TABLE 7. APPROXIMATE UNIT CONDUCTANCES FOR THERMAL CONVECTION FOR SEVERAL FLOW SYSTEMS[a]

Expressed in Convenient Empirical Form

CASE	SYSTEM	UNIT CONDUCTANCE EQUATION[b]
	FORCED CONVECTION	
1.	Longitudinal flow in cylinders, turbulent region. Fluid being heated[c].	$\frac{4hr_H}{k} = 0.0225\left(\frac{4r_HG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{0.4}$ For $\left(\frac{4r_HG}{\mu}\right) > 3000$
2.	For longitudinal air flow in cylinders case 1 reduces to[c].	$h = 0.0036\,G^{0.8}/(4r_H)^{0.2}$ For $\left(\frac{4r_HG}{\mu}\right) > 3000$
3.	For longitudinal water flow in cylinders case 1 reduces to[c].	$h = 0.00486\,(1 + 0.01t)\,\frac{G^{0.8}}{4r_H^{0.2}}$ For $\left(\frac{4r_HG}{\mu}\right) > 3000$
4.	Air flow normal to a single right circular cylinder.	$h = 0.45\left(\frac{k}{D}\right) + 0.178\,G^{0.56}\left(\frac{k}{D}\right)^{0.44}$
5.	Air flow over staggered pipe banks.	$h = 0.061\left(\frac{k}{D}\right)^{0.31} G^{0.69}$
6.	Air flow over single spheres.	$h = 0.040\,\frac{G^{0.52}}{D^{0.48}}$ $0 < t < 250$ F
7.	Air flow over plane surfaces.	$h = 1 + 0.22\,V_s$ For $V_s < 16$ ft per second or $h = 0.53\,V_s^{0.8}$ 16 ft per second $< V_s <$ 100 ft per second
8.	Air flow normal to finned cylinders.	$h = 6.2\left(\frac{G}{3600}\right)^{0.8}\frac{t_f^{0.32}}{D^{0.52}}$ $0 < t < 250$ F
	FREE CONVECTION[d]	
9.	Single horizontal right circular cylinder in air.	$h = 0.23\left(\frac{\Delta t}{D}\right)^{0.25}$
10.	Single vertical right circular cylinder in air.	$h = 0.22\left(\frac{\Delta t}{D}\right)^{0.25}$
11.	Vertical surfaces in air.	$h = 0.3\,(\Delta t)^{0.25}$
12.	Top surface of horizontal plates to air.	$h = 0.4\,(\Delta t)^{0.25}$
13.	Bottom surface of horizontal plates to air.	$h = 0.2\,(\Delta t)^{0.25}$

[a]Heat Transmission, by W. H. McAdams.
[b]Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{surface} + t_{fluid})$ divided by 2.
[c]These expressions are applicable to longitudinal flow in other than right circular cylinders provided the hydraulic radius is employed as the conduit dimension parameter. For right circular cylinders $4r_H = D$.
[d]For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates increases towards 0.33. The following equations employing an exponent equal to 0.25 are applicable in the intermediate range.

NOMENCLATURE AND DIMENSIONS FOR TABLE 7

c_p = fluid unit heat capacity at constant pressure, Btu per pound per degree Fahrenheit.
D = cylinder diameter, feet.
$G = V_s\gamma$ = fluid mass velocity, pounds per hour per square foot of flow cross section.
γ = density, pounds per cubic foot.
h = unit conductance for thermal convection, Btu per hour per square foot per degree Fahrenheit.
k = unit thermal conductivity of the fluid, Btu per hour per square foot per degree Fahrenheit for one foot thickness.
r_H = hydraulic radius of the flow cross section[c].
= flow cross section area per wetted perimeter, feet.
s = fin spacing, feet.
t = average fluid film temperature, degree Fahrenheit.
Δt = temperature difference surface to main fluid, degree Fahrenheit.
V_s = fluid velocity, foot per second.
μ = fluid viscosity, pounds per hour per foot.
viscosity in centipoises × 2.42 = viscosity in pounds per hour per foot.

be calculated by means of Equation 3b. For the purposes of this illustrative problem it will be assumed that the insulated pipe is exposed to (*sees*) surroundings, which exist at 120 F. For this condition case 2, Table 8 is applicable. Then the angle factor, F_A, is unity and for an estimated surface emissivity of 0.9 (see Table 9), $F_e = 0.9$. As a first approximation the insulation surface temperature will be estimated as 20 F lower than the surroundings at 120 F. Thus $t_{ave} = 110$ F and $T_{ave} = (460 + 110)$ degree absolute temperature. (Note that even if the assumed temperature difference is in error by 10 F this fact will only affect T_{ave} by 2 per cent. Then by Equation 3b:

$h_r = 1730 \times 10^{-12} \times 1 \times 0.9 \times 4\,(460 + 110)^3 = 1.15$ Btu per hour per square foot per degree Fahrenheit.

The outside surface area of the insulation associated with 1 ft of pipe length was previously calculated as 1.14 sq ft. Thus:

$$R_r = \frac{1}{1.15 \times 1.14} = 0.76 \text{ hr degree Fahrenheit per Btu.}$$

TABLE 8. ANGLE FACTORS, F_A, AND EMISSIVITY FACTORS, F_E, FOR TWO RADIANT HEAT TRANSFER SYSTEMS[a]

CASE	SYSTEM	F_A	F_E
1.	Large parallel planes	1	$\frac{1}{e_1} + \frac{1}{e_2} - 1$
2.	Completely enclosed body small compared to the enclosing body. Enclosed body of area A_1.	1	e_1

[a]Solutions for other systems are available in Heat Transmission, by W. H. McAdams. e denotes the surface emissivity, see Table 9.