Finned Surface

Thursday, February 29, 2024 11:42 AM

$$\dot{Q}_{conv} = h A_{fin} (T_b - T_\infty)$$
$$= h(pL)(T_b - T_\infty)$$

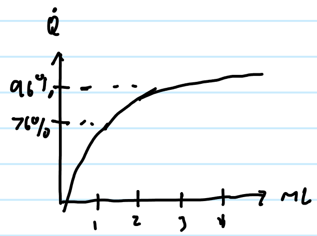
perimeter

$$\dot{Q}_{fin} = \int_0^L h(T(x) - T_\infty)\, p\, dx$$

$$\dot{Q}_{fin} = \sqrt{hpkA_c}\,(T_b - T_\infty)\tanh(mL)$$

$$\tanh(mL) = \frac{e^{mL} - e^{-mL}}{e^{mL} + e^{-mL}}$$

$$m = \sqrt{hp/kA_c}$$



Proper length of fin:

$mL = 1$

Fin efficiency:

$$\eta_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{fin,max}} = \frac{\tanh(mL)}{mL}$$

constant cross-section only

temp is constant through entire fin length

Fin effectiveness:

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{no\ fin}} = \frac{\dot{Q}_{fin}}{hA_b(T_b - T_\infty)} = \frac{A_{fin}}{A_b}\eta_{fin}$$

Convection Fundamentals

Thursday, February 29, 2024 11:06 AM

Newton's Law of Cooling:

$$\dot{Q}_{conv} = h A_s (T_s - T_\infty)$$

$$\dot{q}_{conv} = h (T_s - T_\infty)$$

$$\frac{h L_c}{k} = C\left(\frac{\rho V L_c}{\mu}\right)^b \left(\frac{\mu c_p}{k}\right)^e$$

dynamic viscosity

$$Nu = C\, Re_{L_c}^{\,b}\, Pr^e$$

Nusselt number

Reynold's number $= \dfrac{\text{inertial force}}{\text{viscous force}}$

Prandlts number $= \dfrac{\nu}{\alpha} = \dfrac{\text{kinematic viscosity}}{\text{thermal diffusivity}}$, $\nu = \dfrac{\mu}{\rho}$

External Forced Convection.

evaluate fluid properties at film temperature

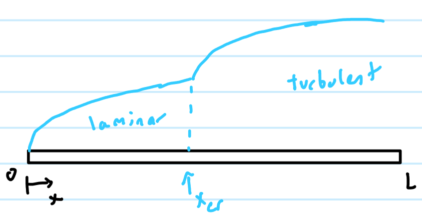
$$T_f = \frac{T_s + T_\infty}{2}$$

Flat plate:

$$Re_{cr} = \frac{V x_{cr}}{\nu} = 5\times10^5$$

$Re_x < 5\times10^5 \rightarrow$ laminar: $Nu_x = \dfrac{hx}{k} = .332\, Re_x^{.5}\, Pr^{.333}$

$Re_x > 5\times10^5 \rightarrow$ turbolent: $Nu_x = \dfrac{hx}{k} = .0296\, Re_x^{.8}\, Pr^{.333}$



$Re_L < 5\times10^5 \rightarrow$ laminar:

$$Nu = \frac{hL}{k} = .664\, Re_L^{.5}\, Pr^{.333}$$

$Re_L > 5\times10^5 \rightarrow$ turbolent + laminar:

$$Nu = \frac{hL}{k} = (.037\, Re_L^{.8} - 871)\, Pr^{.333}$$

EXAMPLE 7–1 Flow of Hot Oil Over a Flat Plate

Engine oil at 60°C flows over the upper surface of a 5-m-long flat plate whose temperature is 20°C with a velocity of 2 m/s (**Fig. 7–13**). Determine the total drag force and the rate of heat transfer per unit width of the entire plate.

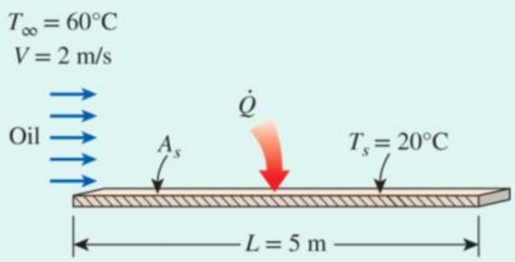


FIGURE 7–13 Schematic for **Example 7–1**.

SOLUTION Engine oil flows over a flat plate. The total drag force and the rate of heat transfer per unit width of the plate are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5 \times 10^5$.

Properties The properties of engine oil at the film temperature of $T_f = (T_s + T_\infty)/2 = (20 + 60)/2 = 40°C$ are (**Table A-13**)

$$\rho = 876 \text{ kg/m}^3 \qquad \text{Pr} = 2962$$
$$k = 0.1444 \text{ W/m·K} \qquad \nu = 2.485 \times 10^{-4} \text{ m}^2/\text{s}$$

Analysis Noting that $L = 5$ m, the Reynolds number at the end of the plate is

$$\text{Re}_L = \frac{\text{VL}}{\nu} = \frac{(2 \text{ m/s})(5 \text{ m})}{2.485 \times 10^{-4} \text{ m}^2/\text{s}} = 4.024 \times 10^4$$

which is less than the critical Reynolds number. Thus we have *laminar flow* over the entire plate, and the average friction coefficient is

$$C_f = 1.33\,\mathrm{Re}_L^{-0.5} = 1.33 \times \left(4.024 \times 10^4\right)^{-0.5} = 0.00663$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for parallel flow over a flat plate, the drag force acting on the plate per unit width becomes

$$F_D = C_f A \frac{\rho V^2}{2} = 0.00663\left(5 \times 1\ \mathrm{m}^2\right)\frac{\left(876\ \mathrm{kg/m^3}\right)\left(2\ \mathrm{m/s}\right)^2}{2}\left(\frac{1\ \mathrm{N}}{1\ \mathrm{kg \cdot m/s^2}}\right) = 58.1\ \mathrm{N}$$

The total drag force acting on the entire plate can be determined by multiplying the value obtained above by the width of the plate.

This force per unit width corresponds to the weight of a mass of about 6 kg. Therefore, a person who applies an equal and opposite force to the plate to keep it from moving will feel like he or she is using as much force as is necessary to hold a 6-kg mass from dropping.

Similarly, the Nusselt number is determined using the laminar flow relations for a flat plate,

$$\mathrm{Nu} = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} = 0.664 \times \left(4.024 \times 10^4\right)^{0.5} \times 2962^{1/3} = 1913$$

Then,

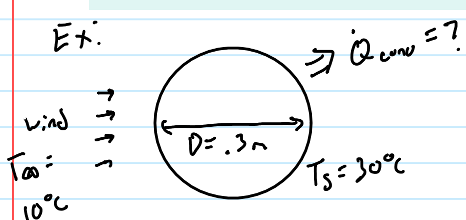
$$h = \frac{k}{L}\mathrm{Nu} = \frac{0.144\ \mathrm{W/m \cdot K}}{5\ \mathrm{m}}(1913) = 55.25\ \mathrm{W/m^2 \cdot K}$$

and

$$\dot{Q} = hA_s(T_\infty - T_s) = \left(55.25\ \mathrm{W/m^2 \cdot K}\right)\left(5 \times 1\ \mathrm{m}^2\right)(60 - 20)^\circ\mathrm{C} = 11{,}050\ \mathrm{W}$$

Discussion Note that heat transfer is always from the higher-temperature medium to the lower-temperature one. In this case, it is from the oil to the plate. The heat transfer rate is per meter width of the plate. The heat transfer for the entire plate can be obtained by multiplying the value obtained by the actual width of the plate.

Ex:



$V = 12\ \frac{km}{hr}$

$\dot{Q}_{conv} = hA_s(T_s - T_\infty)$

$A_s = \pi D^2 = .283\ m^2$

$h = ?$

$Nu = \frac{hD}{k} = 2 + (.4Re^{\frac{1}{2}} + .06Re^{\frac{2}{3}})Pr^{.4}\left(\frac{u_\infty}{u_s}\right)^{\frac{1}{4}}$

$Re = \frac{VD}{\nu} = \frac{\rho VD}{\mu} = \frac{(12{,}000/3600\ m/s)\ .3}{1.426 \times 10^{-5}} = 7.01 \times 10^4$

$$Re = \frac{VD}{\nu} = \frac{\rho VD}{\mu} = \frac{(12{,}000/3600 \text{ m/s}) \, .3}{1.426\times10^{-5}} = 7.01\times10^{4}$$

$$T_{\infty} = 10^{\circ}C \rightarrow \nu, Pr, k, \mu_{\infty}$$

$$T_s = 30^{\circ}C \rightarrow \mu_s$$

HW5

Thursday, February 29, 2024 12:14 PM

HW5

ME 3525 HW5: Finned surfaces and project

Part 1: Watch the lecture video on the project and answer the following questions:

1. Can we use the one-dimensional heat conduction we learned in Chap. 2 to solve for T(x) in this project? Why?

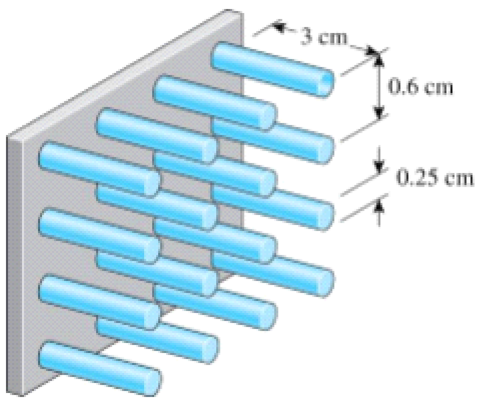
2. What is the meaning of rou_e in the governing equation? What is the unit of m in the analytical solution? How do we calculate the m value?

3. Which four tables and four figures do you need to show in your project report?

4. At which electrical current do we compare the analytical solution and the numerical solutions?

Part 2: Problem on Finned surfaces

A hot surface at 100°C is to be cooled by attaching 3-cm-long, 0.25-cm-diameter aluminum pin fins (k = 237 W/m·K) to it, with a center-to-center distance of 0.6 cm. The temperature of the surrounding medium is 30°C, and the heat transfer coefficient on the surfaces is 35 W/m^2·K. Determine the rate of heat transfer from the surface for a 1-m × 1-m section of the plate.



P1:

1. No, we cannot use the one-dimensional heat conduction equation to solve for T(x) because the heat transfer is not in one direction. There is at least heat transfer in the y and x directions. We have to derive our own equation from the first law of thermodynamics.
2. rou_e (ρ_e) is the electrical resistivity of the copper wire. We need to find the material property using an outside reference. The constant/property m is unitless. It is found using the equation below.
   $m^2 = hp/kA_c$
3. Four tables that compare the numerical solution to the analytical solution at each node are needed. These include the three, five, seven, and eleven node cases. Four figures are also needed. Figure 1 shows the schematic diagram of the wire, and Figure 2 shows the analytical solution for the temperature distribution for an electrical current of 60 A, 40 A, and 0 A. Figure 3 compares the four node cases using the analytical solution and numerical solution, with temperature on vertical axis, and x in cm for the horizontal axis. Figure 4 should show the temperature at the center of the wire on the y axis and the number of nodes on the x axis for both solutions.
4. When the electrical current is 60 Amperes, we obtain the numerical solution to the temperatures at the nodes and validate the numerical results with the prediction from the analytical solution.

P2:

$L_c = L + D/4 = .03 + .0025/4 = .030625$

$A_{fin} = \pi D L_c = \pi D(L + D/4) = .000241$

$m = \sqrt{4h/kD} = \sqrt{\frac{4(35)}{237(.0025)}} = 15.3716$

$\eta_{fin} = \frac{\tanh mL_c}{mL_c} = .761594$

$\dot{Q}_{fin} = \eta_{fin} h A_{fin} (T_b - T_\infty) = .761594(35)(.000241)(100-30)$

$= .448803 \text{ W}$

$\dot{Q}_{no fin} = h A_s (T_b - T_\infty) = 35(1)(1)(100-30)$

$= 2450 \text{ W}$

$\dot{Q}_{tot} = n\dot{Q}_{fin} + \dot{Q}_{no fin}$

$n = \left(\frac{1 \text{ m}}{.006 \text{ m}} + 1\right)^2 = 167^2 = 27889$

$\dot{Q}_{tot} = 27889(.448803) + 2450$

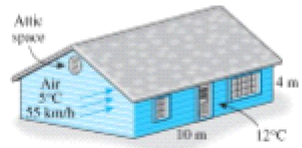
$= \boxed{14966.7 \text{ W}}$

HW6

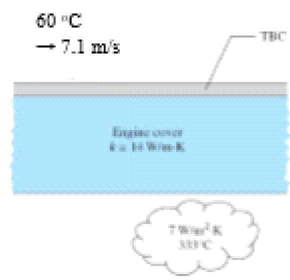
Sunday, March 17, 2024 8:24 PM

HW6

1. During a cold winter day, wind at 55 km/h is blowing parallel to a 4-m-high and 10-m-long wall of a house. If the air outside is at 5°C and the surface temperature of the wall is 12°C, determine the rate of heat loss from that wall by convection. What would your answer be if the wind velocity was doubled?

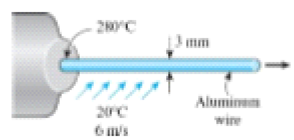


2. The outer surface of an engine is situated in a place where oil leakage can occur. When leaked oil comes in contact with a hot surface that has a temperature above its autoignition temperature, the oil can ignite spontaneously. Consider an engine cover that is made of a stainless steel plate with a thickness of 1 cm and a thermal conductivity of 14 W/m·K. The inner surface of the engine cover is exposed to hot air with a convection heat transfer coefficient of 7 W/m2·K at a temperature of 333°C. The engine outer surface is cooled by air blowing in parallel over the 2-m-long surface at 7.1 m/s, in an environment where the ambient air is at 60°C. To prevent fire hazard in the event of oil leak on the engine cover, a layer of thermal barrier coating (TBC) with a thermal conductivity of 1.1 W/m·K is applied on the engine cover outer surface. Would a TBC layer with a thickness of 4 mm in conjunction with 7.1 m/s air cooling be sufficient to keep the engine cover surface from going above 180°C to prevent fire hazard? Evaluate the air properties at 120°C.

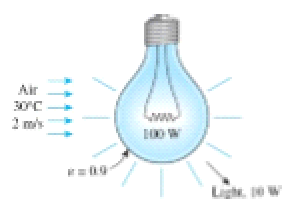


3. A 15-cm × 15-cm circuit board dissipating 20 W of power uniformly is cooled by air, which approaches the circuit board at 20°C with a velocity of 6 m/s. Disregarding any heat transfer from the back surface of the board, determine the surface temperature of the electronic components (*a*) at the leading edge and (*b*) at the end of the board. Assume the flow to be turbulent since the electronic components are expected to act as turbulators. For air properties evaluations assume a film temperature of 35°C. Is this a good assumption?

4. A long aluminum wire of diameter 3 mm is extruded at a temperature of 280°C. The wire is subjected to cross air flow at 20°C at a velocity of 6 m/s. Determine the rate of heat transfer from the wire to the air per meter length when it is first exposed to the air.



5. Consider a 10-cm-diameter 100-W lightbulb cooled by a fan that blows air at 30°C to the bulb at a velocity of 2 m/s. The surrounding surfaces are also at 30°C, and the emissivity of the glass is 0.9. Assuming 10 percent of the energy passes through the glass bulb as light with negligible absorption and the rest of the energy is absorbed and dissipated by the bulb itself, determine the equilibrium temperature of the glass bulb. Assume a surface temperature of 100°C for evaluation of μ_s. Is this a good assumption?



1. $\dot{Q} = hA_s(T_s - T_\infty)$

$$T_f = \frac{12+5}{2} = 8.5\,°C \approx 10°C$$

$$V = 55\ \frac{km}{hr}\ \frac{1hr}{3600s}\ \frac{1000m}{1km} = 15.2778\ \frac{m}{s}$$

Table A-15:

$\rho = 1.246\ kg/m^3$

$k = .02439\ W/m\cdot K$

$Pr = .7336$

$\nu = 1.426\times10^{-5}\ m^2/s$

$$Re_L = \frac{VL}{\nu} = \frac{15.2778(10)}{1.426\times10^{-5}} = 1.07137\times10^7 \quad > 5\times10^5$$

Laminar + Turbulent. $5\times10^5 < Re_L < 10^7$ (close enough)

$$Nu = \frac{hL}{k} = (.037 Re_L^{.8} - 871)\,Pr^{1/3}$$

$$= 14009$$

$$h = Nu\frac{k}{L} = 14009\frac{(.02439)}{10} = 34.168$$

$$\dot{Q} = hA_s(T_s - T_\infty) = 34.168(10)(5)(12-5)$$

$$\boxed{\dot{Q} = 11958.8\ W}$$

$Re_{L,double} = 2.14\times10^7$

$Nu_{double} = 24\,426$

$h_{double} = 59.5424$

$$\boxed{\dot{Q}_{double} = 20\,839.8\ W}$$

2. $T_f = 120°C$

$k = .03235\ W/m\cdot k$

$\nu = 2.527\times10^{-5}\ m^2/s$

$Pr = .7073$

$$Re_L = \frac{VL}{\nu} = \frac{7.1(2)}{2.522\cdot10^{-5}} = 5.63\times10^5 \quad > 5\times10^5$$

$$Nu = \frac{hL}{k} = (.037 Re_L^{.8} - 871)\,Pr^{1/3}$$

$$Nu = 537.681$$

$$h = \frac{Nu\cdot k}{L} = \frac{537.681\cdot .03235}{2} = 8.697\ W/m^2\cdot K$$

$$R_{conv,i} = \frac{1}{h_i A}$$

$$R_{ss} = \frac{L_{ss}}{k_{ss} A}$$

$$R_{TBC} = \frac{L_{TBC}}{K_{TBC} A}$$

$$R_{conv,o} = \frac{1}{h_o A}$$

$$A R_{eq} = \frac{1}{7} + \frac{.1}{14} + \frac{.004}{1.1} + \frac{1}{8.697} = .26862$$

$$A R_{conv,o} = \frac{1}{h_o} = \frac{1}{8.697} = .114982$$

$$\dot{q} = \frac{\dot{Q}}{A} = \frac{T_{\infty,i} - T_{\infty,o}}{A R_{eq}} = \frac{T_{s,o} - T_{\infty,o}}{A R_{conv}} \Rightarrow T_{s,o} = \frac{R_{conv,o}}{R_{eq}}(T_{\infty,i} - T_{\infty,o}) + T_{\infty,o}$$

$$T_{s,o} = \frac{.114982}{.26862}(337 - 60) + 60$$

$$\boxed{T_{s,o} = 176.858\ ^\circ C}$$

$\boxed{\text{Yes, keeps } T_{s,o} < 180^\circ C}$

3. $T_f = 20^\circ C$

Table A-15:

$K = .02\,63\ W/m\,^\circ C$

$\nu = 1.655 \times 10^{-5}\ m^2/s$

$Pr = .7268$

$$Re_x = \frac{Vx}{\nu} = \frac{6(15)}{1.655 \times 10^{-5}} = 5.438 \times 10^4,$$ but assume turbulent.

$$Nu_x = \frac{h_x x}{K} = .0308\, Re_x^{.8} Pr^{1/3} = .0308(5.438 \times 10^4)^{.8}(.7268)^{\frac{1}{3}} = 170.1$$

$$h_x = \frac{kx}{x} Nu = \frac{.02623}{.15}(170.1) = 29.77$$

$$\dot{q} = h_x(T_s - T_\infty) \Rightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20 + \frac{20(.15)^2}{29.77} = 49.9^\circ C$$

$$\boxed{T_{s,end} = 49.9^\circ C}$$

$$\boxed{T_{s,\text{leading edge}} = 20^\circ C, \text{since } h \to \infty}$$

4. $T_f = \frac{280 + 20}{2} = 150^\circ C$

$K = .03447\ W/m\,^\circ C$

$v = 2.86 \times 10^{-5}$

$Pr = .70275$

$Re = \frac{VD}{v} = \frac{6(.003)}{2.86\times10^{-5}} = 629.371$ laminar

$$Nu_{cyl} = .3 + \frac{.62 Re^{\frac{1}{2}} Pr^{\frac{1}{3}}}{\left[1+\left(\frac{.4}{Pr}\right)^{\frac{2}{3}}\right]^{\frac{1}{4}}}\left[1+\left(\frac{Re}{282000}\right)^{\frac{5}{8}}\right]^{\frac{4}{5}}$$

$Nu_{cyl} = 12.648$

$h = \frac{k Nu_{cyl}}{D} = \frac{12.648 \times .02 ...}{.003} = 145.152 \ W/m^2 \cdot K$

$\dot{Q} = hA_s(T_s - T)$

$= hD\pi(T_s - T)$

$= 145.152\,\pi(.003)(280-20)$

$\boxed{= 355.686 \ W/m}$

5.

$T_f = \frac{100+30}{2} = 165°C \approx 160$

From Table A-15 @ 30°C:

$k = .02588$

$v = 1.608 \times 10^{-5}$

$Pr = .7282$

$\mu_\infty = 1.872 \times 10^{-5}$

μ_s @ 160°C $= 2.42\times10^{-5}$

$Re = \frac{VD}{v} = \frac{2(.1)}{1.608\times10^{-5}} = 12437.8$, laminar

$Nu = \frac{hD}{k} = 2 + [.4 Re^{\frac{1}{2}} + .06 Re^{\frac{2}{3}}] Pr^{.4} \left(\frac{\mu_\infty}{\mu_s}\right)^{\frac{1}{4}}$

$Nu = 65.46$

$h = \frac{k}{D} Nu = \frac{.02588}{.1}(65.46) = 16.9408$

$\dot{Q}_{conv} = hA_s(T_s - T_\infty)$

$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{surr}^4)$

$\cdot\, 9\dot{Q} = \dot{Q}_{rad} + \dot{Q}_{conv}$

$T_s = ?$

$90 = .9(5.67\times10^{-8})(T_s^4 - 30^4)(\pi(.1)^2) + 16.9408(\pi(.1)^2)(T_s - 30)$

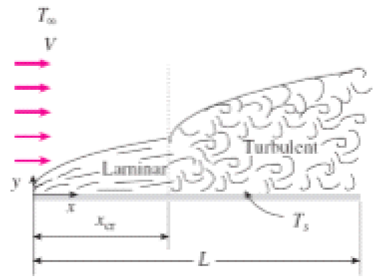
Solving,

$$\boxed{\begin{array}{l} T_s = 194.8\,^\circ C \\ \text{Not a great assumption} \end{array}}$$

Convection Formulas

Tuesday, March 19, 2024 11:07 AM

Formula+Sheet+conv...



<u>External forced convections</u>

$$\dot{Q}_{conv} = hA_s\left(T_s - T_\infty\right)$$

$$\dot{q}_{conv} = h\left(T_s - T_\infty\right)$$

1. Flow over a flat plate

If T_s = constant,

Nusselt numbers for **average** heat transfer coefficients

Laminar: $\quad \text{Nu} = \dfrac{hL}{k} = 0.664\ \text{Re}_L^{0.5}\ \text{Pr}^{1/3} \qquad \text{Re}_L < 5 \times 10^5$

Laminar + turbulent $\quad \text{Nu} = \dfrac{hL}{k} = (0.037\ \text{Re}_L^{0.8} - 871)\text{Pr}^{1/3} \qquad 0.6 \le \text{Pr} \le 60, \quad 5 \times 10^5 \le \text{Re}_L \le 10^7$

If q_s = constant,

Nusselt numbers for **local** heat transfer coefficients ($\text{Nu}_x = h_x x/k$)

Laminar: $\text{Nu}_x = 0.453\ \text{Re}_x^{0.5}\ \text{Pr}^{1/3} \qquad \text{Pr} > 0.6, \qquad \text{Re}_x < 5 \times 10^5$

Turbulent: $\text{Nu}_x = 0.0308\ \text{Re}_x^{0.8}\ \text{Pr}^{1/3} \qquad 0.6 \le \text{Pr} \le 60, \qquad 5 \times 10^5 \le \text{Re}_x \le 10^7$

2.

For flow over a *cylinder*

$$\text{Nu}_{cyl} = \frac{hD}{k} = 0.3 + \frac{0.62\ \text{Re}^{1/2}\ \text{Pr}^{1/3}}{[1 + (0.4/\text{Pr})^{2/3}]^{1/4}}\left[1 + \left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5} \qquad \text{RePr} > 0.2$$

The fluid properties are evaluated at the *film temperature* $T_f = \frac{1}{2}(T_\infty + T_s)$

3.

For flow over a *sphere*

$$\text{Nu}_{sph} = \frac{hD}{k} = 2 + [0.4\ \text{Re}^{1/2} + 0.06\ \text{Re}^{2/3}]\ \text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$3.5 \le \text{Re} \le 80{,}000$ and $0.7 \le \text{Pr} \le 380$

The fluid properties are evaluated at the free-stream temperature T_∞, except for μ_s, which is evaluated at the surface temperature T_s.

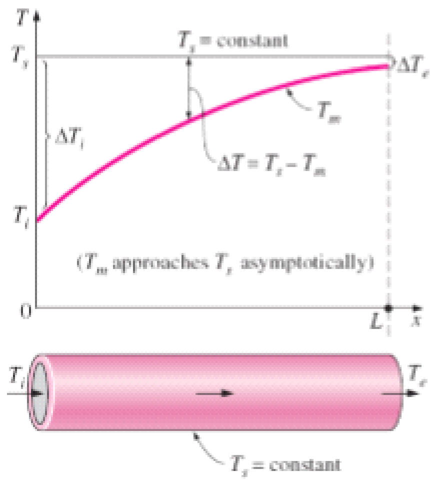
1

Internal forced convections

Entry length: $L_{t,\,\text{laminar}} \approx 0.05\,\text{Re}\,\text{Pr}\,D$

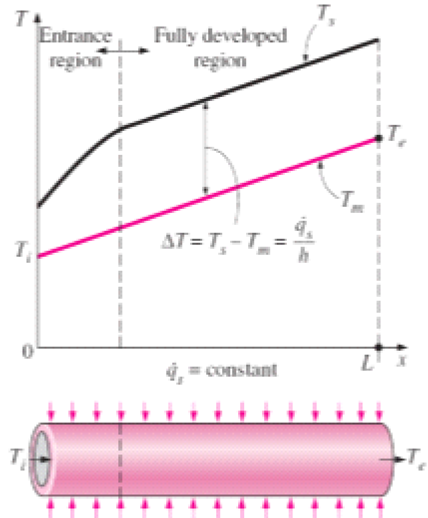
$L_{t,\,\text{turbulent}} \approx 10D$

Two common cases:



$$\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{hA_s}{\dot{m}c_p}$$

$$\dot{Q}_{conv.} = hA_s\Delta T_{lm}$$



Surface temperature:

$$\dot{q}_s = h(T_s - T_m) \longrightarrow T_s = T_m + \frac{\dot{q}_s}{h}$$

(1) Fully developed laminar flow in a circular tube

(a) q_s = const. : Nu =4.36; (b) T_s = const. : Nu = 3.66.

(2) Laminar flow in a circular tube with constant surface temperature and in the entry region:

$$\text{Nu} = 3.66 + \frac{0.065\,(D/L)\,\text{Re}\,\text{Pr}}{1 + 0.04[(D/L)\,\text{Re}\,\text{Pr}]^{2/3}}$$

(3) Fully developed turbulent flow in circular tubes:

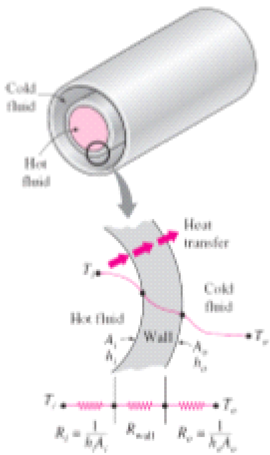
$$\text{Nu} = \frac{(f/8)(\text{Re} - 1000)\,\text{Pr}}{1 + 12.7(f/8)^{0.5}(\text{Pr}^{2/3} - 1)} \qquad \begin{pmatrix} 0.5 \le \text{Pr} \le 2000 \\ 3 \times 10^3 < \text{Re} < 5 \times 10^6 \end{pmatrix}$$

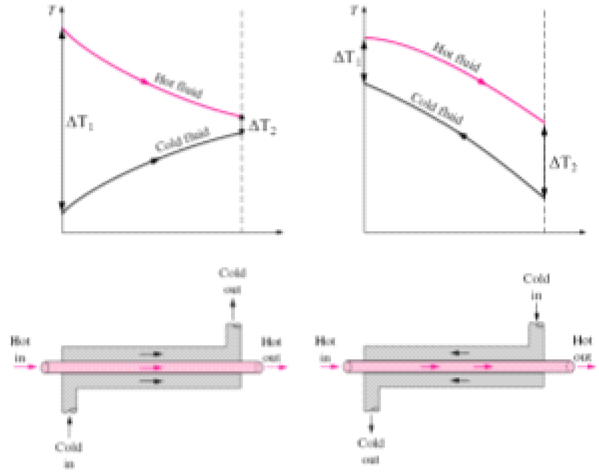
Smooth tubes: $f = (0.790 \ln \text{Re} - 1.64)^{-2} \qquad 3000 < \text{Re} < 5 \times 10^6$

(4) Fully developed turbulent flow (Re > 10,000) in **noncircular** pipes: $D_h = 4A_c/p$

$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^n \qquad n = 0.4$ for *heating* and 0.3 for *cooling* of fluids in pipe

Heat Exchangers





(a) Parallel flow (b) Counter flow

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

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

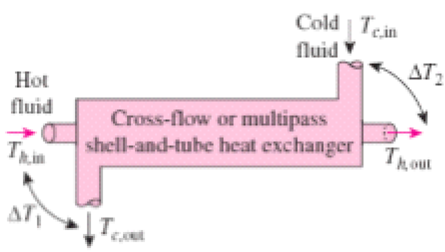
$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,\text{out}} - T_{c,\text{in}})$$

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

$\dot{Q} = \dot{m} h_{fg}$ when phase change occurs

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

where $\Delta T_{\text{lm}} = \dfrac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$



Heat transfer rate:

$\dot{Q} = UA_s F \Delta T_{\text{lm,CF}}$ where $F = f(P, R)$ in Fig. 11-19

Internal Forced Convection

Thursday, March 21, 2024 11:27 AM

EXAMPLE 8–1 Heating of Water in a Tube by Steam

Water enters a 2.5-cm-internal-diameter thin copper tube of a heat exchanger at 15°C at a rate of 0.3 kg/s, and is heated by steam condensing outside at 120°C. If the average heat transfer coefficient is 800 W/m^2·K, determine the length of the tube required in order to heat the water to 115°C (Fig. 8–16). Assume water is at high enough pressure so that it leaves the heat exchanger as a liquid at 115°C.

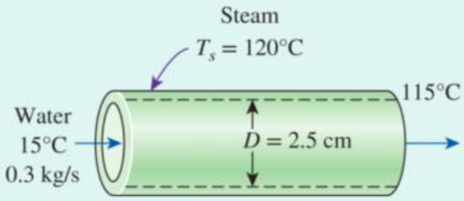


FIGURE 8–16 Schematic for Example 8–1.

SOLUTION Water is heated by steam in a circular tube. The tube length required to heat the water to a specified temperature is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Fluid properties are constant. **3** The convection heat transfer coefficient is constant. **4** The conduction resistance of copper tube is negligible, so the inner surface temperature of the tube is equal to the condensation temperature of steam.

Properties The specific heat of water at the bulk mean temperature of $(15+115)/2 = 65°C$ is 4187 J/kg·K. The heat of condensation of steam at 120°C is 2203 kJ/kg (Table A–9).

Analysis Knowing the inlet and exit temperatures of water, the rate of heat transfer is determined to be

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.3\ \text{kg/s})(4.187\ \text{kJ/kg}\cdot\text{K})(115°\text{C} - 15°\text{C}) = 125.6\ \text{kW}$$

The log mean temperature difference is

Page 503

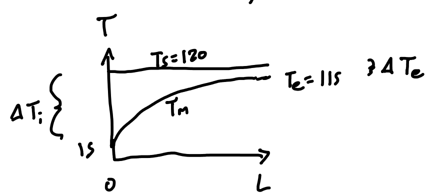
$$\Delta T_e = T_s - T_e = 120°\text{C} - 115°\text{C} = 5°\text{C}$$
$$\Delta T_i = T_s - T_i = 120°\text{C} - 15°\text{C} = 105°\text{C}$$
$$\Delta T_{\text{lm}} = \frac{\Delta T_e - T_i}{\ln(\Delta T_e/\Delta T_i)} = \frac{5-105}{\ln(5/105)} = 32.85°\text{C}$$

The heat transfer surface area is

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow A_s = \frac{\dot{Q}}{h\Delta T_{\text{lm}}} = \frac{125.6\ \text{kW}}{(0.8\ \text{kW/m}^2\cdot\text{K})(32.85°\text{C})} = 4.78\ \text{m}^2$$

Then the required tube length becomes

Alternatively,



$\Delta T_e = \Delta T_i e^{-hA_s/\dot{m}c_p}$ $A_s = \pi DL$

$\dot{Q} = \dot{m}c_p(T_e - T_i)$

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

$L = -\dfrac{\dot{m}c_p \ln(\Delta T_e/\Delta T_i)}{h\pi D}$

$\Delta T_e = 120 - 115 = 5\ °C$

$\Delta T_i = 120 - 15 = 105\ °C$

Table A-9:

$T_b = \dfrac{15+115}{2} = 65\ °C$

$c_p = \cancel{4}187$ J/kg·K

$L = 61$ m

$\Delta T_e - T_i$

Then the required tube length becomes

$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D} = \frac{4.78 \text{ m}^2}{\pi(0.025 \text{ m})} = 61 \text{ m}$$

Discussion The bulk mean temperature of water during this heating process is 65°C, and thus the *arithmetic* mean temperature difference is $\Delta T_{am} = 120 - 65 = 55°C$. Using ΔT_{am} instead of ΔT_{lm} would give $L = 36$ m, which is grossly in error. This shows the importance of using the log mean temperature in calculations.

$$\frac{\Delta T_e - T_i}{\ln(\Delta T_e / T_i)}$$

Finding heat transfer coefficient:

$$Nu = \frac{hD}{k}$$

Check

1. Geometry
2. Boundary Conditions
3. Reynolds Number
4. Flow region

I. Laminar Flow in a circular tube

1. Fully developed
   a. $\dot{q}_s = \text{const} \Rightarrow Nu = \frac{hD}{k} = 4.36$
   b. $T_s = \text{const} \Rightarrow Nu = \frac{hD}{k} = 3.66$
2. Entrance region ($T_s = \text{const}$)

$$Nu = \frac{hD}{k} = 3.66 + \frac{.065(D/L)\,Re\,Pr}{1 + .04[(D/L)\,Re\,Pr]^{2/3}}$$

II Turbulent Flow in Circular Tube

1. Fully developed

$$Nu = \frac{hD}{k} = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)(Pr^{2/3} - 1)}$$

$f = F(Re, \varepsilon/D)$

for smooth tubes,

$f = (.79 \ln Re - 1.64)^{-2}$

EXAMPLE 8–3 Flow of Oil in a Pipeline Through a Lake

Consider the flow of oil at $20°C$ in a 30-cm-diameter pipeline at an average velocity of 2 m/s (Fig. 8–26). A 200-m-long section of the horizontal pipeline passes through icy waters of a lake at $0°C$. Measurements indicate that the surface temperature of the pipe is very nearly $0°C$. Disregarding the thermal resistance of the pipe material, determine (*a*) the temperature of the oil when the pipe leaves the lake, (*b*) the rate of heat transfer from the oil, and (*c*) the pumping power required to overcome the pressure losses and to maintain the flow of the oil in the pipe.

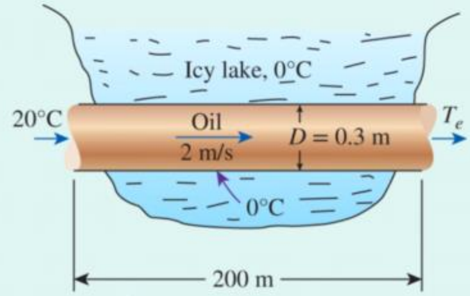
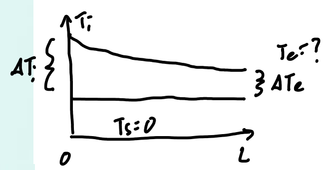


FIGURE 8–26 Schematic for Example 8–3.

SOLUTION Oil flows in a pipeline that passes through icy waters of a lake at $0°C$. The exit temperature of the oil, the rate of heat loss, and the pumping power needed to overcome pressure losses are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is very nearly $0°C$. **3** The thermal resistance of the pipe is negligible. **4** The inner surfaces of the pipeline are smooth. **5** The flow is hydrodynamically developed when the pipeline reaches the lake.



Properties We do not know the exit temperature of the oil, and thus we cannot determine the bulk mean temperature, which is the temperature at which the properties of oil are to be evaluated. The mean temperature of the oil at the inlet is $20°C$, and we expect this temperature to drop somewhat as a result of heat loss to the icy waters of the lake. We evaluate the properties of the oil at the inlet temperature, but we will repeat the calculations, if necessary, using properties at the evaluated bulk mean temperature. At 20°C we read (Table A–13)

$$\rho = 888.1 \text{ kg/m}^3 \quad \nu = 9.429 \times 10^{-4} \text{ m}^2/\text{s}$$
$$k = 0.145 \text{ W/m·K} \quad c_p = 1880 \text{ J/kg·K} \quad \text{Pr} = 10{,}863$$

$\Delta T_e = \Delta T_i e^{-hA_s/\dot{m}C_p}$

$T_s - T_e = (T_s - T_i) e^{-hA_s/\dot{m}C_p}$

$T_e = T_s - (T_s - T_i) e^{-hA_s/\dot{m}C_p}$

$A_s = \pi DL = \pi(.3)(200)\ 188.5 m^2$

$\dot{m} = \rho VA_c = (888.1 \frac{kg}{m^3})(2) \frac{\pi}{4}(.3)^2 = 125.6 \frac{kg}{s}$

$T_b = (T_i + T_e)/2$

Assume T_e is 20°C, iterate

$T_b = 20$

$C_p = 1880$

$\nu = 9.429 \times 10^{-4}$

$k = .145$

$Pr = 10.863$

Analysis (*a*) The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(2 \text{ m/s})(0.3 \text{ m})}{9.429 \times 10^{-4} \text{ m}^2/\text{s}} = 636$$

which is less than the critical Reynolds number of 2300. Therefore, the flow is laminar, and the thermal entry length in this case is roughly

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$$L_t \approx 0.05 \text{ Re Pr } D = 0.05 \times 636 \times 10{,}863 \times (0.3 \text{ m}) \approx 103{,}600 \text{ m}$$

this case is roughly

$$L_t \approx 0.05 \text{ Re Pr } D = 0.05 \times 636 \times 10{,}863 \times (0.3 \text{ m}) \approx 103{,}600 \text{ m}$$

which is much greater than the total length of the pipe. This is typical of fluids with high Prandtl numbers. Therefore, we assume thermally developing flow and determine the Nusselt number from

$$\begin{aligned} \text{Nu} &= \frac{hD}{k} = 3.66 + \frac{0.065(D/L)\text{ Re Pr}}{1 + 0.04\left[(D/L)\text{ Re Pr}\right]^{2/3}} \\ &= 3.66 + \frac{0.065(0.3/200) \times 636 \times 10{,}863}{1 + 0.04\left[(0.3/200) \times 636 \times 10{,}863\right]^{2/3}} \\ &= 37.3 \end{aligned}$$

Note that this Nusselt number is considerably higher than the fully developed value of 3.66. Then,

$$h = \frac{\text{k}}{D}\text{Nu} = \frac{0.145 \text{ W/m}\cdot\text{K}}{0.3 \text{ m}}(37.3) = 18.0 \text{ W/m}^2\cdot\text{K}$$

Also,

$$\begin{aligned} A_s &= \pi D L = \pi(0.3 \text{ m})(200 \text{ m}) = 188.5 \text{ m}^2 \\ \dot{m} &= \rho A_c V_{\text{avg}} = (888.1 \text{ kg/m}^3)\left[\frac{1}{4}\pi(0.3 \text{ m})^2\right](2 \text{ m/s}) = 125.6 \text{ kg/s} \end{aligned}$$

Next we determine the exit temperature of oil,

$$\begin{aligned} T_e &= T_s - (T_s - T_i)\exp\left(-hA_s/\dot{m}c_p\right) \\ &= 0°\text{C} - \left[(0 - 20)°\text{C}\right]\exp\left[-\frac{(18.0 \text{ W/m}^2\cdot\text{K})(188.5 \text{ m}^2)}{(125.6 \text{ kg/s})(1881 \text{ J/kg}\cdot\text{K})}\right] \\ &= \mathbf{19.71°C} \end{aligned}$$

Thus, the mean temperature of oil drops by a mere 0.29°C as it crosses the lake. This makes the bulk mean oil temperature 19.86°C, which is practically identical to the inlet temperature of 20°C. Therefore, we do not need to reevaluate the properties.

(*b*) The log mean temperature difference and the rate of heat loss from the oil are

$$\Delta T_{\text{lm}} = \frac{T_i - T_e}{\ln\frac{T_s - T_e}{T_s - T_i}} = \frac{20 - 19.71}{\ln\frac{0 - 19.71}{0 - 20}} = -19.85°\text{C}$$

$$\dot{Q} = hA_s\,\Delta T_{\text{lm}} = (18.0 \text{ W/m}^2\cdot\text{K})(188.5 \text{ m}^2)(-19.85°\text{C}) = \mathbf{-6.75 \times 10^4\ W}$$

Therefore, the oil will lose heat at a rate of 67.5 kW as it flows through the pipe in the icy waters of the lake. Note that ΔT_{lm} is identical to the arithmetic mean temperature in this case, since $\Delta T_i \approx \Delta T_e$.

(*c*) The laminar flow of oil is hydrodynamically developed. Therefore, the friction factor can be determined from

$$f = \frac{64}{\text{Re}} = \frac{64}{636} = 0.1006$$

Then the pressure drop in the pipe and the required pumping power become

Page 514

$$\Delta P = f\frac{L\rho V^2_{\text{avg}}}{\ldots} = 0.1006\frac{200 \text{ m}(888.1 \text{ kg/m}^3)(2 \text{ m/s})^2}{\ldots} = 1.19 \times 10^5 \text{N/m}^2$$

$$\Delta P = f\frac{L}{D}\frac{\rho V_{avg}^2}{2} = 0.1006\frac{200\ \text{m}}{0.3\ \text{m}}\frac{(888.1\ \text{kg/m}^3)(2\ \text{m/s})^2}{2} = 1.19 \times 10^5\ \text{N/m}^2$$

$$\dot{W}_{pump} = \frac{\dot{m}\Delta P}{\rho} = \frac{(125.6\ \text{kg/s})(1.19 \times 10^5\ \text{N/m}^2)}{888.1\ \text{kg/m}^3} = 16.8\ \text{kW}$$

Discussion We need a 16.8-kW pump just to overcome the friction in the pipe as the oil flows in the 200-m-long pipe through the lake.

EXAMPLE 8–5 Heating of Water by Resistance Heaters with Turbulent Flow in a Tube

Water is to be heated from 15°C to 65°C as it flows through a 3-cm-internal-diameter 5-m-long tube (**Fig. 8-32**). The tube is equipped with an electric resistance heater that provides uniform heating throughout the surface of the tube. The outer surface of the heater is well insulated, so in steady operation, all the heat generated in the heater is transferred to the water in the tube. If the system is to provide hot water at a rate of 10 L/min, determine the power rating of the resistance heater. Also estimate the inner surface temperature of the tube at the exit.

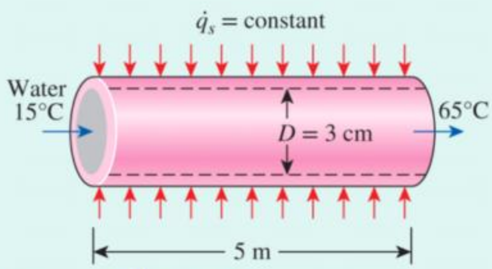
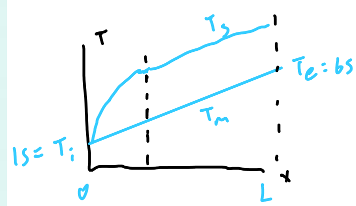


FIGURE 8–32 Schematic for **Example 8–5.**



$\dot{q}_s = \frac{\dot{Q}}{A_s}$

SOLUTION Water is to be heated in a tube equipped with an electric resistance heater on its surface. The power rating of the heater and the inner surface temperature at the exit are to be determined.

a. $\dot{W}_e = \dot{Q} = \dot{m} C_p (T_e - T_i)$

$T_b = \frac{T_i + T_e}{2} = 40$

$C_p = 4179\ J/kg \cdot K$

$\rho = 992.1\ kg/m^3$

$k = .631\ W/m.K$

$Pr = 4.32$

$\nu = .658 \times 10^{-6}\ m^2/s$

$\dot{m} = \rho \dot{V} = .165\ kg/s$

$\dot{W}_e = .165\ (4179)(65-15) = 34.6\ kW$

b. $T_{s,exit} = ?$

$\dot{q}_s = h(T_s - T_m) = h(T_{s,exit} - T_e)$

Assumptions **1** Steady flow conditions exist. **2** The surface heat flux is uniform. **3** The inner surfaces of the tube are smooth.

Properties The properties of water at the bulk mean temperature of $T_b = (T_i + T_e)/2 = (15 + 65)/2 = 40°C$ are (**Table A-9**)

$$\rho = 992.1\ \text{kg/m}^3 \qquad c_p = 4179\ \text{J/kg·K}$$
$$k = 0.631\ \text{W/m·K} \qquad \text{Pr} = 4.32$$
$$\nu = \mu/\rho = 0.658 \times 10^{-6}\ \text{m}^2/\text{s}$$

Analysis The cross-sectional and heat transfer surface areas are

$$A_c = \frac{1}{4}\pi D^2 = \frac{1}{4}\pi(0.03\ \text{m})^2 = 7.069 \times 10^{-4}\ \text{m}^2$$
$$A_s = \pi DL = \pi(0.03\ \text{m})(5\ \text{m}) = 0.471\ \text{m}^2$$

The volume flow rate of water is given as

$\dot{V} = 10\ \text{L/min} = 0.01\ \text{m}^3/\text{min}$. Then the mass flow rate becomes

$$\dot{m} = \rho\dot{V} = (992.1\ \text{kg/m}^3)(0.01\ \text{m}^3/\text{min}) = 9.921\ \text{kg/min} = 0.1654\ \text{kg/s}$$

b. $T_{s,exit} = ?$

$\dot{q}_s = h(T_s - T_m) = h(T_{s,exit} - T_e)$

$$A_c = \frac{1}{4}\pi D^2 = \frac{1}{4}\pi(0.03\text{ m})^2 = 7.069 \times 10^{-4}\text{ m}^2$$
$$A_s = \pi DL = \pi(0.03\text{ m})(5\text{ m}) = 0.471\text{ m}^2$$

The volume flow rate of water is given as

$\dot{V} = 10\text{ L/min} = 0.01\text{ m}^3\text{/min}$. Then the mass flow rate becomes

$$\dot{m} = \rho\dot{V} = (992.1\text{ kg/m}^3)(0.01\text{ m}^3\text{/min}) = 9.921\text{ kg/min} = 0.1654\text{ kg/s}$$

To heat the water at this mass flow rate from 15°C to 65°C, heat must be supplied to the water at a rate of

$$\begin{aligned}\dot{Q} &= \dot{m}c_p(T_e - T_i)\\ &= (0.1654\text{ kg/s})(4.179\text{ kJ/kg·K})(65-15)°\text{C}\\ &= 34.6\text{ kJ/s} = 34.6\text{ kW}\end{aligned}$$

All of this energy must come from the resistance heater. Therefore, the power rating of the heater must be **34.6 kW**.

The surface temperature T_s of the tube at any location can be determined from

$$\dot{q}_s = h(T_s - T_m) \rightarrow T_s = T_m + \frac{\dot{q}_s}{h}$$

where h is the heat transfer coefficient and T_m is the mean temperature of the fluid at that location. The surface heat flux is constant in this case, and its value can be determined from

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{34.6\text{ kW}}{0.471\text{ m}^2} = 73.46\text{ kW/m}^2$$

To determine the heat transfer coefficient, we first need to find the mean velocity of water and the Reynolds number:

$$\begin{aligned}V_{avg} &= \frac{\dot{V}}{A_c} = \frac{0.010\text{ m}^3\text{/min}}{7.069 \times 10^{-4}\text{ m}^2} = 14.15\text{ m/min} = 0.236\text{ m/s}\\ \text{Re} &= \frac{V_{avg}D}{\nu} = \frac{(0.236\text{ m/s})(0.03\text{ m})}{0.658 \times 10^{-6}\text{ m}^2\text{/s}} = 10{,}760\end{aligned}$$

which is greater than 10,000. Therefore, the flow is turbulent, and the entry length is roughly

$$L_h \approx L_t \approx 10D = 10 \times 0.03 = 0.3\text{ m}$$

which is much shorter than the total length of the tube. Therefore, we can assume fully developed turbulent flow in the entire tube and determine the Nusselt number from

$$\text{Nu} = \frac{hD}{k} = 0.023\text{ Re}^{0.8}\text{ Pr}^{0.4} = 0.023(10{,}760)^{0.8}(4.32)^{0.4} = 69.4$$

$Nu = \dfrac{(\frac{f}{8})(Re - 1000)Pr}{1 + 12.7(f/8)^{.5}(Pr^{\frac{2}{3}} - 1)}$

Then,

$$h = \frac{k}{D}\text{Nu} = \frac{0.631\text{ W/m·K}}{0.03\text{ m}}(69.4) = 1460\text{ W/m}^2\text{·K}$$

$f = .790(\ln Re - 1.64)^{-2} = .0308$

and the inner surface temperature of the tube at the exit becomes and the inner surface temperature of the tube at the exit becomes

$$T_s = T_m + \frac{\dot{q}_s}{h} = 65°\text{C} + \frac{73{,}460\ \text{W/m}^2}{1460\ \text{W/m}^2\cdot\text{K}} = 115°\text{C}$$

Discussion Note that the inner surface temperature of the tube will be 50°C higher than the mean water temperature at the tube exit. This temperature difference of 50°C between the water and the surface will remain constant throughout the fully developed flow region.

HW7

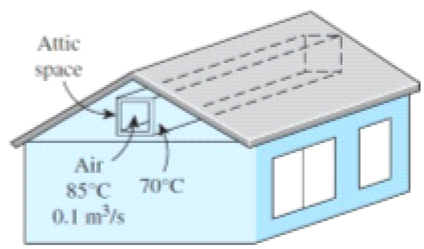
Thursday, April 4, 2024 11:04 AM

HW7

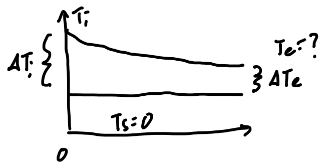
1. Consider the flow of oil at 10°C in a 40-cm-diameter pipeline at an average velocity of 0.5 m/s. A 1500-m-long section of the pipeline passes through icy waters of a lake at 0°C. Measurements indicate that the surface temperature of the pipe is very nearly 0°C. Disregarding the thermal resistance of the pipe material, determine (*a*) the temperature of the oil when the pipe leaves the lake, (*b*) the rate of heat transfer from the oil.

2. Water is to be heated from 10°C to 80°C as it flows through a 2-cm-internal-diameter, 13-m-long tube. The tube is equipped with an electric resistance heater, which provides uniform heating throughout the surface of the tube. The outer surface of the heater is well insulated, so that in steady operation all the heat generated in the heater is transferred to the water in the tube. If the system is to provide hot water at a rate of 5 L/min, determine the power rating of the resistance heater. Also, estimate the inner surface temperature of the pipe at the exit.

3. Hot air at atmospheric pressure and 85°C enters a 10-m-long uninsulated square duct of cross section 0.15 m × 0.15 m that passes through the attic of a house at a rate of 0.1 m^3/s. The duct is observed to be nearly isothermal at 70°C. Determine the exit temperature of the air and the rate of heat loss from the duct to the air space in the attic. Evaluate air properties at a bulk mean temperature of 75°C. Is this a good assumption?



1.



$\Delta T_e = \Delta T_i e^{-hA_s/\dot{m}C_p}$

$T_s - T_e = (T_s - T_i) e^{-hA_s/\dot{m}C_p}$

$T_e = T_s - (T_s - T_i) e^{-hA_s/\dot{m}C_p}$

$A_s = \pi DL = \pi(.4)(1500) = 1884.96$

$\dot{m} = \rho V A_c =$

$T = 10°C$. Table A-13, Linear interpolation

$\rho = 893.55$

$k = .14595$

$v = 2.5825 \times 10^{-3}$

$C_p = 1829$

$Pr = 28749.5$

$Re = \dfrac{V_{avg} D}{v} = \dfrac{.5(.4)}{2.5825 \times 10^{-3}} = 77.44 < 2300$

laminar,

$L_t = .5\, Re\, Pr\, D = .5(77.44)(28749.5)(.4)$

$= 445297 \gg L$

$Nu = \dfrac{hD}{k} = 3.66 + \dfrac{.065(D/L)Re\,Pr}{1 + .04[(D/L)Re\,Pr]^{2/3}}$

$= 36.92$

$h = \dfrac{k}{D} Nu = \dfrac{.14595}{.4}(36.92) = 13.4717$

$A_s = \pi DL = 1884.96$

$\dot{m} = \rho A_c V_{avg} = 893.55(\pi \frac{.4^2}{4})(.5) = 56.14$

$T_e = T_s - (T_s - T_i) e^{-hA_s/\dot{m}c_p}$

$= 0 - (0 - 10) e^{-13.4717(1884.96)/56.14(1829)}$

$\boxed{T_e = 7.8196°C}$

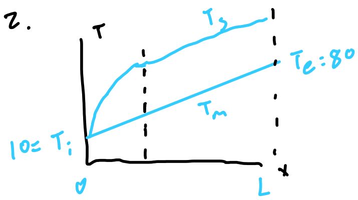
b. $\Delta T_{ln} = \dfrac{T_i - T_e}{\ln \frac{T_s - T_e}{T_s - T_i}} = \dfrac{10 - 7.8196}{\ln \frac{0 - 7.8196}{0 - 10}} = -8.8652$

$\dot{Q} = hA_s \Delta T_{ln}$

$= 13.4717(1884.96)(-8.8652)$

$\boxed{\dot{Q} = -2.2512 \times 10^5 \text{ W}}$

2.



$\dot{q}_s = \frac{\dot{Q}}{A_s}$

a. $\dot{W}_e = \dot{Q} = \dot{m} c_p (T_e - T_i)$

$T_b = \frac{T_i + T_e}{2} = 45$

$c_p = 4180 \ J/kg \cdot K$

$\rho = 990.1 \ kg/m^3$

$k = .637 \ W/m \cdot K$

$Pr = 3.91$

$\nu = \mu/\rho = 596 \cdot 10^{-3}/990.1 = 6.02 \times 10^{-7}$

$\dot{m} = \rho \dot{V} = 990.1 (5) \frac{1}{1000} \frac{1}{60} = .0825$

$\dot{W}_e = .0825 (4180)(80-10)$

$\boxed{\dot{W}_e = 24.1395 \ kW}$

b. $T_{s,exit} = ?$

$\dot{q}_s = h(T_s - T_m) = h(T_{s,exit} - T_e)$

$T_{s,e} = \frac{q_s}{h} + T_m$

$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{24.1395}{\pi (.02)(13)} = 29.55 \ \frac{kW}{m^2}$

$V_{avg} = \frac{\dot{V}}{A_c} = \frac{\frac{5}{60 \cdot 1000}}{\pi (.02)^2/4} = .2652 \ m/s$

$Re = \frac{V_{avg} D}{\nu} = \frac{.2652 (.02)}{6.02 \times 10^{-7}} = 8813.16 < 10,000$

$L_t = .05 \, Re \, Pr \, D = .05 (8813)(3.91)(.02) = 34.46 \ m > L = 15$

Laminar, entry region, constant $\dot{q}_s$?

3.

$T_b = 75°C$

$\rho = .9994$

$k = .02953$

$\nu = 2.097 \times 10^{-5}$

$c_p = 1008$

$Pr = .7154$

$D_h = \frac{4A_c}{P} = a = .15$

$V_{avg} = \frac{\dot{V}}{A_c} = \frac{.1}{(.15)^2} = 4.44 \ m/s$

$Re = \frac{V_{avg} D_h}{\nu} = \frac{4.44(.15)}{2.097 \times 10^{-5}} = 31759.7 > 10{,}000$

$L_h = 10D = 10(.15) = 1.5 < L$

$Nu = \frac{hD_h}{k} = .023 Re^{.8} Pr^{.3} = 83.099$

$h = \frac{k}{D_h} Nu = 83.099 \frac{.02953}{.15} = 16.359$

$A_s = 4aL = 4(.15)(10) = 6 \ m^2$

$\dot{m} = \rho \dot{V} = .9994(.1) = .09994$

$T_e = T_s - (T_s - T_i) e^{-hA_s/\dot{m}c_p}$

$= 70 - (70 - 85) e^{-16.359(6)/.09994(1008)}$

$\boxed{T_e = 32.08}$ < 70, error in calc

Not a good assumption

$\Delta T_{lm} = \frac{T_i - T_e}{\ln \frac{T_s - T_e}{T_s - T_i}} = \frac{85 - 32.08}{\ln \frac{70 - 32.08}{70 - 85}} =$

domain error

Heat Exchangers

Thursday, April 4, 2024 11:15 AM

EXAMPLE 11–2 Effect of Fouling on the Overall Heat Transfer Coefficient

A double-pipe (shell-and-tube) heat exchanger is constructed of a stainless steel $(k = 15.1\ \mathrm{W/m{\cdot}K})$ inner tube of inner diameter $D_i = 1.5$ cm and outer diameter $D_o = 1.9$ cm and an outer shell of inner diameter 3.2 cm. The convection heat transfer coefficient is given to be $h_i = 800\ \mathrm{W/m^2{\cdot}K}$ on the inner surface of the tube and $h_o = 1200\ \mathrm{W/m^2{\cdot}K}$ on the outer surface. For a fouling factor of $R_{f,i} = 0.0004\ \mathrm{m^2{\cdot}K/W}$ on the tube side and $R_{f,o} = 0.0001\ \mathrm{m^2{\cdot}K/W}$ on the shell side, determine (*a*) the thermal resistance of the heat exchanger per unit length and (*b*) the overall heat transfer coefficients U_i and U_o based on the inner and outer surface areas of the tube, respectively.

SOLUTION The heat transfer coefficients and the fouling factors on the tube and shell sides 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 The heat transfer coefficients and the fouling factors are constant and uniform.

Analysis (*a*) The schematic of the heat exchanger is given in **Fig. 11–12**. The thermal resistance for an unfinned shell-and-tube heat exchanger with fouling on both heat transfer surfaces is given by **Eq. 11–8** as

$$R = \frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o} = \frac{1}{h_iA_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{f,o}}{A_o} + \frac{1}{h_oA_o}$$

$$R = \frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o} = \frac{1}{hA_i} + \frac{R_{fi}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{fo}}{A_o} + \frac{1}{h_oA_o}$$

where

$$A_i = \pi D_iL = \pi(0.015\ \mathrm{m})(1\ \mathrm{m}) = 0.0471\ \mathrm{m^2}$$
$$A_o = \pi D_oL = \pi(0.019\ \mathrm{m})(1\ \mathrm{m}) = 0.0597\ \mathrm{m^2}$$

$A_i = \pi D_i L$

$A_o = \pi D_o L$

Substituting, the total thermal resistance is determined to be

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$$\begin{aligned} R &= \frac{1}{(800\ \mathrm{W/m^2{\cdot}K})(0.0471\ \mathrm{m^2})} + \frac{0.0004\ \mathrm{m^2{\cdot}K/W}}{0.0471\ \mathrm{m^2}} \\ &\quad + \frac{\ln(0.019/0.015)}{2\pi(15.1\ \mathrm{W/m{\cdot}K})(1\ \mathrm{m})} \\ &\quad + \frac{0.0001\ \mathrm{m^2{\cdot}K/W}}{0.0597\ \mathrm{m^2}} + \frac{1}{(1200\ \mathrm{W/m^2{\cdot}K})(0.0597\ \mathrm{m^2})} \\ &= (0.02654 + 0.00849 + 0.0025 + 0.00168 + 0.01396)\mathrm{K/W} \\ &= 0.0532^\circ\mathrm{C/W} \end{aligned}$$

Note that about 19 percent of the total thermal resistance in this case is due to fouling and about 5 percent of it is due to the steel tube separating the two fluids. The rest (76 percent) is due to the convection resistances.

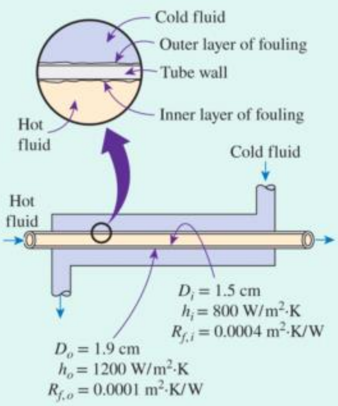

FIGURE 11–12 Schematic for **Example 11–2**.

$U_i = \frac{1}{RA_i}$

(*b*) Knowing the total thermal resistance and the heat transfer surface areas, the overall heat transfer coefficients based on the inner and outer surfaces of the tube are

$U_o = \frac{1}{RA_o}$

FIGURE 11–12 Schematic for Example 11–2.

(*b*) Knowing the total thermal resistance and the heat transfer surface areas, the overall heat transfer coefficients based on the inner and outer surfaces of the tube are

$$U_i = \frac{1}{RA_i} = \frac{1}{(0.0532\ \mathrm{K/W})(0.0471\ \mathrm{m^2})} = 399\ \mathrm{W/m^2 \cdot K}$$

and

$$U_o = \frac{1}{RA_o} = \frac{1}{(0.0532\ \mathrm{K/W})(0.0597\ \mathrm{m^2})} = 315\ \mathrm{W/m^2 \cdot K}$$

DISCUSSION Note that the two overall heat transfer coefficients differ significantly (by 27 percent) in this case because of the considerable difference between the heat transfer surface areas on the inner and the outer sides of the tube. For tubes of negligible thickness, the difference between the two overall heat transfer coefficients would be negligible.

$U_o = \frac{1}{RA_o}$

EXAMPLE 11–4 Heating Water in a Counterflow Heat Exchanger

A counterflow double-pipe heat exchanger is to heat water from 20°C to 80°C at a rate of 1.2 kg/s. The heating is to be accomplished by geothermal water available at 160°C at a mass flow rate of 2 kg/s. The inner tube is thin-walled and has a diameter of 1.5 cm. If the overall heat transfer coefficient of the heat exchanger is 640 W/m²·K, determine the length of the heat exchanger required to achieve the desired heating.

SOLUTION Water is heated in a counterflow double-pipe heat exchanger by geothermal water. The required length 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. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties We take the specific heats of water and geothermal fluid to be 4.18 and 4.31 kJ/kg·K, respectively.

Analysis The schematic of the heat exchanger is given in Fig. 11–21. The rate of heat transfer in the heat exchanger can be determined from

$$\dot{Q} = \left[\dot{m}c_p(T_{out} - T_{in})\right]_{water} = (1.2\ \mathrm{kg/s})(4.18\ \mathrm{kJ/kg \cdot K})(80 - 20)°\mathrm{C} = 301\ \mathrm{kW}$$

Noting that all of this heat is supplied by the geothermal water, the outlet temperature of the geothermal water is determined to be

$$\begin{aligned} \dot{Q} &= \left[\dot{m}c_p(T_{in} - T_{out})\right]_{geothermal} \\ T_{out} &= T_{in} - \frac{\dot{Q}}{\dot{m}c_p} \\ &= 160°\mathrm{C} - \frac{301\ \mathrm{kW}}{(2\ \mathrm{kg/s})(4.31\ \mathrm{kJ/kg \cdot K})} \end{aligned}$$

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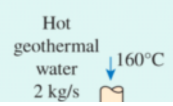
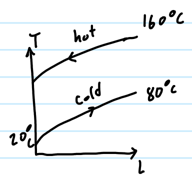
Knowing the inlet and outlet temperatures of both fluids, the logarithmic mean temperature difference for this counterflow heat exchanger becomes

$$\Delta T_1 = T_{h,\,in} - T_{c,\,out} = (160 - 80)°\mathrm{C} = 80°\mathrm{C}$$
$$\Delta T_2 = T_{h,\,out} - T_{c,\,in} = (125 - 20)°\mathrm{C} = 105°\mathrm{C}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{80 - 105}{\ln(80/105)} = 91.9°\mathrm{C}$$





① $\dot{Q} = \dot{m}_c C_{p,c} (T_{c,out} - T_{c,in})$

$T_{b,c} = (T_{c,in} + T_{c,out})/2 = 50°c$

$\dot{Q} = 1.2(4180)(80-20) = 300 kW$

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

② $T_{h,out} = \frac{\dot{Q}_h}{\dot{m}_h C_{p,h}} - T_{h,in}$

$T_{h,out} =$ $T_{b,c} = (T_{h,in} + T_{h,out})/2$, $T_{h,out}$ unknown, so guess and iterate

Guess $T_{h,out} = 140°c$ get $T_{h,out} = 125$

Guess $T_{b,h} = 124.9$

③ $\dot{Q} = UA_s \Delta T_{lm}$

$\Rightarrow L = \frac{\dot{Q}}{U \pi D \Delta T_{lm}}$

$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$ $\Delta T_1 = 124.4 - 20 = 104.4°c$

ΔT_2 $160 - 80 = 80°c$

$\Delta T_{lm} = 919°c$

$L = 109 m$

Knowing the inlet and outlet temperatures of both fluids, the logarithmic mean temperature difference for this counterflow heat exchanger becomes

Page 697

$$\Delta T_1 = T_{h,\ in} - T_{c,\ out} = (160 - 80)°C = 80°C$$
$$\Delta T_2 = T_{h,\ out} - T_{c,\ in} = (125 - 20)°C = 105°C$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{80 - 105}{\ln(80/105)} = 91.9°C$$

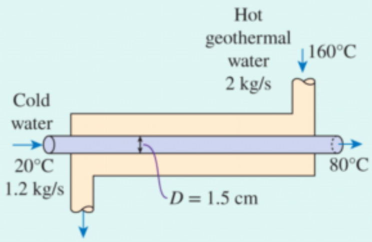


FIGURE 11–21 Schematic for **Example 11–4**.

Then the surface area of the heat exchanger is determined to be

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{301{,}000 \text{ W}}{(640 \text{ W/m}^2\cdot\text{K})(91.9°C)} = 5.12 \text{ m}^2$$

To provide this much heat transfer surface area, the length of the tube must be

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{5.12 \text{ m}^2}{\pi(0.015 \text{ m})} = 109 \text{ m}$$

$\ln(\Delta T_1/\Delta T_2)$

ΔT_2 $160 - 80 = 80°C$

$\Delta T_{lm} = 91.9°C$

$L = 109$ m

The correction factor is less than unity for a crossflow and multipass shell-and-tube heat exchanger. That is, $F \leq 1$. The limiting value of $F = 1$ corresponds to the counterflow heat exchanger. Thus, the correction factor F for a heat exchanger is *a measure of deviation of the* ΔT_{lm} *from the corresponding values for the counterflow case.*

The correction factor F for common crossflow and shell-and-tube heat exchanger configurations is given in **Fig. 11–19** versus two temperature ratios P and R defined as

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad \textbf{(11–27)}$$

and

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}c_p)_{\text{tube side}}}{(\dot{m}c_p)_{\text{shell side}}} \qquad \textbf{(11–28)}$$

where the subscripts 1 and 2 represent the *inlet* and *outlet,* respectively. Note that for a shell-and-tube heat exchanger, T and t represent the *shell-* and *tube-side* temperatures, respectively, as shown in the correction factor charts.It makes no difference whether the hot or the cold fluid flows through the shell or the tube. The determination of the correction factor F requires the availability of the inlet and the outlet temperatures for both the cold and hot fluids.

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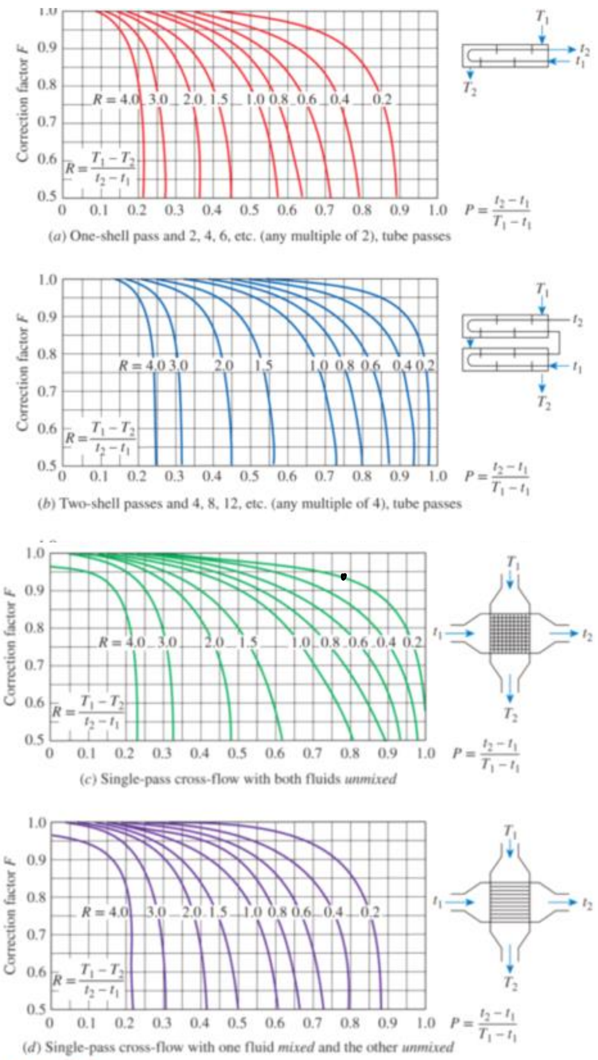


FIGURE 11–19 Correction factor F charts for common shell-and-tube and crossflow heat exchangers.

Source: Bowman, Mueller, and Nagle, 1940.

EXAMPLE 11–6 Cooling of Water in an Automotive Radiator

Page 699

A test is conducted to determine the overall heat transfer coefficient in an automotive radiator that is a compact crossflow water-to-air heat exchanger with both fluids (air and water) unmixed (**Fig. 11–23**). The radiator has 40 tubes of internal diameter 0.5 cm and length 65 cm in a closely spaced plate-finned matrix. Hot water enters the tubes at 90°C at a rate of 0.6 kg/s and leaves at 65°C. Air flows across the radiator through the interfin spaces and is heated from 20°C to 40°C. Determine the overall heat transfer coefficient U_i of this radiator based on the inner surface area of the tubes.

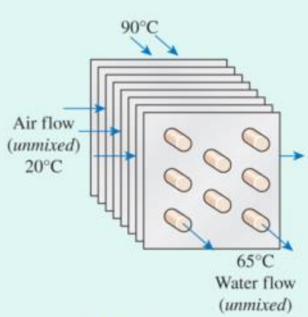
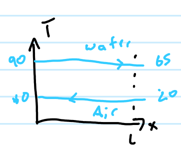


FIGURE 11–23 Schematic for **Example 11–6**.



SOLUTION During an experiment involving an automotive radiator, the inlet and exit temperatures of water and air and the mass flow rate of water are measured. The overall heat transfer coefficient based on the

SOLUTION During an experiment involving an automotive radiator, the inlet and exit temperatures of water and air 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** Changes in the kinetic and potential energies of fluid streams are negligible. **3** Fluid properties are constant.

Properties The specific heat of water at the average temperature of $(90+65)/2 = 77.5°\text{C}$ is 4.195 kJ/kg·K (Table A-9).

Analysis The rate of heat transfer in this radiator from the hot water to the air is determined from an energy balance on water flow,

$$\begin{aligned}\dot{Q} &= \left[\dot{m}c_p(T_{\text{in}} - T_{\text{out}})\right]_{\text{water}} = (0.6\ \text{kg/s})(4.195\ \text{kJ/kg·K})(90-65)°\text{C} \\ &= 62.93\ \text{kW}\end{aligned}$$

The tube-side heat transfer area is the total surface area of the tubes and is determined from (where n is the number of tubes)

$$A_i = n\pi D_i L = (40)\pi(0.005\ \text{m})(0.65\ \text{m}) = 0.408\ \text{m}^2$$

Knowing the rate of heat transfer and the surface area, the overall heat transfer coefficient can be determined from

$$\dot{Q} = U_i A_i F \Delta T_{\text{lm, CF}} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{\text{lm, CF}}}$$

where F is the correction factor and $\Delta T_{\text{lm, CF}}$ is the log mean temperature difference for the counterflow arrangement. These two quantities are found to be

$$\begin{aligned}\Delta T_1 &= T_{h,\text{ in}} - T_{c,\text{ out}} = (90-40)°\text{C} = 50°\text{C} \\ \Delta T_2 &= T_{h,\text{ out}} - T_{c,\text{ in}} = (65-20)°\text{C} = 45°\text{C} \\ \Delta T_{\text{lm, CF}} &= \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{50-45}{\ln(50/45)} = 47.5°\text{C}\end{aligned}$$

and

$$\left.\begin{aligned}P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{65-90}{20-90} = 0.36 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{20-40}{65-90} = 0.80\end{aligned}\right\} F = 0.97$$

(Fig. 11–19c)

Substituting, the overall heat transfer coefficient U_i is determined to be

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$$U_i = \frac{\dot{Q}}{A_i F \Delta T_{\text{lm, CF}}} = \frac{62{,}930\ \text{W}}{(0.408\ \text{m}^2)(0.97)(47.5°\text{C})} = 3347\ \text{W/m}^2\cdot\text{K}$$

Discussion Note that the overall heat transfer coefficient on the air side will be much lower because of the large surface area involved on that side.

HW8

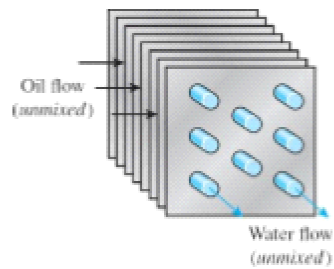
Thursday, April 11, 2024 11:00 AM

HW8

1. A double-pipe heat exchanger is constructed of a copper (k = 380 W/m·K) inner tube of internal diameter D_i = 1.2 cm and external diameter D_o = 1.6 cm and an outer tube of diameter 3.0 cm. The convection heat transfer coefficient is reported to be h_i = 700 W/m^2·K on the inner surface of the tube and h_o = 1400 W/m^2·K on its outer surface. For a fouling factor $R_{f,i}$ = 0.0005 m^2·K/W on the tube side and $R_{f,o}$ = 0.0002 m^2·K/W on the shell side, determine (*a*) the thermal resistance of the heat exchanger per unit length and (*b*) the overall heat transfer coefficients U_i and U_o based on the inner and outer surface areas of the tube, respectively.

2. A **thin-walled** double-pipe counter-flow heat exchanger is to be used to cool oil (c_p = 2200 J/kg·K) from 150°C to 40°C at a rate of 2 kg/s by water (c_p = 4180 J/kg·K) that enters at 22°C at a rate of 1.5 kg/s. The diameter of the tube is 2.5 cm, and its length is 6 m. Determine the overall heat transfer coefficient of this heat exchanger.

3. A single-pass cross-flow heat exchanger with both fluids unmixed has water entering at 16°C and exiting at 33°C, while oil (c_p = 1.93 kJ/kg·K and ρ = 870 kg/m^3) flowing at 0.19 m^3/min enters at 38°C and exits at 29°C. If the surface area of the heat exchanger is 20 m^2, determine the value of the overall heat transfer coefficient.



1.a.

$$R = \frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o} = \frac{1}{h_iA_i} + \frac{R_{fi}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{fo}}{A_o} + \frac{1}{hA_o}$$

1.a.

$$R = \frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o} = \frac{1}{h_iA_i} + \frac{R_{fi}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{fo}}{A_o} + \frac{1}{h_oA_o}$$

$$A_i = \pi D_i L = \pi(.012)(1) = .0377\ m^2$$

$$A_o = \pi D_o L = \pi(.016)(1) = .0503\ m^2$$

$$R = \frac{1}{700(.0377)} + \frac{.0005}{.0377} + \frac{\ln(.016/.012)}{2\pi(380)(1)} + \frac{.0002}{.0503} + \frac{1}{1400(.0503)}$$

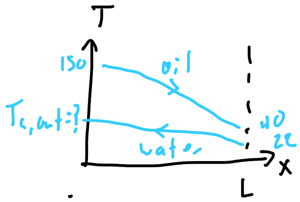
$$\boxed{R = .069467\ m^2 k/W}$$

b.

$$U_i = \frac{1}{RA_i} = \frac{1}{.069467(.0377)} = \boxed{381.85\ W/m^2 \cdot k}$$

$$U_o = \frac{1}{RA_o} = \frac{1}{.069467(.0503)} = \boxed{286.288\ W/m^2 \cdot k}$$

2.



$$\dot{Q} = \dot{m}c_p(T_{h,in} - T_{h,out}) = 2(2200)(150-40) = 484\ kW$$

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

$$T_{c,out} = \frac{\dot{Q}}{\dot{m}c_p} + T_{c,in} = \frac{48.4(1000)}{1.5(4180)} + 22 = 29.72\ ^\circ C$$

$$\Delta T_1 = 150 - 29.72 = 120.28\ ^\circ C$$

$$\Delta T_2 = 40 - 22 = 18\ ^\circ C$$

$$\Delta T_{ln} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = 64.78\ ^\circ C$$

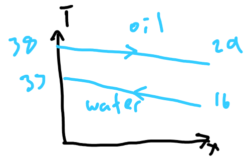
$$A_s = \pi DL = \pi(.025)(6) = .471239$$

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

$$U = \frac{\dot{Q}}{A_s\Delta T_{ln}} = \frac{48.4(1000)}{.471239(64.78)}$$

$$\boxed{U = 1585.54\ W/m^2 \cdot k}$$

3.



$\dot{m}_{oil} = .19\ m^3/min = .003167\ m^3/s$

$\dot{Q} = \dot{m} C_p (T_{in} - T_{out})_{oil} = .003167(1.93)(38-29)$

$= .055$

$A = 20 m^2$

$\dot{Q} = U A F \Delta T_{lm}$

$$U = \frac{\dot{Q}}{A F \Delta T_{lm}}$$

$\Delta T_1 = 38 - 33 = 5$

$\Delta T_2 = 29 - 16 = 13$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{5-13}{\ln(5/13)} = 8.372$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{33-16}{38-16} = .773 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{38-29}{33-16} = 235 \end{aligned}\right\} F = .94$$

$$U = \frac{5500}{20(.94)(8.372)} = \boxed{.349 \quad W/m^2 \cdot K}$$

Thermal Radiation

Thursday, April 11, 2024 11:53 AM

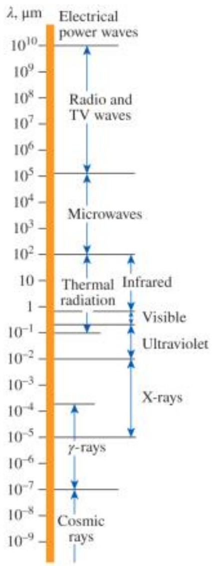


FIGURE 12–3 The electromagnetic wave spectrum.

TABLE 12–1

The wavelength ranges of different colors

Color	Wavelength Band
Violet	0.40 – 0.44 μm
Blue	0.44 – 0.49 μm
Green	0.49 – 0.54 μm
Yellow	0.54 – 0.60 μm
Orange	0.60 – 0.67 μm
Red	0.63 – 0.76 μm

$$E_{b,\lambda}(\lambda, T) = \frac{C_1}{\lambda^5\left[e^{C_2/\lambda T} - 1\right]}$$

Planck's law

↑ blackbody

↑ max plank in 1901

where

$$C_1 = 2\pi h c_0^2 = 3.74177 \times 10^8 \text{ W}\cdot\mu\text{m}^4/\text{m}^2$$
$$C_2 = hc_0/k = 1.43878 \times 10^4 \ \mu\text{m}\cdot\text{K}$$

Also, T is the absolute temperature of the surface, λ is the wavelength of the radiation emitted, and $k = 1.38065 \times 10^{-23}$ J/K is *Boltzmann's constant*. This relation is valid for a surface in a *vacuum* or a *gas*. For other media, it needs to be modified by replacing C_1 with C_1/n^2, where n is the index of refraction of the medium. Note that the term *spectral* indicates dependence on wavelength.

It is left as an exercise to show that integration of the *spectral* blackbody emissive power $E_{b\lambda}$ over the entire wavelength spectrum gives the *total* blackbody emissive power E_b:

$$E_b(T) = \int_0^\infty E_{b\lambda}(\lambda, T)\,d\lambda = \sigma T^4 \quad (\text{W/m}^2) \qquad \textbf{(12–6)}$$

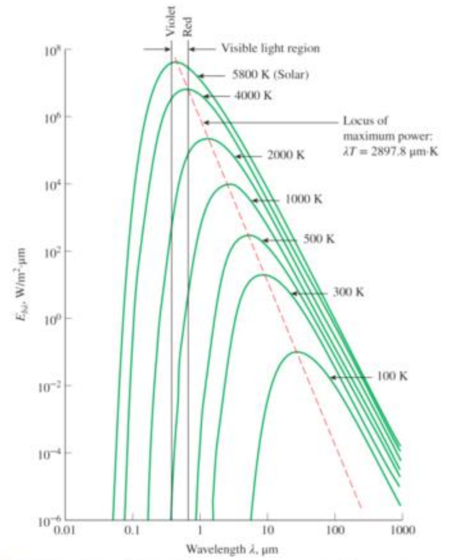


FIGURE 12–9 The variation of the blackbody emissive power with wavelength for several temperature

As the temperature increases, the peak of the curve in Fig. 12–9 shifts toward shorter wavelengths. The wavelength at which the peak occurs for a specified temperature is given by **Wien's displacement law** as

$$(\lambda T)_{\text{max power}} = 2897.8 \ \mu\text{m}\cdot\text{K}$$

Ex:

→ $\lambda_{max} = ?$

$$\lambda_{max} T = 2897.8 \ \mu m K$$
$$T_{sun} = 5780 \text{ K}$$

$$\lambda_{max,surface} = .50 \ \mu m$$

visible light

EXAMPLE 12–2 Light Emitted by the Sun and by a Lightbulb

Charge-coupled device (CCD) image sensors, which are common in modern digital cameras, respond differently to light sources with different spectral distributions. Daylight and incandescent light sources may be approximated as blackbodies at the effective surface temperatures of 5800 K and 2800 K, respectively. Determine the fraction of radiation emitted within the visible spectrum wavelengths, from 0.40 μm (violet) to 0.76 μm (red), for each of the lighting sources.

SOLUTION For specified blackbody temperatures, the fraction of visible radiation emitted by the sun and the filament of an incandescent lightbulb are to be determined.

Assumptions The sun and the incandescent light filament behave as blackbodies.

Analysis The visible range of the electromagnetic spectrum extends from $\lambda_1 = 0.40\ \mu m$ to $\lambda_2 = 0.76\ \mu m$. For the sun at $T = 5800\ K$, the blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined from Table 12-2 to be (Fig. 12-15)

$$\lambda_1 T = (0.40\ \mu m)(5800\ K) = 2320\ \mu m\ .K \quad \rightarrow f_{\lambda_1,\ daylight} = 0.124509$$
$$\lambda_2 T = (0.76\ \mu m)(5800\ K) = 4408\ \mu m\ .K \quad \rightarrow f_{\lambda_2,\ daylight} = 0.550015$$

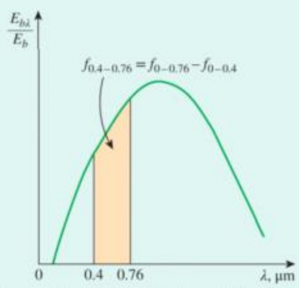


FIGURE 12–15 Graphical representation of the fraction of radiation emitted in the visible ra

Then the fraction of visible radiation emitted by the sun becomes

$$f_{\lambda_1-\lambda_2,\ daylight} = 0.550015 - 0.124509 = 0.426 \text{ or } 42.6 \text{ percent}$$

For an incandescent lightbulb at $T = 2800\ K$, the blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined from Table 12-2 to be

$$\lambda_1 T = (0.40\ \mu m)(2800\ K) = 1120\ \mu m\ .K \rightarrow f_{\lambda_1,\ incandescent} = 0.001409$$
$$\lambda_2 T = (0.76\ \mu m)(5800\ K) = 2128\ \mu m\ .K \rightarrow f_{\lambda_2,\ incandescent} = 0.088590$$

Then the fraction of radiation the lightbulb emits in the visible range becomes

$$f_{\lambda_1-\lambda_2,\ incandescent} = 0.088590 - 0.001409 = 0.087 \text{ or } 8.7 \text{ percent}$$

Discussion Note that almost half of the radiation emitted by the sun is in the visible range, and thus the sun is a very efficient light source. But less than 10 percent of the radiation emitted by the incandescent lightbulb is in the form of visible light, and thus incandescent lightbulbs are inefficient as light sources. Consequently, they are being replaced by the highly efficient fluorescent and LED light sources.

TABLE 12–2

Blackbody radiation functions f_λ

λT, μm·K	f_λ	λT, μm·K	f_λ
200	0.000000	6200	0.754140
400	0.000000	6400	0.769234
600	0.000000	6600	0.783199
800	0.000016	6800	0.796129
1000	0.000321	7000	0.808109
1200	0.002134	7200	0.819217
1400	0.007790	7400	0.829527
1600	0.019718	7600	0.839102
1800	0.039341	7800	0.848005
2000	0.066728	8000	0.856288
2200	0.100888	8500	0.874608
2400	0.140256	9000	0.890029
2600	0.183120	9500	0.903085
2800	0.227897	10,000	0.914199
3000	0.273232	10,500	0.923710
3200	0.318102	11,000	0.931890
3400	0.361735	11,500	0.939959
3600	0.403607	12,000	0.945098
3800	0.443382	13,000	0.955139
4000	0.480877	14,000	0.962898
4200	0.516014	15,000	0.969981
4400	0.548796	16,000	0.973814
4600	0.579280	18,000	0.980860
4800	0.607559	20,000	0.985602
5000	0.633747	25,000	0.992215
5200	0.658970	30,000	0.995340
5400	0.680360	40,000	0.997967
5600	0.701046	50,000	0.998953
5800	0.720158	75,000	0.999713
6000	0.737818	100,000	0.999905

$$\varepsilon_\lambda = \begin{cases} \varepsilon_1 = \text{constant}, & 0 \le \lambda < \lambda_1 \\ \varepsilon_2 = \text{constant}, & \lambda_1 \le \lambda < \lambda_2 \\ \varepsilon_3 = \text{constant}, & \lambda_2 \le \lambda < \infty \end{cases} \quad \textbf{(12–35)}$$

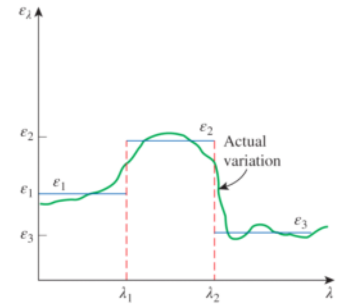


FIGURE 12–24 Approximating the actual variation of emissivity with wavelength by a step function.

Then the average emissivity can be determined from **Eq. 12-34** by breaking the integral into three parts and utilizing the definition of the blackbody radiation function as

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$$\begin{aligned} \varepsilon(T) &= \frac{\varepsilon_1 \int_0^{\lambda_1} E_{b\lambda}\, d\lambda}{E_b} + \frac{\varepsilon_2 \int_{\lambda_1}^{\lambda_2} E_{b\lambda}\, d\lambda}{E_b} + \frac{\varepsilon_3 \int_{\lambda_2}^{\infty} E_{b\lambda}\, d\lambda}{E_b} \\ &= \varepsilon_1 f_{0-\lambda_1}(T) + \varepsilon_2 f_{\lambda_1-\lambda_2}(T) + \varepsilon_3 f_{\lambda_2-\infty}(T) \end{aligned} \quad \textbf{(12–36)}$$

Exam 2 Study guide

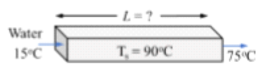
Tuesday, April 16, 2024 11:29 AM

Exam+2+-+ Study+gui...

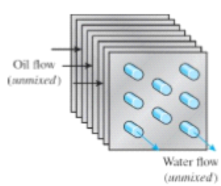
ME 3525 3 Exam 2 Name:

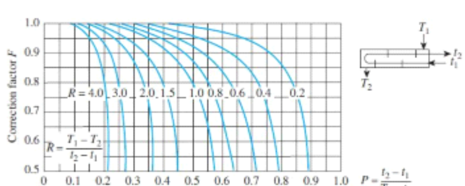
1. Air at a temperature of 20°C is blowing at a speed of 36 km/h across a long electrical line 6 mm in diameter. If an electric current of 60 A is passed through the electrical line and the electrical resistance of the electrical line is 0.002 ohm per meter length, (a) prove that heat flux at the surface of the wire at steady state is 382 W/m²; (b) determine the surface temperature of the wire. (**22.8 °C**)

2. A smooth **rectangular** tube whose surface temperature is maintained at 90°C is used to heat a **water** flow from 15°C (at the entrance) to 75°C (at the outlet). If the cross section of the rectangular tube is 5 cm by 3 cm and the mass flow rate of water in the tube is 0.3 kg/s. Determine the length of tube. (**9.8 m**)

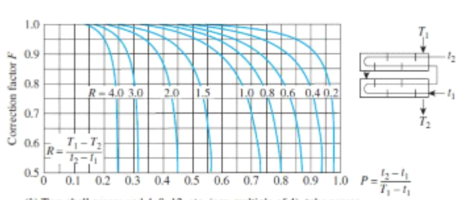


3. Water and oil exchange heat in a single-pass cross-flow heat exchanger as shown in the following figure. Water enters at 18°C and exits at 35°C, and oil (c_p = 1.93 kJ/kg·K and ρ = 870 kg/m³) flows at 0.19 m³/min enters at 40°C and exits at 31°C. If the surface area of the heat exchanger is 19 m², determine the value of the overall heat transfer coefficient. (**354 W/m²K**)

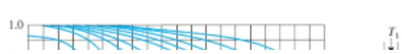




(a) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes



(b) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes



1.

$\dot{q}_s = h(T_s - T_\infty)$

$q_s = \frac{\dot{Q}}{A_s} = \frac{\dot{W}_e}{A_s} = \frac{I^2R_e}{A_s} = \frac{60^2(.002)}{\pi(1)(.006)} = 381.972 \text{ W/m}^2$

$T_s = T_\infty + \frac{q_s}{h}$

$T_f = (T_s + T_\infty)/2$

Guess $T_s = 20$

$T_f = \frac{20+20}{2} = 20$

$k = .02514$ $\nu = 1.516\times10^{-5}$ $Pr = .7309$

$Re = \frac{VD}{\nu} = \frac{36 \quad (.006)}{1.516\times10^{-5}} = 1.4248\times10^4 < 5\times10^5 \Rightarrow$ laminar

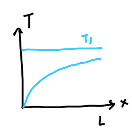
$Nu = \frac{hD}{k} = .453\, Re^{\frac{1}{2}} Pr^{\frac{1}{3}} = .453(1.4248\times10^4)^{\frac{1}{2}}(.7309)^{\frac{1}{3}} = 48.4842$

$h = \frac{Nu\,k}{D} = \frac{48.4842(.02514)}{.006} = 203.149$

$T_s = T_\infty + \frac{q_s}{h} = 20 + \frac{381.972}{203.149}$

$T_s = 21.88$ close enough, no need for second iteration

2.



1. $\Delta T_e = \Delta T_i e^{hA_s/\dot{m}c_p}$
2. $\dot{Q} = \dot{m}c_p(T_e - T_i)$
3. $\dot{Q} = hA_s\Delta T_{lm}$

$T_b = (T_i + T_e)/2 = (75+15)/2 = 45$

$k = .637$ $\rho = 990.1$

$Pr = 3.91$ $c_p = 4180$ $\nu = \frac{\mu}{\rho}$

$\nu = \frac{596\times10^{-3}}{990.1} = 6.02\times10^{-7}$

$D_h = \frac{4A_c}{P} = \frac{4(.05\times.03)}{2(.05)+2(.03)} = .0375 \text{ m}$

$\dot{m} = \rho V A_c \Rightarrow V = \frac{\dot{m}}{A_c\rho} = \frac{.3}{990.1(.05)(.03)} = .202$

$Re = \frac{VD_h}{\nu} = \frac{.0375(.202)}{6.02\times10^{-7}} = 12583.9 > 10{,}000$ Turbulent

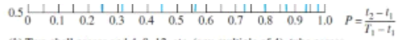
$L_t = 10D_h = 10(.0375) = .375$

assume fully developed

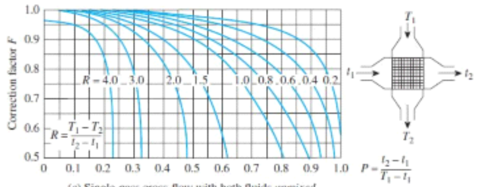
$Nu = \frac{hD_h}{k} = .023\, Re^{.8} Pr^{.4} = 75.5876$

$h = \frac{Nu\,k}{D_h} = \frac{75.5876(.637)}{.0375} = 1283.98$

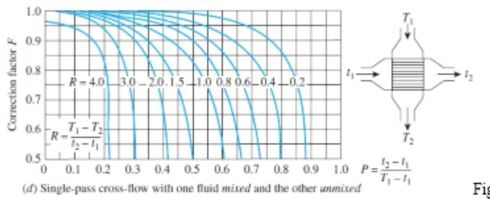
$\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{hA_s}{\dot{m}c_p} = \frac{-h\pi DL}{\dot{m}c_p} \Rightarrow L = -\ln\frac{T_s - T_e}{T_s - T_i}\left(\frac{\dot{m}c_p}{h\pi D}\right)$



(*b*) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes



(*c*) Single-pass cross-flow with both fluids *unmixed*



(*d*) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

Fig. 11-19

2

TABLE A-9

Properties of saturated water

Temp. T, °C	Saturation Pressure P_{sat}, kPa	Density ρ, kg/m³ Liquid	Density Vapor	Enthalpy of Vaporization h_{fg}, kJ/kg	Specific Heat c_p, J/kg·K Liquid	Specific Heat Vapor	Thermal Conductivity k, W/m·K Liquid	Thermal Conductivity Vapor	Dynamic Viscosity μ, kg/m·s Liquid	Dynamic Viscosity Vapor	Prandtl Number Pr Liquid	Prandtl Number Vapor	Volume Expansion Coefficient β, 1/K Liquid
0.01	0.6113	999.8	0.0048	2501	4217	1854	0.561	0.0171	1.792×10^{-3}	0.922×10^{-5}	13.5	1.00	-0.068×10^{-3}
5	0.8721	999.9	0.0068	2490	4205	1857	0.571	0.0173	1.519×10^{-3}	0.934×10^{-5}	11.2	1.00	0.015×10^{-3}
10	1.2276	999.7	0.0094	2478	4194	1862	0.580	0.0176	1.307×10^{-3}	0.946×10^{-5}	9.45	1.00	0.733×10^{-3}
15	1.7051	999.1	0.0128	2466	4185	1863	0.589	0.0179	1.138×10^{-3}	0.959×10^{-5}	8.09	1.00	0.138×10^{-3}
20	2.339	998.0	0.0173	2454	4182	1867	0.598	0.0182	1.002×10^{-3}	0.973×10^{-5}	7.01	1.00	0.195×10^{-3}
25	3.169	997.0	0.0231	2442	4180	1870	0.607	0.0186	0.891×10^{-3}	0.987×10^{-5}	6.14	1.00	0.247×10^{-3}
30	4.246	996.0	0.0304	2431	4178	1875	0.615	0.0189	0.798×10^{-3}	1.001×10^{-5}	5.42	1.00	0.294×10^{-3}
35	5.628	994.0	0.0397	2419	4178	1880	0.623	0.0192	0.720×10^{-3}	1.016×10^{-5}	4.83	1.00	0.337×10^{-3}
40	7.384	992.1	0.0512	2407	4179	1885	0.631	0.0196	0.653×10^{-3}	1.031×10^{-5}	4.32	1.00	0.377×10^{-3}
45	9.593	990.1	0.0655	2395	4180	1892	0.637	0.0200	0.596×10^{-3}	1.046×10^{-5}	3.91	1.00	0.415×10^{-3}
50	12.35	988.1	0.0831	2383	4181	1900	0.644	0.0204	0.547×10^{-3}	1.062×10^{-5}	3.55	1.00	0.451×10^{-3}
55	15.76	985.2	0.1045	2371	4183	1908	0.649	0.0208	0.504×10^{-3}	1.077×10^{-5}	3.25	1.00	0.484×10^{-3}
60	19.94	983.3	0.1304	2359	4185	1916	0.654	0.0212	0.467×10^{-3}	1.093×10^{-5}	2.99	1.00	0.517×10^{-3}
65	25.03	980.4	0.1614	2346	4187	1926	0.659	0.0216	0.433×10^{-3}	1.110×10^{-5}	2.75	1.00	0.548×10^{-3}
70	31.19	977.5	0.1983	2334	4190	1936	0.663	0.0221	0.404×10^{-3}	1.126×10^{-5}	2.55	1.00	0.578×10^{-3}
75	38.58	974.7	0.2421	2321	4193	1948	0.667	0.0225	0.378×10^{-3}	1.142×10^{-5}	2.38	1.00	0.607×10^{-3}
80	47.39	971.8	0.2935	2309	4197	1962	0.670	0.0230	0.355×10^{-3}	1.159×10^{-5}	2.22	1.00	0.653×10^{-3}
85	57.83	968.1	0.3536	2296	4201	1977	0.673	0.0235	0.333×10^{-3}	1.176×10^{-5}	2.08	1.00	0.670×10^{-3}
90	70.14	965.3	0.4235	2283	4206	1993	0.675	0.0240	0.315×10^{-3}	1.193×10^{-5}	1.96	1.00	0.702×10^{-3}
95	84.55	961.5	0.5045	2270	4212	2010	0.677	0.0246	0.297×10^{-3}	1.210×10^{-5}	1.85	1.00	0.716×10^{-3}
100	101.33	957.9	0.5978	2257	4217	2029	0.679	0.0251	0.282×10^{-3}	1.227×10^{-5}	1.75	1.00	0.750×10^{-3}
110	143.27	950.6	0.8263	2230	4229	2071	0.682	0.0262	0.255×10^{-3}	1.261×10^{-5}	1.58	1.00	0.798×10^{-3}
120	198.53	943.4	1.121	2203	4244	2120	0.683	0.0275	0.232×10^{-3}	1.296×10^{-5}	1.44	1.00	0.858×10^{-3}
130	270.1	934.6	1.496	2174	4263	2177	0.684	0.0288	0.213×10^{-3}	1.330×10^{-5}	1.33	1.01	0.913×10^{-3}
140	361.3	921.7	1.965	2145	4286	2244	0.683	0.0301	0.197×10^{-3}	1.365×10^{-5}	1.24	1.02	0.970×10^{-3}
150	475.8	916.6	2.546	2114	4311	2314	0.682	0.0316	0.183×10^{-3}	1.399×10^{-5}	1.16	1.02	1.025×10^{-3}
160	617.8	907.4	3.256	2083	4340	2420	0.680	0.0331	0.170×10^{-3}	1.434×10^{-5}	1.09	1.05	1.145×10^{-3}
170	791.7	897.7	4.119	2050	4370	2490	0.677	0.0347	0.160×10^{-3}	1.468×10^{-5}	1.03	1.05	1.178×10^{-3}
180	1,002.1	887.3	5.153	2015	4410	2590	0.673	0.0364	0.150×10^{-3}	1.502×10^{-5}	0.983	1.07	1.210×10^{-3}
190	1,254.4	876.4	6.388	1979	4460	2710	0.669	0.0382	0.142×10^{-3}	1.537×10^{-5}	0.947	1.09	1.280×10^{-3}
200	1,553.8	864.3	7.852	1941	4500	2840	0.663	0.0401	0.134×10^{-3}	1.571×10^{-5}	0.910	1.11	1.350×10^{-3}
220	2,318	840.3	11.60	1859	4610	3110	0.650	0.0442	0.122×10^{-3}	1.641×10^{-5}	0.865	1.15	1.520×10^{-3}
240	3,344	813.7	16.73	1767	4760	3520	0.632	0.0487	0.111×10^{-3}	1.712×10^{-5}	0.836	1.24	1.720×10^{-3}
260	4,688	783.7	23.69	1663	4970	4070	0.609	0.0540	0.102×10^{-3}	1.788×10^{-5}	0.832	1.35	2.000×10^{-3}
280	6,412	750.8	33.15	1544	5280	4835	0.581	0.0605	0.094×10^{-3}	1.870×10^{-5}	0.854	1.49	2.380×10^{-3}
300	8,581	713.8	46.15	1405	5750	5980	0.548	0.0695	0.086×10^{-3}	1.965×10^{-5}	0.902	1.69	2.950×10^{-3}
320	11,274	667.1	64.57	1239	6540	7900	0.509	0.0836	0.078×10^{-3}	2.084×10^{-5}	1.00	1.97	
340	14,586	610.5	92.62	1028	8240	11,870	0.469	0.110	0.070×10^{-3}	2.255×10^{-5}	1.23	2.43	
360	18,651	528.3	144.0	720	14,690	25,800	0.427	0.178	0.060×10^{-3}	2.571×10^{-5}	2.06	3.73	
374.14	22,090	317.0	317.0	0	—	—	—	—	0.043×10^{-3}	4.313×10^{-5}			

Note 1: Kinematic viscosity ν and thermal diffusivity α can be calculated from their definitions, $\nu = \mu/\rho$ and $\alpha = k/\rho c_p = \nu/\mathrm{Pr}$. The temperatures 0.01°C, 100°C, and 374.14°C are the triple-, boiling-, and critical-point temperatures of water, respectively. The properties listed above (except the vapor density) can be used at any pressure with negligible error except at temperatures near the critical-point value.

Note 2: The unit kJ/kg·°C for specific heat is equivalent to kJ/kg·K, and the unit W/m·°C for thermal conductivity is equivalent to W/m·K.

Source: Viscosity and thermal conductivity data are from J. V. Sengers and J. T. R. Watson, *Journal of Physical and Chemical Reference Data* 15 (1986), pp. 1291–1322. Other data are obtained from various sources or calculated.

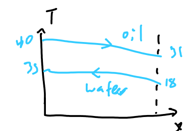
3

$$\ln \frac{T_s - T_e}{T_s - T_i} = -\frac{hA_s}{\dot m c_p} = \frac{-h\pi D L}{\dot m C_p} \Rightarrow L = -\ln \frac{T_s - T_e}{T_s - T_i}\left(\frac{\dot m C_p}{h\pi D}\right)$$

$$L = -\ln \frac{90-75}{90-15}\left(\frac{.3(4180)}{1283\ 98\ (.0775)\pi}\right)$$

$L =$

3.



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

@ $T_b = (T_{h,in} + T_{h,out})/2 = (40+31)/2 = 35.5$ $C_p = 193$

$\dot Q = U A_s \Delta T_{lm} F$

$U =$

$F(P,R)$ Figure 11-1a

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = 8.972$$

$\Delta T_1 = 40 - 35 = 5$
$\Delta T_2 = 31 - 18 = 13$

TABLE A–15

Properties of air at 1 atm pressure

Temp. T, °C	Density ρ, kg/m^3	Specific Heat c_p, J/kg·K	Thermal Conductivity k, W/m·K	Thermal Diffusivity α, m^2/s	Dynamic Viscosity μ, kg/m·s	Kinematic Viscosity ν, m^2/s	Prandtl Number Pr
−150	2.866	983	0.01171	4.158×10^{-6}	8.636×10^{-6}	3.013×10^{-6}	0.7246
−100	2.038	966	0.01582	8.036×10^{-6}	1.189×10^{-5}	5.837×10^{-6}	0.7263
−50	1.582	999	0.01979	1.252×10^{-5}	1.474×10^{-5}	9.319×10^{-6}	0.7440
−40	1.514	1002	0.02057	1.356×10^{-5}	1.527×10^{-5}	1.008×10^{-5}	0.7436
−30	1.451	1004	0.02134	1.465×10^{-5}	1.579×10^{-5}	1.087×10^{-5}	0.7425
−20	1.394	1005	0.02211	1.578×10^{-5}	1.630×10^{-5}	1.169×10^{-5}	0.7408
−10	1.341	1006	0.02288	1.696×10^{-5}	1.680×10^{-5}	1.252×10^{-5}	0.7387
0	1.292	1006	0.02364	1.818×10^{-5}	1.729×10^{-5}	1.338×10^{-5}	0.7362
5	1.269	1006	0.02401	1.880×10^{-5}	1.754×10^{-5}	1.382×10^{-5}	0.7350
10	1.246	1006	0.02439	1.944×10^{-5}	1.778×10^{-5}	1.426×10^{-5}	0.7336
15	1.225	1007	0.02476	2.009×10^{-5}	1.802×10^{-5}	1.470×10^{-5}	0.7323
20	1.204	1007	0.02514	2.074×10^{-5}	1.825×10^{-5}	1.516×10^{-5}	0.7309
25	1.184	1007	0.02551	2.141×10^{-5}	1.849×10^{-5}	1.562×10^{-5}	0.7296
30	1.164	1007	0.02588	2.208×10^{-5}	1.872×10^{-5}	1.608×10^{-5}	0.7282
35	1.145	1007	0.02625	2.277×10^{-5}	1.895×10^{-5}	1.655×10^{-5}	0.7268
40	1.127	1007	0.02662	2.346×10^{-5}	1.918×10^{-5}	1.702×10^{-5}	0.7255
45	1.109	1007	0.02699	2.416×10^{-5}	1.941×10^{-5}	1.750×10^{-5}	0.7241
50	1.092	1007	0.02735	2.487×10^{-5}	1.963×10^{-5}	1.798×10^{-5}	0.7228
60	1.059	1007	0.02808	2.632×10^{-5}	2.008×10^{-5}	1.896×10^{-5}	0.7202
70	1.028	1007	0.02881	2.780×10^{-5}	2.052×10^{-5}	1.995×10^{-5}	0.7177
80	0.9994	1008	0.02953	2.931×10^{-5}	2.096×10^{-5}	2.097×10^{-5}	0.7154
90	0.9718	1008	0.03024	3.086×10^{-5}	2.139×10^{-5}	2.201×10^{-5}	0.7132
100	0.9458	1009	0.03095	3.243×10^{-5}	2.181×10^{-5}	2.306×10^{-5}	0.7111
120	0.8977	1011	0.03235	3.565×10^{-5}	2.264×10^{-5}	2.522×10^{-5}	0.7073
140	0.8542	1013	0.03374	3.898×10^{-5}	2.345×10^{-5}	2.745×10^{-5}	0.7041
160	0.8148	1016	0.03511	4.241×10^{-5}	2.420×10^{-5}	2.975×10^{-5}	0.7014
180	0.7788	1019	0.03646	4.593×10^{-5}	2.504×10^{-5}	3.212×10^{-5}	0.6992
200	0.7459	1023	0.03779	4.954×10^{-5}	2.577×10^{-5}	3.455×10^{-5}	0.6974
250	0.6746	1033	0.04104	5.890×10^{-5}	2.760×10^{-5}	4.091×10^{-5}	0.6946
300	0.6158	1044	0.04418	6.871×10^{-5}	2.934×10^{-5}	4.765×10^{-5}	0.6935
350	0.5664	1056	0.04721	7.892×10^{-5}	3.101×10^{-5}	5.475×10^{-5}	0.6937
400	0.5243	1069	0.05015	8.951×10^{-5}	3.261×10^{-5}	6.219×10^{-5}	0.6948
450	0.4880	1081	0.05298	1.004×10^{-4}	3.415×10^{-5}	6.997×10^{-5}	0.6965
500	0.4565	1093	0.05572	1.117×10^{-4}	3.563×10^{-5}	7.806×10^{-5}	0.6986
600	0.4042	1115	0.06093	1.352×10^{-4}	3.846×10^{-5}	9.515×10^{-5}	0.7037
700	0.3627	1135	0.06581	1.598×10^{-4}	4.111×10^{-5}	1.133×10^{-4}	0.7092
800	0.3289	1153	0.07037	1.855×10^{-4}	4.362×10^{-5}	1.326×10^{-4}	0.7149
900	0.3008	1169	0.07465	2.122×10^{-4}	4.600×10^{-5}	1.529×10^{-4}	0.7206
1000	0.2772	1184	0.07868	2.398×10^{-4}	4.826×10^{-5}	1.741×10^{-4}	0.7260
1500	0.1990	1234	0.09599	3.908×10^{-4}	5.817×10^{-5}	2.922×10^{-4}	0.7478
2000	0.1553	1264	0.11113	5.664×10^{-4}	6.630×10^{-5}	4.270×10^{-4}	0.7539

Note: For ideal gases, the properties c_p, k, μ, and Pr are independent of pressure. The properties ρ, ν, and α at a pressure P (in atm) other than 1 atm are determined by multiplying the values of ρ at the given temperature by P and by dividing ν and α by P.

Source: Data generated from the EES software developed by S. A. Klein and F. L. Alvarado. Original sources: Keenan, Chao, Keyes, Gas Tables, Wiley, 1984; and Thermophysical Properties of Matter. Vol. 3: Thermal Conductivity, Y. S. Touloukian, P. E. Liley, S. C. Saxena, Vol. 11: Viscosity, Y. S. Touloukian, S. C. Saxena, and P. Hestermans, IFI/Plenum, NY, 1970, ISBN 0-306067020-8.

Exam 2

Thursday, April 18, 2024 12:07 PM

ME 3525 Exam 2 Name: Easton Ingram

1. A long, cylindrical, electrical heating element of diameter d = 12 mm is installed in a duct for which air moves in cross flow over the heater at a temperature and velocity of 30 °C and 8 m/s, respectively. If electrical energy is being dissipated at a rate of 1000 W per unit length of the heater (i.e. 1000 W/m), (a) show that the heat flux at the surface of the heater in steady state is 26.5 kW/m^2; and (b) neglecting radiation, estimate the steady-state surface temperature. **(40 pts)**

a. $\dot{q} = \frac{\dot{Q}}{A_s} = \frac{\dot{W}_e}{\pi D} = \frac{1000 \text{ W/m}}{\pi (.012) \text{ m}} = 26.5 \text{ kW/m}^2$

b. $\dot{q}_{conv} = h(T_s - T_\infty)$

$T_f = (T_\infty + T_s)/2$

Guess $T_s = 40\,°C$

$T_f = (30+40)/2 = 35$

$Re = \frac{VD}{\nu} = \frac{8(.012)}{1.657\times10^{-5}}$

$Re = 5800.6 < 5\times10^5$, laminar

Table A-15 @ 30°C

$k = .02625$ W/m·K

$Pr = .728$

$\nu = 1.657\times10^{-5}$ m²/s

Flow over cylinder:

$Re\,Pr = 4222.7$

$$Nu_{cyl} = \frac{hD}{k} = .3 + \frac{.62\,Re^{1/2}\,Pr^{1/3}}{[1+(.4/Pr)^{2/3}]^{1/4}}\left[1+\left(\frac{Re}{282000}\right)^{5/8}\right]^{4/5}$$

$Nu_{cyl} = 40.2761$

$h = \frac{Nu_{cyl}\,k}{D} = \frac{40.2761(.02625)}{.012} = 88.10$ W/m²·K

$T_s = \frac{\dot{q}_{conv}}{h} + T_\infty = \frac{26.5 \text{ kW/m}^2}{88.10 \text{ W/m}^2\text{K}} + 30 = 330.795\,°C$

Iterate with $T_s = 330\,°C$

$T_f = (30+330)/2 = 180\,°C$

$k = .03646$

$Pr = .6992$

$\nu = 3.212\times10^{-5}$

$Re = 2988.79$

$Nu = 27.913$

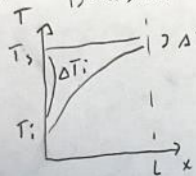
$h = 84.809$

$T_s = \frac{26.5 \text{ kW/m}^2}{84.809 \text{ W/m}^2\text{K}} + 30 = \boxed{342.460\,°C}$

2 Iterations, moving on

2. Glycerin flows through a tube with a mass flow rate of 0.5 kg/s. The flow of glycerin enters the tube at 20°C and exits the tube at 40°C. If the tube surface temperature is constant, and the length and diameter of the tube are 10 m and 2.5 cm, respectively, determine (a) the total rate of heat transfer for the tube and (b) the surface temperature of the tube. **(35 pts)**

a. T_s = constant



$T_b = (20+40)/2 = 30°C$

Table A-9 @ 30°C Glycerin

$C_p = 2447\ J/kg \cdot K$ $\quad P_r = 5631$

$\rho = 1258\ kg/m^3$

$\nu = 5.232 \times 10^{-4}\ m^2/s$ $\quad k = .286\ W/m \cdot K$

$\dot{Q} = \dot{m} C_p (T_e - T_i) = .5(2447)(40-20) = \boxed{24.47\ kW}$

b. $Re = \frac{VD}{\nu} = \frac{.2026(.025)}{5.232 \times 10^{-4}}$ $\quad \dot{m} = \rho V A_c \Rightarrow V = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho \pi D^2} = \frac{.5}{1258\pi(.025)^2}$

$V = .2026\ m/s^2$

$Re = 9.672 < 2300$, laminar

$L_{T_{laminar}} = .5 Re P_r D = .5(9.672)(5631)(.025) = 680 \gg L$, entry region

Laminar flow, circular tube, T_s = constant, entry region: $\quad D = .025\ m$, $L = 10\ m$

$$Nu = 3.66 + \frac{.065\ (D/L) Re P_r}{1 + .04[(D/L) Re P_r]^{2/3}} = 7.9591$$

$$Nu = \frac{hD}{k} \Rightarrow h = \frac{k\,Nu}{D} = \frac{.286(7.9591)}{.025} = 91.05\ W/m^2 \cdot k$$

$\Delta T_e = \Delta T_i\, e^{-hA_s/\dot{m}C_p}$ $\qquad A_s = \pi D L = .7854\ m^2$

$T_s - T_e = e^{-\frac{hA_s}{\dot{m}C_p}}(T_s - T_i)$

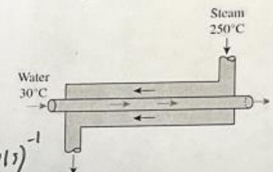
$$T_s - e^{-\frac{hA_s}{\dot{m}C_p}} T_s = -e^{-\frac{hA_s}{\dot{m}C_p}}(T_i) + T_e$$

$$T_s = \frac{T_e - e^{-\frac{hA_s}{\dot{m}C_p}} T_i}{1 - e^{-hA_s/\dot{m}C_p}} = \frac{40 - e^{-91.05(.7854)/.5(2447)}(20)}{1 - e^{-91.05(.7854)/.5(2447)}}$$

2

$\boxed{T_s = 372.276\ °C}$ $\quad$ Seems quite large

3. In an industrial facility a counter flow double pipe heat exchanger uses superheated steam at a temperature of 250°C to heat feed water at 30°C. The superheated steam experiences a **temperature drop** of 70°C as it exits the heat exchanger. The water to be heated flows through the heat exchanger tube of negligible thickness at a constant rate of 3.47 kg/s. The convective heat transfer coefficient on the superheated steam and water side is 850 W/m²·K and 1250 W/m²·K, respectively. To account for the fouling due to chemical impurities that might be present in the feed water, one can use $R_{f,i} = 0.0015$ m²·K/W. Neglecting the fouling resistance on the steam side, determine the heat exchanger area, A_s required to maintain the exit temperature of the water to a minimum of 70°C? (**25 pts**)

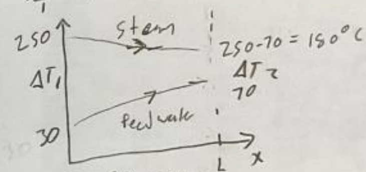


Thin walled:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + R_{f,i}$$

$$U = \left(\frac{1}{h_i} + \frac{1}{h_o} + R_f\right)^{-1} \left(\frac{1}{850} + \frac{1}{1250} + .0015\right)^{-1}$$

$U = 470.263$



$$\dot{Q} = U A_s \, \Delta T_{ln}$$

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

$\Delta T_1 = 250 - 30 = 220$

$\Delta T_2 = 180 - 70 = 110$

$$\Delta T_{ln} = \frac{220 - 110}{\ln \frac{220}{110}} = 158.696$$

$\dot{m}_c = 3.47$ kg/s

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

$T_{b,c} = (30 + 70)/2 = 50$ °C

C_p @ 50°C

Table A-9:

$C_p = 4181$ J/kg·k

$\dot{Q} = 3.47(4181)(70-30)$

$\dot{Q} = 580.3$ kW

$$A_s = \frac{\dot{Q}}{U \Delta T_{lm}}$$

$$A_s = \frac{580.3 \text{ kW}(1000)}{470.263(158.696)}$$

$$\boxed{A_s = 7.7761 \text{ m}^2}$$

3

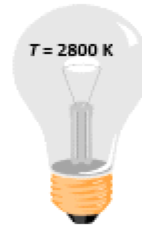
HW9

Tuesday, April 23, 2024 11:02 AM

HW9

ME 3525 HW9

1. The temperature of the filament of an incandescent lightbulb is 2800 K. Treating the filament as a blackbody, determine the fraction of the radiant energy emitted by the filament that falls in the visible range. Also, determine the wavelength at which the emission of radiation from the filament peaks.



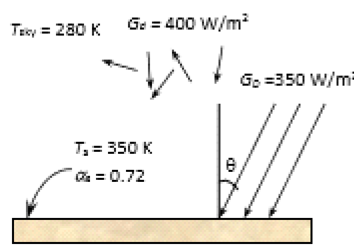
2. An incandescent lightbulb is desired to emit at least 15 percent of its energy at wavelengths shorter than 0.76 μm. Determine the minimum temperature to which the filament of the lightbulb must be heated if the filament behaves as a blackbody.

3. The spectral emissivity function of an opaque surface at 1000 K is approximated as

$$\varepsilon_\lambda = \begin{cases} \varepsilon_1 = 0.4, & 0 \le \lambda < 3\ \mu\text{m} \\ \varepsilon_2 = 0.7, & 3\ \mu\text{m} \le \lambda < 6\ \mu\text{m} \\ \varepsilon_3 = 0.3, & 6\ \mu\text{m} \le \lambda < \infty \end{cases}$$

Determine the average emissivity of the surface and the rate of radiation emission from the surface, in W/m^2.

4. A surface has an absorptivity of $\alpha_s = 0.72$ for solar radiation and an emissivity of $\varepsilon = 0.6$ at room temperature. The surface temperature is observed to be 350 K when the direct and the diffuse components of solar radiation are $G_D = 350$ and $G_d = 400$ W/m^2, respectively, and the direct radiation makes a 30° angle with the normal of the surface. Taking the effective sky temperature to be 280 K, determine the net rate of radiation heat transfer to the surface at that time in unit W/m^2.



1. a.

$\lambda_1 = .4\ \text{um}$

$\lambda_2 = .76\ \text{um}$

$\lambda_1 T = .4(2800) = 1120$

$\lambda_2 T = .76(2800) = 2128$

Table 12.2:

$f_{\lambda_1} = .002134$

$f_{\lambda_2} = .100888$

$\boxed{f_{\lambda_2} - f_{\lambda_1} = .098754}$

b.

Wien's law: $(\lambda T)_{max\,power} = 2897.8\ \mu m \cdot K$

$\lambda_{max\,power} = \frac{2897.8}{2800} = \boxed{1.035\ \mu m}$

2.

$f_\lambda(\lambda t) = \frac{\int_0^\lambda E_{b,\lambda}\, d\lambda}{\sigma T^4} = .15$

$\lambda T = .76\mu m\ T$

Table 12.2 @ .15:

$\lambda T \approx 2450$

$\boxed{T = \frac{2450}{.76} = 3223.7\ K}$

3. $\lambda_1 = 3\ \mu m$

$\lambda_2 = 6\ \mu m$

$T = 1000\ K$

$\lambda_1 T = 3000$

$\lambda_2 T = 6000$

Table 12.2:

$f_{\lambda_1} = .273232$

$f_{\lambda_2} = .737818$

$\varepsilon = \varepsilon_1 \lambda_1 + \varepsilon_2(\lambda_2 - \lambda_1) + \varepsilon_3(1 - \lambda_2)$

$= .4(.273232) + .7(.737818 - .273232) + .3(1 - .737818)$

$\boxed{\varepsilon = .513158}$

$E = \varepsilon \sigma T^4 = .513158(5.67\times10^{-8})(1000)^4$

$\boxed{E = 29096\ W/m^2}$

4

$G_{solar} = G_0 \cos\theta + G_d = 350\cos(30) + 400 = 703.109\ W/m^2$

$\dot{q}_{net,rad} = \alpha_s G_{solar} + \varepsilon\sigma(T_{sky}^4 - T_s^4)$

$= .72(703.19) + .6(5.67\times10^{-8})(280^4 - 350^4)$

$\boxed{\dot{q}_{net,rad} = 204.832\ \frac{W}{m^2}}$

Radiation heat transfer

Tuesday, April 30, 2024 11:53 AM

Chap+13+R
adiation+...

HW10

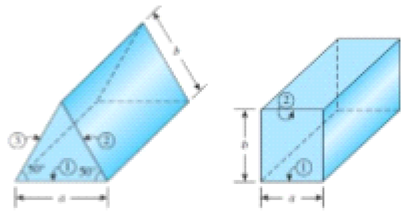
Friday, May 3, 2024 8:43 PM

HW10

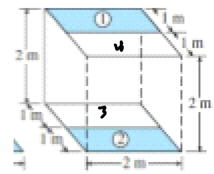
ME 3525 HW10

1. Consider a hemispherical furnace with a flat circular base of diameter D. Determine the view factor from the dome of this furnace to its base.

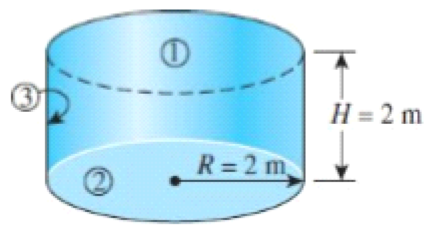
2. Determine the view factors F_{12} and F_{21} for the very long ducts shown in the following figures without using any view factor tables or charts. Neglect end effects.



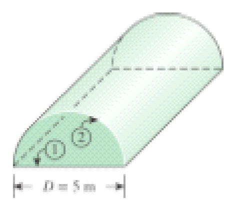
3. Determine the view factor F_{12} between the rectangular surfaces shown in the following figure.



4. A furnace is of cylindrical shape with $R = H = 2$ m. The top (surface 1), base (surface 2) and side (surface 3) surfaces of the furnace are all black and are maintained at uniform temperatures of 700, 1400, and 500 K, respectively. Determine the net rate of radiation heat transfer to or from the top surface during steady operation.



5. A furnace is shaped like a long semicylindrical duct of diameter $D = 5$ m. The base and dome of the furnace have emissivities of 0.5 and 0.9 and are maintained at uniform temperature of 305 K and 1000 K, respectively. Determine the rate of heat transfer from the dome to the base surface per unit length during steady operation.



1.

$F_{11}=0$, flat base

$F_{12}=1$, summation rule

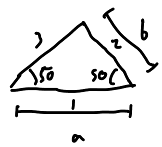
$A_1F_{12}=A_2F_{21} \Rightarrow F_{21}=\frac{A_1}{A_2}F_{12}=\frac{\pi D^2/4}{\pi D^2/2}(1)=\frac{1}{2}$, reciprocity

$\boxed{F_{12}=\frac{1}{2}}$

$A_1 F_{12} = A_2 F_{21} \Rightarrow F_{21} = \frac{A_1}{A_2} F_{12} = \frac{\cdots}{\pi D^2/2}(1) = 2$, reciprocity

$\boxed{F_{12} = \frac{1}{2}}$

2.



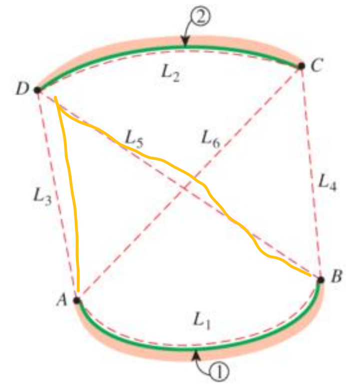
$$F_{1\to 2} = \frac{(L_5 + L_6) - (L_3 + L_4)}{2L_1}$$

$L_6 = L_1 = a$

$L_2 = L_5 = b$

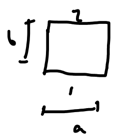
$L_4 = 0$

$L_3 = b$



$$F_{12} = \frac{(b+a) - (b+0)}{2a}$$

$\boxed{F_{12} = \frac{1}{2}}$



$$F_{12} = \frac{2\sqrt{a^2+b^2} - (b+b)}{2a}$$

$$\boxed{F_{12} = \frac{\sqrt{a^2+b^2} - b}{a}}$$

3. $(L_2/D)_{13} = \frac{2}{2} = 1$

$(L_1/D)_{13} = \frac{1}{2}$

$F_{13} = .12$

$F_{(2,3)\to(1,4)} = .2$

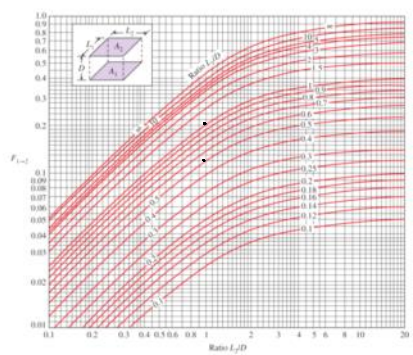


FIGURE 13–5 View factor between two aligned parallel rectangles of equal size.

$F_{(2,3)\to 1} = F_{(2,3)\to 4}$, symmetry

$F_{(2,3)\to(1,4)} = F_{(2,3)\to 1} + F_{(2,3)\to 4}$, superposition

$A_{(2,3)} F_{(2,3)\to 1} = A_1 F_{1\to(2,3)}$, reciprocity

$F_{1\to(2,3)} = F_{12} + F_{13}$ superposition

$F_{12} = F_{1\to(2,3)} + .12$

$$F_{12} = \frac{A_{(2,3)} F_{(2,3)\to 1}}{A_1} + .12$$

$$= \frac{A_{(2,3)}}{A_1} F_{(2,3)\to(1,4)} - F_{(2,3)\to 4} + .12$$

$$= \frac{A_{(2,3)}}{A_1} \frac{1}{2} F_{(2,3)\to(1,4)} + .12$$

$$= \frac{4}{2}\left(\frac{1}{2}\right)(2) + 12$$

$\boxed{F_{12} = .32}$

4.

F_{12} .38

$F_{11} = 0$

$F_{11} + F_{12} + F_{13} = 1$

$F_{13} = 1 - .38 = .62$

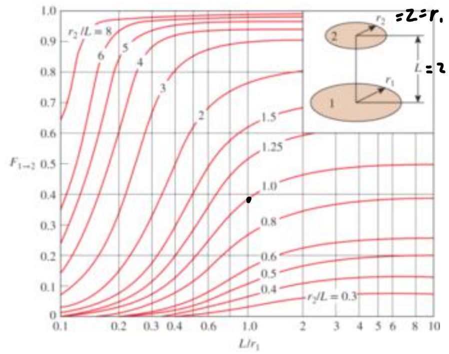


FIGURE 13–7 View factor between two coaxial parallel disks.

$\dot{Q}_1 = \cancel{\dot{Q}_{11}}^{0} + \dot{Q}_{12} + \dot{Q}_{13}$

$\dot{Q}_{12} = A_1 F_{12} \sigma (T_1^4 - T_2^4) = \pi (2)^2 \, .38 (5.67\times10^{-8}) (700^4 - 1400^4)$

$\dot{Q}_{12} = -975.124$ kW

$\dot{Q}_{13} = A_1 F_{13} \sigma (T_1^4 - T_3^4) = \pi (2)^2 (.62)(5.67\times10^{-8})(700^4 - 500^4)$

$\dot{Q}_{13} = 78.4567$ kW

$\dot{Q}_1 = 0 - 975.124 + 78.4567$

$\boxed{\dot{Q}_1 = -896.668 \text{ kW}}$

5.

$F_{11} = 0$

$F_{12} = 1$

$A_1 F_{12} = A_2 F_{21}$

$F_{21} = \frac{A_1}{A_2} F_{12} = \frac{5}{5\pi} = .318$

$F_{22} = .682$

$$\dot{Q}_{12} = \frac{\sigma (T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

$$\dot{Q}_{21} = \frac{5.67\times10^{-8} (1000^4 - 303^4)}{\frac{1-.9}{5\pi(.9)} + \frac{1-.682}{5\pi(.682)} + \frac{1-.6}{5(.6)}}$$

$\boxed{\dot{Q}_{21} = 330.466 \text{ kW}}$