*Solutions Manual*

for

**Introduction to Thermodynamics and Heat Transfer**

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# Chapter 16

# HEAT EXCHANGERS

## Types of Heat Exchangers

**16-1C** 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.

**16-2C** 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.

**16-3C** A heat exchanger is classified as being compact if $\beta > 700\ m^2/m^3$ or ($200\ ft^2/ft^3$) where $\beta$ 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.

**16-4C** 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.

**16-5C** 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.

**16-6C** 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.

**16-7C** Using so many tubes increases the heat transfer surface area which in turn increases the rate of heat transfer.

**16-8C** 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

**16-9C** 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.

**16-10C** 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.

**16-11C** 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 $A_i \cong A_o \cong A_s$.

**16-12C** No, it is not reasonable to say $h_i \approx h_0 \approx h$

**16-13C** 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 $U = (1/h_i + 1/h_o)^{-1}$

**16-14C** None.

**16-15C** 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 $U_i = U_0 = U \cong h_i$.

**16-16C** 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.

**16-17C** 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.

**16-18** 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

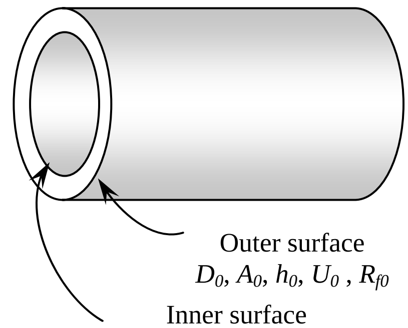
$$R = \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{\ln(D_o / D_i)}{2\pi kL} + \frac{R_{fo}}{A_o} + \frac{1}{h_o A_o}$$

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

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

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

$$= \mathbf{0.0837°C/W}$$




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

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

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

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

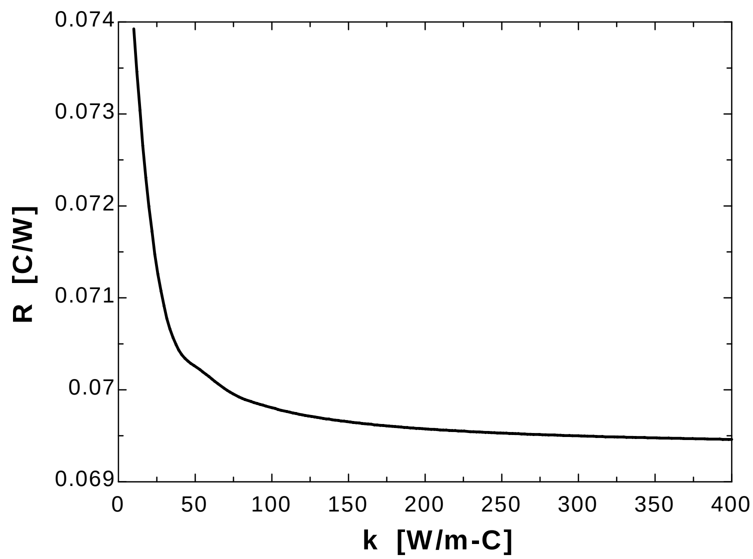
**16-19 EES** Prob. 16-18 is reconsidered. The effects of pipe conductivity and heat transfer coefficients on the thermal resistance of the heat exchanger are to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

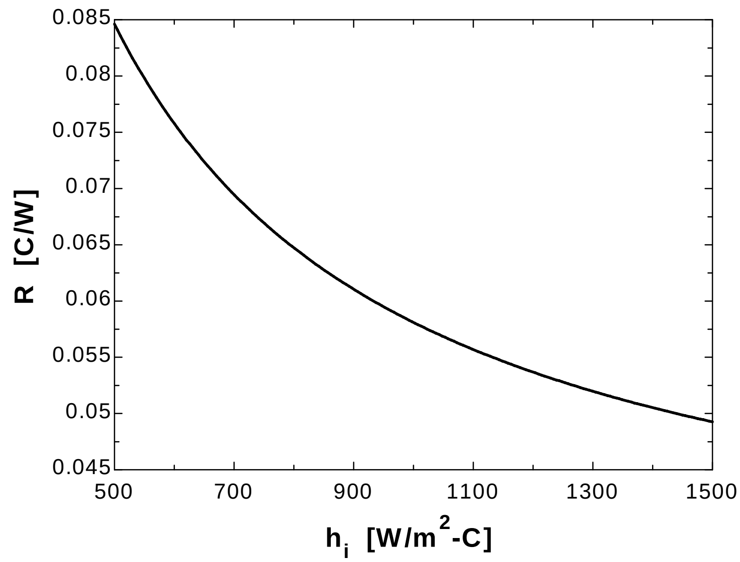
```
"GIVEN"
k=380 [W/m-C]
D_i=0.012 [m]
D_o=0.016 [m]
D_2=0.03 [m]
h_i=700 [W/m^2-C]
h_o=1400 [W/m^2-C]
R_f_i=0.0005 [m^2-C/W]
R_f_o=0.0002 [m^2-C/W]

"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
```

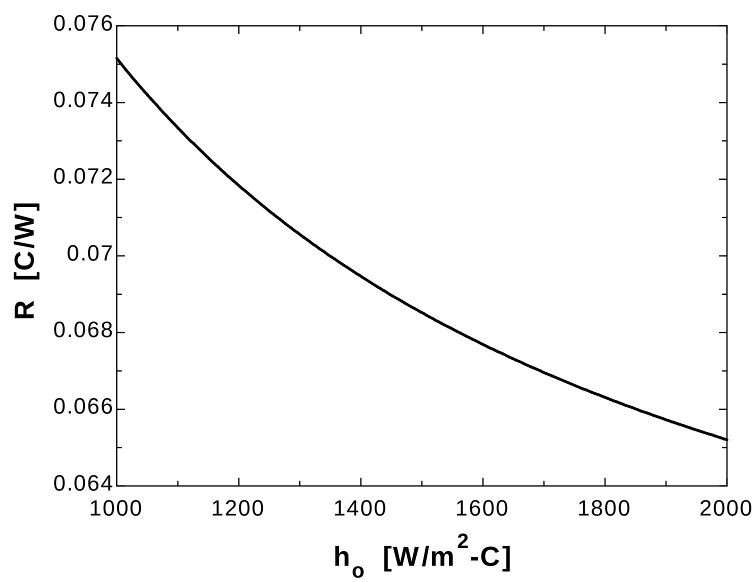
| k [W/m-C] | R [C/W] |
|---|---|
| 10 | 0.07392 |
| 30.53 | 0.07085 |
| 51.05 | 0.07024 |
| 71.58 | 0.06999 |
| 92.11 | 0.06984 |
| 112.6 | 0.06975 |
| 133.2 | 0.06969 |
| 153.7 | 0.06964 |
| 174.2 | 0.06961 |
| 194.7 | 0.06958 |
| 215.3 | 0.06956 |
| 235.8 | 0.06954 |
| 256.3 | 0.06952 |
| 276.8 | 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 |



**16-20** A water stream is heated by a jacketted-agitated vessel, fitted with a turbine agitator. The mass flow rate of water is to be determined.

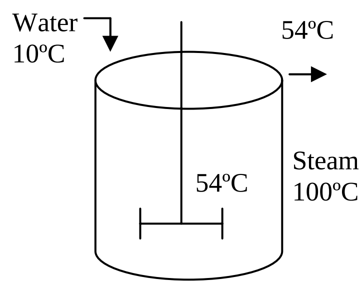
***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The properties of water at 54°C are (Table A-15)

$$k = 0.648 \text{ W/m.°C}$$
$$\rho = 985.8 \text{ kg/m}^3$$
$$\mu = 0.513\times10^{-3} \text{ kg/m}\cdot\text{s}$$
$$\text{Pr} = 3.31$$

The specific heat of water at the average temperature of (10+54)/2=32°C is 4178 J/kg.°C (Table A-15)

***Analysis*** We first determine the heat transfer coefficient on the inner wall of the vessel




$$\text{Re} = \frac{\dot{n}D_a^2\rho}{\mu} = \frac{(60/60\text{ s}^{-1})(0.2\text{ m})^2(985.8\text{ kg/m}^3)}{0.513\times10^{-3}\text{ kg/m}\cdot\text{s}} = 76{,}865$$

$$Nu = 0.76\,\text{Re}^{2/3}\,\text{Pr}^{1/3} = 0.76(76{,}865)^{2/3}(3.31)^{1/3} = 2048$$

$$h_j = \frac{k}{D_t}Nu = \frac{0.648\text{ W/m.°C}}{0.6\text{ m}}(2048) = 2211\text{ W/m}^2\text{.°C}$$

The heat transfer coefficient on the outer side is determined as follows

$$h_o = 13{,}100(T_g - T_w)^{-0.25} = 13{,}100(100 - T_w)^{-0.25}$$

$$h_o(T_g - T_w) = h_j(T_w - 54)$$

$$13{,}100(100 - T_w)^{-0.25}(100 - T_w) = 2211(T_w - 54)$$

$$13{,}100(100 - T_w)^{0.75} = 2211(T_w - 54)$$

$$\rightarrow T_w = 89.2\text{°C}$$

$$h_o = 13{,}100(100 - T_w)^{-0.25} = 13{,}100(100 - 89.2)^{-0.25} = 7226\text{ W/m}^2\cdot\text{C}$$

Neglecting the wall resistance and the thickness of the wall, the overall heat transfer coefficient can be written as

$$U = \left(\frac{1}{h_j} + \frac{1}{h_o}\right)^{-1} = \left(\frac{1}{2211} + \frac{1}{7226}\right)^{-1} = 1694\text{ W/m}^2\cdot\text{C}$$

From an energy balance

$$[\dot{m}c(T_{out} - T_{in})]_{\text{water}} = UA\Delta T$$

$$\dot{m}_w(4178)(54 - 10) = (1694)(\pi\times0.6\times0.6)(100 - 54)$$

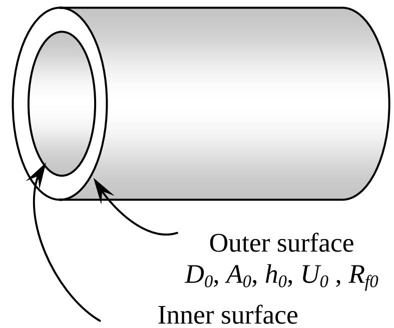
$$\dot{m}_w = 0.479\text{ kg/s} = \mathbf{1725\text{ kg/h}}$$

**16-21** 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 of water at 110°C are (Table A-15)

$$\nu = \mu / \rho = 0.268 \times 10^{-6} \text{ m}^2/\text{s}$$
$$k = 0.682 \text{ W/m}^2.\text{K}$$
$$\text{Pr} = 1.58$$




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3.5 \text{ m/s})(0.01 \text{ m})}{0.268 \times 10^{-6} \text{ m}^2/\text{s}} = 130{,}600$$

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

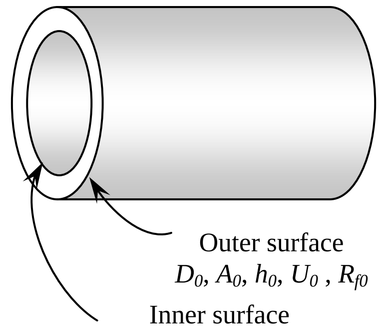
$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.00157\text{°C/W})[\pi(0.01 \text{ m})(5 \text{ m})]} = \mathbf{4055 \text{ W/m}^2.\text{°C}}$$

**16-22** 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 of water at 110°C are (Table A-15)

$$\nu = \mu / \rho = 0.268 \times 10^{-6} \text{ m}^2/\text{s}$$
$$k = 0.682 \text{ W/m}^2.\text{K}$$
$$\text{Pr} = 1.58$$




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3.5 \text{ m/s})(0.01 \text{ m})}{0.268 \times 10^{-6} \text{ m}^2/\text{s}} = 130{,}600$$

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

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

$$R = \frac{1}{(23{,}324 \text{ W/m}^2.\text{°C})[\pi(0.01 \text{ m})(5 \text{ m})]} + \frac{0.0005 \text{ m}^2.\text{°C/W}}{[\pi(0.01 \text{ m})(5 \text{ m})]}$$
$$+ \frac{\ln(1.4/1)}{2\pi(14.2 \text{ W/m.°C})(5 \text{ m})} + \frac{1}{(8400 \text{ W/m}^2.\text{°C})[\pi(0.014 \text{ m})(5 \text{ m})]}$$
$$= 0.00475\text{°C/W}$$

Then,

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.00475\text{°C/W})[\pi(0.01 \text{ m})(5 \text{ m})]} = \mathbf{1340 \text{ W/m}^2.\text{°C}}$$

**16-23 EES** Prob. 16-22 is reconsidered. The overall heat transfer coefficient based on the inner surface as a function of fouling factor is to be plotted.

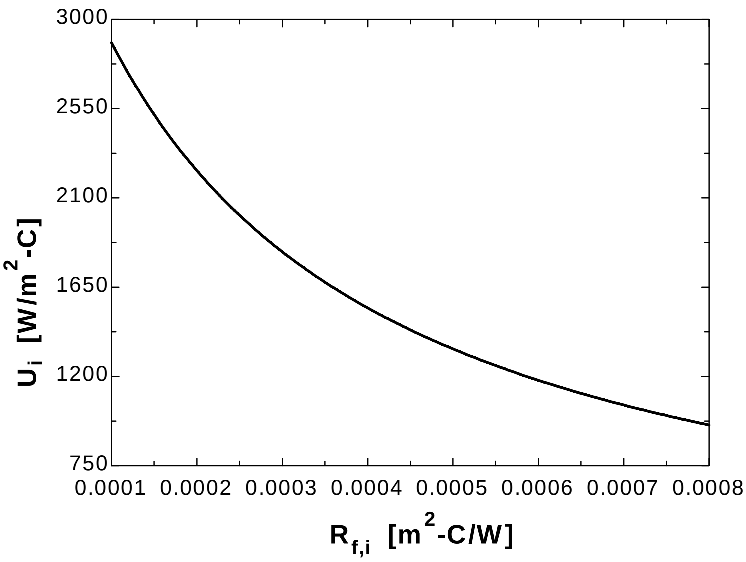
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_w=110 [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]

"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 10,000. 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)
```

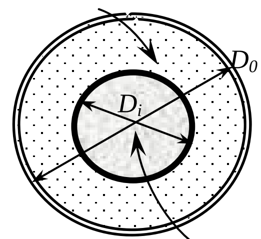
| $R_{f,i}$ [m$^2$-C/W] | $U_i$ [W/m$^2$-C] |
|---|---|
| 0.0001 | 2883 |
| 0.00015 | 2520 |
| 0.0002 | 2238 |
| 0.00025 | 2013 |
| 0.0003 | 1829 |
| 0.00035 | 1675 |
| 0.0004 | 1546 |
| 0.00045 | 1435 |
| 0.0005 | 1339 |
| 0.00055 | 1255 |
| 0.0006 | 1181 |
| 0.00065 | 1115 |
| 0.0007 | 1056 |
| 0.00075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



**16-24** 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 flow are fully developed. **3** Properties of the water and refrigerant-134a are constant.

***Properties*** The properties of water at 20°C are (Table A-15)




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

***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

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

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

which is greater than 4000. 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

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

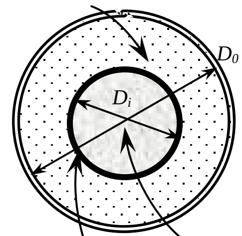
Then the overall heat transfer coefficient becomes

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

**16-25** 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 of water at 20°C are (Table A-15)

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

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

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

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

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

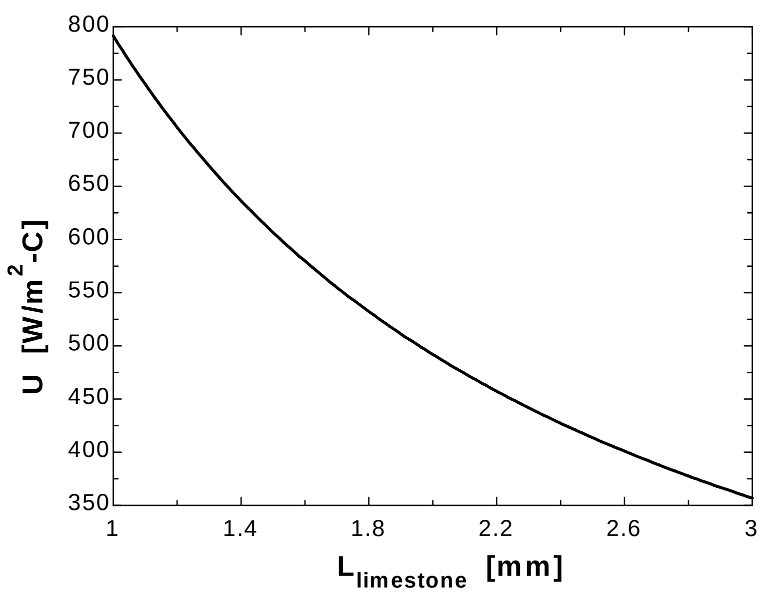
$$U = \frac{1}{\frac{1}{h_i} + \left(\frac{L}{k}\right)_{\text{limeston}} + \frac{1}{h_o}} = \frac{1}{\frac{1}{5000 \text{ W/m}^2\text{.°C}} + \frac{0.002 \text{ m}}{1.3 \text{ W/m.°C}} + \frac{1}{3390 \text{ W/m}^2\text{.°C}}} = \mathbf{493\ W/m^2.°C}$$

**16-26 EES** Prob. 16-25 is reconsidered. The overall heat transfer coefficient as a function of the limestone thickness is to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"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]
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 10,000. 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 |

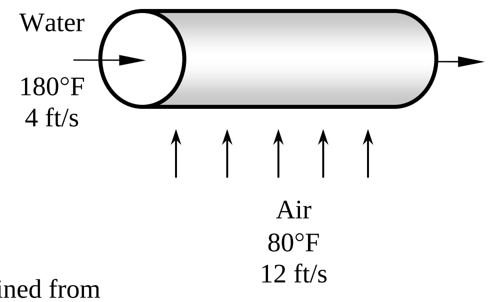


**16-27E** Water is cooled by air in a cross-flow heat exchanger. The overall heat transfer coefficient is to be determined.

***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 of water at 180°F are (Table A-15E)

$$k = 0.388\ \text{Btu/h.ft.°F}$$
$$\nu = 3.825\times10^{-6}\ \text{ft}^2/\text{s}$$
$$\text{Pr} = 2.15$$




The properties of air at 80°F are (Table A-22E)

$$k = 0.01481\ \text{Btu/h.ft.°F}$$
$$\nu = 1.697\times10^{-4}\ \text{ft}^2/\text{s}$$
$$\text{Pr} = 0.7290$$

***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

$$\text{Re} = \frac{V_{avg}D_h}{\nu} = \frac{(4\ \text{ft/s})[0.75/12\ \text{ft}]}{3.825\times10^{-6}\ \text{ft}^2/\text{s}} = 65{,}360$$

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(65{,}360)^{0.8}(2.15)^{0.4} = 222$$

and $$h_i = \frac{k}{D_h}Nu = \frac{0.388\ \text{Btu/h.ft.°F}}{0.75/12\ \text{ft}}(222) = 1378\ \text{Btu/h.ft}^2.\text{°F}$$

The Reynolds number of air is

$$\text{Re} = \frac{VD}{\nu} = \frac{(12\ \text{ft/s})[3/(4\times12)\ \text{ft}]}{1.697\times10^{-4}\ \text{ft}^2/\text{s}} = 4420$$

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

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

$$= 0.3 + \frac{0.62(4420)^{0.5}(0.7290)^{1/3}}{\left[1+(0.4/0.7290)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4420}{282{,}000}\right)^{5/8}\right]^{4/5} = 34.86$$

and $$h_o = \frac{k}{D}Nu = \frac{0.01481\ \text{Btu/h.ft.°F}}{0.75/12\ \text{ft}}(34.86) = 8.26\ \text{Btu/h.ft}^2.\text{°F}$$

Then the overall heat transfer coefficient becomes

$$U = \frac{1}{\dfrac{1}{h_i}+\dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{1378\ \text{Btu/h.ft}^2.\text{°F}}+\dfrac{1}{8.26\ \text{Btu/h.ft}^2.\text{°F}}} = \mathbf{8.21\ Btu/h.ft^2.°F}$$

## Analysis of Heat Exchangers

**16-28C** 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.

**16-29C** That relation is valid under steady operating conditions, constant specific heats, and negligible heat loss from the heat exchanger.

**16-30C** 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.

**16-31C** 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 $R = \dot{Q} / \Delta T$ .

**16-32C** 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.

### The Log Mean Temperature Difference Method

**16-33C** $\Delta T_{\text{lm}}$ is called the log mean temperature difference, and is expressed as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

where

$\Delta T_1 = T_{h,in} - T_{c,in}$ $\Delta T_2 = T_{h,out} - T_{c,out}$ for parallel-flow heat exchangers and

$\Delta T = T_{h,in} - T_{c,out}$ $\Delta T_2 = T_{h,out} - T_{c,in}$ for counter-flow heat exchangers

**16-34C** The temperature difference between the two fluids decreases from $\Delta T_1$ at the inlet to $\Delta T_2$ at the outlet, and arithmetic mean temperature difference is defined as $\Delta T_{\text{am}} = \frac{\Delta T_1 + \Delta T_2}{2}$. The logarithmic mean temperature difference $\Delta T_{\text{lm}}$ is obtained by tracing the actual temperature profile of the fluids along the heat exchanger, and is an exact representation of the average temperature difference between the hot and cold fluids. It truly reflects the exponential decay of the local temperature difference. The logarithmic mean temperature difference is always less than the arithmetic mean temperature.

**16-35C** $\Delta T_{\text{lm}}$ cannot be greater than both $\Delta T_1$ and $\Delta T_2$ because $\Delta T_{\text{ln}}$ is always less than or equal to $\Delta T_{\text{m}}$ (arithmetic mean) which can not be greater than both $\Delta T_1$ and $\Delta T_2$.

**16-36C** No, it cannot. When $\Delta T_1$ is less than $\Delta T_2$ the ratio of them must be less than one and the natural logarithms of the numbers which are less than 1 are negative. But the numerator is also negative in this case. When $\Delta T_1$ is greater than $\Delta T_2$, we obtain positive numbers at the both numerator and denominator.

**16-37C** In the parallel-flow heat exchangers the hot and cold fluids enter the heat exchanger at the same end, and the temperature of the hot fluid decreases and the temperature of the cold fluid increases along the heat exchanger. But the temperature of the cold fluid can never exceed that of the hot fluid. In case of the counter-flow heat exchangers the hot and cold fluids enter the heat exchanger from the opposite ends and the outlet temperature of the cold fluid may exceed the outlet temperature of the hot fluid.

**16-38C** The $\Delta T_{\text{lm}}$ will be greatest for double-pipe counter-flow heat exchangers.

**16-39C** The factor $F$ is called as correction factor which depends on the geometry of the heat exchanger and the inlet and the outlet temperatures of the hot and cold fluid streams. It represents how closely a heat exchanger approximates a counter-flow heat exchanger in terms of its logarithmic mean temperature difference. $F$ cannot be greater than unity.

**16-40C** In this case it is not practical to use the LMTD method because it requires tedious iterations. Instead, the effectiveness-NTU method should be used.

**16-41C** First heat transfer rate is determined from $\dot{Q} = \dot{m}c_p[T_{in} - T_{out}]$, $\Delta T_{\text{ln}}$ from $\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$, correction factor from the figures, and finally the surface area of the heat exchanger from $\dot{Q} = UAFDT_{lm,CF}$

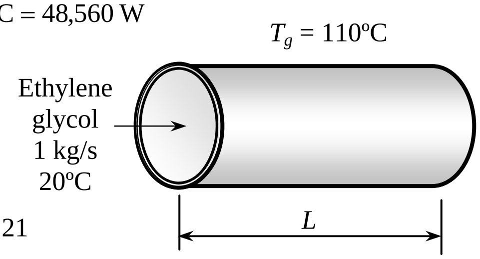
**16-42** Ethylene glycol is heated in a tube while steam condenses on the outside tube surface. The tube length is to be determined.

***Assumptions*** **1** Steady flow conditions exist. **2** The inner surfaces of the tubes are smooth. **3** Heat transfer to the surroundings is negligible.

***Properties*** The properties of ethylene glycol are given to be ρ = 1109 kg/m$^3$, $c_p$ = 2428 J/kg·K, $k$ = 0.253 W/m·K, μ = 0.01545 kg/m·s, Pr = 148.5. The thermal conductivity of copper is given to be 386 W/m·K.

***Analysis*** The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (1\,\text{kg/s})(2428\,\text{J/kg.°C})(40-20)\text{°C} = 48{,}560\,\text{W}$$




The fluid velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{1\,\text{kg/s}}{(1109\,\text{kg/m}^3)\left[\pi(0.02\,\text{m})^2/4\right]} = 2.870\,\text{m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(1109\,\text{kg/m}^3)(2.870\,\text{m/s})(0.02\,\text{m})}{0.01545\,\text{kg/m}\cdot\text{s}} = 4121$$

which is greater than 2300 and smaller than 10,000. Therefore, we have transitional flow. We assume fully developed flow and evaluate the Nusselt number from turbulent flow relation:

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(4121)^{0.8}(148.5)^{0.4} = 132.5$$

Heat transfer coefficient on the inner surface is

$$h_i = \frac{k}{D}Nu = \frac{0.253\,\text{W/m.°C}}{0.02\,\text{m}}(132.5) = 1677\,\text{W/m}^2.\text{°C}$$

Assuming a wall temperature of 100°C, the heat transfer coefficient on the outer surface is determined to be

$$h_o = 9200(T_g - T_w)^{-0.25} = 9200(110-100)^{-0.25} = 5174\,\text{W/m}^2.\text{°C}$$

Let us check if the assumption for the wall temperature holds:

$$h_i A_i (T_w - T_{b,\text{avg}}) = h_o A_o (T_g - T_w)$$

$$h_i \pi D_i L(T_w - T_{b,\text{avg}}) = h_o \pi D_o L(T_g - T_w)$$

$$1677\times 0.02(T_w - 30) = 5174\times 0.025(110 - T_w) \longrightarrow T_w = 93.5\text{°C}$$

Now we assume a wall temperature of 90°C:

$$h_o = 9200(T_g - T_w)^{-0.25} = 9200(110-90)^{-0.25} = 4350\,\text{W/m}^2.\text{°C}$$

Again checking, $1677\times 0.02(T_w - 30) = 4350\times 0.025(110 - T_w) \longrightarrow T_w = 91.1\text{°C}$

which is sufficiently close to the assumed value of 90°C. Now that both heat transfer coefficients are available, we use thermal resistance concept to find overall heat transfer coefficient based on the outer surface area as follows:

$$U_o = \frac{1}{\dfrac{D_o}{h_i D_i} + \dfrac{D_o \ln(D_2/D_1)}{2k_{\text{copper}}} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{0.025}{(1677)(0.02)} + \dfrac{(0.025)\ln(2.5/2)}{2(386)} + \dfrac{1}{4350}} = 1018\,\text{W/m}^2\cdot\text{C}$$

The rate of heat transfer can be expressed as

$$\dot{Q} = U_o A_o \Delta T_{\ln}$$

where the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{(T_g - T_e) - (T_g - T_i)}{\ln\left(\dfrac{T_g - T_e}{T_g - T_i}\right)} = \frac{(110-40)-(110-20)}{\ln\left(\dfrac{110-40}{110-20}\right)} = 79.58\text{°C}$$

Substituting, the tube length is determined to be

$$\dot{Q} = U_o A_o \Delta T_{\ln} \longrightarrow 48{,}560 = (1018)\pi(0.025)L(79.58) \longrightarrow L = \mathbf{7.63\ m}$$

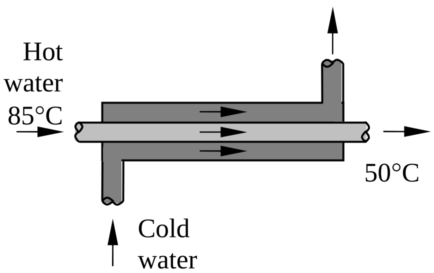
**16-43** Water is heated in a double-pipe, parallel-flow uninsulated heat exchanger by geothermal water. The rate of heat transfer to the cold water and the log mean temperature difference for this heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heat of hot water is given to be 4.25 kJ/kg.°C.

***Analysis*** The rate of heat given up by the hot water is

$$\dot{Q}_h = [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}}$$
$$= (1.4\text{ kg/s})(4.25\text{ kJ/kg.°C})(85\text{°C} - 50\text{°C}) = 208.3\text{ kW}$$




The rate of heat picked up by the cold water is

$$\dot{Q}_c = (1-0.03)\dot{Q}_h = (1-0.03)(208.3\text{ kW}) = \mathbf{202.0\text{ kW}}$$

The log mean temperature difference is

$$\dot{Q} = UA\Delta T_{lm} \longrightarrow \Delta T_{lm} == \frac{\dot{Q}}{UA} = \frac{202.0\text{ kW}}{(1.15\text{ kW/m}^2\cdot\text{°C})(4\text{ m}^2)} = \mathbf{43.9°C}$$

**16-44** A stream of hydrocarbon is cooled by water in a double-pipe counterflow heat exchanger. The overall heat transfer coefficient is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

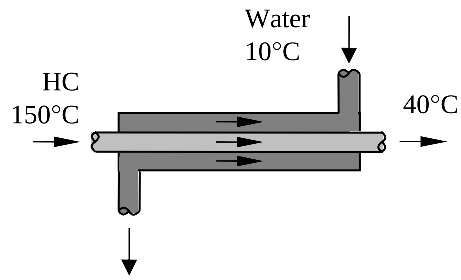
***Properties*** The specific heats of hydrocarbon and water are given to be 2.2 and 4.18 kJ/kg.°C, respectively.

***Analysis*** The rate of heat transfer is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{HC}} = (720/3600\text{ kg/s})(2.2\text{ kJ/kg.°C})(150\text{°C} - 40\text{°C}) = 48.4\text{ kW}$$

The outlet temperature of water is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{w}}$$
$$48.4\text{ kW} = (540/3600\text{ kg/s})(4.18\text{ kJ/kg.°C})(T_{\text{w,out}} - 10\text{°C})$$
$$T_{\text{w,out}} = 87.2\text{ °C}$$




The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 150\text{°C} - 87.2\text{°C} = 62.8\text{°C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 40\text{°C} - 10\text{°C} = 30\text{°C}$$

and $$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{62.8 - 30}{\ln(62.8/30)} = 44.4\text{°C}$$

The overall heat transfer coefficient is determined from

$$\dot{Q} = UA\Delta T_{lm}$$
$$48.4\text{ kW} = U(\pi \times 0.025 \times 6.0)(44.4\text{°C})$$
$$U = \mathbf{2.31\text{ kW/m}^2\cdot\text{K}}$$

**16-45** Oil is heated by water in a 1-shell pass and 6-tube passes heat exchanger. The rate of heat transfer and the heat transfer surface area are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heat of oil is given to be 2.0 kJ/kg.°C.

***Analysis*** The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{oil}} = (10\text{ kg/s})(2.0\text{ kJ/kg.°C})(46°\text{C} - 25°\text{C}) = \mathbf{420\ kW}$$

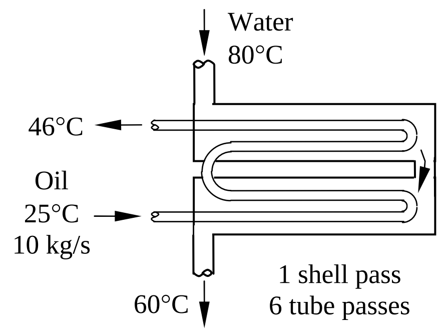
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80°\text{C} - 46°\text{C} = 34°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 25°\text{C} = 35°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{34 - 35}{\ln(34/35)} = 34.5°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{46 - 25}{80 - 25} = 0.38 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{80 - 60}{46 - 25} = 0.95 \end{aligned}\right\} F = 0.94$$




Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm,CF}} = \frac{420\text{ kW}}{(1.0\text{ kW/m}^2.°\text{C})(0.94)(34.5°\text{C})} = \mathbf{13.0\ m^2}$$

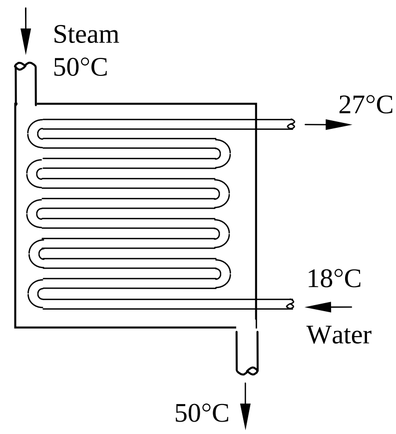
**16-46** Steam is condensed by cooling water in the condenser of a power plant. The mass flow rate of the cooling water and the rate of condensation are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The heat of vaporization of water at 50°C is given to be $h_{fg}$ = 2383 kJ/kg and specific heat of cold water at the average temperature of 22.5°C is given to be $c_p$ = 4180 J/kg.°C.

***Analysis*** The temperature differences between the steam and the cooling water at the two ends of the condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 50°\text{C} - 27°\text{C} = 23°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 50°\text{C} - 18°\text{C} = 32°\text{C}$$




and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{23-32}{\ln(23/32)} = 27.3°\text{C}$$

Then the heat transfer rate in the condenser becomes

$$\dot{Q} = UA_s \Delta T_{lm} = (2400\ \text{W/m}^2.°\text{C})(42\ \text{m}^2)(27.3°\text{C}) = 2752\ \text{kW}$$

The mass flow rate of the cooling water and the rate of condensation of steam are determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{cooling water}}$$

$$\dot{m}_{\substack{\text{cooling}\\\text{water}}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{2752\ \text{kJ/s}}{(4.18\ \text{kJ/kg.°C})(27°\text{C} - 18°\text{C})} = \mathbf{73.1\ kg/s}$$

$$\dot{Q} = (\dot{m}h_{fg})_{steam} \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{2752\ \text{kJ/s}}{2383\ \text{kJ/kg}} = \mathbf{1.15\ kg/s}$$

**16-47** Water is heated in a double-pipe parallel-flow heat exchanger by geothermal water. The required length of tube is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of water and geothermal fluid are given to be 4.18 and 4.31 kJ/kg.°C, respectively.

***Analysis*** The rate of heat transfer in the heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (0.2\text{ kg/s})(4.18\text{ kJ/kg.°C})(60\text{°C} - 25\text{°C}) = 29.26\text{ kW}$$

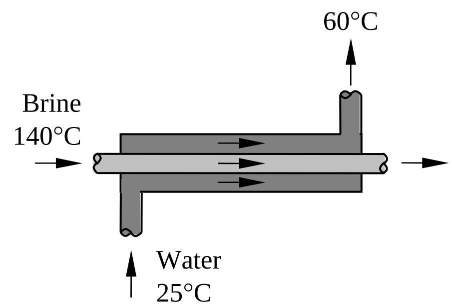
Then the outlet temperature of the geothermal water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{geot.water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 140\text{°C} - \frac{29.26\text{ kW}}{(0.3\text{ kg/s})(4.31\text{ kJ/kg.°C})} = 117.4\text{°C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,in} = 140\text{°C} - 25\text{°C} = 115\text{°C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = 117.4\text{°C} - 60\text{°C} = 57.4\text{°C}$$




and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{115 - 57.4}{\ln(115/57.4)} = 82.9\text{°C}$$

The surface area of the heat exchanger is determined from

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{29.26\text{ kW}}{(0.55\text{ kW/m}^2)(82.9\text{°C})} = 0.642\text{ m}^2$$

Then the length of the tube required becomes

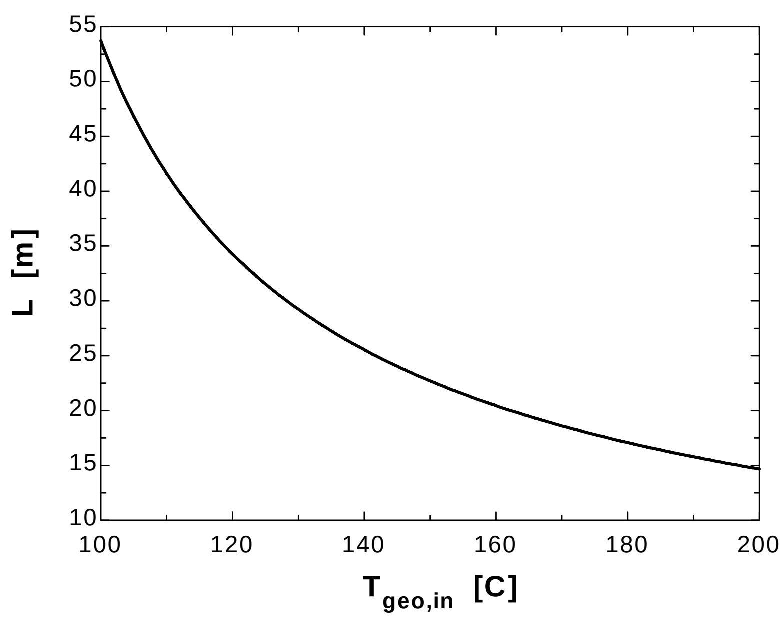
$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.642\text{ m}^2}{\pi(0.008\text{ m})} = \mathbf{25.5\text{ m}}$$

**16-48 EES** Prob. 16-47 is reconsidered. The effects of temperature and mass flow rate of geothermal water on the length of the tube are to be investigated.

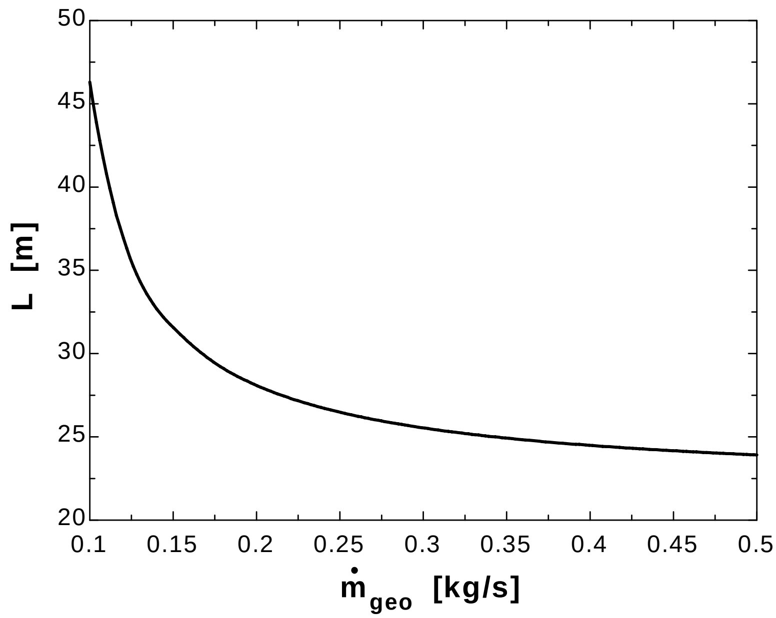
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_w_in=25 [C]
T_w_out=60 [C]
m_dot_w=0.2 [kg/s]
c_p_w=4.18 [kJ/kg-C]
T_geo_in=140 [C]
m_dot_geo=0.3 [kg/s]
c_p_geo=4.31 [kJ/kg-C]
D=0.008 [m]
U=0.55 [kW/m^2-C]

"ANALYSIS"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
Q_dot=m_dot_geo*c_p_geo*(T_geo_in-T_geo_out)
DELTAT_1=T_geo_in-T_w_in
DELTAT_2=T_geo_out-T_w_out
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
A=pi*D*L
```

| $T_{geo,in}$ [C] | L [m] |
|---|---|
| 100 | 53.73 |
| 105 | 46.81 |
| 110 | 41.62 |
| 115 | 37.56 |
| 120 | 34.27 |
| 125 | 31.54 |
| 130 | 29.24 |
| 135 | 27.26 |
| 140 | 25.54 |
| 145 | 24.04 |
| 150 | 22.7 |
| 155 | 21.51 |
| 160 | 20.45 |
| 165 | 19.48 |
| 170 | 18.61 |
| 175 | 17.81 |
| 180 | 17.08 |
| 185 | 16.4 |
| 190 | 15.78 |
| 195 | 15.21 |
| 200 | 14.67 |



| $m_{geo}$ [kg/s] | L [m] |
|---|---|
| 0.1 | 46.31 |
| 0.125 | 35.52 |
| 0.15 | 31.57 |
| 0.175 | 29.44 |
| 0.2 | 28.1 |
| 0.225 | 27.16 |
| 0.25 | 26.48 |
| 0.275 | 25.96 |
| 0.3 | 25.54 |
| 0.325 | 25.21 |
| 0.35 | 24.93 |
| 0.375 | 24.69 |
| 0.4 | 24.49 |
| 0.425 | 24.32 |
| 0.45 | 24.17 |
| 0.475 | 24.04 |
| 0.5 | 23.92 |



**16-49E** Glycerin is heated by hot water in a 1-shell pass and 8-tube passes heat exchanger. The rate of heat transfer for the cases of fouling and no fouling are to be determined.

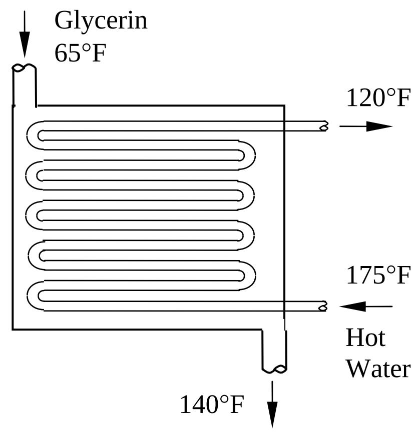
***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Heat transfer coefficients and fouling factors are constant and uniform. **5** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

***Properties*** The specific heats of glycerin and water are given to be 0.60 and 1.0 Btu/lbm.°F, respectively.

***Analysis*** (*a*) The tubes are thin walled and thus we assume the inner surface area of the tube to be equal to the outer surface area. Then the heat transfer surface area of this heat exchanger becomes

$$A_s = n\pi DL = 8\pi(0.5/12\text{ ft})(500\text{ ft}) = 523.6\text{ ft}^2$$

The temperature differences at the two ends of the heat exchanger are




$$\Delta T_1 = T_{h,in} - T_{c,out} = 175°\text{F} - 140°\text{F} = 35°\text{F}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 120°\text{F} - 65°\text{F} = 55°\text{F}$$

and $$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{35-55}{\ln(35/55)} = 44.25°\text{F}$$

The correction factor is

$$\left.\begin{array}{l} P = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{120-175}{65-175} = 0.5 \\ R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{65-140}{120-175} = 1.36 \end{array}\right\} F = 0.70$$

In case of no fouling, the overall heat transfer coefficient is determined from

$$U = \frac{1}{\dfrac{1}{h_i} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{50\text{ Btu/h.ft}^2.°\text{F}} + \dfrac{1}{4\text{ Btu/h.ft}^2.°\text{F}}} = 3.7\text{ Btu/h.ft}^2.°\text{F}$$

Then the rate of heat transfer becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} = (3.7\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)(0.70)(44.25°\text{F}) = \mathbf{60{,}000\ Btu/h}$$

(*b*) The thermal resistance of the heat exchanger with a fouling factor is

$$R = \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{1}{h_o A_o}$$
$$= \frac{1}{(50\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)} + \frac{0.002\text{ h.ft}^2.°\text{F/Btu}}{523.6\text{ ft}^2} + \frac{1}{(4\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)}$$
$$= 0.0005195\text{ h.°F/Btu}$$

The overall heat transfer coefficient in this case is

$$R = \frac{1}{UA_s} \longrightarrow U = \frac{1}{RA_s} = \frac{1}{(0.0005195\text{ h.°F/Btu})(523.6\text{ ft}^2)} = 3.68\text{ Btu/h.ft}^2.°\text{F}$$

Then rate of heat transfer becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} = (3.68\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)(0.70)(44.25°\text{F}) = \mathbf{59{,}680\ Btu/h}$$

**16-50** During an experiment, the inlet and exit temperatures of water and oil and the mass flow rate of water are measured. The overall heat transfer coefficient based on the inner surface area is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

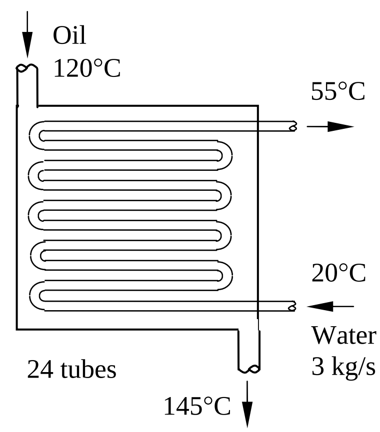
***Properties*** The specific heats of water and oil are given to be 4180 and 2150 J/kg.°C, respectively.

***Analysis*** The rate of heat transfer from the oil to the water is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (3\text{ kg/s})(4.18\text{ kJ/kg.°C})(55°\text{C} - 20°\text{C}) = 438.9\text{ kW}$$

The heat transfer area on the tube side is

$$A_i = n\pi D_i L = 24\pi(0.012\text{ m})(2\text{ m}) = 1.8\text{ m}^2$$




The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 120°\text{C} - 55°\text{C} = 65°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 45°\text{C} - 20°\text{C} = 25°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{65 - 25}{\ln(65/25)} = 41.9°\text{C}$$

$$\left.\begin{array}{l} P = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{55 - 20}{120 - 20} = 0.35 \\ R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{120 - 45}{55 - 20} = 2.14 \end{array}\right\} F = 0.70$$

Then the overall heat transfer coefficient becomes

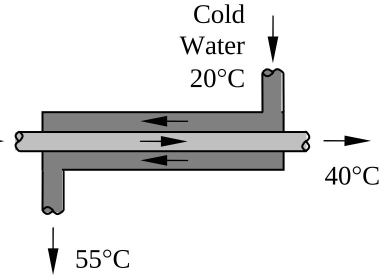
$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{lm,CF}} = \frac{438.9\text{ kW}}{(1.8\text{ m}^2)(0.70)(41.9°\text{C})} = \mathbf{8.31\ kW/m^2.°C}$$

**16-51** Ethylene glycol is cooled by water in a double-pipe counter-flow heat exchanger. The rate of heat transfer, the mass flow rate of water, and the heat transfer surface area on the inner side of the tubes are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg.°C, respectively.

***Analysis*** (*a*) The rate of heat transfer is




$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{glycol}}$$
$$= (3.5 \text{ kg/s})(2.56 \text{ kJ/kg.°C})(80°\text{C} - 40°\text{C})$$
$$= \mathbf{358.4 \text{ kW}}$$

(*b*) The rate of heat transfer from water must be equal to the rate of heat transfer to the glycol. Then,

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})}$$
$$= \frac{358.4 \text{ kJ/s}}{(4.18 \text{ kJ/kg.°C})(55°\text{C} - 20°\text{C})} = \mathbf{2.45 \text{ kg/s}}$$

(*c*) The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80°\text{C} - 55°\text{C} = 25°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°\text{C} - 20°\text{C} = 20°\text{C}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25 - 20}{\ln(25/20)} = 22.4°\text{C}$$

Then the heat transfer surface area becomes

$$\dot{Q} = U_i A_i \Delta T_{lm} \longrightarrow A_i = \frac{\dot{Q}}{U_i \Delta T_{lm}} = \frac{358.4 \text{ kW}}{(0.25 \text{ kW/m}^2.°\text{C})(22.4°\text{C})} = \mathbf{64.0 \text{ m}^2}$$

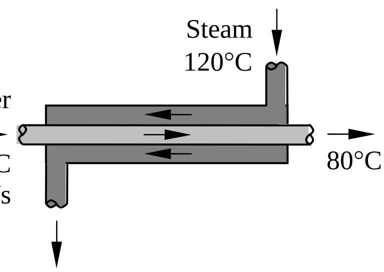
**16-52** Water is heated by steam in a double-pipe counter-flow heat exchanger. The required length of the tubes is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heat of water is given to be 4.18 kJ/kg.°C. The heat of condensation of steam at 120°C is given to be 2203 kJ/kg.

***Analysis*** The rate of heat transfer is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}}$$
$$= (3\text{ kg/s})(4.18\text{ kJ/kg.°C})(80°\text{C} - 17°\text{C})$$
$$= 790.02\text{ kW}$$




The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 120°\text{C} - 80°\text{C} = 40°\text{C}$$
$$\Delta T_2 = T_{h,in} - T_{c,in} = 120°\text{C} - 17°\text{C} = 103°\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 103}{\ln(40/103)} = 66.6°\text{C}$$

The heat transfer surface area is

$$\dot{Q} = U_i A_i \Delta T_{lm} \longrightarrow A_i = \frac{\dot{Q}}{U_i \Delta T_{lm}} = \frac{790.02\text{ kW}}{(1.5\text{ kW/m}^2.°\text{C})(66.6°\text{C})} = 7.9\text{ m}^2$$

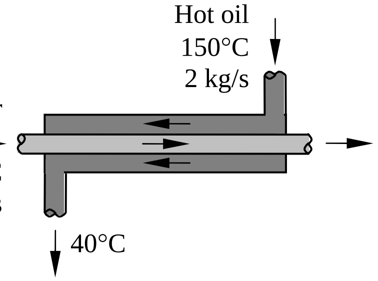
Then the length of tube required becomes

$$A_i = \pi D_i L \longrightarrow L = \frac{A_i}{\pi D_i} = \frac{7.9\text{ m}^2}{\pi(0.025\text{ m})} = \mathbf{100.6\text{ m}}$$

**16-53** Oil is cooled by water in a thin-walled double-pipe counter-flow heat exchanger. The overall heat transfer coefficient of the heat exchanger is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

***Properties*** The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg.°C, respectively.




***Analysis*** The rate of heat transfer from the water to the oil is

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} = (2\text{ kg/s})(2.2\text{ kJ/kg.°C})(150°\text{C} - 40°\text{C}) = 484\text{ kW}$$

The outlet temperature of the water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p}$$

$$= 22°\text{C} + \frac{484\text{ kW}}{(1.5\text{ kg/s})(4.18\text{ kJ/kg.°C})} = 99.2°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 150°\text{C} - 99.2°\text{C} = 50.8°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°\text{C} - 22°\text{C} = 18°\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{50.8 - 18}{\ln(50.8/18)} = 31.6°\text{C}$$

Then the overall heat transfer coefficient becomes

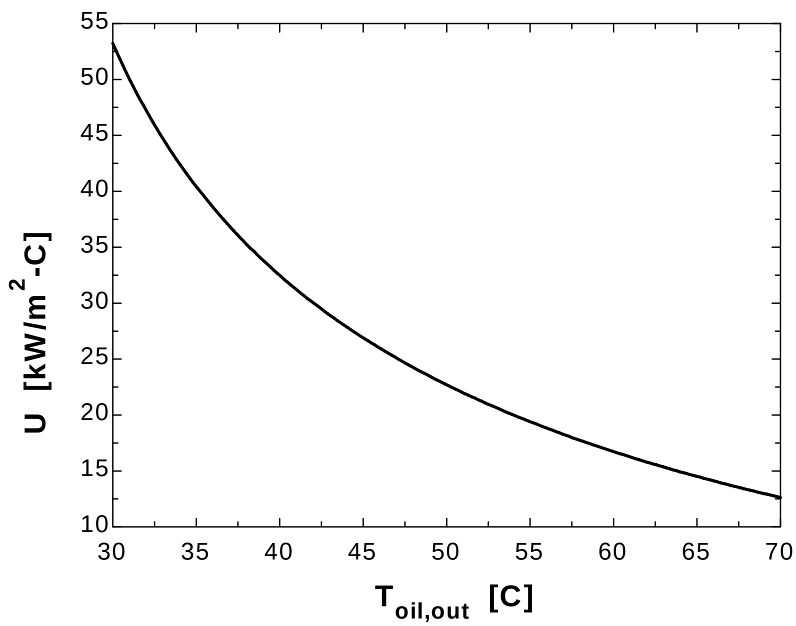
$$U = \frac{\dot{Q}}{A_s \Delta T_{lm}} = \frac{484\text{ kW}}{\pi(0.025\text{ m})(6\text{ m})(31.6°\text{C})} = \mathbf{32.5\text{ kW/m}^2.°\text{C}}$$

**16-54 EES** Prob. 16-53 is reconsidered. The effects of oil exit temperature and water inlet temperature on the overall heat transfer coefficient of the heat exchanger are to be investigated.

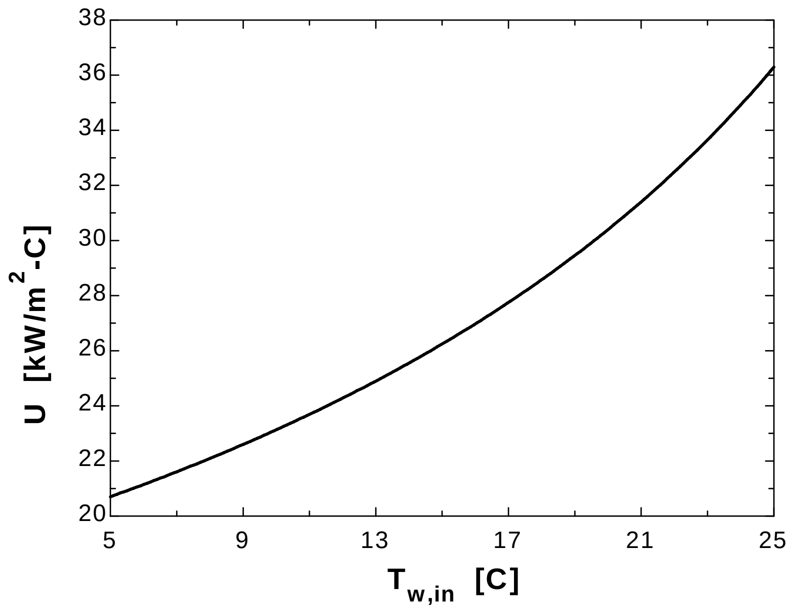
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_oil_in=150 [C]
T_oil_out=40 [C]
m_dot_oil=2 [kg/s]
c_p_oil=2.20 [kJ/kg-C]
T_w_in=22 [C]
m_dot_w=1.5 [kg/s]
C_p_w=4.18 [kJ/kg-C]
D=0.025 [m]
L=6 [m]

"ANALYSIS"
Q_dot=m_dot_oil*c_p_oil*(T_oil_in-T_oil_out)
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
DELTAT_1=T_oil_in-T_w_out
DELTAT_2=T_oil_out-T_w_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
A=pi*D*L
```

| $T_{oil,out}$ [C] | U [kW/m$^2$-C] |
|---|---|
| 30 | 53.22 |
| 32.5 | 45.94 |
| 35 | 40.43 |
| 37.5 | 36.07 |
| 40 | 32.49 |
| 42.5 | 29.48 |
| 45 | 26.9 |
| 47.5 | 24.67 |
| 50 | 22.7 |
| 52.5 | 20.96 |
| 55 | 19.4 |
| 57.5 | 18 |
| 60 | 16.73 |
| 62.5 | 15.57 |
| 65 | 14.51 |
| 67.5 | 13.53 |
| 70 | 12.63 |



| $T_{w,in}$ [C] | U [kW/m$^2$C] |
|---|---|
| 5 | 20.7 |
| 6 | 21.15 |
| 7 | 21.61 |
| 8 | 22.09 |
| 9 | 22.6 |
| 10 | 23.13 |
| 11 | 23.69 |
| 12 | 24.28 |
| 13 | 24.9 |
| 14 | 25.55 |
| 15 | 26.24 |
| 16 | 26.97 |
| 17 | 27.75 |
| 18 | 28.58 |
| 19 | 29.46 |
| 20 | 30.4 |
| 21 | 31.4 |
| 22 | 32.49 |
| 23 | 33.65 |
| 24 | 34.92 |
| 25 | 36.29 |



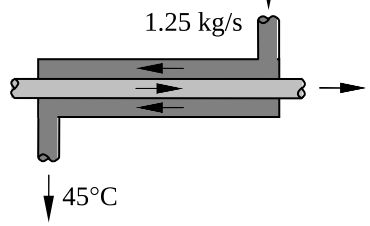
**16-55** The inlet and outlet temperatures of the cold and hot fluids in a double-pipe heat exchanger are given. It is to be determined whether this is a parallel-flow or counter-flow heat exchanger.

***Analysis*** In parallel-flow heat exchangers, the temperature of the cold water can never exceed that of the hot fluid. In this case $T_{\text{cold out}}$ = 50°C which is greater than $T_{\text{hot out}}$ = 45°C. Therefore this must be a counter-flow heat exchanger.

**16-56** Cold water is heated by hot water in a double-pipe counter-flow heat exchanger. The rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

***Properties*** The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.




***Analysis*** The rate of heat transfer in this heat exchanger is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_{\text{cold water}} \\ &= (1.25\text{ kg/s})(4.18\text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) \\ &= \mathbf{156.8\ kW}\end{aligned}$$

The outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 100°\text{C} - \frac{156.8\text{ kW}}{(3\text{ kg/s})(4.19\text{ kJ/kg.°C})} = 87.5°\text{C}$$

The temperature differences at the two ends of the heat exchanger are

$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{c,out} = 100°\text{C} - 45°\text{C} = 55°\text{C} \\ \Delta T_2 &= T_{h,out} - T_{c,in} = 87.5°\text{C} - 15°\text{C} = 72.5°\text{C}\end{aligned}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{55 - 72.5}{\ln(55/72.5)} = 63.3°\text{C}$$

Then the surface area of this heat exchanger becomes

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{156.8\text{ kW}}{(0.880\text{ kW/m}^2.°\text{C})(63.3°\text{C})} = \mathbf{2.81\ m^2}$$

**16-57** Engine oil is heated by condensing steam in a condenser. The rate of heat transfer and the length of the tube required are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

***Properties*** The specific heat of engine oil is given to be 2.1 kJ/kg.°C. The heat of condensation of steam at 130°C is given to be 2174 kJ/kg.

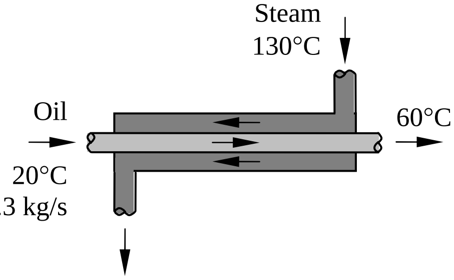
***Analysis*** The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{oil}} = (0.3\text{ kg/s})(2.1\text{ kJ/kg.°C})(60°\text{C} - 20°\text{C}) = \mathbf{25.2\ kW}$$

The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 130°\text{C} - 60°\text{C} = 70°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 130°\text{C} - 20°\text{C} = 110°\text{C}$$




and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{70 - 110}{\ln(70/110)} = 88.5°\text{C}$$

The surface area is

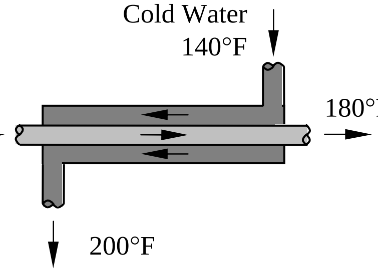
$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{25.2\text{ kW}}{(0.65\text{ kW/m}^2.°\text{C})(88.5°\text{C})} = 0.44\text{ m}^2$$

Then the length of the tube required becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.44\text{ m}^2}{\pi(0.02\text{ m})} = \mathbf{7.0\ m}$$

**16-58E** Water is heated by geothermal water in a double-pipe counter-flow heat exchanger. The mass flow rate of each fluid and the total thermal resistance of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.
**3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.




***Properties*** The specific heats of water and geothermal fluid are given to be 1.0 and 1.03 Btu/lbm.°F, respectively.

***Analysis*** The mass flow rate of each fluid is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}}$$

$$\dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{40 \text{ Btu/s}}{(1.0 \text{ Btu/lbm.°F})(200°\text{F} - 140°\text{F})} = \mathbf{0.667 \text{ lbm/s}}$$

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{geo. water}}$$

$$\dot{m}_{\text{geo. water}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{40 \text{ Btu/s}}{(1.03 \text{ Btu/lbm.°F})(270°\text{F} - 180°\text{F})} = \mathbf{0.431 \text{ lbm/s}}$$

The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 270°\text{F} - 200°\text{F} = 70°\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 180°\text{F} - 140°\text{F} = 40°\text{F}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{70 - 40}{\ln(70/40)} = 53.61°\text{F}$$

Then

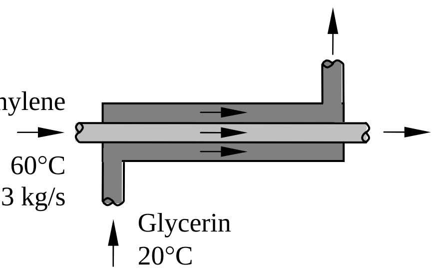
$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow UA_s = \frac{\dot{Q}}{\Delta T_{lm}} = \frac{40 \text{ Btu/s}}{53.61°\text{F}} = 0.7462 \text{ Btu/s.}^{\circ}\text{F}$$

$$U = \frac{1}{RA_s} \longrightarrow R = \frac{1}{UA_s} = \frac{1}{0.7462 \text{ Btu/s.°F}} = \mathbf{1.34 \text{ s} \cdot °\text{F/Btu}}$$

**16-59** Glycerin is heated by ethylene glycol in a thin-walled double-pipe parallel-flow heat exchanger. The rate of heat transfer, the outlet temperature of the glycerin, and the mass flow rate of the ethylene glycol are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

***Properties*** The specific heats of glycerin and ethylene glycol are given to be 2.4 and 2.5 kJ/kg.°C, respectively.




***Analysis*** (*a*) The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,in} = 60°\text{C} - 20°\text{C} = 40°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = T_{h,out} - (T_{h,out} - 15°\text{C}) = 15°\text{C}$$

and $$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 15}{\ln(40/15)} = 25.5°\text{C}$$

Then the rate of heat transfer becomes

$$\dot{Q} = UA_s \Delta T_{lm} = (240 \text{ W/m}^2.°\text{C})(3.2 \text{ m}^2)(25.5°\text{C}) = 19{,}584 \text{ W} = \mathbf{19.58\ kW}$$

(*b*) The outlet temperature of the glycerin is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{glycerin}} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{19.584 \text{ kW}}{(0.3 \text{ kg/s})(2.4 \text{ kJ/kg.°C})} = \mathbf{47.2°C}$$

(*c*) Then the mass flow rate of ethylene glycol becomes

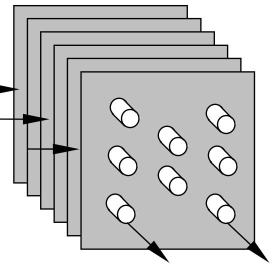
$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{ethylene glycol}}$$

$$\dot{m}_{\text{ethylene glycol}} = \frac{\dot{Q}}{c_p(T_{in} - T_{out})} = \frac{19.584 \text{ kJ/s}}{(2.5 \text{ kJ/kg.°C})[(47.2 + 15)°\text{C} - 60°\text{C}]} = \mathbf{3.56\ kg/s}$$

**16-60** Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the outlet temperature of the air are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of air and combustion gases are given to be 1005 and 1100 J/kg.°C, respectively.




***Analysis*** The rate of heat transfer is

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{gas.}$$
$$= (1.1\,\text{kg/s})(1.1\,\text{kJ/kg.°C})(180°\text{C} - 95°\text{C})$$
$$= \mathbf{103\,kW}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95\,\text{kPa})(0.8\,\text{m}^3/\text{s})}{(0.287\,\text{kPa.m}^3/\text{kg.K}) \times 293\,\text{K}} = 0.904\,\text{kg/s}$$

Then the outlet temperature of the air becomes

$$\dot{Q} = \dot{m}c_p(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{103 \times 10^3\ \text{W}}{(0.904\,\text{kg/s})(1005\,\text{J/kg.°C})} = \mathbf{133°C}$$

**16-61** Water is heated by hot oil in a 2-shell passes and 12-tube passes heat exchanger. The heat transfer surface area on the tube side is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

***Analysis*** The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (4.5\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = 940.5\text{ kW}$$

The outlet temperature of the oil is determined from

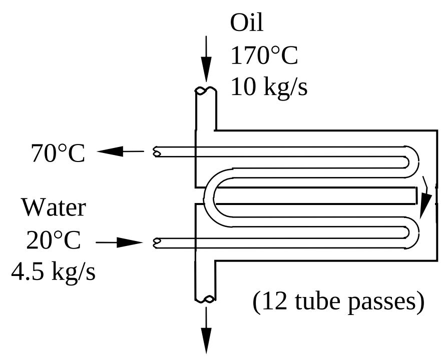
$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5\text{ kW}}{(10\text{ kg/s})(2.3\text{ kJ/kg.°C})} = 129°\text{C}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 170°\text{C} - 70°\text{C} = 100°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 129°\text{C} - 20°\text{C} = 109°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{100 - 109}{\ln(100/109)} = 104.4°\text{C}$$




$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 20}{170 - 20} = 0.33 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{170 - 129}{70 - 20} = 0.82 \end{aligned}\right\} F = 1.0$$

Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm,CF}} = \frac{940.5\text{ kW}}{(0.350\text{ kW/m}^2.°\text{C})(1.0)(104.4°\text{C})} = \mathbf{25.7\ m^2}$$

**16-62** Water is heated by hot oil in a 2-shell passes and 12-tube passes heat exchanger. The heat transfer surface area on the tube side is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

***Analysis*** The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (2\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = 418\text{ kW}$$

The outlet temperature of the oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{418\text{ kW}}{(10\text{ kg/s})(2.3\text{ kJ/kg.°C})} = 151.8°\text{C}$$

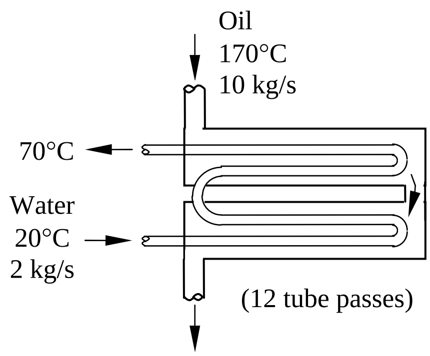
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 170°\text{C} - 70°\text{C} = 100°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 151.8°\text{C} - 20°\text{C} = 131.8°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{100 - 131.8}{\ln(100/131.8)} = 115.2°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 20}{170 - 20} = 0.33 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{170 - 151.8}{70 - 20} = 0.36 \end{aligned}\right\} F = 1.0$$




Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{418\text{ kW}}{(0.350\text{ kW/m}^2.°\text{C})(1.0)(115.2°\text{C})} = \mathbf{10.4\ m^2}$$

**16-63** Ethyl alcohol is heated by water in a 2-shell passes and 8-tube passes heat exchanger. The heat transfer surface area of the heat exchanger is to be determined.

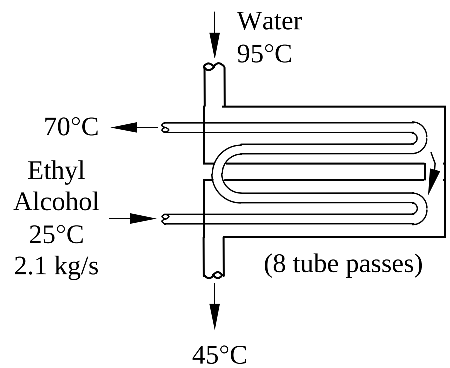
***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of water and ethyl alcohol are given to be 4.19 and 2.67 kJ/kg.°C, respectively.

***Analysis*** The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{ethyl alcohol}} = (2.1\text{ kg/s})(2.67\text{ kJ/kg.°C})(70°\text{C} - 25°\text{C}) = 252.3\text{ kW}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are




$$\Delta T_1 = T_{h,in} - T_{c,out} = 95°\text{C} - 70°\text{C} = 25°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 45°\text{C} - 25°\text{C} = 20°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25 - 20}{\ln(25/20)} = 22.4°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 25}{95 - 25} = 0.64 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{95 - 45}{70 - 25} = 1.1 \end{aligned}\right\} F = 0.82$$

Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{252.3\text{ kW}}{(0.950\text{ kW/m}^2.°\text{C})(0.82)(22.4°\text{C})} = \mathbf{14.5\ m^2}$$

**16-64** Water is heated by ethylene glycol in a 2-shell passes and 12-tube passes heat exchanger. The rate of heat transfer and the heat transfer surface area on the tube side are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of water and ethylene glycol are given to be 4.18 and 2.68 kJ/kg.°C, respectively.

***Analysis*** The rate of heat transfer in this heat exchanger is :

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (0.8\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 22°\text{C}) = \mathbf{160.5\ kW}$$

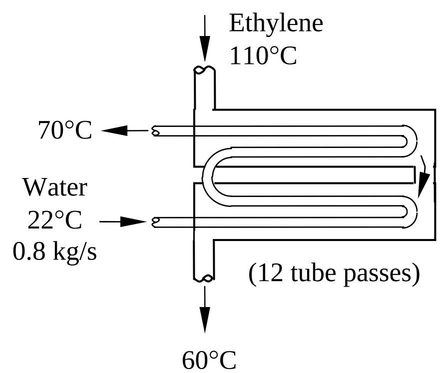
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 110°\text{C} - 70°\text{C} = 40°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 22°\text{C} = 38°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{40-38}{\ln(40/38)} = 39°\text{C}$$

$$\left.\begin{array}{l} P = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{70-22}{110-22} = 0.55 \\ R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{110-60}{70-22} = 1.04 \end{array}\right\} F = 0.92$$




Then the heat transfer surface area on the tube side becomes

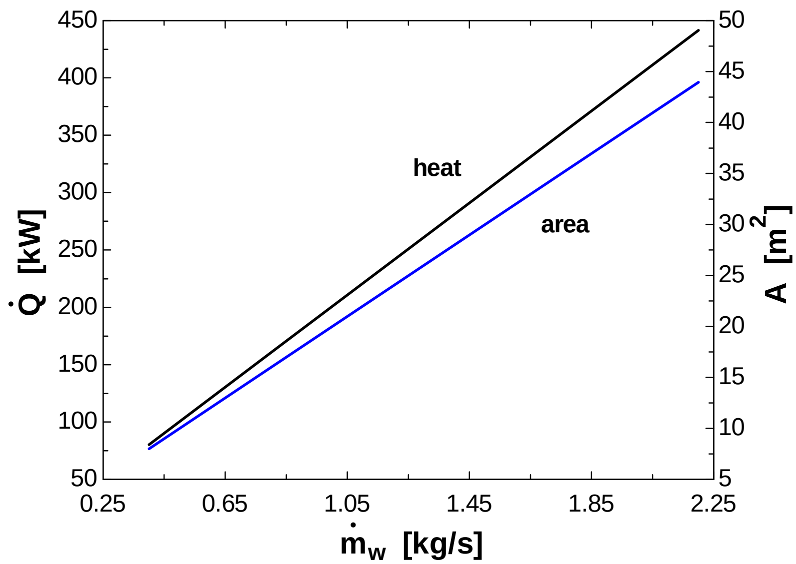
$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{160.5\text{ kW}}{(0.28\text{ kW/m}^2\text{.°C})(0.92)(39°\text{C})} = \mathbf{16.0\ m^2}$$

**16-65 EES** Prob. 16-64 is reconsidered. The effect of the mass flow rate of water on the rate of heat transfer and the tube-side surface area is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_w_in=22 [C]
T_w_out=70 [C]
m_dot_w=0.8 [kg/s]
c_p_w=4.18 [kJ/kg-C]
T_glycol_in=110 [C]
T_glycol_out=60 [C]
c_p_glycol=2.68 [kJ/kg-C]
U=0.28 [kW/m^2-C]

"ANALYSIS"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
Q_dot=m_dot_glycol*c_p_glycol*(T_glycol_in-T_glycol_out)
DELTAT_1=T_glycol_in-T_w_out
DELTAT_2=T_glycol_out-T_w_in
DELTAT_lm_CF=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
P=(T_w_out-T_w_in)/(T_glycol_in-T_w_in)
R=(T_glycol_in-T_glycol_out)/(T_w_out-T_w_in)
F=0.92 "from Fig. 16-18b of the text at the calculated P and R"
Q_dot=U*A*F*DELTAT_lm_CF
```

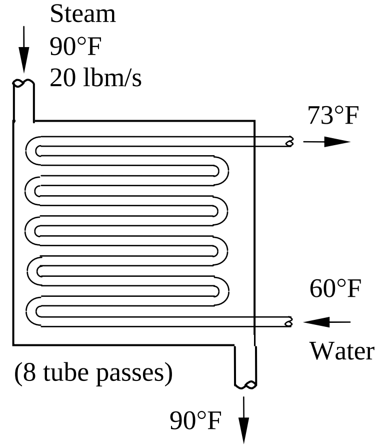
| $m_w$ [kg/s] | Q [kW] | A [$m^2$] |
|---|---|---|
| 0.4 | 80.26 | 7.99 |
| 0.5 | 100.3 | 9.988 |
| 0.6 | 120.4 | 11.99 |
| 0.7 | 140.4 | 13.98 |
| 0.8 | 160.5 | 15.98 |
| 0.9 | 180.6 | 17.98 |
| 1 | 200.6 | 19.98 |
| 1.1 | 220.7 | 21.97 |
| 1.2 | 240.8 | 23.97 |
| 1.3 | 260.8 | 25.97 |
| 1.4 | 280.9 | 27.97 |
| 1.5 | 301 | 29.96 |
| 1.6 | 321 | 31.96 |
| 1.7 | 341.1 | 33.96 |
| 1.8 | 361.2 | 35.96 |
| 1.9 | 381.2 | 37.95 |
| 2 | 401.3 | 39.95 |
| 2.1 | 421.3 | 41.95 |
| 2.2 | 441.4 | 43.95 |



**16-66E** Steam is condensed by cooling water in a condenser. The rate of heat transfer, the rate of condensation of steam, and the mass flow rate of cold water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

***Properties*** We take specific heat of water are given to be 1.0 Btu/lbm.°F. The heat of condensation of steam at 90°F is 1043 Btu/lbm.




***Analysis*** (*a*) The log mean temperature difference is determined from

$$\Delta T_1 = T_{h,in} - T_{c,out} = 90°\text{F} - 73°\text{F} = 17°\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 90°\text{F} - 60°\text{F} = 30°\text{F}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{17 - 30}{\ln(17/30)} = 22.9°\text{F}$$

The heat transfer surface area is

$$A_s = 8n\pi DL = 8 \times 50 \times \pi(3/48\ \text{ft})(5\ \text{ft}) = 392.7\ \text{ft}^2$$

and

$$\dot{Q} = UA_s\Delta T_{lm} = (600\ \text{Btu/h.ft}^2.°\text{F})(392.7\ \text{ft}^2)(22.9°\text{F}) = \mathbf{5.396 \times 10^6\ Btu/h}$$

(*b*) The rate of condensation of the steam is

$$\dot{Q} = (\dot{m}h_{fg})_{steam} \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{5.396 \times 10^6\ \text{Btu/h}}{1043\ \text{Btu/lbm}} = \mathbf{5173\ lbm/h = 1.44\ lbm/s}$$

(*c*) Then the mass flow rate of cold water becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{cold water}}$$

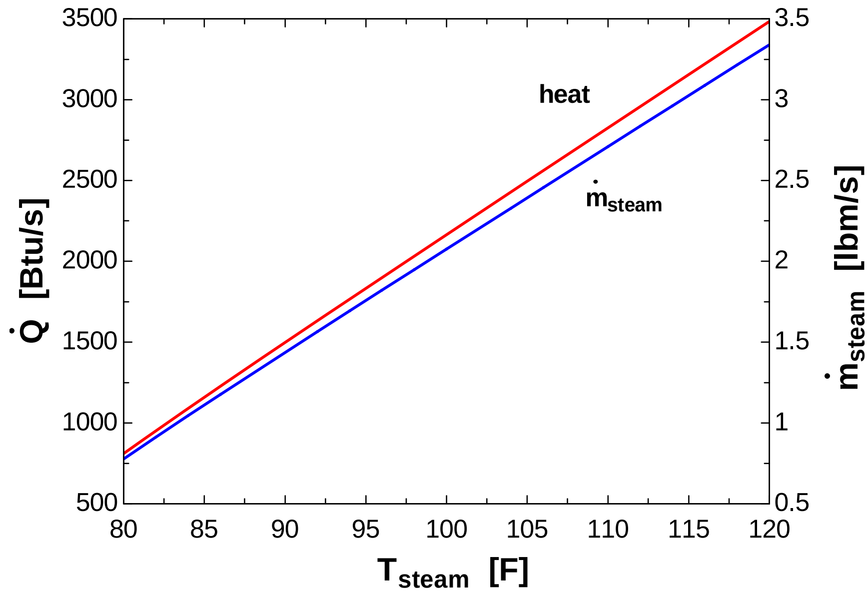
$$\dot{m}_{\text{cold water}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{5.396 \times 10^6\ \text{Btu/h}}{(1.0\ \text{Btu/lbm.°F})(73°\text{F} - 60°\text{F}]} = \mathbf{4.15 \times 10^5\ lbm/h} = \mathbf{115\ lbm/s}$$

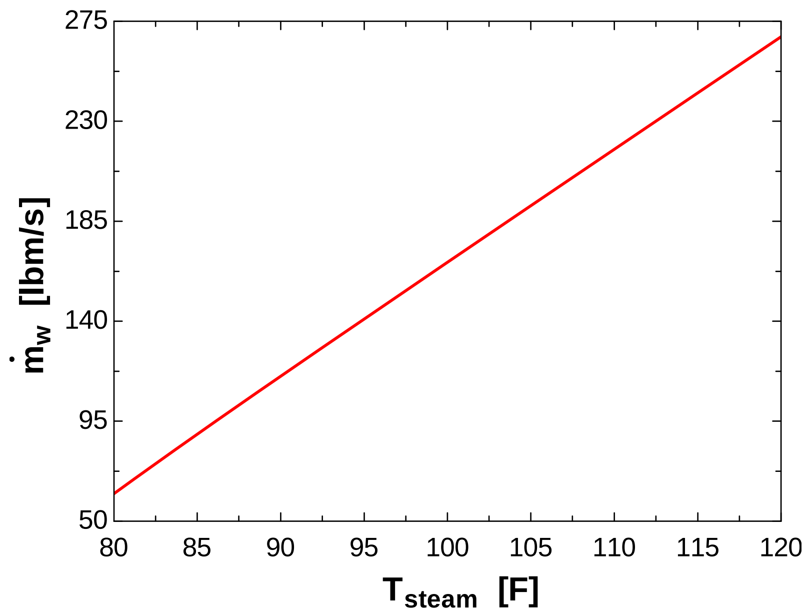
**16-67E EES** Prob. 16-66E is reconsidered. The effect of the condensing steam temperature on the rate of heat transfer, the rate of condensation of steam, and the mass flow rate of cold water is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
N_pass=8
N_tube=50
T_steam=90 [F]
h_fg_steam=1043 [Btu/lbm]
T_w_in=60 [F]
T_w_out=73 [F]
c_p_w=1.0 [Btu/lbm-F]
D=3/4*1/12 [ft]
L=5 [ft]
U=600 [Btu/h-ft^2-F]

"ANALYSIS"
"(a)"
DELTAT_1=T_steam-T_w_out
DELTAT_2=T_steam-T_w_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
A=N_pass*N_tube*pi*D*L
Q_dot=U*A*DELTAT_lm*Convert(Btu/h, Btu/s)
"(b)"
Q_dot=m_dot_steam*h_fg_steam
"(c)"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
```

| $T_{steam}$ [F] | Q [Btu/s] | $m_{steam}$[lbm/s] | $m_w$ [lbm/s] |
|---|---|---|---|
| 80 | 810.5 | 0.7771 | 62.34 |
| 82 | 951.9 | 0.9127 | 73.23 |
| 84 | 1091 | 1.046 | 83.89 |
| 86 | 1228 | 1.177 | 94.42 |
| 88 | 1363 | 1.307 | 104.9 |
| 90 | 1498 | 1.436 | 115.2 |
| 92 | 1632 | 1.565 | 125.6 |
| 94 | 1766 | 1.693 | 135.8 |
| 96 | 1899 | 1.821 | 146.1 |
| 98 | 2032 | 1.948 | 156.3 |
| 100 | 2165 | 2.076 | 166.5 |
| 102 | 2297 | 2.203 | 176.7 |
| 104 | 2430 | 2.329 | 186.9 |
| 106 | 2562 | 2.456 | 197.1 |
| 108 | 2694 | 2.583 | 207.2 |
| 110 | 2826 | 2.709 | 217.4 |
| 112 | 2958 | 2.836 | 227.5 |
| 114 | 3089 | 2.962 | 237.6 |
| 116 | 3221 | 3.088 | 247.8 |
| 118 | 3353 | 3.214 | 257.9 |
| 120 | 3484 | 3.341 | 268 |


heat
$\dot{m}_{steam}$
$\dot{Q}$ [Btu/s]
$\dot{m}_{steam}$ [lbm/s]
$T_{steam}$ [F]
3500
3000
2500
2000
1500
1000
500
3.5
3
2.5
2
1.5
1
0.5
80
85
90
95
100
105
110
115
120


$\dot{m}_w$ [lbm/s]
$T_{steam}$ [F]
275
230
185
140
95
50
80
85
90
95
100
105
110
115
120

**16-68** Glycerin is heated by hot water in a 1-shell pass and 20-tube passes heat exchanger. The mass flow rate of glycerin and the overall heat transfer coefficient of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heat of glycerin is given to be are given to be 2.48 kJ/kg.°C and that of water is taken to be 4.18 kJ/kg.°C.

***Analysis*** The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{water}} = (0.5\text{ kg/s})(4.18\text{ kJ/kg.°C})(100°\text{C} - 55°\text{C}) = 94.05\text{ kW}$$

The mass flow rate of the glycerin is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{glycerin}}$$

$$\dot{m}_{\text{glycerin}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{94.05\text{ kJ/s}}{(2.48\text{ kJ/kg.°C})[(55°\text{C} - 15°\text{C}]} = \mathbf{0.95\text{ kg/s}}$$

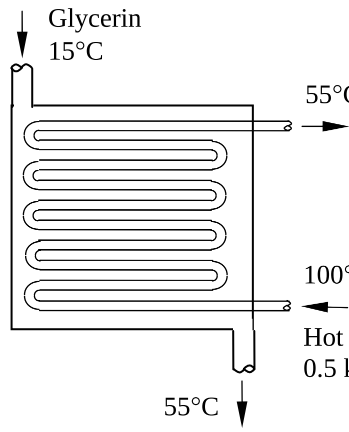
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 100°\text{C} - 55°\text{C} = 45°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 55°\text{C} - 15°\text{C} = 40°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{45 - 40}{\ln(45/40)} = 42.5°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{55 - 100}{15 - 100} = 0.53 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{15 - 55}{55 - 100} = 0.89 \end{aligned}\right\} F = 0.77$$




The heat transfer surface area is

$$A_s = n\pi DL = 20\pi(0.04\text{ m})(2\text{ m}) = 5.027\text{ m}^2$$

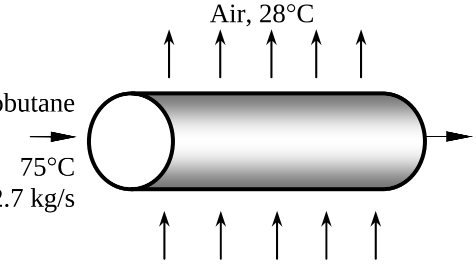
Then the overall heat transfer coefficient of the heat exchanger is determined to be

$$\dot{Q} = UA_s F\Delta T_{lm,CF} \longrightarrow U = \frac{\dot{Q}}{A_s F\Delta T_{lm,CF}} = \frac{94.05\text{ kW}}{(5.027\text{ m}^2)(0.77)(42.5°\text{C})} = \mathbf{0.572\text{ kW/m}^2.°\text{C}}$$

**16-69** Isobutane is condensed by cooling air in the condenser of a power plant. The mass flow rate of air and the overall heat transfer coefficient are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The heat of vaporization of isobutane at 75°C is given to be $h_{fg}$ = 255.7 kJ/kg and specific heat of air is taken to be $c_p$ = 1005 J/kg.°C.




***Analysis*** First, the rate of heat transfer is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\text{isobutane}}$$
$$= (2.7\text{ kg/s})(255.7\text{ kJ/kg}) = 690.4\text{ kW}$$

The mass flow rate of air is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} \longrightarrow \dot{m}_{\text{air}} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{690.4\text{ kJ/s}}{(1.005\text{ kJ/kg.°C})(28°\text{C} - 21°\text{C})} = \mathbf{98.1\ kg/s}$$

The temperature differences between the isobutane and the air at the two ends of the condenser are

$$\Delta T_1 = T_{\text{h,in}} - T_{\text{c,out}} = 75°\text{C} - 21°\text{C} = 54°\text{C}$$
$$\Delta T_2 = T_{\text{h,out}} - T_{\text{c,in}} = 75°\text{C} - 28°\text{C} = 47°\text{C}$$

and

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{54 - 47}{\ln(54/47)} = 50.4°\text{C}$$

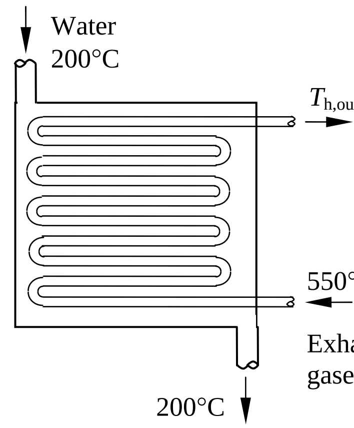
Then the overall heat transfer coefficient is determined from

$$\dot{Q} = UA_s\Delta T_{\text{lm}} \longrightarrow 690{,}400\text{ W} = U(24\text{ m}^2)(50.4°\text{C}) \longrightarrow U = \mathbf{571\ W/m^2.°C}$$

**16-70** Water is evaporated by hot exhaust gases in an evaporator. The rate of heat transfer, the exit temperature of the exhaust gases, and the rate of evaporation of water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The heat of vaporization of water at 200°C is given to be $h_{fg}$ = 1941 kJ/kg and specific heat of exhaust gases is given to be $c_p$ = 1051 J/kg.°C.




***Analysis*** The temperature differences between the water and the exhaust gases at the two ends of the evaporator are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 550°C - 200°C = 350°C$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = (T_{h,out} - 200)°C$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{350 - (T_{h,out} - 200)}{\ln[350/(T_{h,out} - 200)]}$$

Then the rate of heat transfer can be expressed as

$$\dot{Q} = UA_s \Delta T_{lm} = (1.780 \text{ kW/m}^2.°C)(0.5 \text{ m}^2)\frac{350 - (T_{h,out} - 200)}{\ln[350/(T_{h,out} - 200)]} \quad \text{(Eq. 1)}$$

The rate of heat transfer can also be expressed as in the following forms

$$\dot{Q} = [\dot{m}c_p(T_{h,in} - T_{h,out})]_{\substack{\text{exhaust}\\\text{gases}}} = (0.25 \text{ kg/s})(1.051 \text{ kJ/kg.°C})(550°C - T_{h,out}) \quad \text{(Eq. 2)}$$

$$\dot{Q} = (\dot{m}h_{fg})_{water} = \dot{m}_{water}(1941 \text{ kJ/kg}) \quad \text{(Eq. 3)}$$

We have three equations with three unknowns. Using an equation solver such as EES, the unknowns are determined to be

$$\dot{Q} = \mathbf{88.85\ kW}$$

$$T_{h,out} = \mathbf{211.8°C}$$

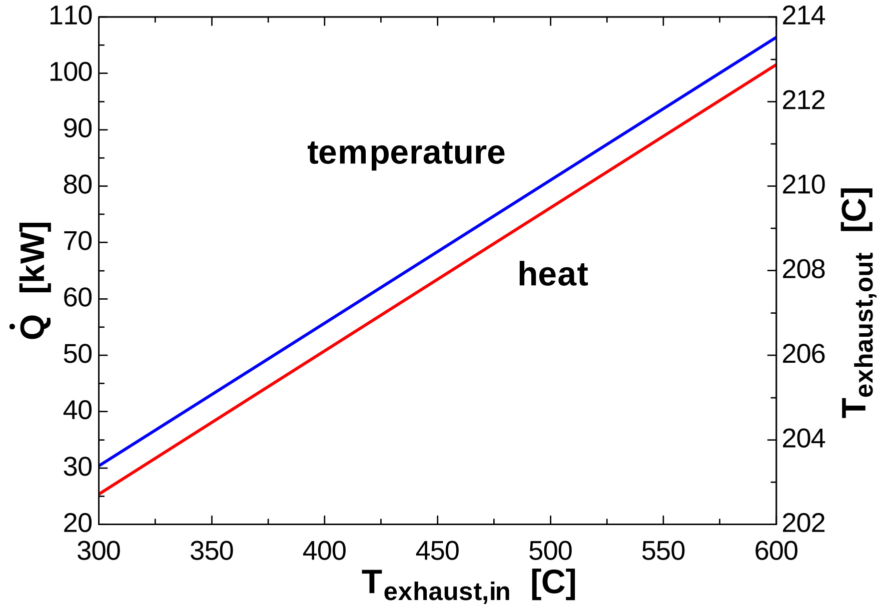
$$\dot{m}_{water} = \mathbf{0.0458\ kg/s}$$

**16-71 EES** Prob. 16-70 is reconsidered. The effect of the exhaust gas inlet temperature on the rate of heat transfer, the exit temperature of exhaust gases, and the rate of evaporation of water is to be investigated.

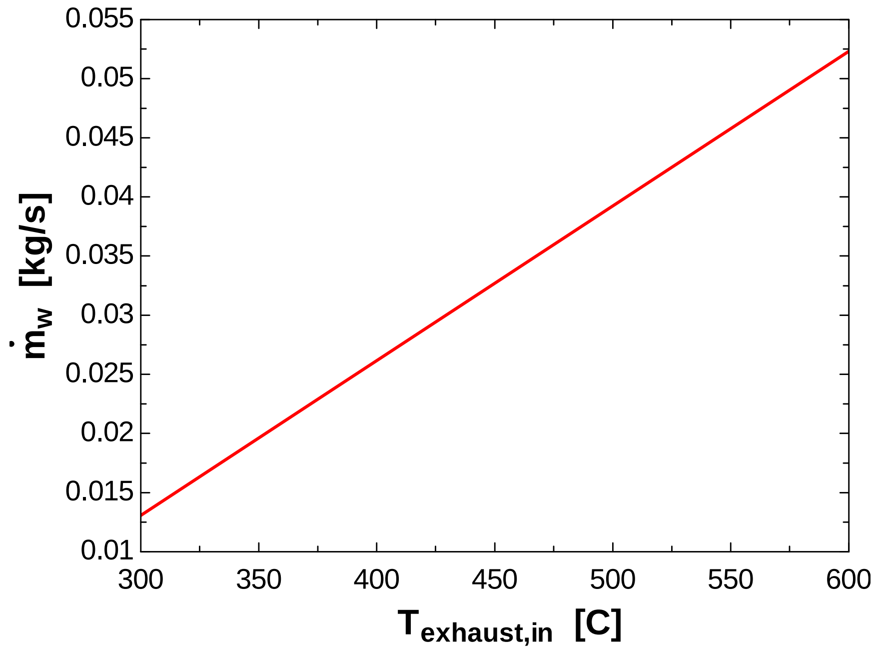
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_exhaust_in=550 [C]
c_p_exhaust=1.051 [kJ/kg-C]
m_dot_exhaust=0.25 [kg/s]
T_w=200 [C]
h_fg_w=1941 [kJ/kg]
A=0.5 [m^2]
U=1.780 [kW/m^2-C]

"ANALYSIS"
DELTAT_1=T_exhaust_in-T_w
DELTAT_2=T_exhaust_out-T_w
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_exhaust*c_p_exhaust*(T_exhaust_in-T_exhaust_out)
Q_dot=m_dot_w*h_fg_w
```

| $T_{exhaust,in}$ [C] | Q [kW] | $T_{exhaust,out}$ [C] | $m_w$ [kg/s] |
|---|---|---|---|
| 300 | 25.39 | 203.4 | 0.01308 |
| 320 | 30.46 | 204.1 | 0.0157 |
| 340 | 35.54 | 204.7 | 0.01831 |
| 360 | 40.62 | 205.4 | 0.02093 |
| 380 | 45.7 | 206.1 | 0.02354 |
| 400 | 50.77 | 206.8 | 0.02616 |
| 420 | 55.85 | 207.4 | 0.02877 |
| 440 | 60.93 | 208.1 | 0.03139 |
| 460 | 66.01 | 208.8 | 0.03401 |
| 480 | 71.08 | 209.5 | 0.03662 |
| 500 | 76.16 | 210.1 | 0.03924 |
| 520 | 81.24 | 210.8 | 0.04185 |
| 540 | 86.32 | 211.5 | 0.04447 |
| 560 | 91.39 | 212.2 | 0.04709 |
| 580 | 96.47 | 212.8 | 0.0497 |
| 600 | 101.5 | 213.5 | 0.05232 |



110
100
90
80
70
60
50
40
30
20
214
212
210
208
206
204
202
300
350
400
450
500
550
600
temperature
heat
Q̇ [kW]
T exhaust,out [C]
T exhaust,in [C]

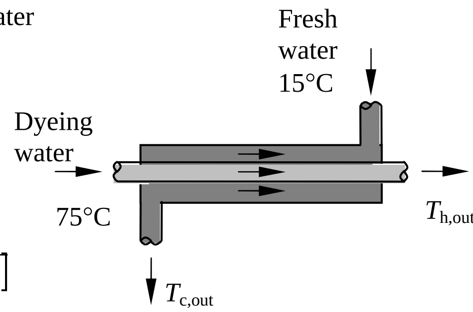
0.055
0.05
0.045
0.04
0.035
0.03
0.025
0.02
0.015
0.01
300
350
400
450
500
550
600
ṁw [kg/s]
T exhaust,in [C]

**16-72** The waste dyeing water is to be used to preheat fresh water. The outlet temperatures of each fluid and the mass flow rate are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

***Properties*** The specific heats of waste dyeing water and the fresh water are given to be $c_p$ = 4295 J/kg.°C and $c_p$ = 4180 J/kg.°C, respectively.

***Analysis*** The temperature differences between the dyeing water and the fresh water at the two ends of the heat exchanger are




$$\Delta T_1 = T_{h,in} - T_{c,out} = 75 - T_{c,out}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = T_{h,out} - 15$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{(75 - T_{c,out}) - (T_{h,out} - 15)}{\ln[(75 - T_{c,out})/(T_{h,out} - 15)]}$$

Then the rate of heat transfer can be expressed as

$$\dot{Q} = UA_s \Delta T_{lm}$$

$$35\text{ kW} = (0.625\text{ kW/m}^2.°\text{C})(1.65\text{ m}^2)\frac{(75 - T_{c,out}) - (T_{h,out} - 15)}{\ln[(75 - T_{c,out})/(T_{h,out} - 15)]} \quad \text{(Eq. 1)}$$

The rate of heat transfer can also be expressed as

$$\dot{Q} = [\dot{m}c_p(T_{h,in} - T_{h,out})]_{\substack{\text{dyeing}\\\text{water}}} \longrightarrow 35\text{ kW} = \dot{m}(4.295\text{ kJ/kg.°C})(75°\text{C} - T_{h,out}) \quad \text{(Eq. 2)}$$

$$\dot{Q} = [\dot{m}c_p(T_{h,in} - T_{h,out})]_{\text{water}} \longrightarrow 35\text{ kW} = \dot{m}(4.18\text{ kJ/kg.°C})(T_{c,out} - 15°\text{C}) \quad \text{(Eq. 3)}$$

We have three equations with three unknowns. Using an equation solver such as EES, the unknowns are determined to be

$$T_{c,out} = \mathbf{41.4°C}$$

$$T_{h,out} = \mathbf{49.3°C}$$

$$\dot{m} = \mathbf{0.317\ kg/s}$$

## The Effectiveness-NTU Method

**16-73C** When the heat transfer surface area A of the heat exchanger is known, but the outlet temperatures are not, the effectiveness-NTU method is definitely preferred.

**16-74C** The effectiveness of a heat exchanger is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate and represents how closely the heat transfer in the heat exchanger approaches to maximum possible heat transfer. Since the actual heat transfer rate can not be greater than maximum possible heat transfer rate, the effectiveness can not be greater than one. The effectiveness of a heat exchanger depends on the geometry of the heat exchanger as well as the flow arrangement.

**16-75C** For a specified fluid pair, inlet temperatures and mass flow rates, the counter-flow heat exchanger will have the highest effectiveness.

**16-76C** Once the effectiveness $\varepsilon$ is known, the rate of heat transfer and the outlet temperatures of cold and hot fluids in a heat exchanger are determined from

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = \varepsilon C_{\min}(T_{h,in} - T_{c,in})$$

$$\dot{Q} = \dot{m}_c c_{p,c}(T_{c,out} - T_{c,in})$$

$$\dot{Q} = \dot{m}_h c_{p,h}(T_{h,in} - T_{h,out})$$

**16-77C** The heat transfer in a heat exchanger will reach its maximum value when the hot fluid is cooled to the inlet temperature of the cold fluid. Therefore, the temperature of the hot fluid cannot drop below the inlet temperature of the cold fluid at any location in a heat exchanger.

**16-78C** The heat transfer in a heat exchanger will reach its maximum value when the cold fluid is heated to the inlet temperature of the hot fluid. Therefore, the temperature of the cold fluid cannot rise above the inlet temperature of the hot fluid at any location in a heat exchanger.

**16-79C** The fluid with the lower mass flow rate will experience a larger temperature change. This is clear from the relation

$$\dot{Q} = \dot{m}_c c_p \Delta T_{cold} = \dot{m}_h c_p \Delta T_{hot}$$

**16-80C** The maximum possible heat transfer rate is in a heat exchanger is determined from

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in})$$

where $C_{\min}$ is the smaller heat capacity rate. The value of $\dot{Q}_{\max}$ does not depend on the type of heat exchanger.

**16-81C** The longer heat exchanger is more likely to have a higher effectiveness.

**16-82C** The increase of effectiveness with NTU is not linear. The effectiveness increases rapidly with NTU for small values (up to abo ut NTU = 1.5), but rather slowly for larger values. Therefore, the effectiveness will not double when the length of heat exchanger is doubled.

**16-83C** A heat exchanger has the smallest effectiveness value when the heat capacity rates of two fluids are identical. Therefore, reducing the mass flow rate of cold fluid by half will increase its effectiveness.

**16-84C** When the capacity ratio is equal to zero and the number of transfer units value is greater than 5, a counter-flow heat exchanger has an effectiveness of one. In this case the exit temperature of the fluid with smaller capacity rate will equal to inlet temperature of the other fluid. For a parallel-flow heat exchanger the answer would be the same.

**16-85C** The NTU of a heat exchanger is defined as $NTU = \dfrac{UA_s}{C_{\min}} = \dfrac{UA_s}{(\dot{m}c_p)_{\min}}$ where $U$ is the overall heat transfer coefficient and $A_s$ is the heat transfer surface area of the heat exchanger. For specified values of $U$ and $C_{\min}$, the value of NTU is a measure of the heat exchanger surface area $A_s$. Because the effectiveness increases slowly for larger values of NTU, a large heat exchanger cannot be justified economically. Therefore, a heat exchanger with a very large NTU is not necessarily a good one to buy.

**16-86C** The value of effectiveness increases slowly with a large values of NTU (usually larger than 3). Therefore, doubling the size of the heat exchanger will not save much energy in this case since the increase in the effectiveness will be very small.

**16-87C** The value of effectiveness increases rapidly with small values of NTU (up to about 1.5). Therefore, tripling the NTU will cause a rapid increase in the effectiveness of the heat exchanger, and thus saves energy. I would support this proposal.

**16-88** Hot water coming from the engine of an automobile is cooled by air in the radiator. The outlet temperature of the air and the rate of heat transfer are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

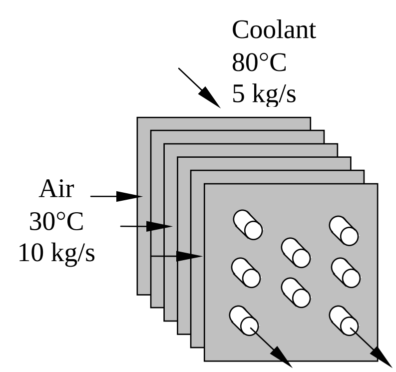
***Properties*** The specific heats of water and air are given to be 4.00 and 1.00 kJ/kg.°C, respectively.

***Analysis*** (*a*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (5\text{ kg/s})(4.00\text{ kJ/kg.°C}) = 20\text{ kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (10\text{ kg/s})(1.00\text{ kJ/kg.°C}) = 10\text{ kW/°C}$$

Therefore $\quad C_{\min} = C_c = 10\text{ kW/°C}$




which is the smaller of the two heat capacity rates. Noting that the heat capacity rate of the air is the smaller one, the outlet temperature of the air is determined from the effectiveness relation to be

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_{\min}(T_{a,out} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} \longrightarrow 0.4 = \frac{(T_{a,out} - 30)\text{°C}}{(80-30)\text{°C}} \longrightarrow T_{a,out} = \mathbf{50°C}$$

(*b*) The rate of heat transfer is determined from

$$\dot{Q} = C_{air}(T_{a,out} - T_{a,in}) = (10\text{ kW/°C})(50\text{°C} - 30\text{°C}) = \mathbf{200\ kW}$$

**16-89** The inlet and exit temperatures and the volume flow rates of hot and cold fluids in a heat exchanger are given. The rate of heat transfer to the cold water, the overall heat transfer coefficient, the fraction of heat loss, the heat transfer efficiency, the effectiveness, and the NTU of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Changes in the kinetic and potential energies of fluid streams are negligible. **3** Fluid properties are constant.

***Properties*** The densities of hot water and cold water at the average temperatures of (71.5+58.2)/2 = 64.9°C and (19.7+27.8)/2 = 23.8°C are 980.5 and 997.3 kg/m$^3$, respectively. The specific heat at the average temperature is 4187 J/kg.°C for hot water and 4180 J/kg.°C for cold water (Table A-15).

***Analysis*** (a) The mass flow rates are

$$\dot{m}_h = \rho_h \dot{V}_h = (980.5\ \text{kg/m}^3)(0.00105/60\ \text{m}^3/\text{s}) = 0.0172\ \text{kg/s}$$

$$\dot{m}_c = \rho_c \dot{V}_c = (997.3\ \text{kg/m}^3)(0.00155/60\ \text{m}^3/\text{s}) = 0.0258\ \text{kg/s}$$

The rates of heat transfer from the hot water and to the cold water are

$$\dot{Q}_h = [\dot{m}c_p(T_{in} - T_{out})]_\text{h} = (0.0172\ \text{kg/s})(4187\ \text{kJ/kg.°C})(71.5°\text{C} - 58.2°\text{C}) = 957.8\ \text{W}$$

$$\dot{Q}_c = [\dot{m}c_p(T_{out} - T_{in})]_\text{c} = (0.0258\ \text{kg/s})(4180\ \text{kJ/kg.°C})(27.8°\text{C} - 19.2°\text{C}) = \mathbf{873.5\ W}$$

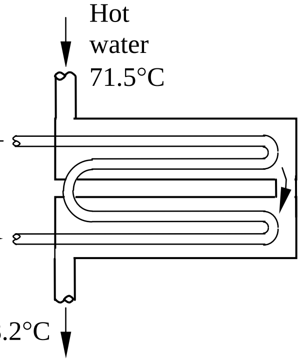
(b) The number of shell and tubes are not specified in the problem. Therefore, we take the correction factor to be unity in the following calculations. The logarithmic mean temperature difference and the overall heat transfer coefficient are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 71.5°\text{C} - 27.8°\text{C} = 43.7°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 58.2°\text{C} - 19.7°\text{C} = 38.5°\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{43.7 - 38.5}{\ln\left(\frac{43.7}{38.5}\right)} = 41.0°\text{C}$$

$$U = \frac{\dot{Q}_{hc,m}}{A\Delta T_{lm}} = \frac{(957.8 + 873.5)/2\ \text{W}}{(0.02\ \text{m}^2)(41.0°\text{C})} = \mathbf{1117\ W/m^2 \cdot C}$$




Note that we used the average of two heat transfer rates in calculations.

(c) The fraction of heat loss and the heat transfer efficiency are

$$f_{loss} = \frac{\dot{Q}_h - \dot{Q}_c}{\dot{Q}_h} = \frac{957.8 - 873.5}{957.8} = 0.088 = \mathbf{8.8\%}$$

$$\eta = \frac{\dot{Q}_c}{\dot{Q}_h} = \frac{873.5}{957.8} = 0.912 = \mathbf{91.2\%}$$

(d) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.0172\ \text{kg/s})(4187\ \text{kJ/kg.°C}) = 72.02\ \text{W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.0258\ \text{kg/s})(4180\ \text{kJ/kg.°C}) = 107.8\ \text{W/°C}$$

Therefore

$$C_{\min} = C_h = 72.02\ \text{W/°C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (72.02\ \text{W/°C})(71.5°\text{C} - 19.7°\text{C}) = 3731\ \text{W}$$

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{(957.8 + 873.5)/2\ \text{kW}}{3731\ \text{kW}} = 0.245 = \mathbf{24.5\%}$$

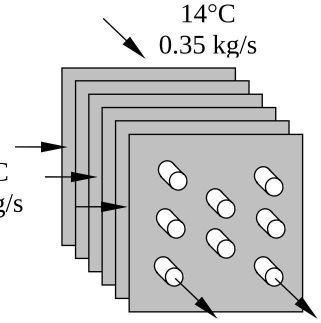
One again we used the average heat transfer rate. We could have used the smaller or greater heat transfer rates in calculations. The NTU of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{\min}} = \frac{(1117\ \text{W/m}^2 \cdot \text{C})(0.02\ \text{m}^2)}{72.02\ \text{W/°C}} = \mathbf{0.310}$$

**16-90** Water is heated by a hot water stream in a heat exchanger. The maximum outlet temperature of the cold water and the effectiveness of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of water and air are given to be 4.18 and 1.0 kJ/kg.°C.




***Analysis*** The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.8\text{ kg/s})(1.0\text{ kJ/kg.°C}) = 0.8\text{ kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.35\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 1.463\text{ kW/°C}$$

Therefore

$$C_{\min} = C_h = 0.8\text{ kW/°C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (0.8\text{ kW/°C})(65\text{°C} - 14\text{°C}) = 40.80\text{ kW}$$

The maximum outlet temperature of the cold fluid is determined to be

$$\dot{Q}_{\max} = C_c(T_{c,out,\max} - T_{c,in}) \longrightarrow T_{c,out,\max} = T_{c,in} + \frac{\dot{Q}_{\max}}{C_c} = 14\text{°C} + \frac{40.80\text{ kW}}{1.463\text{ kW/°C}} = \mathbf{41.9°C}$$

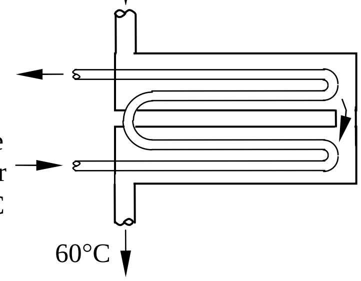
The actual rate of heat transfer and the effectiveness of the heat exchanger are

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = (0.8\text{ kW/°C})(65\text{°C} - 25\text{°C}) = 32\text{ kW}$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{32\text{ kW}}{40.8\text{ kW}} = \mathbf{0.784}$$

**16-91** Lake water is used to condense steam in a shell and tube heat exchanger. The outlet temperature of the water and the required tube length are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.




***Properties*** The properties of water are given in problem statement. The enthalpy of vaporization of water at 60°C is 2359 kJ/kg (Table A-15).

***Analysis*** (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}h_{fg} = (2.5\text{ kg/s})(2359\text{ kJ/kg}) = 5898\text{ kW}$$

The outlet temperature of water is determined from

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_c} = 20°\text{C} + \frac{5898\text{ kW}}{(200\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})} = \mathbf{27.1°C}$$

(b) The Reynold number is

$$\text{Re} = \frac{4\dot{m}}{N_{tube}\pi D\mu} = \frac{4(200\text{ kg/s})}{(300)\pi(0.025\text{ m})(8\times10^{-4}\text{ kg/m}\cdot\text{s})} = 42{,}440$$

which is greater than 10,000. Therefore, we have turbulent flow. We assume fully developed flow and evaluate the Nusselt number from

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(42{,}440)^{0.8}(6)^{0.4} = 237.2$$

Heat transfer coefficient on the inner surface of the tubes is

$$h_i = \frac{k}{D}Nu = \frac{0.6\text{ W/m.°C}}{0.025\text{ m}}(237.2) = 5694\text{ W/m}^2.°\text{C}$$

Disregarding the thermal resistance of the tube wall the overall heat transfer coefficient is determined from

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{5694} + \frac{1}{8500}} = 3410\text{ W/m}^2\cdot°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 60°\text{C} - 27.1°\text{C} = 32.9°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 20°\text{C} = 40°\text{C}$$

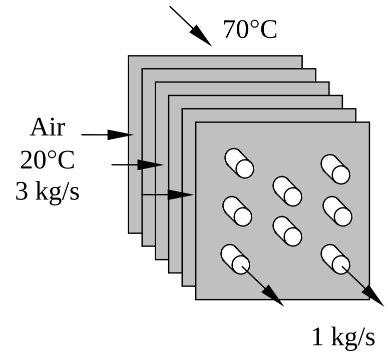
$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{32.9 - 40}{\ln\left(\frac{32.9}{40}\right)} = 36.3°\text{C}$$

Noting that each tube makes two passes and taking the correction factor to be unity, the tube length per pass is determined to be

$$\dot{Q} = UAF\Delta T_{lm} \to L = \frac{\dot{Q}}{U(\pi D)F\Delta T_{lm}} = \frac{5898\text{ kW}}{(3.410\text{ kW/m}^2\cdot\text{C})\left[2\times300\times\pi\times0.025\text{ m}^2\right](1)(36.3°\text{C})} = \mathbf{1.01\,m}$$

**16-92** Air is heated by a hot water stream in a cross-flow heat exchanger. The maximum heat transfer rate and the outlet temperatures of the cold and hot fluid streams are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.




***Properties*** The specific heats of water and air are given to be 4.19 and 1.005 kJ/kg.°C.

***Analysis*** The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (1\,\text{kg/s})(4190\,\text{J/kg.°C}) = 4190\,\text{W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (3\,\text{kg/s})(1005\,\text{J/kg.°C}) = 3015\,\text{W/°C}$$

Therefore

$$C_{\min} = C_c = 3015\,\text{W/°C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (3015\,\text{W/°C})(70°\text{C} - 20°\text{C}) = 150{,}750\,\text{W} = \mathbf{150.8\,kW}$$

The outlet temperatures of the cold and the hot streams in this limiting case are determined to be

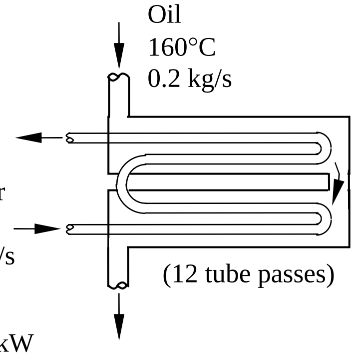
$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20°\text{C} + \frac{150.75\,\text{kW}}{3.015\,\text{kW/°C}} = \mathbf{70°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 70°\text{C} - \frac{150.75\,\text{kW}}{4.19\,\text{kW/°C}} = \mathbf{34.0°C}$$

**16-93** Hot oil is to be cooled by water in a heat exchanger. The mass flow rates and the inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined. √

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The thickness of the tube is negligible since it is thin-walled. **5** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg.°C, respectively.




***Analysis*** The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.2\text{ kg/s})(2200\text{ J/kg.°C}) = 440\text{ W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.1\text{ kg/s})(4180\text{ J/kg.°C}) = 418\text{ W/°C}$$

Therefore, $C_{\min} = C_c = 418\text{ W/°C}$

and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{418}{440} = 0.95$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (418\text{ W/°C})(160\text{°C} - 18\text{°C}) = 59.36\text{ kW}$$

The heat transfer surface area is

$$A_s = n(\pi DL) = (12)(\pi)(0.018\text{ m})(3\text{ m}) = 2.04\text{ m}^2$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(340\text{ W/m}^2.\text{°C})(2.04\text{ m}^2)}{418\text{ W/°C}} = 1.659$$

Then the effectiveness of this heat exchanger corresponding to $c$ = 0.95 and NTU = 1.659 is determined from Fig. 16-26d to be

$$\varepsilon = 0.61$$

Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.61)(59.36\text{ kW}) = \mathbf{36.2\text{ kW}}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 18\text{°C} + \frac{36.2\text{ kW}}{0.418\text{ kW/°C}} = \mathbf{104.6\text{°C}}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 160\text{°C} - \frac{36.2\text{ kW}}{0.44\text{ kW/°C}} = \mathbf{77.7\text{°C}}$$

**16-94** Inlet and outlet temperatures of the hot and cold fluids in a double-pipe heat exchanger are given. It is to be determined whether this is a parallel-flow or counter-flow heat exchanger and the effectiveness of it.

***Analysis*** This is a counter-flow heat exchanger because in the parallel-flow heat exchangers the outlet temperature of the cold fluid (55°C in this case) cannot exceed the outlet temperature of the hot fluid, which is (45°C in this case). Noting that the mass flow rates of both hot and cold oil streams are the same, we have $C_{\min} = C_{\max}$. Then the effectiveness of this heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{C_h(T_{h,in} - T_{h,out})}{C_h(T_{h,in} - T_{c,in})} = \frac{80\text{°C} - 45\text{°C}}{80\text{°C} - 20\text{°C}} = \mathbf{0.583}$$

**16-95E** Inlet and outlet temperatures of the hot and cold fluids in a double-pipe heat exchanger are given. It is to be determined the fluid, which has the smaller heat capacity rate and the effectiveness of the heat exchanger.

***Analysis*** Hot water has the smaller heat capacity rate since it experiences a greater temperature change. The effectiveness of this heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{C_h(T_{h,in} - T_{h,out})}{C_h(T_{h,in} - T_{c,in})} = \frac{190°\text{F} - 100°\text{F}}{190°\text{F} - 70°\text{F}} = \mathbf{0.75}$$

**16-96** A chemical is heated by water in a heat exchanger. The mass flow rates and the inlet temperatures are given. The outlet temperatures of both fluids are to be determined.

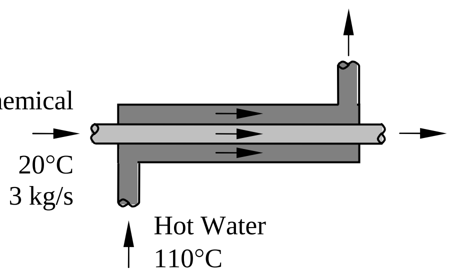
***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The thickness of the tube is negligible since tube is thin-walled. **5** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the water and chemical are given to be 4.18 and 1.8 kJ/kg.°C, respectively.

***Analysis*** The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (2\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 8.36\text{ kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (3\text{ kg/s})(1.8\text{ kJ/kg.°C}) = 5.40\text{ kW/°C}$$




Therefore, $C_{\min} = C_c = 5.4\text{ kW/°C}$

and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{5.40}{8.36} = 0.646$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (5.4\text{ kW/°C})(110°\text{C} - 20°\text{C}) = 486\text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(1.2\text{ kW/m}^2.°\text{C})(7\text{ m}^2)}{5.4\text{ kW/°C}} = 1.556$$

Then the effectiveness of this parallel-flow heat exchanger corresponding to $c$ = 0.646 and NTU=1.556 is determined from

$$\varepsilon = \frac{1 - \exp[-NTU(1+c)]}{1+c} = \frac{1 - \exp[-1.556(1+0.646)]}{1+0.646} = 0.56$$

Then the actual rate of heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.56)(486\text{ kW}) = 272.2\text{ kW}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20°\text{C} + \frac{272.2\text{ kW}}{5.4\text{ kW / °C}} = \mathbf{70.4°C}$$

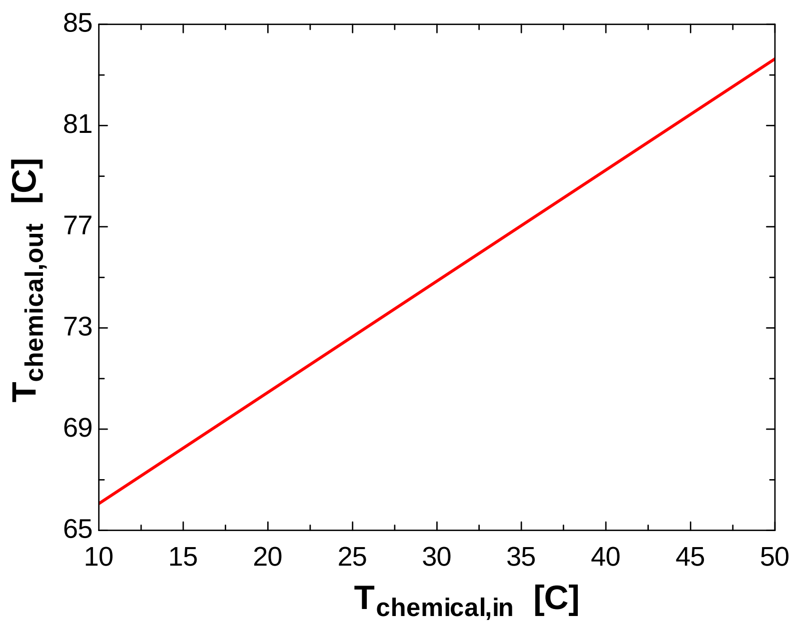
$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 110°\text{C} - \frac{272.2\text{ kW}}{8.36\text{ kW/°C}} = \mathbf{77.4°C}$$

**16-97 EES** Prob. 16-96 is reconsidered. The effects of the inlet temperatures of the chemical and the water on their outlet temperatures are to be investigated.

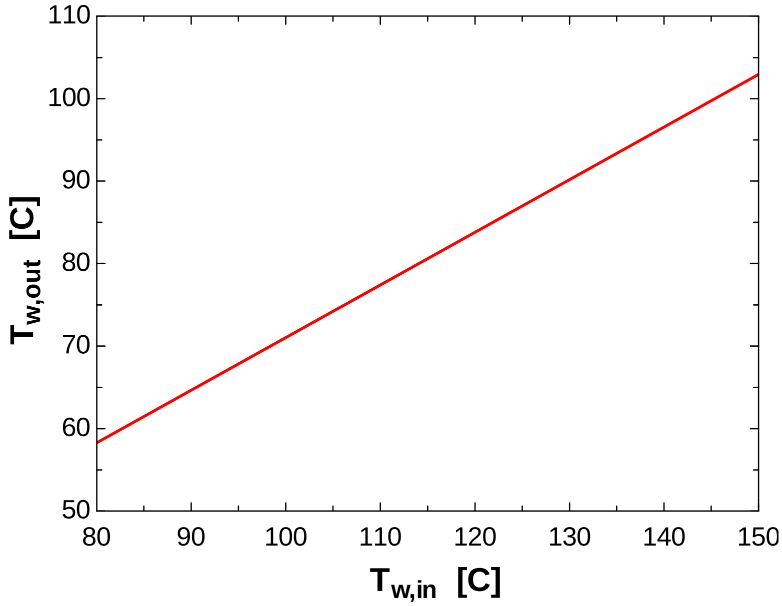
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_chemical_in=20 [C]
c_p_chemical=1.8 [kJ/kg-C]
m_dot_chemical=3 [kg/s]
T_w_in=110 [C]
m_dot_w=2 [kg/s]
c_p_w=4.18 [kJ/kg-C]
A=7 [m^2]
U=1.2 [kW/m^2-C]

"ANALYSIS"
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below,
we use LMTD method. Both methods give the same results."
DELTAT_1=T_w_in-T_chemical_in
DELTAT_2=T_w_out-T_chemical_out
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_chemical*c_p_chemical*(T_chemical_out-T_chemical_in)
Q_dot=m_dot_w*c_p_w*(T_w_in-T_w_out)
```

| $T_{chemical, in}$ [C] | $T_{chemical, out}$ [C] |
|---|---|
| 10 | 66.06 |
| 12 | 66.94 |
| 14 | 67.82 |
| 16 | 68.7 |
| 18 | 69.58 |
| 20 | 70.45 |
| 22 | 71.33 |
| 24 | 72.21 |
| 26 | 73.09 |
| 28 | 73.97 |
| 30 | 74.85 |
| 32 | 75.73 |
| 34 | 76.61 |
| 36 | 77.48 |
| 38 | 78.36 |
| 40 | 79.24 |
| 42 | 80.12 |
| 44 | 81 |
| 46 | 81.88 |
| 48 | 82.76 |
| 50 | 83.64 |



| $T_{w, in}$ [C] | $T_{w, out}$ [C] |
|---|---|
| 80 | 58.27 |
| 85 | 61.46 |
| 90 | 64.65 |
| 95 | 67.84 |
| 100 | 71.03 |
| 105 | 74.22 |
| 110 | 77.41 |
| 115 | 80.6 |
| 120 | 83.79 |
| 125 | 86.98 |
| 130 | 90.17 |
| 135 | 93.36 |
| 140 | 96.55 |
| 145 | 99.74 |
| 150 | 102.9 |



**16-98** Water is heated by hot air in a heat exchanger. The mass flow rates and the inlet temperatures are given. The heat transfer surface area of the heat exchanger on the water side is to be determined.

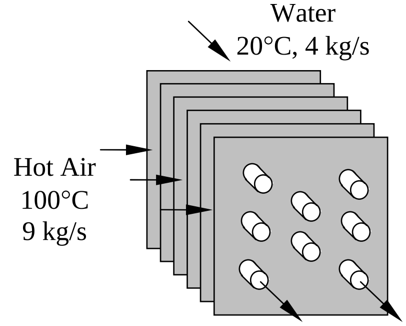
***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the water and air are given to be 4.18 and 1.01kJ/kg.°C, respectively.

***Analysis*** The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (4\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 16.72\text{ kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (9\text{ kg/s})(1.01\text{ kJ/kg.°C}) = 9.09\text{ kW/°C}$$




Therefore, $C_{\min} = C_c = 9.09\text{ kW/°C}$

and $C = \dfrac{C_{\min}}{C_{\max}} = \dfrac{9.09}{16.72} = 0.544$

Then the NTU of this heat exchanger corresponding to $c = 0.544$ and $\varepsilon = 0.65$ is determined from Fig. 16-26 to be

NTU = 1.5

Then the surface area of this heat exchanger becomes

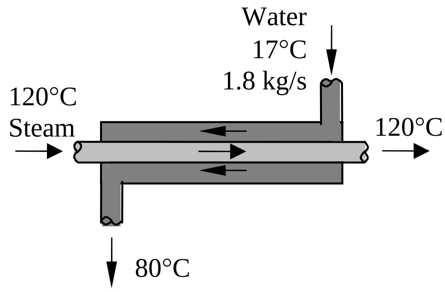
$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{\text{NTU}\, C_{\min}}{U} = \frac{(1.5)(9.09\text{ kW/°C})}{0.260\text{ kW/m}^2\text{.°C}} = \mathbf{52.4\ m^2}$$

**16-99** Water is heated by steam condensing in a condenser. The required length of the tube is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heat of the water is given to be 4.18 kJ/kg.°C. The heat of vaporization of water at 120°C is given to be 2203 kJ/kg.

***Analysis*** (*a*) The temperature differences between the steam and the water at the two ends of the condenser are




$$\Delta T_1 = T_{h,in} - T_{c,out} = 120°\text{C} - 80°\text{C} = 40°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 120°\text{C} - 17°\text{C} = 103°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 103}{\ln(40/103)} = 66.6°\text{C}$$

The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) = (1.8 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(80°\text{C} - 17°\text{C}) = 474.0 \text{ kW}$$

The surface area of heat transfer is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{474.0 \text{ kW}}{0.7 \text{ kW/m}^2.°\text{C})(66.6°\text{C})} = 10.17 \text{ m}^2$$

The length of tube required then becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{10.17 \text{ m}^2}{\pi(0.025 \text{ m})} = \mathbf{129.5 \text{ m}}$$

(*b*) The maximum rate of heat transfer rate is

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in}) = (1.8 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(120°\text{C -17°C}) = 775.0 \text{ kW}$$

Then the effectiveness of this heat exchanger becomes

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{474 \text{ kW}}{775 \text{ kW}} = 0.612$$

The NTU of this heat exchanger is determined using the relation in Table 16-5 to be

$$\text{NTU} = -\ln(1-\varepsilon) = -\ln(1-0.612) = 0.947$$

The surface area is

$$NTU = \frac{UA_s}{C_{min}} \longrightarrow A_s = \frac{NTU\ C_{min}}{U} = \frac{(0.947)(1.8 \text{ kg/s})(4.18 \text{ kJ/kg.°C})}{0.7 \text{ kW/m}^2.°\text{C}} = 10.18 \text{ m}^2$$

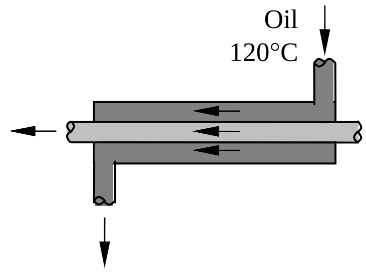
Finally, the length of tube required is

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{10.18 \text{ m}^2}{\pi(0.025 \text{ m})} = \mathbf{129.6 \text{ m}}$$

**16-100** Ethanol is vaporized by hot oil in a double-pipe parallel-flow heat exchanger. The outlet temperature and the mass flow rate of oil are to be determined using the LMTD and NTU methods.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heat of oil is given to be 2.2 kJ/kg.°C. The heat of vaporization of ethanol at 78°C is given to be 846 kJ/kg.




***Analysis*** (*a*) The rate of heat transfer is

$$\dot{Q} = \dot{m}h_{fg} = (0.03\ \text{kg/s})(846\ \text{kJ/kg}) = 25.38\ \text{kW}$$

The log mean temperature difference is

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow \Delta T_{lm} = \frac{\dot{Q}}{UA_s} = \frac{25{,}380\ \text{W}}{(320\ \text{W/m}^2.°\text{C})(6.2\ \text{m}^2)} = 12.8°\text{C}$$

The outlet temperature of the hot fluid can be determined as follows

$$\Delta T_1 = T_{h,in} - T_{c,in} = 120°\text{C} - 78°\text{C} = 42°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = T_{h,out} - 78°\text{C}$$

and $$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{42 - (T_{h,out} - 78)}{\ln[42/(T_{h,out} - 78)]} = 12.8°\text{C}$$

whose solution is $T_{h,out} = \mathbf{79.8°C}$

Then the mass flow rate of the hot oil becomes

$$\dot{Q} = \dot{m}c_p(T_{h,in} - T_{h,out}) \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{h,in} - T_{h,out})} = \frac{25{,}380\ \text{W}}{(2200\ \text{J/kg.°C})(120°\text{C} - 79.8°\text{C})} = \mathbf{0.287\ kg/s}$$

(*b*) The heat capacity rate $C = \dot{m}c_p$ of a fluid condensing or evaporating in a heat exchanger is infinity, and thus $C = C_{\min}/C_{\max} = 0$.

The effectiveness in this case is determined from $\varepsilon = 1 - e^{-NTU}$

where $$NTU = \frac{UA_s}{C_{\min}} = \frac{(320\ \text{W/m}^2.°\text{C})(6.2\ \text{m}^2)}{(\dot{m},\ \text{kg/s})(2200\ \text{J/kg.°C})}$$

and $$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in})$$

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_{\min}(T_{h,in} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{120 - T_{h,out}}{120 - 78}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = 25{,}380\ \text{W}$$

$$\dot{Q} = \dot{m} \times 2200(120 - T_{h,out}) = 25{,}380\ \text{W} \qquad (1)$$

Also $$\frac{120 - T_{h,out}}{120 - 78} = 1 - e^{-\frac{6.2\times 320}{\dot{m}\times 2200}} \qquad (2)$$

Solving (1) and (2) simultaneously gives

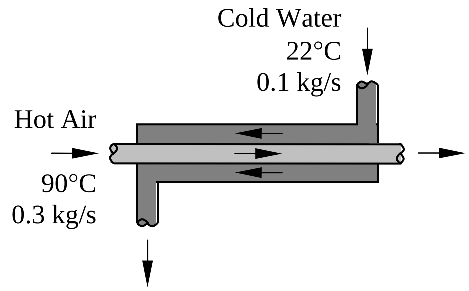
$\dot{m}_h = \mathbf{0.287\ kg/s}$ and $T_{h,out} = \mathbf{79.8°C}$

**16-101** Water is heated by solar-heated hot air in a heat exchanger. The mass flow rates and the inlet temperatures are given. The outlet temperatures of the water and the air are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the water and air are given to be 4.18 and 1.01 kJ/kg.°C, respectively.

***Analysis*** The heat capacity rates of the hot and cold fluids are




$$C_h = \dot{m}_h c_{ph} = (0.3\text{kg/s})(1010\ \text{J/kg.°C}) = 303\ \text{W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.1\ \text{kg/s})(4180\ \text{J/kg.°C}) = 418\ \text{W/°C}$$

Therefore, $C_{\min} = C_c = 303\ \text{W/°C}$

and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{303}{418} = 0.725$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (303\ \text{W/°C})(90\text{°C - }22\text{°C}) = 20{,}604\ \text{kW}$$

The heat transfer surface area is

$$A_s = \pi DL = (\pi)(0.012\ \text{m})(12\ \text{m}) = 0.45\ \text{m}^2$$

Then the NTU of this heat exchanger becomes

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(80\ \text{W/m}^2.\text{°C})\ (0.45\ \text{m}^2)}{303\ \text{W/°C}} = 0.119$$

The effectiveness of this counter-flow heat exchanger corresponding to $c$ = 0.725 and NTU = 0.119 is determined using the relation in Table 16-4 to be

$$\varepsilon = \frac{1-\exp[-NTU(1-c)]}{1-c\exp[-NTU(1-c)]} = \frac{1-\exp[-0.119(1-0.725)]}{1-0.725\exp[-0.119(1-0.725)]} = 0.108$$

Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.108)(20{,}604\ \text{W}) = 2225.2\ \text{W}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 22\text{°C} + \frac{2225.2\ \text{W}}{418\ \text{W / °C}} = \mathbf{27.3°C}$$

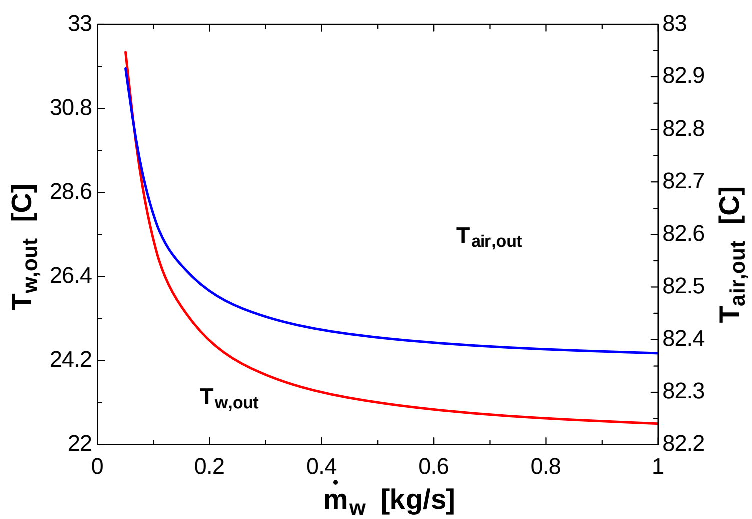
$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 90\text{°C} - \frac{2225.2\ \text{W}}{303\ \text{W/°C}} = \mathbf{82.7°C}$$

**16-102 EES** Prob. 16-101 is reconsidered. The effects of the mass flow rate of water and the tube length on the outlet temperatures of water and air are to be investigated.

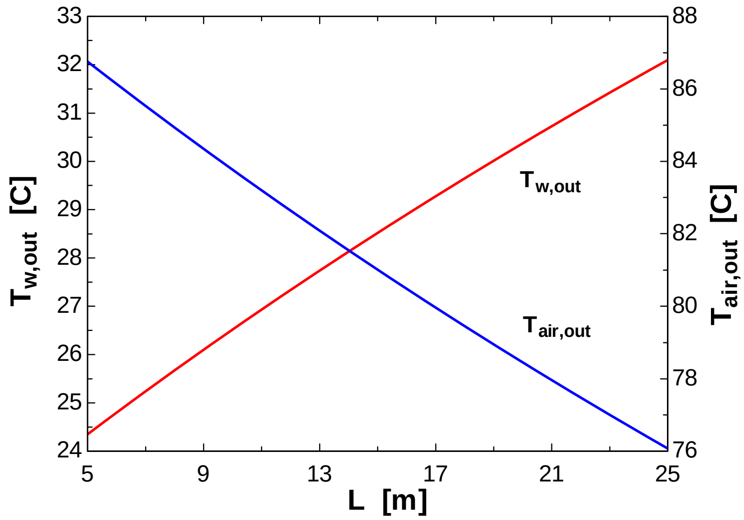
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_air_in=90 [C]
m_dot_air=0.3 [kg/s]
c_p_air=1.01 [kJ/kg-C]
T_w_in=22 [C]
m_dot_w=0.1 [kg/s]
c_p_w=4.18 [kJ/kg-C]
U=0.080 [kW/m^2-C]
L=12 [m]
D=0.012 [m]

"ANALYSIS"
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below,
we use LMTD method. Both methods give the same results."
DELTAT_1=T_air_in-T_w_out
DELTAT_2=T_air_out-T_w_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
A=pi*D*L
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_air*c_p_air*(T_air_in-T_air_out)
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
```

| $m_w$ [kg/s] | $T_{w,\,out}$ [C] | $T_{air,\,out}$ [C] |
|---|---|---|
| 0.05 | 32.27 | 82.92 |
| 0.1 | 27.34 | 82.64 |
| 0.15 | 25.6 | 82.54 |
| 0.2 | 24.72 | 82.49 |
| 0.25 | 24.19 | 82.46 |
| 0.3 | 23.83 | 82.44 |
| 0.35 | 23.57 | 82.43 |
| 0.4 | 23.37 | 82.42 |
| 0.45 | 23.22 | 82.41 |
| 0.5 | 23.1 | 82.4 |
| 0.55 | 23 | 82.4 |
| 0.6 | 22.92 | 82.39 |
| 0.65 | 22.85 | 82.39 |
| 0.7 | 22.79 | 82.39 |
| 0.75 | 22.74 | 82.38 |
| 0.8 | 22.69 | 82.38 |
| 0.85 | 22.65 | 82.38 |
| 0.9 | 22.61 | 82.38 |
| 0.95 | 22.58 | 82.38 |
| 1 | 22.55 | 82.37 |



| L [m] | $T_{w, out}$ [C] | $T_{air, out}$ [C] |
|---|---|---|
| 5 | 24.35 | 86.76 |
| 6 | 24.8 | 86.14 |
| 7 | 25.24 | 85.53 |
| 8 | 25.67 | 84.93 |
| 9 | 26.1 | 84.35 |
| 10 | 26.52 | 83.77 |
| 11 | 26.93 | 83.2 |
| 12 | 27.34 | 82.64 |
| 13 | 27.74 | 82.09 |
| 14 | 28.13 | 81.54 |
| 15 | 28.52 | 81.01 |
| 16 | 28.9 | 80.48 |
| 17 | 29.28 | 79.96 |
| 18 | 29.65 | 79.45 |
| 19 | 30.01 | 78.95 |
| 20 | 30.37 | 78.45 |
| 21 | 30.73 | 77.96 |
| 22 | 31.08 | 77.48 |
| 23 | 31.42 | 77 |
| 24 | 31.76 | 76.53 |
| 25 | 32.1 | 76.07 |



**16-103E** Oil is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient of this heat exchanger is to be determined using both the LMTD and NTU methods.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The thickness of the tube is negligible since it is thin-walled.

***Properties*** The specific heats of the water and oil are given to be 1.0 and 0.525 Btu/lbm.°F, respectively.

***Analysis*** (*a*) The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in-} T_{h,out}) = (5\text{ lbm/s})(0.525\text{ Btu/lbm.°F})(300 - 105\text{°F}) = 511.9\text{ Btu/s}$$

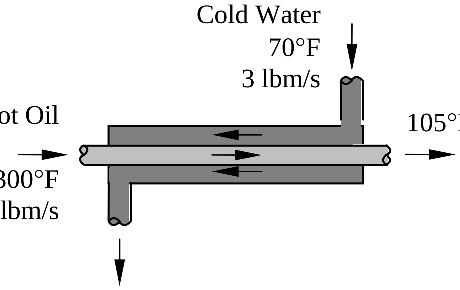
The outlet temperature of the cold fluid is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 70\text{°F} + \frac{511.9\text{ Btu/s}}{(3\text{ lbm/s})(1.0\text{ Btu/lbm.°F})} = 240.6\text{°F}$$

The temperature differences between the two fluids at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 300\text{°F} - 240.6\text{°F} = 59.4\text{°F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 105\text{°F} - 70\text{°F} = 35\text{°F}$$




The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{59.4 - 35}{\ln(59.4/35)} = 46.1\text{°F}$$

Then the overall heat transfer coefficient becomes

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow U = \frac{\dot{Q}}{A_s\Delta T_{lm}} = \frac{511.9\text{ Btu/s}}{\pi(5/12\text{ m})(200\text{ ft})(46.1\text{°F})} = \mathbf{0.0424\ Btu/s.ft^2.°F}$$

(*b*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (5\text{ lbm/s})(0.525\text{ Btu/lbm.°F}) = 2.625\text{ Btu/s.°F}$$

$$C_c = \dot{m}_c c_{pc} = (3\text{ lbm/s})(1.0\text{ Btu/lbm.°F}) = 3.0\text{ Btu/s.°F}$$

Therefore, $C_{\min} = C_h = 2.625\text{ Btu/s.°F}$ and $c = \frac{C_{\min}}{C_{\max}} = \frac{2.625}{3.0} = 0.875$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (2.625\text{ Btu/s.°F})(300\text{°F - }70\text{°F}) = 603.75\text{ Btu/s}$$

The actual rate of heat transfer and the effectiveness are

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = (2.625\text{ Btu/s.°F})(300\text{°F -}105\text{°F}) = 511.9\text{ Btu/s}$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{511.9}{603.75} = 0.85$$

The NTU of this heat exchanger is determined using the relation in Table 16-5 to be

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) = \frac{1}{0.875-1}\ln\left(\frac{0.85-1}{0.85\times 0.875-1}\right) = 4.28$$

The heat transfer surface area of the heat exchanger is

$$A_s = \pi DL = \pi(5/12\text{ ft})(200\text{ ft}) = 261.8\text{ ft}^2$$

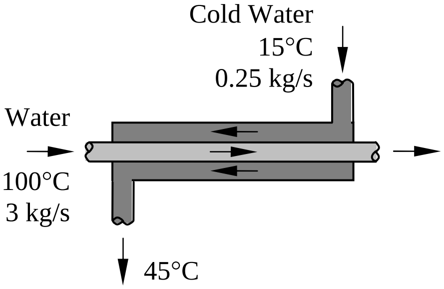
and $$NTU = \frac{UA_s}{C_{\min}} \longrightarrow U = \frac{NTU\ C_{\min}}{A_s} = \frac{(4.28)(2.625\text{ Btu/s.°F})}{261.8\text{ ft}^2} = \mathbf{0.0429\ Btu/s.ft^2.°F}$$

**16-104** Cold water is heated by hot water in a heat exchanger. The net rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

***Properties*** The specific heats of the cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

***Analysis*** The heat capacity rates of the hot and cold fluids are




$$C_h = \dot{m}_h c_{ph} = (0.25\text{ kg/s})(4180\text{ J/kg.°C}) = 1045\text{ W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (3\text{ kg/s})(4190\text{ J/kg.°C}) = 12{,}570\text{ W/°C}$$

Therefore, $C_{\min} = C_c = 1045\text{ W/°C}$

and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{1045}{12{,}570} = 0.083$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (1045\text{ W/°C})(100°\text{C - }15°\text{C}) = 88{,}825\text{ W}$$

The actual rate of heat transfer is

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = (1045\text{ W/°C})(45°\text{C} - 15°\text{C}) = \mathbf{31{,}350\text{ W}}$$

Then the effectiveness of this heat exchanger becomes

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{31{,}350}{88{,}825} = 0.35$$

The NTU of this heat exchanger is determined using the relation in Table 16-5 to be

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) = \frac{1}{0.083-1}\ln\left(\frac{0.35-1}{0.35\times 0.083-1}\right) = 0.438$$

Then the surface area of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{\min}} \longrightarrow A = \frac{NTU\ C_{\min}}{U} = \frac{(0.438)(1045\text{ W/°C})}{950\text{ W/m}^2\text{.°C}} = \mathbf{0.482\text{ m}^2}$$

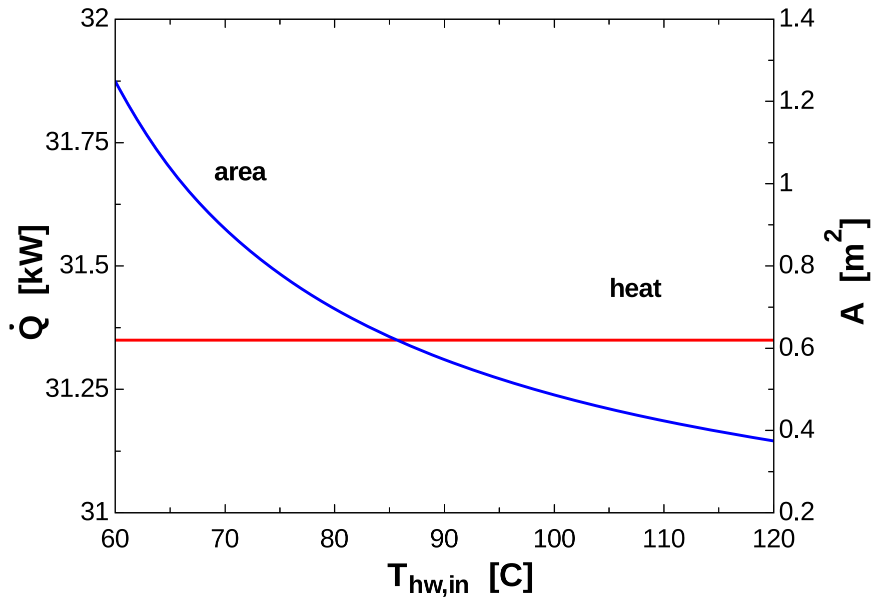
**16-105 EES** Prob. 16-104 is reconsidered. The effects of the inlet temperature of hot water and the heat transfer coefficient on the rate of heat transfer and the surface area are to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

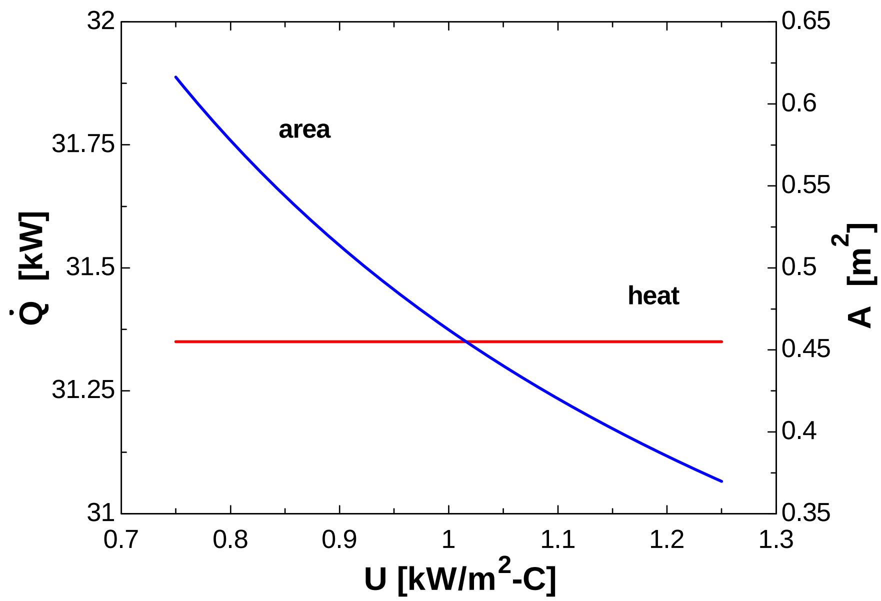
```
"GIVEN"
T_cw_in=15 [C]
T_cw_out=45 [C]
m_dot_cw=0.25 [kg/s]
c_p_cw=4.18 [kJ/kg-C]
T_hw_in=100 [C]
m_dot_hw=3 [kg/s]
c_p_hw=4.19 [kJ/kg-C]
U=0.95 [kW/m^2-C]

"ANALYSIS"
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use LMTD method. Both methods give the same results."
DELTAT_1=T_hw_in-T_cw_out
DELTAT_2=T_hw_out-T_cw_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_hw*c_p_hw*(T_hw_in-T_hw_out)
Q_dot=m_dot_cw*c_p_cw*(T_cw_out-T_cw_in)
```

| $T_{hw, in}$ [C] | Q [kW] | A [m$^2$] |
|---|---|---|
| 60 | 31.35 | 1.25 |
| 65 | 31.35 | 1.038 |
| 70 | 31.35 | 0.8903 |
| 75 | 31.35 | 0.7807 |
| 80 | 31.35 | 0.6957 |
| 85 | 31.35 | 0.6279 |
| 90 | 31.35 | 0.5723 |
| 95 | 31.35 | 0.5259 |
| 100 | 31.35 | 0.4865 |
| 105 | 31.35 | 0.4527 |
| 110 | 31.35 | 0.4234 |
| 115 | 31.35 | 0.3976 |
| 120 | 31.35 | 0.3748 |

| U [kW/m$^2$-C] | Q [kW] | A [m$^2$] |
|---|---|---|
| 0.75 | 31.35 | 0.6163 |
| 0.8 | 31.35 | 0.5778 |
| 0.85 | 31.35 | 0.5438 |
| 0.9 | 31.35 | 0.5136 |
| 0.95 | 31.35 | 0.4865 |
| 1 | 31.35 | 0.4622 |
| 1.05 | 31.35 | 0.4402 |
| 1.1 | 31.35 | 0.4202 |
| 1.15 | 31.35 | 0.4019 |
| 1.2 | 31.35 | 0.3852 |
| 1.25 | 31.35 | 0.3698 |



32
31.75
31.5
31.25
31
1.4
1.2
1
0.8
0.6
0.4
0.2
60
70
80
90
100
110
120
area
heat
Q̇ [kW]
A [m2]
Thw,in [C]

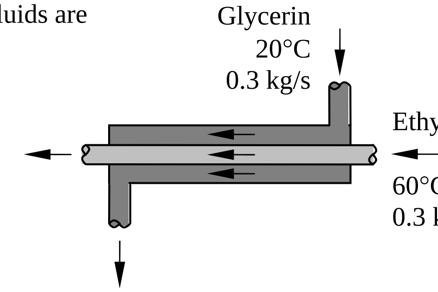
32
31.75
31.5
31.25
31
0.65
0.6
0.55
0.5
0.45
0.4
0.35
0.7
0.8
0.9
1
1.1
1.2
1.3
area
heat
Q̇ [kW]
A [m2]
U [kW/m2-C]

**16-106** Glycerin is heated by ethylene glycol in a heat exchanger. Mass flow rates and inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

***Properties*** The specific heats of the glycerin and ethylene glycol are given to be 2.4 and 2.5 kJ/kg.°C, respectively.

***Analysis*** (*a*) The heat capacity rates of the hot and cold fluids are




$$C_h = \dot{m}_h c_{ph} = (0.3\text{ kg/s})(2400\text{ J/kg.°C}) = 720\text{ W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.3\text{ kg/s})(2500\text{ J/kg.°C}) = 750\text{ W/°C}$$

Therefore, $C_{\min} = C_h = 720\text{ W/°C}$

and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{720}{750} = 0.96$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (720\text{ W/°C})(60\text{°C} - 20\text{°C}) = 28.8\text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(380\text{ W/m}^2.\text{°C})(5.3\text{ m}^2)}{720\text{ W/°C}} = 2.797$$

Effectiveness of this heat exchanger corresponding to $c$ = 0.96 and NTU = 2.797 is determined using the proper relation in Table 16-4

$$\varepsilon = \frac{1 - \exp[-NTU(1+c)]}{1+c} = \frac{1 - \exp[-2.797(1+0.96)]}{1+0.96} = 0.508$$

Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.508)(28.8\text{ kW}) = \mathbf{14.63\text{ kW}}$$

(*b*) Finally, the outlet temperatures of the cold and the hot fluid streams are determined from

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20\text{°C} + \frac{14.63\text{ kW}}{0.72\text{ kW / °C}} = \mathbf{40.3\text{°C}}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 60\text{°C} - \frac{14.63\text{ kW}}{0.75\text{ kW/°C}} = \mathbf{40.5\text{°C}}$$

**16-107** Water is heated by hot air in a cross-flow heat exchanger. Mass flow rates and inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

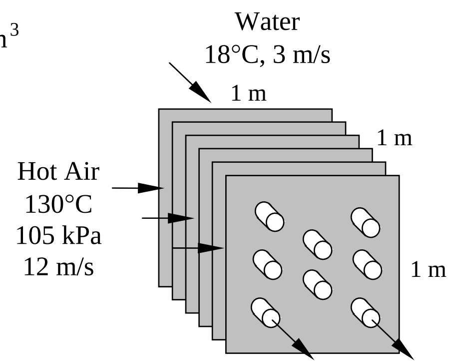
***Properties*** The specific heats of the water and air are given to be 4.18 and 1.01 kJ/kg.°C, respectively.

***Analysis*** The mass flow rates of the hot and the cold fluids are

$$\dot{m}_c = \rho V A_c = (1000\ \text{kg/m}^3)(3\ \text{m/s})[80\pi(0.03\ \text{m})^2/4] = 169.6\ \text{kg/s}$$

$$\rho_{air} = \frac{P}{RT} = \frac{105\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})\times(130+273\ \text{K})} = 0.908\ \text{kg/m}^3$$

$$\dot{m}_h = \rho V A_c = (0.908\ \text{kg/m}^3)(12\ \text{m/s})(1\ \text{m})^2 = 10.90\ \text{kg/s}$$




The heat transfer surface area and the heat capacity rates are

$$A_s = n\pi DL = 80\pi(0.03\ \text{m})(1\ \text{m}) = 7.540\ \text{m}^2$$

$$C_c = \dot{m}_c c_{pc} = (169.6\ \text{kg/s})(4.18\ \text{kJ/kg.°C}) = 708.9\ \text{kW/°C}$$

$$C_h = \dot{m}_h c_{ph} = (10.9\ \text{kg/s})(1.010\ \text{kJ/kg.°C}) = 11.01\ \text{kW/°C}$$

Therefore, $C_{\min} = C_c = 11.01\ \text{kW/°C}$ and $c = \frac{C_{\min}}{C_{\max}} = \frac{11.01}{708.9} = 0.01553$

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (11.01\ \text{kW/°C})(130°\text{C} - 18°\text{C}) = 1233\ \text{kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(130\ \text{W/m}^2.°\text{C})\ (7.540\ \text{m}^2)}{11{,}010\ \text{W/°C}} = 0.08903$$

Noting that this heat exchanger involves mixed cross-flow, the fluid with $C_{\min}$ is mixed, $C_{\max}$ unmixed, effectiveness of this heat exchanger corresponding to $c$ = 0.01553 and NTU =0.08903 is determined using the proper relation in Table 16-4 to be

$$\varepsilon = 1 - \exp\left[-\frac{1}{c}(1 - e^{-cNTU})\right] = 1 - \exp\left[-\frac{1}{0.01553}(1 - e^{-0.01553\times 0.08903})\right] = 0.08513$$

Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.08513)(1233\ \text{kW}) = \mathbf{105.0\ kW}$$

Finally, the outlet temperatures of the cold and the hot fluid streams are determined from

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 18°\text{C} + \frac{105.0\ \text{kW}}{708.9\ \text{kW/°C}} = \mathbf{18.15°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 130°\text{C} - \frac{105.0\ \text{kW}}{11.01\ \text{kW/°C}} = \mathbf{120.5°C}$$

**16-108 CD EES** Ethyl alcohol is heated by water in a shell-and-tube heat exchanger. The heat transfer surface area of the heat exchanger is to be determined using both the LMTD and NTU methods.

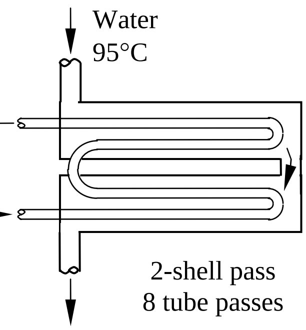
***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the ethyl alcohol and water are given to be 2.67 and 4.19 kJ/kg.°C, respectively.

***Analysis*** (*a*) The temperature differences between the two fluids at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 95°\text{C} - 70°\text{C} = 25°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 25°\text{C} = 35°\text{C}$$




The logarithmic mean temperature difference and the correction factor are

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25-35}{\ln(25/35)} = 29.7°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70-25}{95-25} = 0.64 \\ R &= \frac{T_2 - T_1}{t_1 - t_1} = \frac{95-60}{70-25} = 0.78 \end{aligned}\right\} F = 0.93$$

The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (2.1\,\text{kg/s})(2.67\text{ kJ/kg.°C})(70°\text{C} - 25°\text{C}) = 252.3\,\text{kW}$$

The surface area of heat transfer is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm}} = \frac{252.3\,\text{kW}}{0.8\,\text{kW/m}^2\text{.°C})(0.93)(29.7°\text{C})} = \mathbf{11.4\ m^2}$$

(*b*) The rate of heat transfer is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (2.1\,\text{kg/s})(2.67\text{ kJ/kg.°C})(70°\text{C} - 25°\text{C}) = 252.3\,\text{kW}$$

The mass flow rate of the hot fluid is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) \rightarrow \dot{m}_h = \frac{\dot{Q}}{c_{ph}(T_{h,in} - T_{h,out})} = \frac{252.3\,\text{kW}}{(4.19\text{ kJ/kg.°C})(95°\text{C} - 60°\text{C})} = 1.72\text{ kg/s}$$

The heat capacity rates of the hot and the cold fluids are

$$C_h = \dot{m}_h c_{ph} = (1.72\text{ kg/s})(4.19\text{ kJ/kg.°C}) = 7.21\,\text{kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (2.1\,\text{kg/s})(2.67\text{ kJ/kg.°C}) = 5.61\,\text{kW/°C}$$

Therefore, $C_{\min} = C_c = 5.61\,\text{W/°C}$ and $c = \frac{C_{\min}}{C_{\max}} = \frac{5.61}{7.21} = 0.78$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (5.61\,\text{W/°C})(95°\text{C} - 25°\text{C}) = 392.7\text{ kW}$$

The effectiveness of this heat exchanger is $\varepsilon = \frac{Q}{Q_{\max}} = \frac{252.3}{392.7} = 0.64$

The NTU of this heat exchanger corresponding to this emissivity and $c$ = 0.78 is determined from Fig. 16-26d to be NTU = 1.7. Then the surface area of heat exchanger is determined to be

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{NTU\ C_{\min}}{U} = \frac{(1.7)(5.61\,\text{kW/°C})}{0.8\,\text{kW/m}^2\text{.°C}} = \mathbf{11.9\ m^2}$$

The small difference between the two results is due to the reading error of the chart.

**16-109** Steam is condensed by cooling water in a shell-and-tube heat exchanger. The rate of heat transfer and the rate of condensation of steam are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

***Properties*** The specific heat of the water is given to be 4.18 kJ/kg.°C. The heat of condensation of steam at 30°C is given to be 2430 kJ/kg.

***Analysis*** (*a*) The heat capacity rate of a fluid condensing in a heat exchanger is infinity. Therefore,

$$C_{\min} = C_c = \dot{m}_c c_{pc} = (0.5\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 2.09\text{ kW/°C}$$

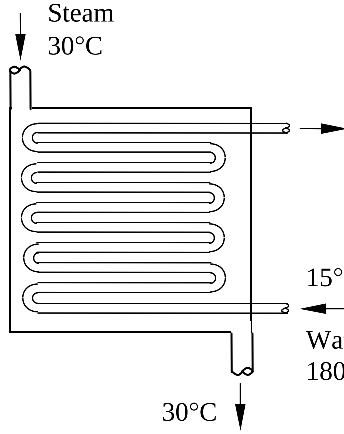
and $c = 0$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (2.09\text{ kW/°C})(30\text{°C} - 15\text{°C}) = 31.35\text{ kW}$$

and

$$A_s = 8n\pi DL = 8\times 50\pi(0.015\text{ m})(2\text{ m}) = 37.7\text{ m}^2$$




The NTU of this heat exchanger

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(3\text{ kW/m}^2.\text{°C})\,(37.7\text{ m}^2)}{2.09\text{ kW/°C}} = 54.11$$

Then the effectiveness of this heat exchanger corresponding to $c = 0$ and NTU = 54.11 is determined using the proper relation in Table 16-5

$$\varepsilon = 1 - \exp(-\text{NTU}) = 1 - \exp(-54.11) = 1$$

Then the actual heat transfer rate becomes

$$\dot{Q} = \varepsilon\dot{Q}_{\max} = (1)(31.35\text{ kW}) = \mathbf{31.35\text{ kW}}$$

(*b*) Finally, the rate of condensation of the steam is determined from

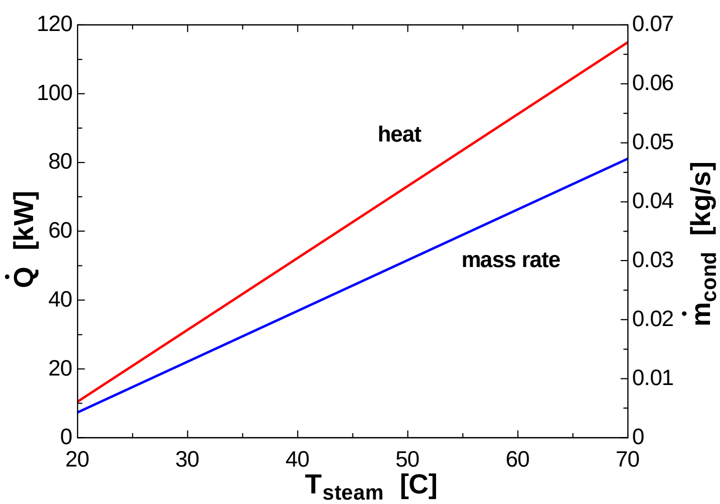
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{31.35\text{ kJ/s}}{2430\text{ kJ/kg}} = \mathbf{0.0129\text{ kg/s}}$$

**16-110 EES** Prob. 16-109 is reconsidered. The effects of the condensing steam temperature and the tube diameter on the rate of heat transfer and the rate of condensation of steam are to be investigated.

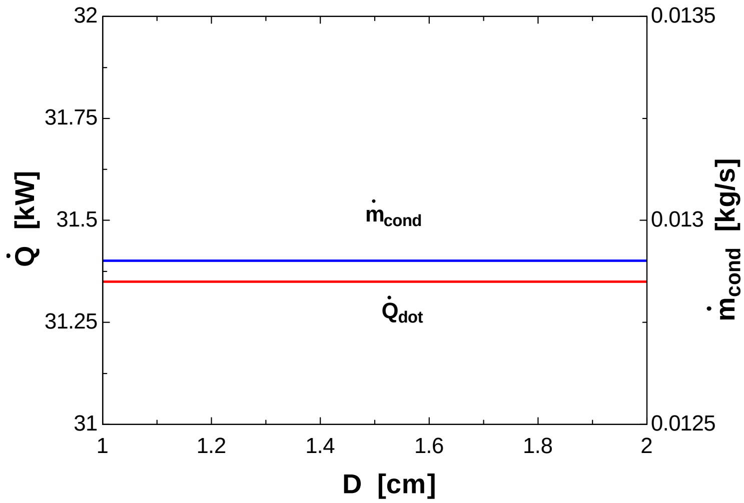
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
N_pass=8
N_tube=50
T_steam=30 [C]
h_fg_steam=2430 [kJ/kg]
T_w_in=15 [C]
m_dot_w=1800[kg/h]*Convert(kg/h, kg/s)
c_p_w=4.18 [kJ/kg-C]
D=1.5 [cm]
L=2 [m]
U=3 [kW/m^2-C]

"ANALYSIS"
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below,
we use NTU method. Both methods give the same results."
C_min=m_dot_w*c_p_w
c=0 "since the heat capacity rate of a fluid condensing is infinity"
Q_dot_max=C_min*(T_steam-T_w_in)
A=N_pass*N_tube*pi*D*L*Convert(cm, m)
NTU=(U*A)/C_min
epsilon=1-exp(-NTU) "from Table 16-4 of the text with c=0"
Q_dot=epsilon*Q_dot_max
Q_dot=m_dot_cond*h_fg_steam
```

| $T_{steam}$ [C] | Q [kW] | $m_{cond}$ [kg/s] |
|---|---|---|
| 20 | 10.45 | 0.0043 |
| 22.5 | 15.68 | 0.006451 |
| 25 | 20.9 | 0.008601 |
| 27.5 | 26.12 | 0.01075 |
| 30 | 31.35 | 0.0129 |
| 32.5 | 36.58 | 0.01505 |
| 35 | 41.8 | 0.0172 |
| 37.5 | 47.03 | 0.01935 |
| 40 | 52.25 | 0.0215 |
| 42.5 | 57.47 | 0.02365 |
| 45 | 62.7 | 0.0258 |
| 47.5 | 67.93 | 0.02795 |
| 50 | 73.15 | 0.0301 |
| 52.5 | 78.38 | 0.03225 |
| 55 | 83.6 | 0.0344 |
| 57.5 | 88.82 | 0.03655 |
| 60 | 94.05 | 0.0387 |
| 62.5 | 99.27 | 0.04085 |
| 65 | 104.5 | 0.043 |
| 67.5 | 109.7 | 0.04515 |
| 70 | 114.9 | 0.0473 |



| D [cm] | Q [kW] | $m_{cond}$ [kg/s] |
|---|---|---|
| 1 | 31.35 | 0.0129 |
| 1.05 | 31.35 | 0.0129 |
| 1.1 | 31.35 | 0.0129 |
| 1.15 | 31.35 | 0.0129 |
| 1.2 | 31.35 | 0.0129 |
| 1.25 | 31.35 | 0.0129 |
| 1.3 | 31.35 | 0.0129 |
| 1.35 | 31.35 | 0.0129 |
| 1.4 | 31.35 | 0.0129 |
| 1.45 | 31.35 | 0.0129 |
| 1.5 | 31.35 | 0.0129 |
| 1.55 | 31.35 | 0.0129 |
| 1.6 | 31.35 | 0.0129 |
| 1.65 | 31.35 | 0.0129 |
| 1.7 | 31.35 | 0.0129 |
| 1.75 | 31.35 | 0.0129 |
| 1.8 | 31.35 | 0.0129 |
| 1.85 | 31.35 | 0.0129 |
| 1.9 | 31.35 | 0.0129 |
| 1.95 | 31.35 | 0.0129 |
| 2 | 31.35 | 0.0129 |



**16-111** Cold water is heated by hot oil in a shell-and-tube heat exchanger. The rate of heat transfer is to be determined using both the LMTD and NTU methods.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg.°C, respectively.

***Analysis*** (*a*) The LMTD method in this case involves iterations, which involves the following steps:

1) Choose $T_{h,out}$

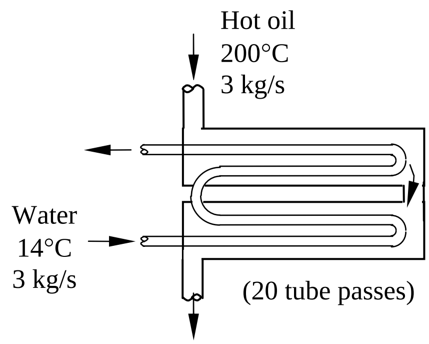
2) Calculate $\dot{Q}$ from $\dot{Q} = \dot{m}_h c_p (T_{h,out} - T_{h,in})$

3) Calculate $T_{h,out}$ from $\dot{Q} = \dot{m}_h c_p (T_{h,out} - T_{h,in})$

4) Calculate $\Delta T_{lm,CF}$

5) Calculate $\dot{Q}$ from $\dot{Q} = UA_s F \Delta T_{lm,CF}$

6) Compare to the $\dot{Q}$ calculated at step 2, and repeat until reaching the same result




Result: **651 kW**

(*b*) The heat capacity rates of the hot and the cold fluids are

$$C_h = \dot{m}_h c_{ph} = (3\text{ kg/s})(2.2\text{ kJ/kg.°C}) = 6.6\text{ kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (3\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 12.54\text{ kW/°C}$$

Therefore, $C_{\min} = C_h = 6.6\text{ kW/°C}$ and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{6.6}{12.54} = 0.53$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (6.6\text{ kW/°C})(200\text{°C} - 14\text{°C}) = 1228\text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.3\text{ kW/m}^2.\text{°C})(20\text{ m}^2)}{6.6\text{ kW/°C}} = 0.91$$

Then the effectiveness of this heat exchanger corresponding to $c$ = 0.53 and NTU = 0.91 is determined from Fig. 16-26d to be

$$\varepsilon = 0.53$$

The actual rate of heat transfer then becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.53)(1228\text{ kW}) = \mathbf{651\text{ kW}}$$

### Selection of the Heat Exchangers

**16-112C** 1) Calculate heat transfer rate, 2) select a suitable type of heat exchanger, 3) select a suitable type of cooling fluid, and its temperature range, 4) calculate or select U, and 5) calculate the size (surface area) of heat exchanger

**16-113C** The first thing we need to do is determine the life expectancy of the system. Then we need to evaluate how much the larger will save in pumping cost, and compare it to the initial cost difference of the two units. If the larger system saves more than the cost difference in its lifetime, it should be preferred.

**16-114C** In the case of automotive and aerospace industry, where weight and size considerations are important, and in situations where the space availability is limited, we choose the smaller heat exchanger.

**16-115** Oil is to be cooled by water in a heat exchanger. The heat transfer rating of the heat exchanger is to be determined and a suitable type is to be proposed.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

***Properties*** The specific heat of the oil is given to be 2.2 kJ/kg.°C.

***Analysis*** The heat transfer rate of this heat exchanger is

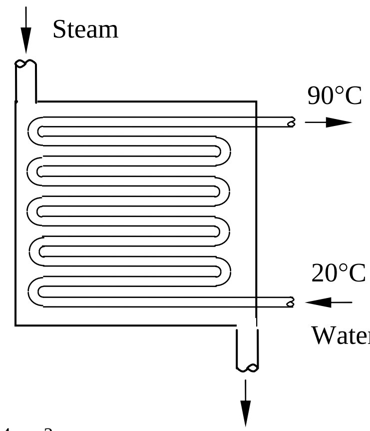
$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (13\text{ kg/s})(2.2\text{ kJ/kg.°C})(120\text{°C} - 50\text{°C}) = \mathbf{2002\ kW}$$

We propose a compact heat exchanger (like the car radiator) if air cooling is to be used, or a tube-and-shell or plate heat exchanger if water cooling is to be used.

**3-116** Water is to be heated by steam in a shell-and-tube process heater. The number of tube passes need to be used is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

***Properties*** The specific heat of the water is given to be 4.19 kJ/kg.°C.




***Analysis*** The mass flow rate of the water is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\dot{m} = \frac{\dot{Q}}{c_{pc}(T_{c,out} - T_{c,in})}$$

$$= \frac{600 \text{ kW}}{(4.19 \text{ kJ/kg.°C})(90°\text{C} - 20°\text{C})}$$

$$= 2.046 \text{ kg/s}$$

The total cross-section area of the tubes corresponding to this mass flow rate is

$$\dot{m} = \rho V A_c \rightarrow A_c = \frac{\dot{m}}{\rho V} = \frac{2.046 \text{ kg/s}}{(1000 \text{ kg/m}^3)(3 \text{ m/s})} = 6.82 \times 10^{-4} \text{ m}^2$$

Then the number of tubes that need to be used becomes

$$A_s = n\frac{\pi D^2}{4} \longrightarrow n = \frac{4A_s}{\pi D^2} = \frac{4(6.82 \times 10^{-4} \text{ m}^2)}{\pi (0.01 \text{ m})^2} = 8.68 \cong \mathbf{9}$$

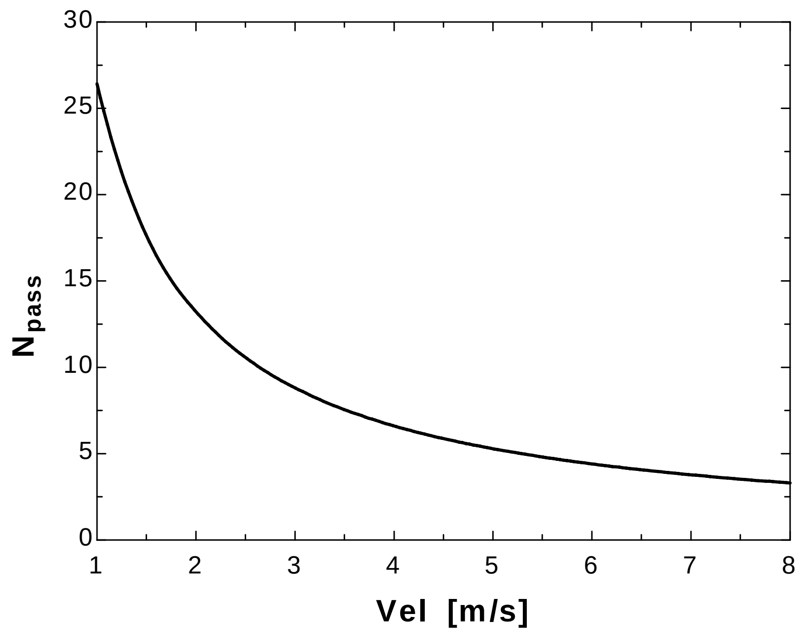
Therefore, we need to use at least 9 tubes entering the heat exchanger.

**16-117 EES** Prob. 16-116 is reconsidered. The number of tube passes as a function of water velocity is to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"GIVEN"
c_p_w=4.19 [kJ/kg-C]
T_w_in=20 [C]
T_w_out=90 [C]
Q_dot=600 [kW]
D=0.01 [m]
Vel=3 [m/s]

"PROPERTIES"
rho=density(water, T=T_ave, P=100)
T_ave=1/2*(T_w_in+T_w_out)

"ANALYSIS"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
m_dot_w=rho*A_c*Vel
A_c=N_pass*pi*D^2/4
```

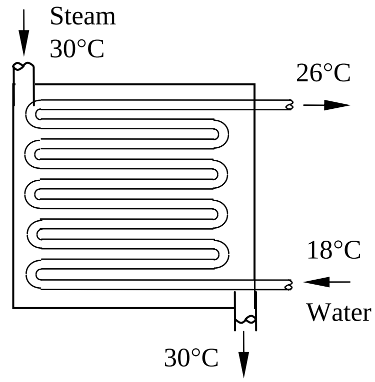
| Vel [m/s] | $N_{pass}$ |
|---|---|
| 1 | 26.42 |
| 1.5 | 17.62 |
| 2 | 13.21 |
| 2.5 | 10.57 |
| 3 | 8.808 |
| 3.5 | 7.55 |
| 4 | 6.606 |
| 4.5 | 5.872 |
| 5 | 5.285 |
| 5.5 | 4.804 |
| 6 | 4.404 |
| 6.5 | 4.065 |
| 7 | 3.775 |
| 7.5 | 3.523 |
| 8 | 3.303 |



**16-118** Cooling water is used to condense the steam in a power plant. The total length of the tubes required in the condenser is to be determined and a suitable HX type is to be proposed.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heat of the water is given to be 4.18 kJ/kg.°C. The heat of condensation of steam at 30°C is given to be 2431 kJ/kg.




***Analysis*** The temperature differences between the steam and the water at the two ends of condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 30°\text{C} - 26°\text{C} = 4°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 30°\text{C} - 18°\text{C} = 12°\text{C}$$

and the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{4-12}{\ln(4/12)} = 7.28°\text{C}$$

The heat transfer surface area is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{500\times10^6 \text{ W}}{(3500 \text{ W/m}^2.°\text{C})(7.28°\text{C})} = 1.96\times10^4 \text{ m}^2$$

The total length of the tubes required in this condenser then becomes

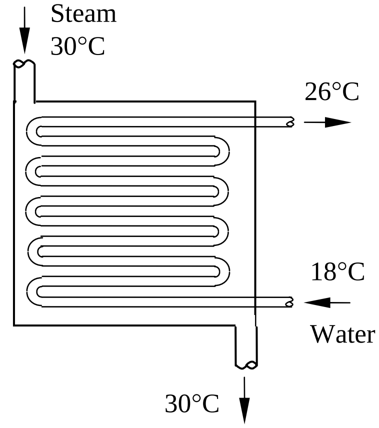
$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{1.96\times10^4 \text{ m}^2}{\pi(0.02 \text{ m})} = 3.123\times10^5 \text{ m} = \textbf{312.3 km}$$

A multi-pass shell-and-tube heat exchanger is suitable in this case.

**16-119** Cold water is heated by hot water in a heat exchanger. The net rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.




***Analysis*** The temperature differences between the steam and the water at the two ends of condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 30°\text{C} - 26°\text{C} = 4°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 30°\text{C} - 18°\text{C} = 12°\text{C}$$

and the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{4-12}{\ln(4/12)} = 7.28°\text{C}$$

The heat transfer surface area is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{50\times10^6 \text{ W}}{(3500 \text{ W/m}^2.°\text{C})(7.28°\text{C})} = 1962 \text{ m}^2$$

The total length of the tubes required in this condenser then becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{1962 \text{ m}^2}{\pi(0.02 \text{ m})} = 31{,}231 \text{ m} = \mathbf{31.23 \text{ km}}$$

A multi-pass shell-and-tube heat exchanger is suitable in this case.

## Review Problems

**16-120** The inlet conditions of hot and cold fluid streams in a heat exchanger are given. The outlet temperatures of both streams are to be determined using LMTD and the effectiveness-NTU methods.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of hot and cold fluid streams are given to be 2.0 and 4.2 kJ/kg.°C, respectively.

***Analysis*** (a) The rate of heat transfer can be expressed as

$$\dot{Q} = \dot{m}c_p(T_{h,in} - T_{h,out}) = (2700/3600\text{ kg/s})(2.0\text{ kJ/kg.°C})(120 - T_{h,out}) = 1.5(120 - T_{h,out}) \quad (1)$$

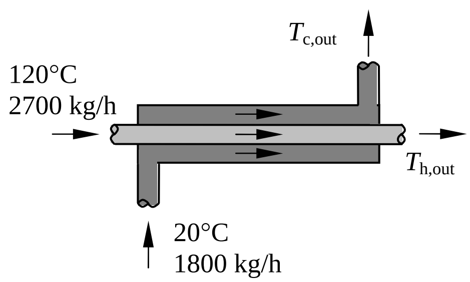
$$\dot{Q} = \dot{m}c_p(T_{c,out} - T_{c,in}) = (1800/3600\text{ kg/s})(4.2\text{ kJ/kg.°C})(T_{c,out} - 20) = 2.1(T_{c,out} - 20) \quad (2)$$

The heat transfer can also be expressed using the logarithmic mean temperature difference as

$$\Delta T_1 = T_{h,in} - T_{c,in} = 120°\text{C} - 20°\text{C} = 100°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)}$$




$$\dot{Q} = UA\Delta T_{lm} = \frac{\dot{Q}_{hc,m}}{A\Delta T_{lm}}$$

$$= (2.0\text{ kW/m}^2\cdot°\text{C})(0.50\text{ m}^2)\frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)} = \frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)} \quad (3)$$

Now we have three expressions for heat transfer with three unknowns: $\dot{Q}$, $T_{h,out}$, $T_{c,out}$. Solving them using an equation solver such as EES, we obtain

$$\dot{Q} = 59.6\text{ kW}$$

$$T_{h,out} = \mathbf{80.3°C}$$

$$T_{c,out} = \mathbf{48.4°C}$$

(b) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (2700/3600\text{ kg/s})(2.0\text{ kJ/kg.°C}) = 1.5\text{ kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (1800/3600\text{ kg/s})(4.2\text{ kJ/kg.°C}) = 2.1\text{ kW/°C}$$

Therefore

$$C_{\min} = C_h = 1.5\text{ kW/°C}$$

which is the smaller of the two heat capacity rates. The heat capacity ratio and the NTU are

$$c = \frac{C_{\min}}{C_{\max}} = \frac{1.5}{2.1} = 0.714$$

$$NTU = \frac{UA}{C_{\min}} = \frac{(2.0\ \text{kW/m}^2 \cdot \text{C})(0.50\ \text{m}^2)}{1.5\ \text{kW/°C}} = 0.667$$

The effectiveness of this parallel-flow heat exchanger is

$$\varepsilon = \frac{1 - \exp\left[-NTU(1+c)\right]}{1+c} = \frac{1 - \exp\left[-(0.667)(1+0.714)\right]}{1+0.714} = 0.397$$

The maximum heat transfer rate is

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (1.5\ \text{kW/°C})(120\text{°C} - 20\text{°C}) = 150\ \text{kW}$$

The actual heat transfer rate is

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.397)(150) = 59.6\ \text{kW}$$

Then the outlet temperatures are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20\text{°C} + \frac{59.6\ \text{kW}}{2.1\ \text{kW/°C}} = \mathbf{48.4°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 120\text{°C} - \frac{59.6\ \text{kW}}{1.5\ \text{kW/°C}} = \mathbf{80.3°C}$$

***Discussion*** The results obtained by two methods are same as expected. However, the effectiveness-NTU method is easier for this type of problems.

**16-121** Water is used to cool a process stream in a shell and tube heat exchanger. The tube length is to be determined for one tube pass and four tube pass cases.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The properties of process stream and water are given in problem statement.

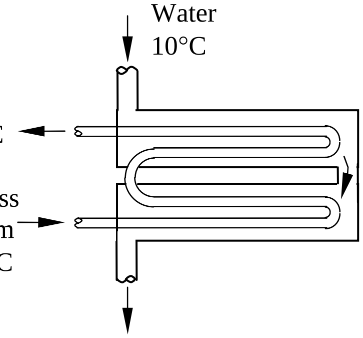
***Analysis*** (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (47 \text{ kg/s})(3.5 \text{ kJ/kg}\cdot°\text{C})(160 - 100)°\text{C} = 9870 \text{ kW}$$

The outlet temperature of water is determined from

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in})$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c C_c} = 10°\text{C} + \frac{9870 \text{ kW}}{(66 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot°\text{C})} = 45.8°\text{C}$$




The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 160°\text{C} - 45.8°\text{C} = 114.2°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 100°\text{C} - 10°\text{C} = 90°\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{114.2 - 90}{\ln\left(\frac{114.2}{90}\right)} = 101.6°\text{C}$$

The Reynolds number is

$$V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{N_{tube}\rho\pi D^2/4} = \frac{(47 \text{ kg/s})}{(100)(950 \text{ kg/m}^3)\pi(0.025 \text{ m})^2/4} = 1.008 \text{ m/s}$$

$$\text{Re} = \frac{VD\rho}{\mu} = \frac{(1.008 \text{ m/s})(0.025 \text{ m})(950 \text{ kg/m}^3)}{0.002 \text{ kg/m}\cdot\text{s}} = 11{,}968$$

which is greater than 10,000. Therefore, we have turbulent flow. We assume fully developed flow and evaluate the Nusselt number from

$$\text{Pr} = \frac{\mu c_p}{k} = \frac{(0.002 \text{ kg/m}\cdot\text{s})(3500 \text{ J/kg}\cdot°\text{C})}{0.50 \text{ W/m}\cdot°\text{C}} = 14$$

$$Nu = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(11{,}968)^{0.8}(14)^{0.3} = 92.9$$

Heat transfer coefficient on the inner surface of the tubes is

$$h_i = \frac{k}{D} Nu = \frac{0.50 \text{ W/m.°C}}{0.025 \text{ m}}(92.9) = 1858 \text{ W/m}^2\text{.°C}$$

Disregarding the thermal resistance of the tube wall the overall heat transfer coefficient is determined from

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{1858} + \frac{1}{4000}} = 1269 \text{ W/m}^2\cdot°\text{C}$$

The correction factor for one shell pass and one tube pass heat exchanger is $F = 1$. The tube length is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$

$$9870 \text{ kW} = (1.269 \text{ kW/m}^2 \cdot \text{C})[100\pi(0.025 \text{ m})L](1)(101.6°\text{C})$$

$$L = \mathbf{9.75 \ m}$$

(b) For 1 shell pass and 4 tube passes, there are 100/4=25 tubes per pass and this will increase the velocity fourfold. We repeat the calculations for this case as follows:

$$V = 4 \times 1.008 = 4.032 \text{ m/s}$$

$$\text{Re} = 4 \times 11{,}968 = 47{,}872$$

$$Nu = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(47{,}872)^{0.8}(14)^{0.3} = 281.6$$

$$h_i = \frac{k}{D} Nu = \frac{0.50 \text{ W/m.°C}}{0.025 \text{ m}}(281.6) = 5632 \text{ W/m}^2\text{.°C}$$

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{5632} + \frac{1}{4000}} = 2339 \text{ W/m}^2 \cdot °\text{C}$$

The correction factor is determined from Fig. 16-18:

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{100 - 160}{10 - 160} = 0.4 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{10 - 45.8}{100 - 160} = 0.60 \end{aligned}\right\} F = 0.96$$

The tube length is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$

$$9870 \text{ kW} = (2.339 \text{ kW/m}^2 \cdot \text{C})[100\pi(0.025 \text{ m})L](0.96)(101.6°\text{C})$$

$$L = \mathbf{5.51 \ m}$$

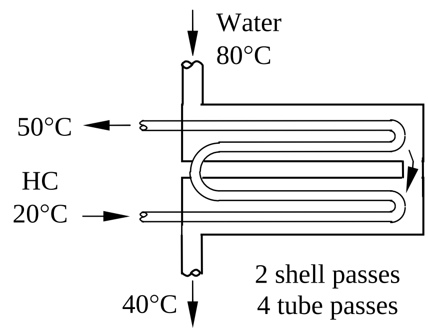
**16-122** A hydrocarbon stream is heated by a water stream in a 2-shell passes and 4-tube passes heat exchanger. The rate of heat transfer and the mass flow rates of both fluid streams and the fouling factor after usage are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heat of HC is given to be 2 kJ/kg.°C. The specific heat of water is taken to be 4.18 kJ/kg.°C.

***Analysis*** (a) The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80°\text{C} - 50°\text{C} = 30°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°\text{C} - 20°\text{C} = 20°\text{C}$$
$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{30-20}{\ln(30/20)} = 24.66°\text{C}$$
$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{50-20}{80-20} = 0.5 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{80-40}{50-20} = 1.33 \end{aligned}\right\} F = 0.90 \ \text{(Fig. 16-18)}$$




The overall heat transfer coefficient of the heat exchanger is

$$U = \frac{1}{\dfrac{1}{h_i} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{1600} + \dfrac{1}{2500}} = 975.6 \text{ W/m}^2 \cdot °\text{C}$$

The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s F \Delta T_{lm,CF} = (975.6 \text{ W/m}^2.°\text{C})[160\pi(0.02 \text{ m})(1.5 \text{ m})](0.90)(24.66°\text{C}) = 3.265 \times 10^5 \text{ W} = \mathbf{326.5 \text{ kW}}$$

The mass flow rates of fluid streams are

$$\dot{m}_c = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{326.5 \text{ kW}}{(2.0 \text{ kJ/kg.°C})(50°\text{C} - 20°\text{C})} = \mathbf{5.44 \text{ kg/s}}$$
$$\dot{m}_h = \frac{\dot{Q}}{c_p(T_{in} - T_{out})} = \frac{326.5 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(80°\text{C} - 40°\text{C})} = \mathbf{1.95 \text{ kg/s}}$$

(b) The rate of heat transfer in this case is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_\text{c} = (5.44 \text{ kg/s})(2.0 \text{ kJ/kg.°C})(45°\text{C} - 20°\text{C}) = 272 \text{ kW}$$

This corresponds to a 17% decrease in heat transfer. The outlet temperature of the hot fluid is

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_\text{h}$$
$$272 \text{ kW} = (1.95 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(80°\text{C} - T_{h,out})$$
$$T_{h,out} = 46.6°\text{C}$$

The logarithmic temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80°\text{C} - 45°\text{C} = 35°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 46.6°\text{C} - 20°\text{C} = 26.6°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{35 - 26.6}{\ln(35/26.6)} = 30.61°\text{C}$$

$$\left. \begin{array}{l} P = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{45 - 20}{80 - 20} = 0.42 \\ R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{80 - 46.6}{45 - 20} = 1.34 \end{array} \right\} F = 0.97 \text{ (Fig. 16-18)}$$

The overall heat transfer coefficient is

$$\dot{Q} = UA_s F \Delta T_{lm,CF}$$

$$272{,}000 \text{ W} = U\left[160\pi(0.02 \text{ m})(1.5 \text{ m})\right](0.97)(30.61°\text{C})$$

$$U = 607.5 \text{ W/m}^2.°\text{C}$$

The fouling factor is determined from

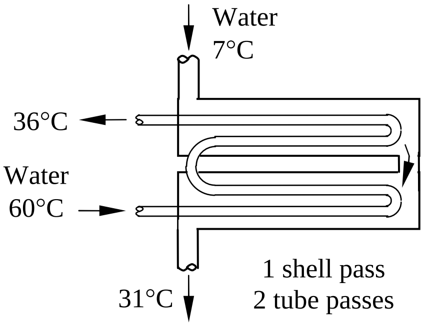
$$R_f = \frac{1}{U_{dirty}} - \frac{1}{U_{clean}} = \frac{1}{607.5} - \frac{1}{975.6} = \mathbf{6.21 \times 10^{-4} \text{ m}^2 \cdot °\text{C/W}}$$

**16-123** Hot water is cooled by cold water in a 1-shell pass and 2-tube passes heat exchanger. The mass flow rates of both fluid streams are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There is no fouling.

***Properties*** The specific heats of both cold and hot water streams are taken to be 4.18 kJ/kg.°C.

***Analysis*** The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are




$$\Delta T_1 = T_{h,in} - T_{c,out} = 60°\text{C} - 31°\text{C} = 29°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 36°\text{C} - 7°\text{C} = 29°\text{C}$$

Since $\Delta T_1 = \Delta T_2$, we have $\Delta T_{lm,CF} = 29°\text{C}$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{31-60}{7-60} = 0.45 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{7-31}{36-60} = 1.0 \end{aligned}\right\} F = 0.88 \text{ (Fig. 16-18)}$$

The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s F\Delta T_{lm,CF} = (950 \text{ W/m}^2.°\text{C})(15 \text{ m}^2)(0.88)(29°\text{C}) = 3.64\times10^5 \text{ W} = 364 \text{ kW}$$

The mass flow rates of fluid streams are

$$\dot{m}_c = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{364 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(60°\text{C} - 36°\text{C})} = \mathbf{3.63 \text{ kg/s}}$$

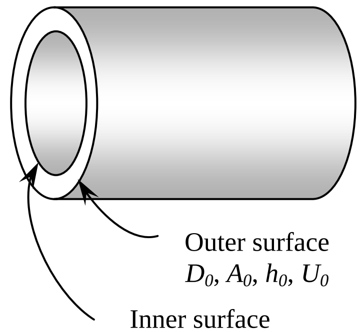
$$\dot{m}_h = \frac{\dot{Q}}{c_p(T_{in} - T_{out})} = \frac{364 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(31°\text{C} - 7°\text{C})} = \mathbf{3.63 \text{ kg/s}}$$

**16-124** Hot oil is cooled by water in a multi-pass shell-and-tube heat exchanger. The overall heat transfer coefficient based on the inner surface is to be determined.

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

***Properties*** The properties of water at 25°C are (Table A-15)

$$k = 0.607 \text{ W/m.°C}$$
$$\nu = \mu / \rho = 0.894\times10^{-6} \text{ m}^2\text{/s}$$
$$\text{Pr} = 6.14$$




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}}D}{\nu} = \frac{(3\text{ m/s})(0.013\text{ m})}{0.894\times10^{-6}\text{ m}^2\text{/s}} = 43{,}624$$

which is greater than 10,000. Therefore, we assume fully developed turbulent flow, and determine Nusselt number from

$$Nu = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4}\; 0.023(43{,}624)^{0.8}(6.14)^{0.4} = 245$$

and

$$h_i = \frac{k}{D}Nu = \frac{0.607\text{ W/m.°C}}{0.013\text{ m}}(245) = 11{,}440\text{ W/m}^2\text{.°C}$$

The inner and the outer surface areas of the tube are

$$A_i = \pi D_i L = \pi(0.013\text{ m})(1\text{ m}) = 0.04084\text{ m}^2$$
$$A_o = \pi D_o L = \pi(0.015\text{ m})(1\text{ m}) = 0.04712\text{ m}^2$$

The total thermal resistance of this heat exchanger per unit length is

$$\begin{aligned} R &= \frac{1}{h_i A_i} + \frac{\ln(D_o / D_i)}{2\pi kL} + \frac{1}{h_o A_o} \\ &= \frac{1}{(11{,}440\text{ W/m}^2\text{.°C})(0.04084\text{ m}^2)} + \frac{\ln(1.5/1.3)}{2\pi(110\text{ W/m.°C})(1\text{ m})} + \frac{1}{(35\text{ W/m}^2\text{.°C})(0.04712\text{ m}^2)} \\ &= 0.609\text{°C/W} \end{aligned}$$

Then the overall heat transfer coefficient of this heat exchanger based on the inner surface becomes

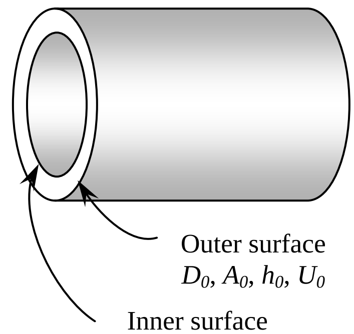
$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.609\text{°C/W})(0.04084\text{ m}^2)} = \mathbf{40.2\ W/m^2.°C}$$

**16-125** Hot oil is cooled by water in a multi-pass shell-and-tube heat exchanger. The overall heat transfer coefficient based on the inner surface is to be determined.

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

***Properties*** The properties of water at 25°C are (Table A-15)

$$k = 0.607 \text{ W/m.°C}$$
$$\nu = \mu / \rho = 0.894\times10^{-6} \text{ m}^2\text{/s}$$
$$\text{Pr} = 6.14$$




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(3\text{ m/s})(0.013\text{ m})}{0.894\times10^{-6}\text{ m}^2\text{/s}} = 43{,}624$$

which is greater than 10,000. Therefore, we assume fully developed turbulent flow, and determine Nusselt number from

$$Nu = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4}\; 0.023(43{,}624)^{0.8}(6.14)^{0.4} = 245$$

and

$$h_i = \frac{k}{D}Nu = \frac{0.607\text{ W/m.°C}}{0.013\text{ m}}(245) = 11{,}440\text{ W/m}^2\text{.°C}$$

The inner and the outer surface areas of the tube are

$$A_i = \pi D_i L = \pi(0.013\text{ m})(1\text{ m}) = 0.04084\text{ m}^2$$
$$A_o = \pi D_o L = \pi(0.015\text{ m})(1\text{ m}) = 0.04712\text{ m}^2$$

The total thermal resistance of this heat exchanger per unit length of it with a fouling factor is

$$R = \frac{1}{h_i A_i} + \frac{\ln(D_o / D_i)}{2\pi kL} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o}$$
$$= \frac{1}{(11{,}440\text{ W/m}^2\text{.°C})(0.04084\text{ m}^2)} + \frac{\ln(1.5/1.3)}{2\pi(110\text{ W/m.°C})(1\text{ m})}$$
$$+ \frac{0.0004\text{ m}^2\text{.°C/W}}{0.04712\text{ m}^2} + \frac{1}{(35\text{ W/m}^2\text{.°C})(0.04712\text{ m}^2)}$$
$$= 0.617\text{°C/W}$$

Then the overall heat transfer coefficient of this heat exchanger based on the inner surface becomes

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.617\text{°C/W})(0.04084\text{ m}^2)} = \mathbf{39.7\ W/m^2.°C}$$

**16-126** Water is heated by hot oil in a multi-pass shell-and-tube heat exchanger. The rate of heat transfer and the heat transfer surface area on the outer side of the tube are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg.°C, respectively.

***Analysis*** (*a*)The rate of heat transfer in this heat exchanger is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = (3\text{ kg/s})(2.2\text{ kJ/kg.°C})(130\text{°C} - 60\text{°C}) = \mathbf{462\ kW}$$

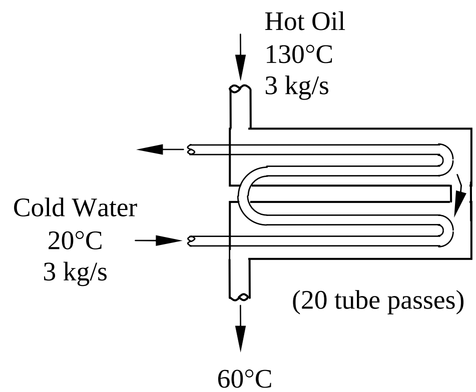
(*b*) The outlet temperature of the cold water is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 20\text{°C} + \frac{462\text{ kW}}{(3\text{ kg/s})(4.18\text{ kJ/kg.°C})} = 56.8\text{°C}$$

The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 130\text{°C} - 56.8\text{°C} = 73.2\text{°C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60\text{°C} - 20\text{°C} = 40\text{°C}$$




The logarithmic mean temperature difference is

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{73.2 - 40}{\ln(73.2/40)} = 54.9\text{°C}$$

and

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{56.8 - 20}{130 - 20} = 0.335 \\ R &= \frac{T_2 - T_1}{t_2 - t_1} = \frac{130 - 60}{56.8 - 20} = 1.90 \end{aligned}\right\} F = 0.96$$

The heat transfer surface area on the outer side of the tube is then determined from

$$\dot{Q} = UA_s F\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm}} = \frac{462\text{ kW}}{(0.22\text{ kW/m}^2.\text{°C})(0.96)(54.9\text{°C})} = \mathbf{39.8\ m^2}$$

**16-127E** Water is heated by solar-heated hot air in a double-pipe counter-flow heat exchanger. The required length of the tube is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the water and air are given to be 1.0 and 0.24 Btu/lbm.°F, respectively.

***Analysis*** The rate of heat transfer in this heat exchanger is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = (0.7\text{ lbm/s})(0.24\text{ Btu/lbm.°F})(190\text{°F} - 135\text{°F}) = 9.24\text{ Btu/s}$$

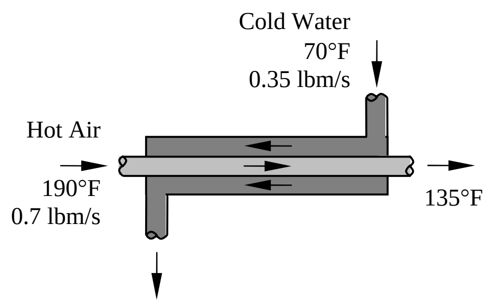
The outlet temperature of the cold water is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 70\text{°F} + \frac{9.24\text{ Btu/s}}{(0.35\text{ lbm/s})(1.0\text{ Btu/lbm.°F})} = 96.4\text{°F}$$

The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 190\text{°F} - 96.4\text{°F} = 93.6\text{°F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 135\text{°F} - 70\text{°F} = 65\text{°F}$$




The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{93.6 - 65}{\ln(93.6/65)} = 78.43\text{°F}$$

The heat transfer surface area on the outer side of the tube is determined from

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{9.24\text{ Btu/s}}{(20/3600\text{ Btu/s.ft}^2\text{.°F})(78.43\text{°F})} = 21.21\text{ ft}^2$$

Then the length of the tube required becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{21.21\text{ ft}^2}{\pi(0.5/12\text{ ft})} = \mathbf{162.0\text{ ft}}$$

**16-128** It is to be shown that when $\Delta T_1 = \Delta T_2$ for a heat exchanger, the $\Delta T_{lm}$ relation reduces to $\Delta T_{lm} = \Delta T_1 = \Delta T_2$.

***Analysis*** When $\Delta T_1 = \Delta T_2$, we obtain

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{0}{0}$$

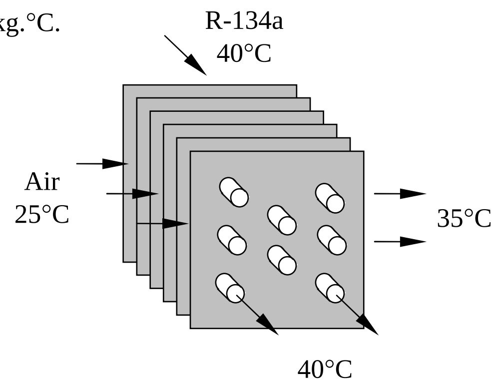
This case can be handled by applying L'Hospital's rule (taking derivatives of nominator and denominator separately with respect to $\Delta T_1$ or $\Delta T_2$ ). That is,

$$\Delta T_{lm} = \frac{d(\Delta T_1 - \Delta T_2)/d\Delta T_1}{d[\ln(\Delta T_1 / \Delta T_2)]/d\Delta T_1} = \frac{1}{1/\Delta T_1} = \Delta T_1 = \Delta T_2$$

**16-129** Refrigerant-134a is condensed by air in the condenser of a room air conditioner. The heat transfer area on the refrigerant side is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heat of air is given to be 1.005 kJ/kg.°C.




***Analysis*** The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 40°\text{C} - 35°\text{C} = 5°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°\text{C} - 25°\text{C} = 15°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{5 - 15}{\ln(5/15)} = 9.1°\text{C}$$

The heat transfer surface area on the outer side of the tube is determined from

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{(15{,}000/3600)\text{ kW}}{(0.150\text{ kW/m}^2.°\text{C})(9.1°\text{C})} = \mathbf{3.05\ m^2}$$

**16-130** Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of air and combustion gases are given to be 1.005 and 1.1 kJ/kg.°C, respectively.

***Analysis*** The rate of heat transfer is simply

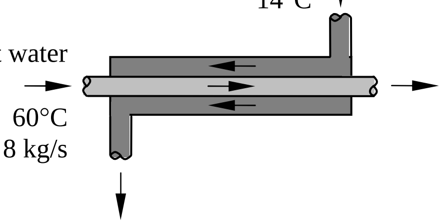
$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{gas.}} = (0.65\text{ kg/s})(1.1\text{ kJ/kg.°C})(180°\text{C} - 95°\text{C}) = \mathbf{60.8\ kW}$$

**16-131** A water-to-water heat exchanger is proposed to preheat the incoming cold water by the drained hot water in a plant to save energy. The heat transfer rating of the heat exchanger and the amount of money this heat exchanger will save are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

***Properties*** The specific heat of the hot water is given to be 4.18 kJ/kg.°C.

***Analysis*** The maximum rate of heat transfer is




$$\begin{aligned}\dot{Q}_{\max} &= \dot{m}_h c_{ph}(T_{h,in} - T_{c,in})\\ &= (8/60\ \text{kg/s})(4.18\ \text{kJ/kg.°C})(60°\text{C} - 14°\text{C})\\ &= 25.6\ \text{kW}\end{aligned}$$

Noting that the heat exchanger will recover 72% of it, the actual heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.72)(25.6\ \text{kJ/s}) = 18.43\ \text{kW}$$

which is the heat transfer rating. The operating hours per year are

The annual operating hours = (8 h/day)(5 days/week)(52 week/year) = 2080 h/year

The energy saved during the entire year will be

Energy saved = (heat transfer rate)(operating time)

= (18.43 kJ/s)(2080 h/year)(3600 s/h)

= $1.38\text{x}10^8$ kJ/year

Then amount of fuel and money saved will be

$$\text{Fuel saved} = \frac{\text{Energy saved}}{\text{Furnace efficiency}} = \frac{1.38\times10^8\ \text{kJ/year}}{0.78}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = 1677\ \text{therms/year}$$

Money saved = (fuel saved)(the price of fuel)

= (1677 therms/year)($1.00/therm) = **$1677/year**

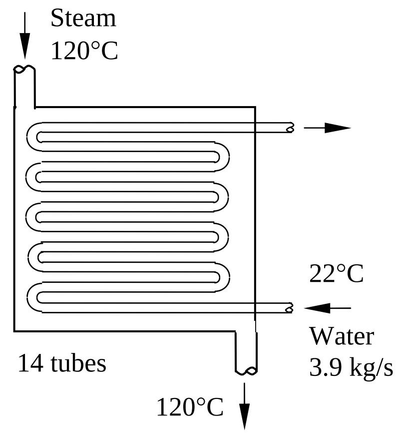
**16-132** A shell-and-tube heat exchanger is used to heat water with geothermal steam condensing. The rate of heat transfer, the rate of condensation of steam, and the overall heat transfer coefficient are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The heat of vaporization of geothermal water at 120°C is given to be $h_{fg}$ = 2203 kJ/kg and specific heat of water is given to be $c_p$ = 4180 J/kg.°C.

***Analysis*** (*a*) The outlet temperature of the water is

$$T_{\rm c,out} = T_{\rm h,out} - 46 = 120°{\rm C} - 46°{\rm C} = 74°{\rm C}$$




Then the rate of heat transfer becomes

$$\dot{Q} = [\dot{m}c_p(T_{\rm out} - T_{\rm in})]_{\rm water}$$
$$= (3.9\ {\rm kg/s})(4.18\ {\rm kJ/kg.°C})(74°{\rm C} - 22°{\rm C})$$
$$= \mathbf{847.7\ kW}$$

(*b*) The rate of condensation of steam is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\substack{\rm geothermal \\ \rm steam}}$$

$$847.7\ {\rm kW} = \dot{m}(2203\ {\rm kJ/kg}) \longrightarrow \dot{m} = \mathbf{0.385\ kg/s}$$

(*c*) The heat transfer area is

$$A_i = n\pi D_i L = 14\pi(0.024\ {\rm m})(3.2\ {\rm m}) = 3.378\ {\rm m}^2$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{\rm h,in} - T_{\rm c,out} = 120°{\rm C} - 74°{\rm C} = 46°{\rm C}$$
$$\Delta T_2 = T_{\rm h,out} - T_{\rm c,in} = 120°{\rm C} - 22°{\rm C} = 98°{\rm C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{46 - 98}{\ln(46/98)} = 68.8°{\rm C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{74 - 22}{120 - 22} = 0.53 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{120 - 120}{74 - 22} = 0 \end{aligned}\right\} F = 1$$

Then the overall heat transfer coefficient is determined to be

$$\dot{Q} = U_i A_i F \Delta T_{\rm lm,CF} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{\rm lm,CF}} = \frac{847{,}700\ {\rm W}}{(3.378\ {\rm m}^2)(1)(68.8°{\rm C})} = \mathbf{3650\ W/m^2.°C}$$

**16-133** Water is heated by geothermal water in a double-pipe counter-flow heat exchanger. The mass flow rate of the geothermal water and the outlet temperatures of both fluids are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the geothermal water and the cold water are given to be 4.25 and 4.18 kJ/kg.°C, respectively.

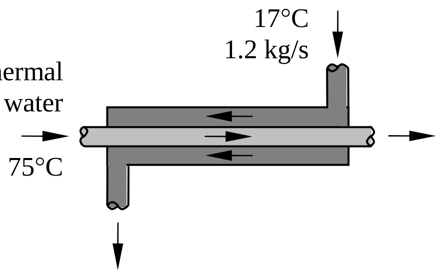
***Analysis*** The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = \dot{m}_h (4.25 \text{ kJ/kg.°C}) = 4.25\dot{m}_h$$

$$C_c = \dot{m}_c c_{pc} = (1.2 \text{ kg/s})(4.18 \text{ kJ/kg.°C}) = 5.016 \text{ kW/°C}$$

$$C_{\min} = C_c = 5.016 \text{ kW/°C}$$

and $$c = \frac{C_{\min}}{C_{\max}} = \frac{5.016}{4.25\dot{m}_h} = \frac{1.1802}{\dot{m}_h}$$




The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.480 \text{ kW/m}^2.\text{°C})(25 \text{ m}^2)}{5.016 \text{ kW/°C}} = 2.392$$

Using the effectiveness relation, we find the capacity ratio

$$\varepsilon = \frac{1-\exp[-\text{NTU}(1-c)]}{1-c\exp[-\text{NTU}(1-c)]} \longrightarrow 0.823 = \frac{1-\exp[-2.392(1-c)]}{1-c\exp[-2.392(1-c)]} \longrightarrow c = 0.494$$

Then the mass flow rate of geothermal water is determined from

$$c = \frac{1.1802}{\dot{m}_h} \longrightarrow 0.494 = \frac{1.1802}{\dot{m}_h} \longrightarrow \dot{m}_h = \mathbf{2.39 \text{ kg/s}}$$

The maximum heat transfer rate is

$$\dot{Q}_{\max} = C_{\min}(T_{\text{h,in}} - T_{\text{c,in}}) = (5.016 \text{ kW/°C})(75\text{°C} - 17\text{°C}) = 290.9 \text{ kW}$$

Then the actual rate of heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.823)(290.9 \text{ kW}) = 239.4 \text{ kW}$$

The outlet temperatures of the geothermal and cold waters are determined to be

$$\dot{Q} = C_c(T_{\text{c,out}} - T_{\text{c,in}}) \longrightarrow 239.4 \text{ kW} = (5.016 \text{ kW/°C})(T_{\text{c,out}} - 17) \longrightarrow T_{\text{c,out}} = \mathbf{64.7°C}$$

$$\dot{Q} = \dot{m}_h c_{ph}(T_{\text{h,in}} - T_{\text{h,out}})$$

$$239.4 \text{ kW} = (2.39 \text{ kg/s})(4.25 \text{ kJ/kg.°C})(75 - T_{\text{h,out}}) \longrightarrow T_{\text{h,out}} = \mathbf{51.4°C}$$

**16-134** Air is to be heated by hot oil in a cross-flow heat exchanger with both fluids unmixed. The effectiveness of the heat exchanger, the mass flow rate of the cold fluid, and the rate of heat transfer are to be determined.

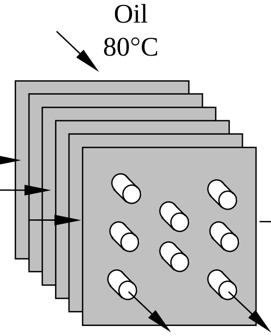
.***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of the air and the oil are given to be 1.006 and 2.15 kJ/kg.°C, respectively.

***Analysis*** (*a*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = 0.5\dot{m}_c(2.15\ \text{kJ/kg.°C}) = 1.075\dot{m}_c$$

$$C_c = \dot{m}_c c_{pc} = \dot{m}_c(1.006\ \text{kJ/kg.°C}) = 1.006\dot{m}_c$$




Therefore, $C_{\min} = C_c = 1.006\dot{m}_c$

and $$c = \frac{C_{\min}}{C_{\max}} = \frac{1.006\dot{m}_c}{1.075\dot{m}_c} = 0.936$$

The effectiveness of the heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{\text{c,out}} - T_{\text{c,in}})}{C_c(T_{\text{h,in}} - T_{\text{c,in}})} = \frac{58-18}{80-18} = \mathbf{0.645}$$

(*b*) The NTU of this heat exchanger is expressed as

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.750\ \text{kW/°C})}{1.006\dot{m}_c} = \frac{0.7455}{\dot{m}_c}$$

The NTU of this heat exchanger can also be determined from

$$NTU = -\frac{\ln[c\ln(1-\varepsilon)+1]}{c} = -\frac{\ln[0.936\times\ln(1-0.645)+1]}{0.936} = 3.724$$

Then the mass flow rate of the air is determined to be

$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow 3.724 = \frac{(0.750\ \text{kW/°C})}{1.006\dot{m}_c} \longrightarrow \dot{m}_c = \mathbf{0.20\ kg/s}$$

(*c*) The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc}(T_{\text{c,out}} - T_{\text{c,in}}) = (0.20\ \text{kg/s})(1.006\ \text{kJ/kg.°C})(58-18)\text{°C} = \mathbf{8.05\ kW}$$

**16-135** A water-to-water counter-flow heat exchanger is considered. The outlet temperature of the cold water, the effectiveness of the heat exchanger, the mass flow rate of the cold water, and the heat transfer rate are to be determined.

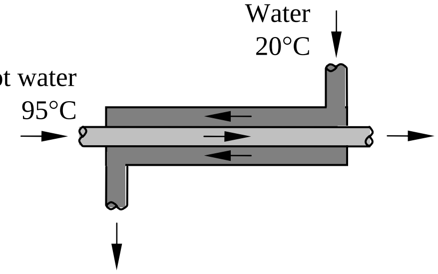
.***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

***Properties*** The specific heats of both the cold and the hot water are given to be 4.18 kJ/kg.°C.

***Analysis*** (*a*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = 1.5\dot{m}_c(4.18\text{ kJ/kg.°C}) = 6.27\dot{m}_c$$

$$C_c = \dot{m}_c c_{pc} = \dot{m}_c(4.18\text{ kJ/kg.°C}) = 4.18\dot{m}_c$$




Therefore, $C_{\min} = C_c = 4.18\dot{m}_c$

and $C = \dfrac{C_{\min}}{C_{\max}} = \dfrac{4.18\dot{m}_c}{6.27\dot{m}_c} = 0.667$

The rate of heat transfer can be expressed as

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) = (4.18\dot{m}_c)(T_{c,out} - 20)$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = (6.27\dot{m}_c)\left[95 - (T_{c,out} + 15)\right] = (6.27\dot{m}_c)(80 - T_{c,out})$$

Setting the above two equations equal to each other we obtain the outlet temperature of the cold water

$$\dot{Q} = 4.18\dot{m}_c(T_{c,out} - 20) = 6.27\dot{m}_c(80 - T_{c,out}) \longrightarrow T_{c,out} = \mathbf{56°C}$$

(*b*) The effectiveness of the heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,out} - T_{c,in})}{C_c(T_{h,in} - T_{c,in})} = \frac{4.18\dot{m}_c(56-20)}{4.18\dot{m}_c(95-20)} = \mathbf{0.48}$$

(*c*) The NTU of this heat exchanger is determined from

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) = \frac{1}{0.667-1}\ln\left(\frac{0.48-1}{0.48\times 0.667 - 1}\right) = 0.805$$

Then, from the definition of NTU, we obtain the mass flow rate of the cold fluid:

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow 0.805 = \frac{1.400\text{ kW/°C}}{4.18\dot{m}_c} \longrightarrow \dot{m}_c = \mathbf{0.416\ kg/s}$$

(*d*) The rate of heat transfer is determined from

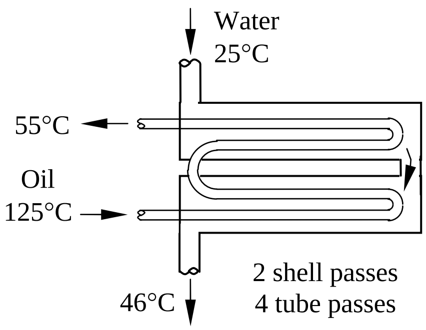
$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (0.416\text{ kg/s})(4.18\text{ kJ/kg.°C})(56-20)\text{°C} = \mathbf{62.6\ kW}$$

**16-136** Oil is cooled by water in a 2-shell passes and 4-tube passes heat exchanger. The mass flow rate of water and the surface area are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There is no fouling.

***Properties*** The specific heat of oil is given to be 2 kJ/kg.°C. The specific heat of water is taken to be 4.18 kJ/kg.°C.

***Analysis*** The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are




$$\Delta T_1 = T_{h,in} - T_{c,out} = 125°\text{C} - 46°\text{C} = 79°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 55°\text{C} - 25°\text{C} = 30°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{79 - 30}{\ln(79/30)} = 50.61°\text{C}$$

$$\left.\begin{array}{l} P = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{55 - 125}{25 - 125} = 0.7 \\ R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{25 - 46}{55 - 125} = 0.3 \end{array}\right\} F = 0.97 \text{ (Fig. 16-18)}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (10 \text{ kg/s})(2.0 \text{ kJ/kg}\cdot°\text{C})(125 - 55)°\text{C} = 1400 \text{ kW}$$

The mass flow rate of water is

$$\dot{m}_w = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{1400 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(46°\text{C} - 25°\text{C})} = \mathbf{15.9 \text{ kg/s}}$$

The surface area of the heat exchanger is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$

$$1400 \text{ kW} = (0.9 \text{ kW/m}^2 \cdot \text{C}) A_s (0.97)(50.61°\text{C})$$

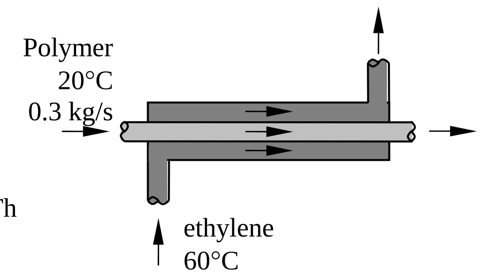
$$A_s = \mathbf{31.7 \text{ m}^2}$$

**16-137** A polymer solution is heated by ethylene glycol in a parallel-flow heat exchanger. The rate of heat transfer, the outlet temperature of polymer solution, and the mass flow rate of ethylene glycol are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There is no fouling.

***Properties*** The specific heats of polymer and ethylene glycol are given to be 2.0 and 2.5 kJ/kg.°C, respectively.

***Analysis*** (a) The logarithmic mean temperature difference is




$$\Delta T_1 = T_{h,in} - T_{c,in} = 60°\text{C} - 20°\text{C} = 40°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = 15°\text{C}$$

$$\Delta T_{lm,PF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40-15}{\ln(40/15)} = 25.49°\text{C}$$

The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s \Delta T_{lm} = (240\ \text{W/m}^2.°\text{C})(0.8\ \text{m}^2)(25.49°\text{C}) = \mathbf{4894\ W}$$

(b) The outlet temperatures of both fluids are

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in}) \rightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_c} = 20°\text{C} + \frac{4894\ \text{W}}{(0.3\ \text{kg/s})(2000\ \text{J/kg}\cdot°\text{C})} = 28.2°\text{C}$$

$$T_{h,out} = \Delta T_{out} + T_{c,out} = 15°\text{C} + 28.2°\text{C} = \mathbf{43.2°C}$$

(c) The mass flow rate of ethylene glycol is determined from

$$\dot{m}_{ethylene} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{4894\ \text{W}}{(2500\ \text{kJ/kg.°C})(60°\text{C} - 43.2°\text{C})} = \mathbf{0.117\ kg/s}$$

**16-138** The inlet and exit temperatures and the volume flow rates of hot and cold fluids in a heat exchanger are given. The rate of heat transfer to the cold water, the overall heat transfer coefficient, the fraction of heat loss, the heat transfer efficiency, the effectiveness, and the NTU of the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Changes in the kinetic and potential energies of fluid streams are negligible. **3** Fluid properties are constant.

***Properties*** The densities of hot water and cold water at the average temperatures of (38.9+27.0)/2 = 33.0°C and (14.3+19.8)/2 = 17.1°C are 994.8 and 998.6 kg/m$^3$, respectively. The specific heat at the average temperature is 4178 J/kg.°C for hot water and 4184 J/kg.°C for cold water (Table A-15).

***Analysis*** (a) The mass flow rates are

$$\dot{m}_h = \rho_h \dot{V}_h = (994.8\ \text{kg/m}^3)(0.0025/60\ \text{m}^3/\text{s}) = 0.04145\ \text{kg/s}$$

$$\dot{m}_c = \rho_c \dot{V}_c = (998.6\ \text{kg/m}^3)(0.0045/60\ \text{m}^3/\text{s}) = 0.07490\ \text{kg/s}$$

The rates of heat transfer from the hot water and to the cold water are

$$\dot{Q}_h = [\dot{m}c_p(T_{in} - T_{out})]_\text{h} = (0.04145\ \text{kg/s})(4178\ \text{kJ/kg.°C})(38.9\text{°C} - 27.0\text{°C}) = 2061\ \text{W}$$

$$\dot{Q}_c = [\dot{m}c_p(T_{out} - T_{in})]_\text{c} = (0.07490\ \text{kg/s})(4184\ \text{kJ/kg.°C})(19.8\text{°C} - 14.3\text{°C}) = \mathbf{1724\ W}$$

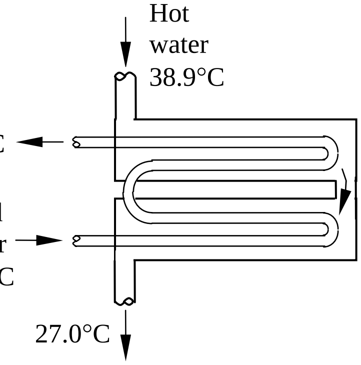
(b) The logarithmic mean temperature difference and the overall heat transfer coefficient are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 38.9\text{°C} - 19.8\text{°C} = 19.1\text{°C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 27.0\text{°C} - 14.3\text{°C} = 12.7\text{°C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{19.1 - 12.7}{\ln\left(\frac{19.1}{12.7}\right)} = 15.68\text{°C}$$

$$U = \frac{\dot{Q}_{hc,m}}{A\Delta T_{lm}} = \frac{(1724 + 2061)/2\ \text{W}}{(0.04\ \text{m}^2)(15.68\text{°C})} = \mathbf{3017\ W/m^2 \cdot C}$$




Note that we used the average of two heat transfer rates in calculations.

(c) The fraction of heat loss and the heat transfer efficiency are

$$f_{loss} = \frac{\dot{Q}_h - \dot{Q}_c}{\dot{Q}_h} = \frac{2061 - 1724}{2061} = 0.164 = \mathbf{16.4\%}$$

$$\eta = \frac{\dot{Q}_c}{\dot{Q}_h} = \frac{1724}{2061} = 0.836 = \mathbf{83.6\%}$$

(d) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.04145\ \text{kg/s})(4178\ \text{kJ/kg.°C}) = 173.2\ \text{W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.07490\ \text{kg/s})(4184\ \text{kJ/kg.°C}) = 313.4\ \text{W/°C}$$

Therefore

$$C_{\min} = C_h = 173.2\ \text{W/°C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (173.2\ \text{W/°C})(38.9\text{°C} - 14.3\text{°C}) = 4261\ \text{W}$$

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{(1724 + 2061)/2\ \text{kW}}{4261\ \text{kW}} = 0.444 = \mathbf{44.4\%}$$

One again we used the average heat transfer rate. We could have used the smaller or greater heat transfer rates in calculations. The NTU of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{\min}} = \frac{(3017\ \text{W/m}^2 \cdot \text{C})(0.04\ \text{m}^2)}{173.2\ \text{W/°C}} = \mathbf{0.697}$$

**16-139 . . . 16-143 Design and Essay Problems**

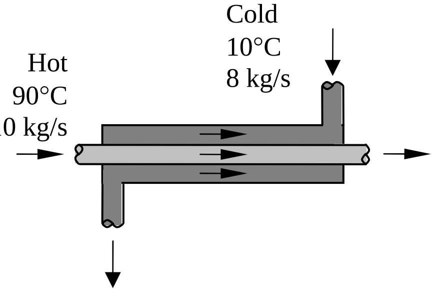
**16-143** A counter flow double-pipe heat exchanger is used for cooling a liquid stream by a coolant. The rate of heat transfer and the outlet temperatures of both fluids are to be determined. Also, a replacement proposal is to be analyzed.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There is no fouling.

***Properties*** The specific heats of hot and cold fluids are given to be 3.15 and 4.2 kJ/kg.°C, respectively.

***Analysis*** (a) The overall heat transfer coefficient is

$$U = \frac{600}{\dfrac{1}{\dot m_c^{0.8}} + \dfrac{2}{\dot m_h^{0.8}}} = \frac{600}{\dfrac{1}{8^{0.8}} + \dfrac{2}{10^{0.8}}} = 1185\ \text{W/m}^2.\text{K}$$




The rate of heat transfer may be expressed as

$$\dot Q = \dot m_c c_c (T_{c,out} - T_{c,in}) = (8)(4200)(T_{c,out} - 10) \qquad (1)$$

$$\dot Q = \dot m_h c_h (T_{h,in} - T_{h,out}) = (10)(3150)(90 - T_{h,out}) \qquad (2)$$

It may also be expressed using the logarithmic mean temperature difference as

$$\dot Q = UA\Delta T_{lm} = UA\frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = (1185)(9)\frac{(90 - T_c) - (T_h - 10)}{\ln\left(\dfrac{90 - T_c}{T_h - 10}\right)} \qquad (3)$$

We have three equations with three unknowns, solving an equation solver such as EES, we obtain

$$\dot Q = \mathbf{6.42 \times 10^5\ W}, \quad T_{c,out} = \mathbf{29.1°C}, \quad T_{h,out} = \mathbf{69.6°C}$$

(b) The overall heat transfer coefficient for each unit is

$$U = \frac{600}{\dfrac{1}{\dot m_c^{0.8}} + \dfrac{2}{\dot m_h^{0.8}}} = \frac{600}{\dfrac{1}{4^{0.8}} + \dfrac{2}{5^{0.8}}} = 680.5\ \text{W/m}^2.\text{K}$$

Then

$$\dot Q = \dot m_c c_c (T_{c,out} - T_{c,in}) = (2\times 4)(4200)(T_{c,out} - 10) \qquad (1)$$

$$\dot Q = \dot m_h c_h (T_{h,in} - T_{h,out}) = (2\times 5)(3150)(90 - T_{h,out}) \qquad (2)$$

$$\dot Q = UA\Delta T_{lm} = UA\frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = (680.5)(2\times 5)\frac{(90 - T_c) - (T_h - 10)}{\ln\left(\dfrac{90 - T_c}{T_h - 10}\right)} \qquad (3)$$

Once again, we have three equations with three unknowns, solving an equation solver such as EES, we obtain

$$\dot Q = \mathbf{4.5 \times 10^5\ W}, \quad T_{c,out} = \mathbf{23.4°C}, \quad T_{h,out} = \mathbf{75.7°C}$$

***Discussion*** Despite a higher heat transfer area, the new heat transfer is about 30% lower. This is due to much lower *U*, because of the halved flow rates. So, the vendor's recommendation is not acceptable. The vendor's unit will do the job provided that they are connected in series. Then the two units will have the same *U* as in the existing unit.