Chapter 13
HEAT EXCHANGERS

Types of Heat Exchangers

13-1C Heat exchangers are classified according to the flow type as parallel flow, counter flow, and cross-flow arrangement. In parallel flow, both the hot and cold fluids enter the heat exchanger at the same end and move in the same direction. In counter-flow, the hot and cold fluids enter the heat exchanger at opposite ends and flow in opposite direction. In cross-flow, the hot and cold fluid streams move perpendicular to each other.

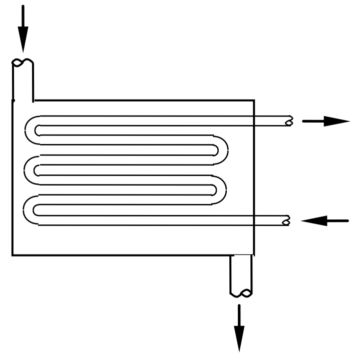
13-2C In terms of construction type, heat exchangers are classified as compact, shell and tube and regenerative heat exchangers. Compact heat exchangers are specifically designed to obtain large heat transfer surface areas per unit volume. The large surface area in compact heat exchangers is obtained by attaching closely spaced thin plate or corrugated fins to the walls separating the two fluids. Shell and tube heat exchangers contain a large number of tubes packed in a shell with their axes parallel to that of the shell. Regenerative heat exchangers involve the alternate passage of the hot and cold fluid streams through the same flow area. In compact heat exchangers, the two fluids usually move perpendicular to each other.

13-3C A heat exchanger is classified as being compact if $\beta > 700\ m^2/m^3$ or ($200\ ft^2/ft^3$) where β is the ratio of the heat transfer surface area to its volume which is called the area density. The area density for double-pipe heat exchanger can not be in the order of 700. Therefore, it can not be classified as a compact heat exchanger.

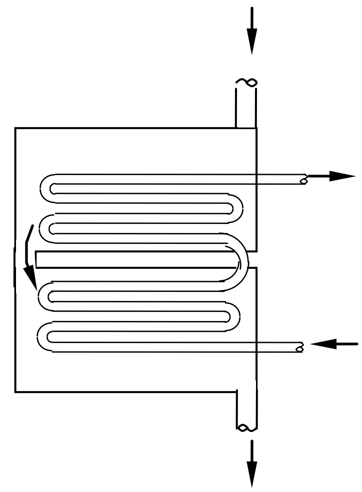
13-4C In counter-flow heat exchangers, the hot and the cold fluids move parallel to each other but both enter the heat exchanger at opposite ends and flow in opposite direction. In cross-flow heat exchangers, the two fluids usually move perpendicular to each other. The cross-flow is said to be unmixed when the plate fins force the fluid to flow through a particular interfin spacing and prevent it from moving in the transverse direction. When the fluid is free to move in the transverse direction, the cross-flow is said to be mixed.

13-5C In the shell and tube exchangers, baffles are commonly placed in the shell to force the shell side fluid to flow across the shell to enhance heat transfer and to maintain uniform spacing between the tubes. Baffles disrupt the flow of fluid, and an increased pumping power will be needed to maintain flow. On the other hand, baffles eliminate dead spots and increase heat transfer rate.

13-6C Using six-tube passes in a shell and tube heat exchanger increases the heat transfer surface area, and the rate of heat transfer increases. But it also increases the manufacturing costs.



13-7C Using so many tubes increases the heat transfer surface area which in turn increases the rate of heat transfer.



13-8C Regenerative heat exchanger involves the alternate passage of the hot and cold fluid streams through the same flow area. The static type regenerative heat exchanger is basically a porous mass which has a large heat storage capacity, such as a ceramic wire mash. Hot and cold fluids flow through this porous mass alternately. Heat is transferred from the hot fluid to the matrix of the regenerator during the flow of the hot fluid and from the matrix to the cold fluid. Thus the matrix serves as a temporary heat storage medium. The dynamic type regenerator involves a rotating drum and continuous flow of the hot and cold fluid through different portions of the drum so that any portion of the drum passes periodically through the hot stream, storing heat and then through the cold stream, rejecting this stored heat. Again the drum serves as the medium to transport the heat from the hot to the cold fluid stream.

The Overall Heat Transfer Coefficient

13-9C Heat is first transferred from the hot fluid to the wall by convection, through the wall by conduction and from the wall to the cold fluid again by convection.

13-10C When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, which is usually the case, the thermal resistance of the tube is negligible.

13-11C The heat transfer surface areas are $A_i = \pi D_1 L$ and $A_o = \pi D_2 L$. When the thickness of inner tube is small, it is reasonable to assume $A_i \cong A_o \cong A_s$.

13-12C No, it is not reasonable to say $h_i \approx h_0 \approx h$

13-13C When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, the thermal resistance of the tube is negligible and the inner and the outer surfaces of the tube are almost identical ($A_i \cong A_o \cong A_s$). Then the overall heat transfer coefficient of a heat exchanger can be determined to from $U = (1/h_i + 1/h_o)^{-1}$

13-14C None.

13-15C When one of the convection coefficients is much smaller than the other $h_i << h_o$, and $A_i \approx A_0 \approx A_s$. Then we have ($1/h_i >> 1/h_o$) and thus $U_i = U_0 = U \cong h_i$.

13-16C The most common type of fouling is the precipitation of solid deposits in a fluid on the heat transfer surfaces. Another form of fouling is corrosion and other chemical fouling. Heat exchangers may also be fouled by the growth of algae in warm fluids. This type of fouling is called the biological fouling. Fouling represents additional resistance to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease, and the pressure drop to increase.

13-17C The effect of fouling on a heat transfer is represented by a fouling factor R_f. Its effect on the heat transfer coefficient is accounted for by introducing a thermal resistance R_f/A_s. The fouling increases with increasing temperature and decreasing velocity.

13-18 The heat transfer coefficients and the fouling factors on tube and shell side of a heat exchanger are given. The thermal resistance and the overall heat transfer coefficients based on the inner and outer areas are to be determined.

Assumptions **1** The heat transfer coefficients and the fouling factors are constant and uniform.

Analysis (*a*) The total thermal resistance of the heat exchanger per unit length is

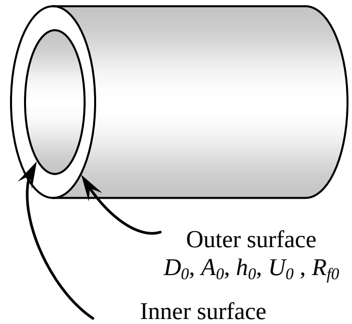
$$R = \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{\ln(D_o / D_i)}{2\pi kL} + \frac{R_{fo}}{A_o} + \frac{1}{h_o A_o}$$

$$R = \frac{1}{(700\,\text{W/m}^2.°\text{C})[\pi(0.012\,\text{m})(1\text{m})]} + \frac{(0.0005\,\text{m}^2.°\text{C/W})}{[\pi(0.012\,\text{m})(1\text{m})]}$$

$$+ \frac{\ln(1.6/1.2)}{2\pi(380\,\text{W/m.°C})(1\,\text{m})} + \frac{(0.0002\,\text{m}^2.°\text{C/W})}{[\pi(0.016\,\text{m})(1\text{m})]}$$

$$+ \frac{1}{(700\,\text{W/m}^2.°\text{C})[\pi(0.016\,\text{m})(1\text{m})]}$$

$$= \mathbf{0.0837\,°C/W}$$



(*b*) The overall heat transfer coefficient based on the inner and the outer surface areas of the tube per length are

$$R = \frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o}$$

$$U_i = \frac{1}{RA_i} = \frac{1}{(0.0837°\text{C/W})[\pi(0.012\,\text{m})(1\text{m})]} = \mathbf{317\ W/m^2.°C}$$

$$U_o = \frac{1}{RA_o} = \frac{1}{(0.0837°\text{C/W})[\pi(0.016\,\text{m})(1\text{m})]} = \mathbf{238\ W/m^2.°C}$$

13-19 "!PROBLEM 13-19"

```
"GIVEN"
k=380 "[W/m-C], parameter to be varied"
D_i=0.012 "[m]"
D_o=0.016 "[m]"
D_2=0.03 "[m]"
h_i=700 "[W/m^2-C], parameter to be varied"
h_o=1400 "[W/m^2-C], parameter to be varied"
R_f_i=0.0005 "[m^2-C/W]"
R_f_o=0.0002 "[m^2-C/W]"

"ANALYSIS"
R=1/(h_i*A_i)+R_f_i/A_i+ln(D_o/D_i)/(2*pi*k*L)+R_f_o/A_o+1/(h_o*A_o)
L=1 "[m], a unit length of the heat exchanger is considered"
A_i=pi*D_i*L
A_o=pi*D_o*L
```

k [W/m-C]	R [C/W]
10	0.07392
30.53	0.07085
51.05	0.07024
71.58	0.06999
92.11	0.06984
112.6	0.06975
133.2	0.06969
153.7	0.06964
174.2	0.06961
194.7	0.06958
215.3	0.06956
235.8	0.06954
256.3	0.06952
276.8	0.06951
297.4	0.0695
317.9	0.06949
338.4	0.06948
358.9	0.06947
379.5	0.06947
400	0.06946

h_i [W/m^2-C]	R [C/W]
500	0.08462
550	0.0798
600	0.07578
650	0.07238
700	0.06947
750	0.06694
800	0.06473
850	0.06278
900	0.06105
950	0.05949
1000	0.0581
1050	0.05684
1100	0.05569
1150	0.05464
1200	0.05368
1250	0.05279
1300	0.05198
1350	0.05122
1400	0.05052
1450	0.04987
1500	0.04926

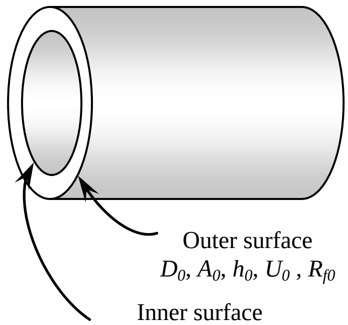
h_o [W/m^2-C]	R [C/W]
1000	0.07515
1050	0.0742
1100	0.07334
1150	0.07256
1200	0.07183
1250	0.07117
1300	0.07056
1350	0.06999
1400	0.06947
1450	0.06898
1500	0.06852
1550	0.06809
1600	0.06769
1650	0.06731
1700	0.06696
1750	0.06662
1800	0.06631
1850	0.06601
1900	0.06573
1950	0.06546
2000	0.0652

13-20 Water flows through the tubes in a boiler. The overall heat transfer coefficient of this boiler based on the inner surface area is to be determined.

Assumptions **1** Water flow is fully developed. **2** Properties of the water are constant.

Properties The properties water at 107°C ≈ 110°C are (Table A-9)

$$\upsilon = \mu / \rho = 0.268\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.682\ \text{W/m}^2.\text{K}$$
$$\text{Pr} = 1.58$$



Analysis The Reynolds number is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(3.5\ \text{m/s})(0.01\text{m})}{0.268\times10^{-6}\ \text{m}^2/\text{s}} = 130{,}600$$

which is greater than 10,000. Therefore, the flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(130{,}600)^{0.8}(1.58)^{0.4} = 342$$

and $$h = \frac{k}{D_h}Nu = \frac{0.682\ \text{W/m.°C}}{0.01\ \text{m}}(342) = 23{,}324\ \text{W/m}^2.\text{°C}$$

The total resistance of this heat exchanger is then determined from

$$R = R_{total} = R_i + R_{wall} + R_o = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_o A_o}$$
$$= \frac{1}{(23{,}324\ \text{W/m}^2.\text{°C})[\pi(0.01\text{m})(5\text{m})]} + \frac{\ln(1.4/1)}{[2\pi(14.2\ \text{W/m.°C})(5\ \text{m})]}$$
$$+ \frac{1}{(8400\ \text{W/m}^2.\text{°C})[\pi(0.014\text{m})(5\text{m})]}$$
$$= 0.00157\text{°C/W}$$

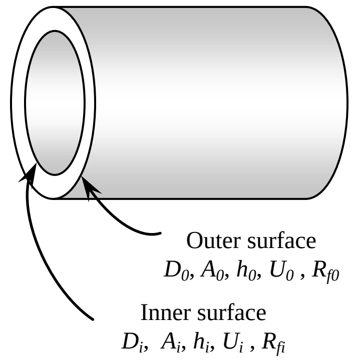
and $$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.00157\text{°C/W})[\pi(0.01\text{m})(5\text{m})]} = \mathbf{4055\ W/m^2.°C}$$

13-21 Water is flowing through the tubes in a boiler. The overall heat transfer coefficient of this boiler based on the inner surface area is to be determined.

Assumptions **1** Water flow is fully developed. **2** Properties of water are constant. **3** The heat transfer coefficient and the fouling factor are constant and uniform.

Properties The properties water at 107°C ≈ 110°C are (Table A-9)

$$\upsilon = \mu / \rho = 0.268\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.682\ \text{W/m}^2.\text{K}$$
$$\text{Pr} = 1.58$$



Analysis The Reynolds number is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(3.5\ \text{m/s})(0.01\text{m})}{0.268\times10^{-6}\ \text{m}^2/\text{s}} = 130{,}600$$

which is greater than 10,000. Therefore, the flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\ \text{Pr}^{0.4} = 0.023(130{,}600)^{0.8}(1.58)^{0.4} = 342$$

and $$h = \frac{k}{D_h} Nu = \frac{0.682\ \text{W/m.°C}}{0.01\ \text{m}}(342) = 23{,}324\ \text{W/m}^2.\text{°C}$$

The thermal resistance of heat exchanger with a fouling factor of $R_{f,i} = 0.0005\ \text{m}^2.\text{° C / W}$ is determined from

$$R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o / D_i)}{2\pi kL} + \frac{1}{h_o A_o}$$

$$R = \frac{1}{(23{,}324\ \text{W/m}^2.\text{°C})[\pi(0.01\,\text{m})(5\,\text{m})]} + \frac{0.0005\text{m}^2.\text{°C/W}}{[\pi(0.01\,\text{m})(5\,\text{m})]}$$
$$+ \frac{\ln(1.4/1)}{2\pi(14.2\ \text{W/m.°C})(5\,\text{m})} + \frac{1}{(8400\,\text{W/m}^2.\text{°C})[\pi(0.014\,\text{m})(5\,\text{m})]}$$
$$= 0.00476\text{°C/W}$$

Then,

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.00476\text{°C/W})[\pi(0.01\text{m})(5\,\text{m})]} = \mathbf{1337\ W/m^2.°C}$$

13-22 "!PROBLEM 13-22"

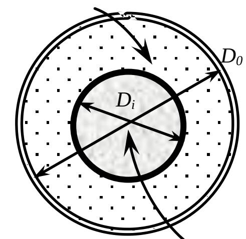
```
"GIVEN"
T_w=107 "[C]"
Vel=3.5 "[m/s]"
L=5 "[m]"
k_pipe=14.2 "[W/m-C]"
D_i=0.010 "[m]"
D_o=0.014 "[m]"
h_o=8400 "[W/m^2-C]"
"R_f_i=0.0005 [m^2-C/W], parameter to be varied"

"PROPERTIES"
k=conductivity(Water, T=T_w, P=300)
Pr=Prandtl(Water, T=T_w, P=300)
rho=density(Water, T=T_w, P=300)
mu=viscosity(Water, T=T_w, P=300)
nu=mu/rho

"ANALYSIS"
Re=(Vel*D_i)/nu
"Re is calculated to be greater than 4000. Therefore, the flow is turbulent."
Nusselt=0.023*Re^0.8*Pr^0.4
h_i=k/D_i*Nusselt
A_i=pi*D_i*L
A_o=pi*D_o*L
R=1/(h_i*A_i)+R_f_i/A_i+ln(D_o/D_i)/(2*pi*k_pipe*L)+1/(h_o*A_o)
U_i=1/(R*A_i)
```

$R_{f,i}$ [m^2-C/W]	U_i [W/m^2-C]
0.0001	2883
0.00015	2520
0.0002	2238
0.00025	2013
0.0003	1829
0.00035	1675
0.0004	1546
0.00045	1435
0.0005	1339
0.00055	1255
0.0006	1181
0.00065	1115
0.0007	1056
0.00075	1003
0.0008	955.2

13-23 Refrigerant-134a is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions **1** The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. **2** Both the water and refrigerant-134a flow are fully developed. **3** Properties of the water and refrigerant-134a are constant.

Properties The properties water at 20°C are (Table A-9)



$$\rho = 998\,\text{kg/m}^3$$
$$\upsilon = \mu/\rho = 1.004\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.598\,\text{W/m.°C}$$
$$\text{Pr} = 7.01$$

Analysis The hydraulic diameter for annular space is

$$D_h = D_o - D_i = 0.025 - 0.01 = 0.015\ \text{m}$$

The average velocity of water in the tube and the Reynolds number are

$$V_m = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho\left(\pi\dfrac{D_o^{\,2} - D_i^{\,2}}{4}\right)} = \frac{0.3\,\text{kg/s}}{(998\,\text{kg/m}^3)\left(\pi\dfrac{(0.025\,\text{m})^2 - (0.01\,\text{m})^2}{4}\right)} = 0.729\,\text{m/s}$$

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(0.729\,\text{m/s})(0.015\text{m})}{1.004\times10^{-6}\ \text{m}^2/\text{s}} = 10{,}890$$

which is greater than 10,000. Therefore flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(10{,}890)^{0.8}(7.01)^{0.4} = 85.0$$

and

$$h_o = \frac{k}{D_h}Nu = \frac{0.598\,\text{W/m.°C}}{0.015\,\text{m}}(85.0) = 3390\,\text{W/m}^2.\text{°C}$$

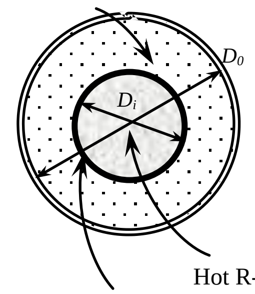
Then the overall heat transfer coefficient becomes

$$U = \frac{1}{\dfrac{1}{h_i} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{5000\,\text{W/m}^2.\text{°C}} + \dfrac{1}{3390\,\text{W/m}^2.\text{°C}}} = \mathbf{2020\ W/m^2.°C}$$

13-24 Refrigerant-134a is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions **1** The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. **2** Both the water and refrigerant-134a flows are fully developed. **3** Properties of the water and refrigerant-134a are constant. **4** The limestone layer can be treated as a plain layer since its thickness is very small relative to its diameter.

Properties The properties water at 20°C are (Table A-9)



$$\rho = 998\,\text{kg/m}^3$$
$$\upsilon = \mu/\rho = 1.004\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.598\,\text{W/m.°C}$$
$$\text{Pr} = 7.01$$

Analysis The hydraulic diameter for annular space is

$$D_h = D_o - D_i = 0.025 - 0.01 = 0.015\ \text{m}$$

The average velocity of water in the tube and the Reynolds number are

$$V_m = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho\left(\pi\dfrac{D_o^{\,2} - D_i^{\,2}}{4}\right)} = \frac{0.3\,\text{kg/s}}{(998\,\text{kg/m}^3)\left(\pi\dfrac{(0.025\,\text{m})^2 - (0.01\,\text{m})^2}{4}\right)} = 0.729\,\text{m/s}$$

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(0.729\,\text{m/s})(0.015\text{m})}{1.004\times10^{-6}\ \text{m}^2/\text{s}} = 10{,}890$$

which is greater than 10,000. Therefore flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(10{,}890)^{0.8}(7.01)^{0.4} = 85.0$$

and

$$h_o = \frac{k}{D_h}Nu = \frac{0.598\,\text{W/m.°C}}{0.015\,\text{m}}(85.0) = 3390\,\text{W/m}^2\text{.°C}$$

Disregarding the curvature effects, the overall heat transfer coefficient is determined to be

$$U = \frac{1}{\dfrac{1}{h_i} + \left(\dfrac{L}{k}\right)_{\text{limeston}} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{5000\,\text{W/m}^2\text{.°C}} + \dfrac{0.002\,\text{m}}{1.3\,\text{W/m.°C}} + \dfrac{1}{3390\,\text{W/m}^2\text{.°C}}} = \mathbf{493\ W/m^2.°C}$$

13-25 "!PROBLEM 13-25"

```
"GIVEN"
D_i=0.010 "[m]"
D_o=0.025 "[m]"
T_w=20 "[C]"
h_i=5000 "[W/m^2-C]"
m_dot=0.3 "[kg/s]"
"L_limestone=2 [mm], parameter to be varied"
k_limestone=1.3 "[W/m-C]"

"PROPERTIES"
k=conductivity(Water, T=T_w, P=100)
Pr=Prandtl(Water, T=T_w, P=100)
rho=density(Water, T=T_w, P=100)
mu=viscosity(Water, T=T_w, P=100)
nu=mu/rho

"ANALYSIS"
D_h=D_o-D_i
Vel=m_dot/(rho*A_c)
A_c=pi*(D_o^2-D_i^2)/4
Re=(Vel*D_h)/nu
"Re is calculated to be greater than 4000. Therefore, the flow is turbulent."
Nusselt=0.023*Re^0.8*Pr^0.4
h_o=k/D_h*Nusselt
U=1/(1/h_i+(L_limestone*Convert(mm, m))/k_limestone+1/h_o)
```

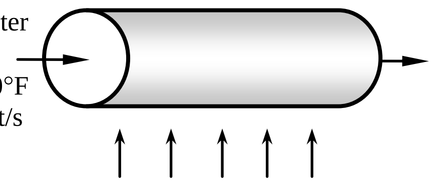
$L_{limestone}$ [mm]	U [W/m^2-C]
1	791.4
1.1	746
1.2	705.5
1.3	669.2
1.4	636.4
1.5	606.7
1.6	579.7
1.7	554.9
1.8	532.2
1.9	511.3
2	491.9
2.1	474
2.2	457.3
2.3	441.8
2.4	427.3
2.5	413.7
2.6	400.9
2.7	388.9
2.8	377.6
2.9	367
3	356.9

13-26E Water is cooled by air in a cross-flow heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions **1** The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. **2** Both the water and air flow are fully developed. **3** Properties of the water and air are constant.

Properties The properties water at 140°F are (Table A-9E)

$$k = 0.378\ \text{Btu/h.ft.°F}$$
$$\upsilon = 5.11\times10^{-6}\ \text{ft}^2/\text{s}$$
$$\Pr = 2.98$$



The properties of air at 80°F are (Table A-18E)

$$k = 0.0150\ \text{Btu/h.ft.°F}$$
$$\upsilon = 0.17\times10^{-3}\ \text{ft}^2/\text{s}$$
$$\Pr = 0.72$$

Analysis The overall heat transfer coefficient can be determined from

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$$

The Reynolds number of water is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(8\ \text{ft/s})[0.75/12\ \text{ft}]}{5.11\times10^{-6}\ \text{ft}^2/\text{s}} = 97{,}850$$

which is greater than 10,000. Therefore the flow of water is turbulent. Assuming the flow to be fully developed, the Nusselt number is determined from

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\Pr^{0.4} = 0.023(97{,}850)^{0.8}(2.98)^{0.4} = 350$$

and $$h_i = \frac{k}{D_h}Nu = \frac{0.378\ \text{Btu/h.ft.°F}}{0.75/12\ \text{ft}}(350) = 2117\ \text{Btu/h.ft}^2\text{.°F}$$

The Reynolds number of air is

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{(12\ \text{ft/s})[3/(4\times12)\ \text{ft}]}{0.17\times10^{-3}\ \text{ft}^2/\text{s}} = 4412$$

The flow of air is across the cylinder. The proper relation for Nusselt number in this case is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\Pr^{1/3}}{\left[1+(0.4/\Pr)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(4412)^{0.5}(0.729)^{1/3}}{\left[1+(0.4/0.729)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4412}{282{,}000}\right)^{5/8}\right]^{4/5} = 34.8$$

and $$h_o = \frac{k}{D}Nu = \frac{0.01481\ \text{Btu/h.ft.°F}}{0.75/12\ \text{ft}}(34.8) = 8.25\ \text{Btu/h.ft}^2\text{.°F}$$

Then the overall heat transfer coefficient becomes

$$U = \frac{1}{\dfrac{1}{h_i}+\dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{2117\ \text{Btu/h.ft}^2\text{.°F}}+\dfrac{1}{8.25\ \text{Btu/h.ft}^2\text{.°F}}} = \mathbf{8.22\ Btu/h.ft^2.°F}$$

Analysis of Heat Exchangers

13-27C The heat exchangers usually operate for long periods of time with no change in their operating conditions, and then they can be modeled as steady-flow devices. As such , the mass flow rate of each fluid remains constant and the fluid properties such as temperature and velocity at any inlet and outlet remain constant. The kinetic and potential energy changes are negligible. The specific heat of a fluid can be treated as constant in a specified temperature range. Axial heat conduction along the tube is negligible. Finally, the outer surface of the heat exchanger is assumed to be perfectly insulated so that there is no heat loss to the surrounding medium and any heat transfer thus occurs is between the two fluids only.

13-28C That relation is valid under steady operating conditions, constant specific heats, and negligible heat loss from the heat exchanger.

13-29C The product of the mass flow rate and the specific heat of a fluid is called the heat capacity rate and is expressed as $C = \dot{m}C_p$. When the heat capacity rates of the cold and hot fluids are equal, the temperature change is the same for the two fluids in a heat exchanger. That is, the temperature rise of the cold fluid is equal to the temperature drop of the hot fluid. A heat capacity of infinity for a fluid in a heat exchanger is experienced during a phase-change process in a condenser or boiler.

13-30C The mass flow rate of the cooling water can be determined from $\dot{Q} = (\dot{m}C_p\Delta T)_{\text{cooling water}}$. The rate of condensation of the steam is determined from $\dot{Q} = (\dot{m}h_{fg})_{\text{steam}}$, and the total thermal resistance of the condenser is determined from $R = \dot{Q} / \Delta T$.

13-31C When the heat capacity rates of the cold and hot fluids are identical, the temperature rise of the cold fluid will be equal to the temperature drop of the hot fluid.