g) \qquad \text{(Metric)} \qquad \text{Eq. (18)M}$$

where:
- q_{fl} = Bottom heat loss, Btu/h (W)
- D_b = Tank (base) diameter, ft (m)
- T_s = Floor plate temperature of uninsulated tank or temperature of the face of floor insulation which is in contact with the ground, °F (°C)
- T_g = Temperature of the ground, °F (°C)
- k_g = Thermal conductivity of the ground, Btu/h ft^2 °F/in. (W/m °C)

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ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT (Cont)

The tank temperature in contact with the ground is usually not known. However, **Eq. (18)** can be written in the resistance form:

$$q_{fl} = \frac{T_s - T_g}{\left(\dfrac{1}{2\ D_b\ (k_g\ /\ 12)}\right)} = \frac{T_s - T_g}{R_g} \qquad \text{(Customary)} \qquad \text{Eq. (19)}$$

$$q_{fl} = \frac{T_s - T_g}{\left(\dfrac{1}{2\ D_b\ (k_g)}\right)} = \frac{T_s - T_g}{R_g} \qquad \text{(Metric)} \qquad \text{Eq. (19)M}$$

where: The resistance is $\dfrac{1}{2\ D_b\ (k_g/12)}$ (Customary Units), $\dfrac{1}{2\ D_b\ (k_g)}$ (Metric Units)

This resistance can be used in series with the resistances of the insulation, pipe wall, and product film to determine the energy loss.

For refrigerated storage, the soils are often such as to require the installation of heating coils in the ground below the tank to prevent frost heaving. The heat loss can be predicted using the plane geometry formulas, since the coils are designed to maintain a uniform temperature below the tank.

Heat Loss From a Buried Cylinder or Pipe - The heat loss per hour per foot (meter) of length is given by:

$$q' = \frac{2\pi\ (T_s - T_e)\ (k_g\ /\ 12)}{\cosh^{-1}\ (H\ /\ r_s)} \qquad \text{(Customary)} \qquad \text{Eq. (20)}$$

$$q' = \frac{2\pi\ (T_s - T_e)\ (k_g)}{\cosh^{-1}\ (H\ /\ r_s)} \qquad \text{(Metric)} \qquad \text{Eq. (20)M}$$

where:
- q' = Heat loss, Btu/h ft of length (W/m of length)
- T_s = Temperature at r_s, °F (°C)
- T_e = Isothermal surface temperature of the soil, °F (°C)
- H = Depth below grade of the axis of the cylinder, in. (mm)
- r_s = Outer radius of the pipe (if insulation present, then use the outer radius of the insulation), in. (mm)

The equivalent environment-side heat transfer coefficient is:

$$h_e = \frac{k_g}{6\ D\ \cosh^{-1}\ (H\ /\ r_s)} \qquad \text{(Customary)} \qquad \text{Eq. (21)}$$

$$h_e \frac{k_g}{2\ D\ \cosh^{-1}\ (H\ /\ r_s)} \qquad \text{(Metric)} \qquad \text{Eq. (21)M}$$

where:
- h_e = Environment-side heat transfer coefficient, Btu/h ft^2 °F (W/m^2 °C)
- D = Diameter of buried cylinder or pipe, ft (m)

Other terms are as defined previously.

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HEAT TRANSFER RESISTANCE OF INSULATION

Although thermal insulations are not homogeneous solids, they can be treated as such for the purpose of calculating heat transfer. The exception to this rule is reflective type insulation, which is not discussed here.

The equation for conduction through a flat sheet of insulation is:

$$q = \left(\frac{k}{t}\right) A\ (T_i - T_s) \qquad \text{(Customary)} \qquad \text{Eq. (22)}$$

$$q = 10^3 \left(\frac{k}{t}\right) A\ (T_i - T_s) \qquad \text{(Metric)} \qquad \text{Eq. (22)M}$$

where: q = Heat transferred, Btu/h (W)
k = Thermal conductivity of the material, Btu/h ft^2 °F/in. (W/m °C)
t = Thickness of plate, in. (mm)
A = Cross-sectional area of plate perpendicular to direction of heat flow, ft^2 (m^2)
$(T_i - T_s)$ = Temperature drop across the plate, °F (°C)

For conduction through cylindrical insulation, the heat loss per ft is:

$$q' = \frac{2\ \pi\ (k\ /\ 12)\ \ (T_i - T_s)}{\ln\ (1 + t\ /\ r_i)} \qquad \text{(Customary)} \qquad \text{Eq. (23)}$$

$$q' = \frac{2\ \pi\ (k)\ \ (T_i - T_s)}{\ln\ (1 + t\ /\ r_i)} \qquad \text{(Metric)} \qquad \text{Eq. (23)M}$$

where: r_i = Inside radius of cylinder, in. (mm)
Other terms are as previously defined.

The thermal conductivity, k, is usually not a constant, but varies with temperature. Because of usually large temperature drops across insulation layers, this variation must be taken into account by use of a value for conductivity based on the mean temperature of the insulation, $(T_i + T_s) / 2$.

OVERALL STEADY-STATE HEAT TRANSFER COEFFICIENT

Plane Geometry (Large Equipment, Vessels, etc.)

For n layers of metal and insulation, the heat loss q", in Btu/h ft^2 °F/in. (W/m^2), can be calculated by dividing the temperature differences by the sum of all the resistances:

$$q'' = \frac{T_f - T_e}{\frac{1}{h_f} + \sum_j \frac{t_j}{k_j} + \frac{1}{h_e}} \qquad \text{(Customary)} \qquad \text{Eq. (24)}$$

$$q'' = \frac{T_f - T_e}{\frac{1}{h_f} + \sum_j (10^{-3})\ \frac{t_j}{k_j} + \frac{1}{h_e}} \qquad \text{(Metric)} \qquad \text{Eq. (24)M}$$

See **Figure A** on the next page.

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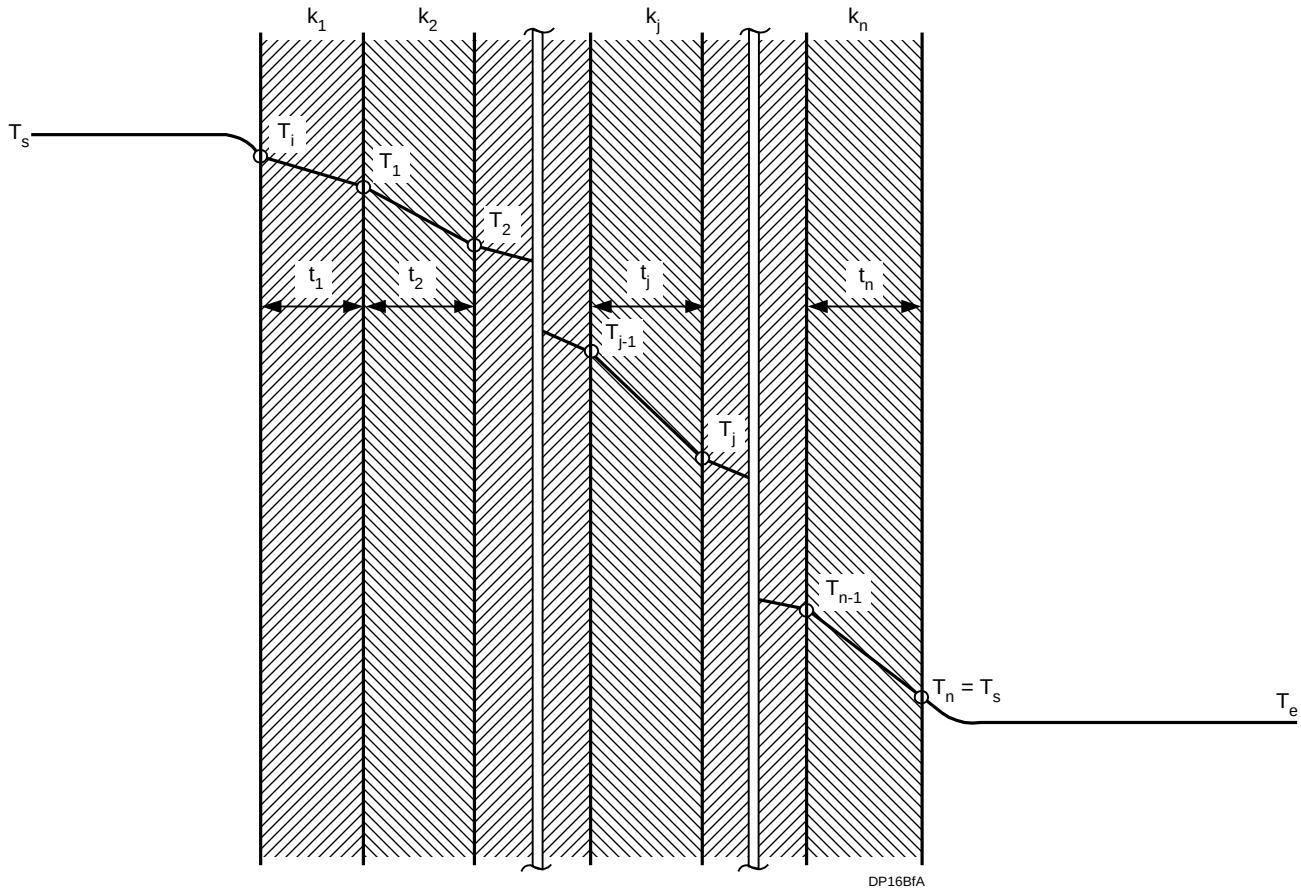
HEAT TRANSFER RESISTANCE OF INSULATION (Cont)

FIGURE A
GEOMETRIC AND PHYSICAL CONVENTIONS FOR FLAT INSULATION



In **Eq. (24)**, t_j and k_j represent the thickness and conductivity, respectively, of each layer. Contact resistances between layers have been neglected. Since h_f, h_e, and the conductivity of each layer depend upon their respective surface and average temperatures, a trial-and-error procedure is necessary to solve this expression. From assumed values for T_j, the conductivities for each layer can be calculated, and from these the total energy loss. With this energy loss, the boundary temperatures can be recalculated as follows:

$$T_i = T_f - q'' / h_f \quad (T_i = \text{Process-side surface temperature})$$
$$T_1 = T_i - q'' (t_1 / K_1)$$
$$T_2 = T_1 - q'' (t_2 / k_2)$$
$$T_j = T_{j-1} - q'' (t_j / k_j)$$
$$T_s = T_n = T_{n-1} - q'' (t_n / k_n) \quad (T_s = \text{Environment-side surface temperature})$$
$$= T_e + q'' / h_e$$

If these temperatures match the assumed values within 10% of the overall temperature drop, $T_i - T_s$, then the assumed values are correct. If not, the calculated temperatures are used for the next trial.

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HEAT TRANSFER RESISTANCE OF INSULATION (Cont)

For a single layer of insulation of thickness t, conductivity k, a high process-side heat transfer coefficient, and negligible metal resistance, **Eq. (24)** reduces to the following:

$$q'' = \frac{T_f - T_e}{\frac{t}{k} + \frac{1}{h_e}} \qquad \text{(Customary)} \qquad \text{Eq. (25)}$$

$$q'' = \frac{T_f - T_e}{(10^{-3})\frac{t}{k} + \frac{1}{h_e}} \qquad \text{(Metric)} \qquad \text{Eq. (25)M}$$

Calculation Form No. 1 outlines the trial-and-error procedure for calculating the energy loss from large [> 36 in. (914 mm) diameter] equipment. The procedure is a slight modification of one for flat plates, and is an acceptable approximation. The process side heat transfer resistance is assumed to be negligible in the procedure.

The thickness required to limit the heat losses through a plane (or shallow cylindrical) surface to a given value can be expressed as:

$$t = k\left(\frac{T_f - T_e}{q''} - \frac{1}{h_e}\right) \qquad \text{Eq. (26)}$$

The surface temperature can be calculated as follows:

$$T_s = T_e + \frac{q''}{h_e} = T_e + \left(\frac{T_f - T_e}{1 + \frac{t\,h_e}{k}}\right) \qquad \text{(Customary)} \qquad \text{Eq. (27)}$$

$$T_s = T_e + \frac{q''}{h_e} = T_e + \left(\frac{T_f - T_e}{1 + (10^3)\frac{t\,h_e}{k}}\right) \qquad \text{(Metric)} \qquad \text{Eq. (27)M}$$

The thickness required to limit the surface temperature to a given value can be expressed as:

$$t = \left(\frac{T_f - T_s}{T_s - T_e}\right)\left(\frac{k}{h_e}\right) \qquad \text{(Customary)} \qquad \text{Eq. (28)}$$

$$t = \left(\frac{T_f - T_s}{T_s - T_e}\right)\left(\frac{k}{h_e}\right) \times 10^3 \qquad \text{(Metric)} \qquad \text{Eq. (28)M}$$

Cylindrical Geometry (Pipe Insulation: See Figure B, on the next page)

For n layers of insulation, the heat loss per linear foot, q', can be expressed as:

$$q' = \frac{2\pi\,(T_f - T_e)}{\frac{1}{h_f\,(r_i\,/\,12)} + \sum_j \left[\frac{1}{(k_j\,/\,12)}\ln\left(1 + t_j\,/\,r_{j-1}\right)\right] + \frac{1}{h_e\,(r_s\,/\,12)}} \qquad \text{(Customary)} \qquad \text{Eq. (29)}$$

$$q' = \frac{2\pi\,(T_f - T_e)}{\frac{10^3}{h_f\,(r_i)} + \sum_j \left[\frac{1}{(k_j)}\ln\left(1 + t_j\,/\,r_{j-1}\right)\right] + \frac{10^3}{h_e\,(r_s)}} \qquad \text{(Metric)} \qquad \text{Eq. (29)M}$$

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HEAT TRANSFER RESISTANCE OF INSULATION (Cont)

The temperatures at the insulation boundaries can be found by:

$$T_i = T_f - \frac{q'}{2\pi\,(r_i\ /\ 12)\,h_f}$$ (Customary) Eq. (30)

$$T_i = T_f - \frac{(10^3)q'}{2\pi\,(r_i)h_f}$$ (Metric) Eq. (30)M

$$T_1 = T_i - \frac{q'\ \ln(1 + t_1\ /\ r_i)}{2\pi\,(k_1\ /\ 12)}$$ (Customary) Eq. (31)

$$T_1 = T_i - \frac{q'\ \ln(1 + t_1\ /\ r_i)}{2\pi\,(k_1)}$$ (Metric) Eq. (31)M

$$T_j = T_{j-1} - \frac{q'\ \ln(1 + t_j\ /\ r_{j-1})}{2\pi\,(k_j\ /\ 12)}$$ (Customary) Eq. (32)

$$T_j = T_{j-1} - \frac{q'\ \ln(1 + t_j\ /\ r_{j-1})}{2\pi\,(k_j)}$$ (Metric) Eq. (32)M

$$T_s = T_n = T_{n-1} - \frac{q'\ \ln(1 + t_n\ /\ r_{n-1})}{2\pi\,(k_n\ /\ 12)}$$ (Customary) Eq. (33a)

$$= T_e + \frac{q'}{2\pi\,(r_s\ /\ 12)\,h_e}$$ (Customary) Eq. (33b)

$$T_s = T_n = T_{n-1} - \frac{q'\ \ln(1 + t_n\ /\ r_{n-1})}{2\pi\,(k_n)}$$ (Metric) Eq. (33a)M

$$= T_e + \frac{(10^3)\,q'}{2\pi\,(r_s)\,h_e}$$ (Metric) Eq. (33b)M

A trial-and-error solution is necessary to calculate q', since h_f and h_e are usually functions of the process-side and environment-side surface temperatures.

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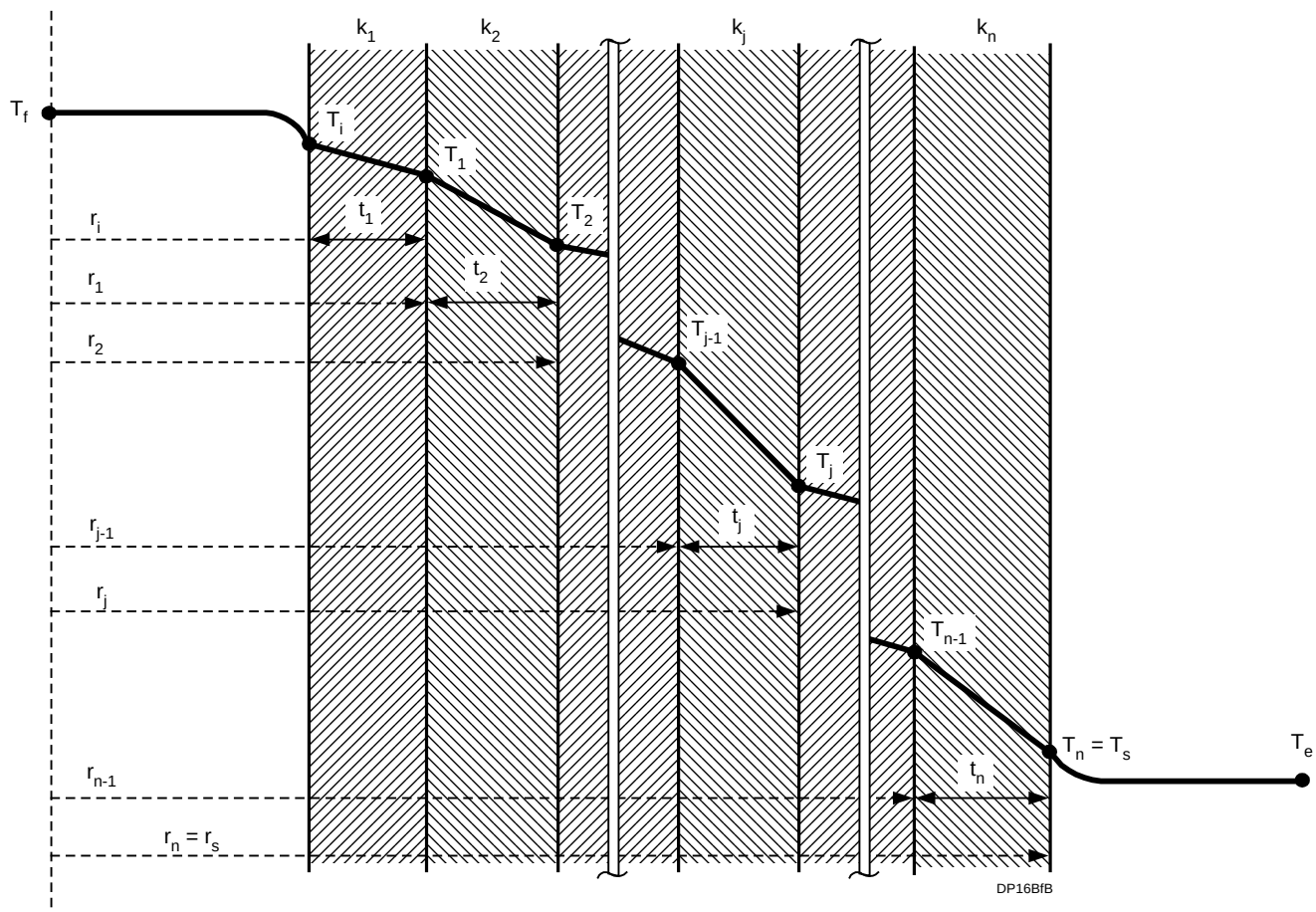
HEAT TRANSFER RESISTANCE OF INSULATION (Cont)

FIGURE B
GEOMETRIC AND PHYSICAL CONVENTIONS FOR PIPE INSULATION



For a single pipe insulation of thickness t, inside radius r, and insulation conductivity k, and for a high process-side heat transfer coefficient, **Eq. (29)** reduces to:

$$q' = \frac{2\pi\left(T_f - T_e\right)}{\left(\frac{1}{k\ /\ 12}\ \ln\left(1 + t\ /\ r\right) + \frac{1}{h_e\left(r + t\right)\ /\ 12}\right)} \qquad \text{(Customary)} \qquad \text{Eq. (34)}$$

$$q' = \frac{2\pi\left(T_f - T_e\right)}{\left(\frac{1}{k}\right)\ \ln\left(1 + t\ /\ r\right) + \frac{10^3}{h_e\ \left(r + t\right)}} \qquad \text{(Metric)} \qquad \text{Eq. (34)M}$$

In **Eq. (34)**, the metal resistance has been neglected and the heat losses are expressed per linear ft (m).

Calculation Form No. 2 gives the trial-and-error computation of heat losses from insulated pipes.

For pipe insulation, nominal insulation diameters and thicknesses often differ from the actual dimensions by about 2 to 5%. This small difference is not significant for calculation purposes.

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HEAT TRANSFER RESISTANCE OF INSULATION (Cont)

Figures 8.1, **8.2**, **9.1**, **9.2**, **10.1** and **10.2** give heat losses per foot (meter) for typical hot, cellular glass and polyurethane insulation systems. In all cases, the air is assumed to be at 50°F (10°C) and both 0 and 10 mph (4.5 m/s) wind conditions are covered. The curves for insulated pipes can be used with fair accuracy for air temperatures between 0 and 100°F (0 and 38°C) for steel or aluminum jacketing, and for wind velocities up to 20 mph (9 m/s); while the curves for bare pipes can be considered accurate only for the conditions specified. Heat losses from pipes can also be calculated by Computer Programs T3680 and T3681.

The thickness required to limit the heat losses to a given value can be found by:

$$\ln(1 + t/r) = (k/12)\left[\frac{2\pi(T_f - T_e)}{q'} - \frac{1}{h_e(r + t)/12}\right] \qquad \text{(Customary)} \qquad \text{Eq. (35)}$$

$$\ln(1 + t/r) = (k)\left[\frac{2\pi(T_f - T_e)}{q'} - \frac{10^3}{h_e(r + t)}\right] \qquad \text{(Metric)} \qquad \text{Eq. (35)M}$$

The surface temperature is given by:

$$T_s = T_e + \frac{q'}{A' h_e} = T_e + \frac{T_f - T_e}{(h_e/k)(r + t)\ln(1 + t/r) + 1} \qquad \text{(Customary)} \qquad \text{Eq. (36)}$$

$$T_s = T_e + \frac{q'}{A' h_e} = T_e + \frac{T_f - T_e}{(h_e/k)\left(\frac{r + t}{10^3}\right)\ln(1 + t/r) + 1} \qquad \text{(Metric)} \qquad \text{Eq. (36)M}$$

where: A' = Surface area of pipe or cylinder, ft^2/ft (m^2/m)

The thickness required to limit the surface temperature to a given value can be found by:

$$(r + t)\ln(1 + t/r) = \frac{(T_f - T_s)k}{(T_s - T_e)h_e} \qquad \text{(Customary)} \qquad \text{Eq. (37)}$$

$$(r + t)\ln(1 + t/r) = \frac{(T_f - T_s)k}{(T_s - T_e)h_e}(10^{-3}) \qquad \text{(Metric)} \qquad \text{Eq. (37)M}$$

STORAGE TANKS

The heat loss from oil storage tanks can be calculated by Computer Program T3681 or hand calculated using the method in Calculation Form No. 3.

SELECTION OF INSULATION THICKNESS

The thickness of an insulation system is usually selected to satisfy one of four criteria:

- To limit heat leak.
- To limit surface temperature.
- To provide the optimum economic configuration.
- To protect the equipment against fire for a specified period of time.

The first two of these criteria are discussed in this section.

DESIGN TO LIMIT HEAT LEAK

This method is used when a process is sensitive to heat leak or when a guaranteed heat leak must be met. For a single layer of insulation and a high process-side heat transfer coefficient, the thickness required to limit the heat losses to a given value can be calculated in a trial-and-error procedure, by **Eq. (26)** for flat surfaces and **Eq. (34)** for piping. The environment-side heat transfer coefficient, h_e, is calculated by the appropriate methods described under **ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT.**

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SELECTION OF INSULATION THICKNESS (Cont)

DESIGN TO LIMIT SURFACE TEMPERATURE

The option of designing for a particular surface temperature is used where it is necessary to protect personnel from accidental burns, limit condensation or icing on cold surfaces, or maintain the shell material at an acceptable temperature (internal insulation). For a single layer of insulation and a high process-side heat transfer coefficient, use **Eq. (24)** to determine the insulation thickness for flat surfaces and **Eq. (29)** for piping. The environment side heat transfer coefficient, h_e, is evaluated using the methods described under **ENVIRONMENT-SIDE HEAT TRANSFER COEFFICIENT**.

OTHER CONSIDERATIONS

Effect of Rainfall - Additional heat losses due to rainfall can be sufficient magnitude to create process upsets.

For *insulated* equipment, the effect of rain will be only to lower the outside surface temperature of the insulation to the temperature of the raindrops. Since the major portion of the thermal resistance is in the insulation, the outside surface temperature is normally close to the atmospheric temperature, and therefore the effect of the rainfall is usually small.

However, for *uninsulated* equipment, three cases may be considered:

- Very high 1000 Btu/h ft^2 °F (6000 W/m °C), process-side heat transfer coefficient and light rain.
- Very low 10 Btu/h ft^2 °F (60 W/m °C), process-side heat transfer coefficient.
- Average 50 - 200 Btu/h ft^2 °F (300 - 1200 W/m °C), process-side heat transfer coefficient, or high coefficient and very heavy rain.

In the case of light rain and a very high process-side heat transfer coefficient, the energy loss due to the rain being in contact with the uninsulated vessel wall is equal to the energy required to raise the temperature of the rain water to the temperature of the process fluid.

For process fluid at temperature lower than 212°F (100°C), the energy losses are:

$$q'' = 5.2\ R_r\ (T_f - T_r) \qquad \text{(Customary)} \qquad \text{Eq. (38)}$$

$$q'' = 1.16\ R_r\ (T_f - T_r) \qquad \text{(Metric)} \qquad \text{Eq. (38)M}$$

where: R_r = Rate of rainfall, in./h (mm/h)
T_f = Process temperature, °F (°C)
T_r = Temperature of the raindrops, °F (°C)

For process fluid at a temperature above 212°F (100°C) and assuming complete vaporization of the rain:

$$q'' = 5.2\ R_r\ (1182 - T_r) \qquad \text{(Customary)} \qquad \text{Eq. (39)}$$

$$q'' = 1.16\ R_r\ (636 - T_r) \qquad \text{(Metric)} \qquad \text{Eq. (39)M}$$

In the case of a very low process-side heat transfer coefficient, the outside metal temperature may be assumed to be equal to the rain temperature and the amount of energy loss may be calculated by including only the process-side heat transfer and process wall resistances. If neither of these assumptions is valid, an energy balance must be made to find the metal temperature T_s:

$$h_r\ (T_f - T_s) = h_c\ (T_s - T_a) + h_r\ (T_s - T_{sur}) + 5.2\ R_r\ (T_s - T_r) \qquad \text{(Customary)} \qquad \text{Eq. (40)}$$

$$h_r\ (T_f - T_s) = h_c\ (T_s - T_a) + h_r\ (T_s - T_{sur}) + 1.16\ R_r\ (T_s - T_r) \qquad \text{(Metric)} \qquad \text{Eq. (40)M}$$

Eq. (40) is used to solve for T_s; then energy loss can be found from:

$$q'' = h_f\ (T_f - T_s) \qquad \text{Eq. (41)}$$

If T_s is found to be greater than 212°F (100°C), the energy loss should be based on the process-side heat transfer coefficient and a metal temperature of 212°F (100°C) (case of partial vaporization of the rain).

Effect of Icing - Observations have shown that atmospheric moisture can solidify on a surface, either as a thin layer of light frost or as solid ice. However, there is no way of predicting which of these two forms of condensation will occur. Therefore, structural calculations should be based on supporting the weight of solid ice; but the layer of ice cannot be relied upon to reduce the heat transfer, because the moisture can also deposit as a thin frost which has essentially no insulating value. In addition, for pipelines, the possibility exists that a buildup of solid ice will actually increase the amount of heat transferred, by increasing the outside surface area.

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SAMPLE PROBLEMS

PROBLEM 1 - CALCULATION OF OVERALL HEAT TRANSFER COEFFICIENT

Given: A large, vertical plate, maintained at T_s = 200°F (93°C) in calm air at T_a = 50°F (10°C), radiating with an emissivity ε = 0.9 to surroundings at 150°F (66°C) and absorbing G_{abs} = 50 Btu/h ft^2 (158 W/m^2) from the sun.

Find: Overall heat transfer coefficient, h_e, and average environmental temperature, T_{eav}.

Solution:

(Customary)

$T_s - T_a$ = 200°F – 50°F = 150°F

From **Figure 1**, using curve for 30-in. (762 mm) diameter cylinder and correction factor of 1.0 for flat plates.

h_{nc} = (1.0) (1.12) = 1.12 Btu/h ft^2 °F

From **Figure 5**,

h_r = (0.9) (1.75) = 1.58 Btu/h ft^2 °F

From **Eq. (16)**,

$h_e = \sum_j h_j$ = 1.12 + 1.58 = **2.70 Btu/h ft^2 °F**

$\sum_j h_j T_j$ = (1.12) (50) + (1.58) (150) = 293 Btu/h ft^2

From **Eq. (17)**,

$$T_{eav} = \frac{\sum_j h_j T_j + G_{abs}}{h_e} = \frac{293 + 50}{2.7} = \mathbf{127°F}$$

If the surface were considered to be radiating to the air and the effect of solar radiation were ignored, one would have:

$h_e = h_{nc} + h_r$ = 1.12 + (0.9) (1.4) = 2.38 Btu/h ft^2 °F

$T_e = T_a$ = 50°F

(Metric)

$T_s - T_a$ = 93°C – 10°C = 83°C

h_{nc} = (1.0) (6.36) = 6.36 W/m^3 °C

h_r = (0.9) (9.9) = 8.9 W/m^2 °C

$h_e = \sum_j h_j$ = 6.4 + 8.9 = **15.3 W/m^2 °C**

$\sum_j h_j T_j$ = (6.4) (10) + (8.9) (66) = 651 W/m^2

$$T_{eav} = \frac{\sum_j h_j T_j + G_{abs}}{h_e} = \frac{651 + 158}{15.3} = \mathbf{53°C}$$

$h_e = h_{nc} + h_r$ = 6.4 + (0.9) (7.9) = 13.5 W/m^2 °C

$T_e = T_a$ = 10°C

This substantial difference in the value of h_e points up the marked effect of the temperature of the surroundings.

PROBLEM 2 - DESIGN TO LIMIT HEAT LEAK

Given: An 8 in. (200 mm) horizontal pipe is carrying liquid at T_f = 500°F (260°C). The Air temperature T_a = 50°F (10°C) and the wind speed V is 10 mph (16 km/h).

Find: The thickness of calcium silicate insulation with aluminum jacketing required to limit the heat loss to 200 Btu/h per foot (192 W per meter) of pipe.

Solution:

(Customary)

For the first trial, assume:

t = 2 in.

(Metric)

t = 51 mm

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