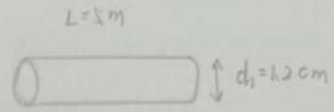
MAE-308 Heat transfer

DDL: 5.19



$T_i = 10°C$ $\quad$ 4 m/s $\quad$ $R = 0.6cm = 0.006\ m$

1. Cooling water available at 10°C is used to condense steam at 30°C in the condenser of a power plant at a rate of 0.15 kg/s by circulating the cooling water through a bank of 5-m-long 1.2-cm internal diameter thin copper tubes. Water enters the tubes at a mean velocity of 4 m/s and leaves at a temperature of 24°C. The tubes are nearly isothermal at 30°C. Determine the average heat transfer coefficient between the water, the tubes, and the number of tubes needed to achieve the indicated heat transfer rate in the condenser.

$T_e = 24°C$ $\quad$ $T_s = 30°C$ $\quad$ h $\quad$?

解: $T_b = \frac{1}{2}(T_i + T_e) = 17°C$

From Table A-9

$\rho = 999.1 + \frac{2}{5}\times(998-999.1) = 998.66\ kg/m^3$

$C_p = 4185 + \frac{2}{5}\times(4182-4185) = 4184\ J/kg\cdot K$

$\dot{m}_w = \rho \cdot V_m \pi R^2 = 0.4518\ kg/s$

$\Delta T_{avg} = \dfrac{T_i - T_e}{\ln \frac{T_s - T_e}{T_s - T_i}} = 11.63°C$

$\Phi = h A_s \Delta T_{avg} = \dot{m}_w C_p (T_e - T_i)$

$\Rightarrow h = \dfrac{\dot{m}_w \cdot C_p (T_e - T_i)}{A_s \cdot \Delta T_{avg}} = 12072\ W/m^2\cdot K$ $\boxed{ANS}$

for the steam at 30°C

from Table A-9, $h_{fg} = 2431\ kJ/kg$

$\dot{m}_{steam} \cdot h_{fg} = n \cdot \dot{m}_w C_p (T_e - T_i)$

$\Rightarrow n = \dfrac{\dot{m}_{steam} \cdot h_{fg}}{\dot{m}_w C_p (T_e - T_i)} = 13.78 \approx 14$ $\boxed{ANS}$

2. Inside a condenser, there is a bank of seven copper tubes with cooling water flowing in them. Steam condenses at a rate of 0.6 kg/s on the outer surfaces of the tubes that are at a constant temperature of 68°C. Each copper tube is 5-m long and has an inner diameter of 25 mm. Cooling water enters each tube at 5°C and exits at 60°C. Determine the average heat transfer coefficient of the cooling water flowing inside each tube and the cooling water mean velocity needed to achieve the indicated heat transfer rate in the condenser.

steam rate ? $\quad$ $T_s =$ $\quad$ $d = 25mm = 25\times10^{-3} m$ $\quad$ T_i $\quad$ T_e $\quad$ V_m

解: $T_b = \frac{1}{2}(T_i + T_e) = 32.5°C$

From Table A-9

$\rho = 996 + \frac{32.5-30}{35-30}\times(994-996) = 995\ kg/m^3$

$C_p = 4178\ J/kg\cdot K$

$\dot{m}_w = \rho V_m \pi R^2 \cdot 7$

$\Delta T_{avg} = \dfrac{T_i - T_e}{\ln \frac{T_s - T_e}{T_s - T_i}} = 26.65°C$

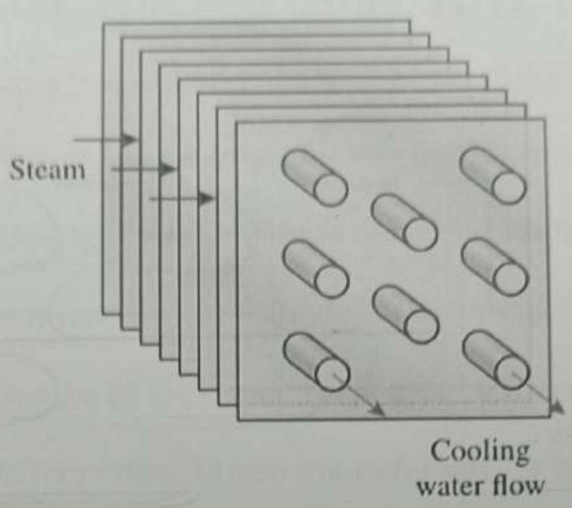
for the steam at 68°C

from Table A-9, $h_{fg} = 2346 + \frac{68-65}{70-65}\times(2334-2346) = 2339\ kJ/kg$

$\dot{m}_{steam} h_{fg} = h A_s \Delta T_{avg} = \dot{m}_w C_p (T_e - T_i)$

$\Rightarrow h = \dfrac{\dot{m}_{steam} h_{fg}}{A_s \Delta T_{avg}} = \dfrac{0.6\times 2339\times10^3}{(\pi\times25\times10^{-3}\times5)\times7\times26.65} = 19157\ W/m^2\cdot K$ $\boxed{ANS}$

$V_m = \dfrac{\dot{m}_{steam} h_{fg}}{C_p (T_e - T_i)} \cdot \dfrac{1}{\rho \pi R^2 \times 7} = 1.786\ m/s$ $\boxed{ANS}$



解:

3. Hot carbon dioxide exhaust gas at 1 atm is being cooled by flat plates. The gas at 220°C flows in parallel over the upper and lower surfaces of a 1.5-m-long flat plate at a velocity of 3 m/s. If the flat plate surface temperature (T_s) is maintained at 80°C, determine (a) the local convection heat (h_x) transfer coefficient at 1 m from the leading edge, (b) the average convection heat transfer coefficient over the entire plate, and (c) the total heat flux transfer to the plate.

in transition

$T_f = \frac{1}{2}(220°C + 80°C) = 150°C$

from Table A-13, $\rho = 1.2675\ kg/m^3$, $\mu = 2.063\times10^{-5}\ kg/m\cdot s$, $P_r = 0.7445$, $\lambda = 0.02652\ W/m\cdot K$

$Re_L = \frac{\rho V L}{\mu} = \frac{1.2675\times3\times1.5}{2.063\times10^{-5}}$

$= 2.765\times10^5 < 5\times10^5$

laminar flow

a) $h_x = \frac{\lambda}{x}\times0.332 P_r^{1/3} Re_x^{1/2}$

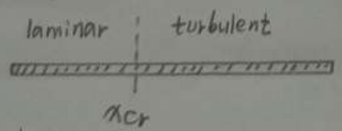
$= \frac{0.02652}{1}\times0.332\times0.7445^{1/3}\times\left(\frac{1.2675\times3\times1}{2.063\times10^{-5}}\right)^{1/2}$

$= 3.426\ W/m^2\cdot K$

$\boxed{ANS}$

b)

$Re_{cr} = 5\times10^5 = \frac{\rho x_{cr} V}{\mu} \Rightarrow x_{cr} = \frac{5\times10^5\times2.063\times10^{-5}}{1.2675\times3} = 2.7\ m$



neglect transition part

$L = 1.5\ m < x_{cr} = 2.7\ m$

$Nu = \frac{hL}{\lambda} = 0.664 P_r^{1/3} Re_L^{1/2}$

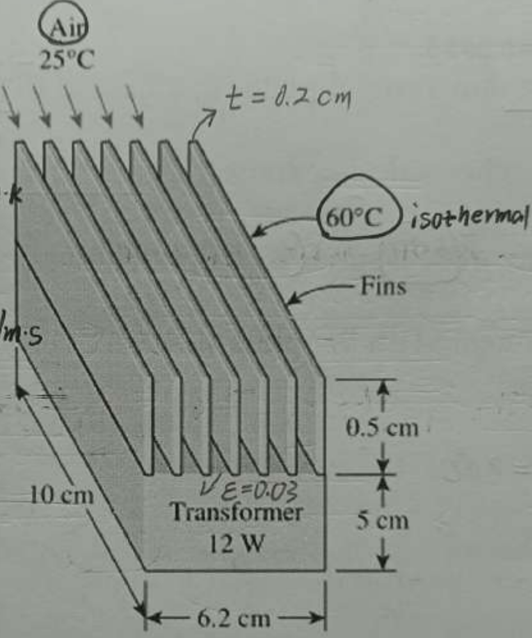
$\Rightarrow h = \frac{\lambda}{L} 0.664 P_r^{1/3} Re_L^{1/2} = 5.5963\ W/(m^2\cdot K)$ $\boxed{ANS}$

c) $q = h(T_g - T_s) = 783.49\ W/m^2$ $\boxed{ANS}$

4. A transformer that is 10 cm long, 6.2 cm wide, and5 cm high is to be cooled by attaching a 10-cm×6.2-cm-wide polished aluminum heat sink (emissivity = 0.03) to its top surface. The heat sink has seven fins, which are 5 mm high, 2 mm thick, and 10 cm long. A fan blows air at 25°C parallel

$\varepsilon = 0.03$ Radiation

to the passages between the fins. The heat sink is to dissipate 12 W of heat and the base temperature of the heat sink is not to exceed 60°C. Assuming the fins and the base plate to be nearly isothermal and the radiation heat transfer to be negligible, determine the minimum free-stream velocity (V) the fan needs to supply to avoid overheating ($T_{base} = 60°C$). Assume the flow is laminar over the entire finned surface of the transformer.



解: $T_f = \frac{1}{2}(25°C + 60°C) = 42.5°C$

from Table A-15

$\rho = 1.127 + \frac{42.5-40}{45-40} \times (1.109 - 1.127) = 1.118\ kg/m^3$

$\lambda = 0.02662 + \frac{1}{2} \times (0.02699 - 0.02662) = 0.026805\ W/m \cdot k$

$P_r = 0.7255 + \frac{1}{2} \times (0.7241 - 0.7255) = 0.7248$

$\mu = [1.918 + \frac{1}{2} \times (1.941 - 1.918)] \times 10^{-5} = 1.930 \times 10^{-5}\ kg/m \cdot s$

laminar flow

$Nu = \frac{hL}{\lambda} = 0.664 P_r^{1/3} Re_L^{1/2}$ ①

$Re_L = \frac{\rho V L}{\mu}$ ②

heat transfer.

$\Phi = h A_s (T_{base} - T_{air}) = 12\ W$

$A_s = 7 \times (0.5 \times 10) \times 2 = 70\ cm^2$

$\Rightarrow h = 48.98\ W/m^2 \cdot k$

combine ①②: $V = (\frac{hL}{\lambda})^2 \frac{1}{(0.664 P_r^{1/3})^2} \cdot \frac{\mu}{\rho L} = 16.20\ m/s$ $\boxed{ANS}$

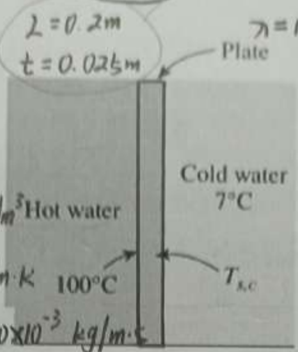
0.025 m λ

5. A 0.2-m-long and 25-mm-thick vertical plate (k = 15 W/m·K) separates the hot water from the cold water. The plate surface exposed to the hot water has a temperature of 100°C and the temperature of the cold water is 7°C. Determine the temperature of the plate surface exposed to the cold water ($T_{s,c}$). Hint: The $T_{s,c}$ has to be found iteratively. Start the iteration

process with an initial guess of 53.5°C for the $T_{s,c}$.

解: initial guess

$T_{s.c} = 53.5^\circ C$



$T_{avg} = \frac{1}{2}(53.5+7) = 30.25^\circ C$

From Table A-9:

$\rho = 996 + \frac{30.25-30}{35-30} \times (994-996) = 996\ kg/m^3$

$\lambda = 0.615 + \frac{0.25}{5} \times (0.623-0.615) = 0.62\ W/m \cdot K$

$\mu = [0.798 + \frac{0.25}{5} \times (0.72-0.798)] \times 10^{-3} = 0.800 \times 10^{-3}\ kg/m \cdot s$

$Pr = 5.42 + \frac{0.25}{5} \times (4.83-5.42) = 5.39$

$\beta = \frac{1}{T_{avg}} = \frac{1}{303.25k} = 0.0033$

$Gr = \frac{\beta g (T_{s.c} - T_c) L^3}{\nu^2} = 1.865 \times 10^{10}$

$Ra = Gr \cdot Pr = 1.01 \times 10^{11}$

$Nu = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}} \right\}^2$

$= 649.5$

$h \cdot (T_{sc} - T_c) = \lambda_p \frac{T_{sh} - T_{sc}}{\delta}$

where $h = \frac{Nu \cdot \lambda}{L}$

$\Rightarrow T_{sc} = \frac{\frac{\lambda_p}{\delta} T_{sh} + h T_c}{\frac{\lambda_p}{\delta} + h} = 28.35^\circ C$

it's half of the guess value $53.5^\circ C$

Now guess $T_{sc} = 33^\circ C$

$T_{avg} = \frac{1}{2}(33+7) = 20^\circ C$, $\frac{1}{\beta} = \frac{1}{293}$

From Table A-9:

$\rho = 998\ kg/m^3$, $\lambda = 0.598\ W/m \cdot k$, $Pr = 7.01$, $\mu = 1.002 \times 10^{-3}\ kg/m \cdot s$

$Gr = 7.17 \times 10^9$

$Ra = 5.03 \times 10^{10}$

$Nu = 527.6$

Similarly, $T_{sc} = \frac{\frac{\lambda_p}{\delta} T_{sh} + \frac{Nu \lambda}{L} T_c}{\frac{\lambda_p}{\delta} + \frac{Nu \lambda}{L}}$

$= 32.6^\circ C \approx 33^\circ C$

$\boxed{ANS}$

6. A 0.5-m-long thin vertical plate is subjected to uniform heat flux on one side, while the other side is exposed to cool air at 5°C. The plate surface has an emissivity of 0.73, and its midpoint temperature is 55°C. Determine the heat flux subjected on the plate surface.

constant q. ε ?

解: $T_{L/2} = 55^\circ C$

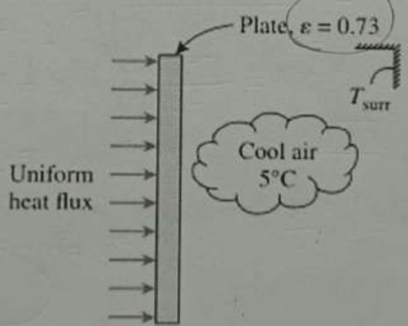
$T_{avg} = \frac{1}{2}(T_{L/2} + T_{air}) = 30^\circ C$

From Table A-15

$\lambda = 0.02588\ W/m \cdot k$

$Pr = 0.7282$

$\beta = \frac{1}{T_{avg}} = \frac{1}{30+273} = \frac{1}{303}$



$\nu = 1.608 \times 10^{-5}\ m^2/s$

$Gr = \frac{\beta g (T_{L/2} - T_{air}) L^3}{\nu^2} = 782590024 = 7.826 \times 10^8$

$Ra = Gr \cdot Pr = 5.699 \times 10^8$

$Nu = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}} \right\}^2 = 103.7 = \frac{q_s L}{\lambda (T_{L/2} - T_\infty)} \Rightarrow q_s = 268.4\ W/m^2$

Consider the radiation

$q_R = \varepsilon \sigma (T_{L/2}^4 - T_c^4)$

$= 0.73 \times 5.67 \times 10^{-8} \times [(55+273)^4 - (5+273)^4]$

$= 231.9\ W/m^2$

$q_{total} = q_s + q_R = 500.3\ W/m^2$ $\boxed{ANS}$

7. Consider a 3-m-high rectangular enclosure consisting of two surfaces separated by a 0.1-m air gap at 1 atm. If the surface temperatures across the air gap are 30°C and −10°C, determine the ratio of the heat transfer rate

for the horizontal orientation (with hotter surface at the bottom) to that for vertical orientation.

解: $T_{avg} = \frac{1}{2}(30-10) = 10^\circ C$

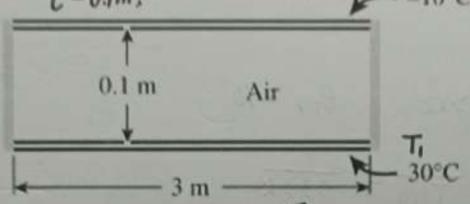
From Table A-15

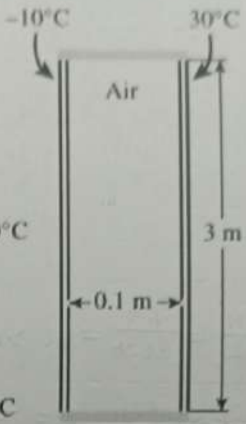
$P_r = 0.7336$

$\nu = 1.426 \times 10^{-5}\ m^2/s$, $\lambda = 0.02439\ W/m\cdot k$

$\beta = \frac{1}{T_{avg}} = \frac{1}{283}$

$l = 0.1m$,





$$Ra = \frac{\beta g (T_1 - T_2) l^3}{\nu^2} P_r$$

$$= 5 \times 10^6$$

For horizontal:

$$Nu_1 = 1 + 1.44\left[1 - \frac{1708}{Ra}\right]^+ + \left[\frac{Ra^{1/3}}{18} - 1\right]^+ = 10.94$$

$$\Phi_h = h A_s (T_1 - T_2) = \frac{\lambda Nu_1}{l} A_s (T_1 - T_2)$$

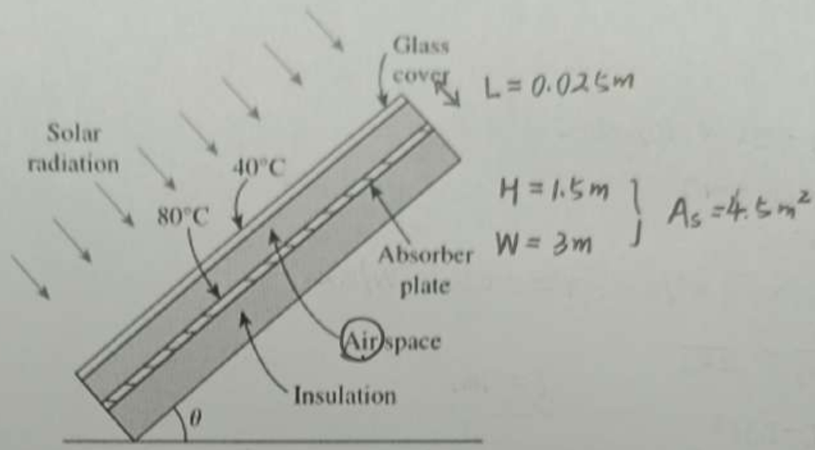
For vertical:

$$\left.\begin{array}{l} 10 < \frac{H}{L} = 30 < 40 \\ 10^4 < Ra = 5\times10^6 < 10^7 \end{array}\right\} \Rightarrow Nu_2 = 0.42 Ra^{1/4} P_r^{0.012} \left(\frac{H}{L}\right)^{-0.3} = 7.132$$

$$\Phi_v = \frac{\lambda Nu_2}{l} A_s (T_1 - T_2)$$

$$\therefore Ratio = \frac{Nu_1}{Nu_2} = 1.534 \ \boxed{ANS}$$

8. Flat-plate solar collectors are often tilted up toward the sun in order to intercept a greater amount of direct solar radiation. The tilt angle from the horizontal also affects the rate of heat loss from the collector. Consider a $H = 1.5m$ 1.5-m-high and $W = 3m$ 3-m-wide solar collector that is tilted at an angle θ from the horizontal. The back side of the absorber is heavily insulated. The absorber plate and the glass cover, which are spaced $L = 0.025m$ 2.5 cm from each other, are maintained at temperatures of 80°C and 40°C, respectively. Determine the rate of heat loss from the absorber plate by natural convection for $\theta = 30^\circ, 60^\circ$ and 90°



解: aspect ratio $\frac{H}{L} = \frac{1.5}{0.025} = 60 > 12$, $\theta_{cr} = 70^\circ$

$T_{avg} = \frac{1}{2}(40+80) = 60^\circ C$, $\beta = \frac{1}{T_{avg}} = \frac{1}{333K}$

From Table A-15

$P_r = 0.7202$, $\nu = 1.896 \times 10^{-5}\ m^2/s$, $\lambda = 0.02808\ W/m\cdot K$

$$Ra = \frac{\beta g (T_1 - T_2) L^3}{\nu^2} P_r = 36888 < 10^5$$

$$\therefore\ Nu = 1 + 1.44\left[1 - \frac{1708}{Ra}\right]^+ \left[1 - \frac{1708(\sin 1.8\theta)^{1.6}}{Ra\cos\theta}\right] + \left[\frac{(Ra\cos\theta)^{1/3}}{18} - 1\right]^+$$

$\theta = 30^\circ$, $Nu = 3.084$, $\Phi = \lambda Nu A_s \frac{(T_1 - T_2)}{L} = 623.5\ W$ $\boxed{ANS}$

$\theta = 60^\circ$, $Nu = 2.724$, $\Phi = 550.7\ W$ $\boxed{ANS}$

$\theta = 90^\circ$, vertical case, $Nu = 0.42 Ra^{1/4} P_r^{0.012} \left(\frac{H}{L}\right)^{-0.3} = 1.698$, $\Phi = 343.2\ W$ $\boxed{ANS}$