ChE 306: HEAT TRANSFER
FALL 2010
Homework #9: Ch 10 & 11
(80 points + 15 bonus points)
DUE: FRIDAY, NOVEMBER 12

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1. A 15 ^{o}C copper rectangular surface measuring 0.6 m by 1.5 m and coated with Rain-X is oriented horizontally above a pool of saturated steam kept at a pressure of 0.51 bars so that it condenses in a dropwise form.
A. What is the rate of heat transfer from the vapor to the copper surface? (W)
B. How fast does liquid water form (use h_{fg}')? (kg/s)
C. If the Rain-X were stripped away, would the heat transfer rate increase, decrease or stay the same? Why?

A. Here, dropwise condensation has a convection coefficient h, determined by equations 10.49 or 10.50. Because the steam is at 0.51 bars (sub-atmospheric pressure), T_{sat} is below 100 ^{o}C, so equation 10.49 applies. To find T_{sat}, use the steam table A.6. Here, at 0.51 bars, the saturation temperature is 355 K (approx 82 ^{o}C).

h_{DC} = 51104 + 2044 * 82 = 218712 W/m^2-K

q = h A (Tsat - Ts) (switched order to give a positive q for heat transferred from saturated steam to surface)
= 218712 W/m2-K * (0.6 * 1.5 m2) * (82 - 15 K)
= 1.32 * 10^7 W

B. What is m-dot? m-dot = q/h_{fg}' = 1.32 * 10^7 W / (h_{fg} + 0.68 $c_{p,l}$ (T_{sat} - T_s))
From Table A.6: at Tsat: hfg = 2304 kJ/kg (or 2304000 J/kg)
at Tfilm cp,l = 4.184 kJ/kg-K (or 4199 J/kg-K)

m-dot = 1.32 * 10^7 J/s / (2304000 J/kg + 0.68 (4184 J/kg-K) (82-15 K))
m-dot = 5.3 kg/s

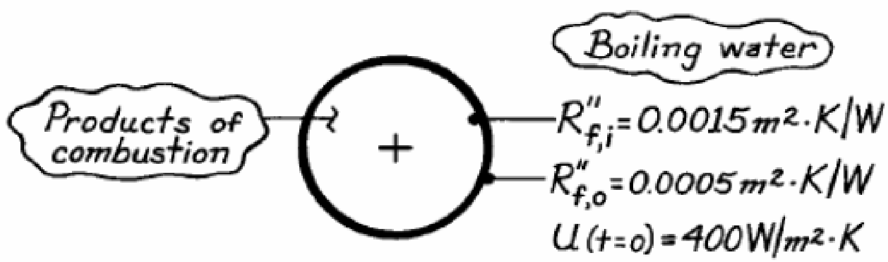
C. If there were no rain-X, the surface would be more wettable, and drops would adhere to the surface longer, reducing the value of h. Thus the heat transfer rate (and the rate of condensation) would decrease.

2. A fire-tube boiler uses the hot products of a combustion reaction (flowing in the tubes) to boil water flowing over the outside of the tubes. When the boiler was first installed U was found to be 400 W/m^2-K. After one year, both sides of the tubes have become fouled. $R_{f,i}$" = 0.0015 and $R_{f,o}$" = 0.0005 m^2-K/W. Should the boiler be cleaned to remove the fouling? You may consider the resistance to conduction through the pipe to be negligible.

KNOWN: Initial overall heat transfer coefficient of a fire-tube boiler. Fouling factors following one year's application.

FIND: Whether cleaning should be scheduled.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible tube wall conduction resistance, (2) Negligible changes in h_c and h_h.

ANALYSIS: From Equation 11.1, the overall heat transfer coefficient after one year is

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + R''_{f,i} + R''_{f,o}.$$

Since the first two terms on the right-hand side correspond to the reciprocal of the initial overall coefficient,

$$\frac{1}{U} = \frac{1}{400\ \mathrm{W/m^2 \cdot K}} + (0.0015 + 0.0005)\,\mathrm{m^2 \cdot K/W} = 0.0045\ \mathrm{m^2 \cdot K/W}$$

$$U = 222\ \mathrm{W/m^2 \cdot K}.$$

COMMENTS: Periodic cleaning of the tube inner surfaces is essential to maintaining efficient fire-tube boiler operations.

The answer is YES! A reduction in U by nearly HALF can cause big heat transfer problems!

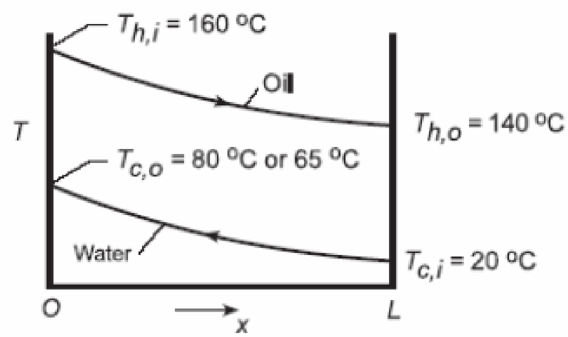
3. Work Incropera and DeWitt Problem 11.16

(C) Calculate the required length of the heat exchanger if operated in co-current flow, and compare your answer to (A).

KNOWN: Inner tube diameter (D = 0.02 m) and fluid inlet and outlet temperatures corresponding to design conditions for a counterflow, concentric tube heat exchanger. Overall heat transfer coefficient (U = 500 $W/m^2 \cdot K$) and desired heat rate (q = 3000 W). Cold fluid outlet temperature after three years of operation.

FIND: (a) Required heat exchanger length, (b) Heat rate, hot fluid outlet temperature, overall heat transfer coefficient, and fouling factor after three years.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat loss to the surroundings, (2) Negligible tube wall conduction resistance, (3) Constant properties.

ANALYSIS: (a) The tube length needed to achieve the prescribed conditions may be obtained from Eqs. 11.14 and 11.15 where $\Delta T_1 = T_{h,i} - T_{c,o} = 80°C$ and $\Delta T_2 = T_{h,o} - T_{c,i} = 120°C$. Hence, $\Delta T_{lm} = (120 - 80)°C/\ln(120/80) = 98.7°C$ and

$$L = \frac{q}{(\pi D) U \Delta T_{lm}} = \frac{3000\,W}{(\pi \times 0.02\,m) 500\,W/m^2 \cdot K \times 98.7°C} = 0.968\,m \qquad <$$

(b) With $q = C_c(T_{c,o} - T_{c,i})$, the following ratio may be formed in terms of the design and 3 year conditions.

$$\frac{q}{q_3} = \frac{C_c\left(T_{c,o} - T_{c,i}\right)}{C_c\left(T_{c,o} - T_{c,i}\right)_3} = \frac{60°C}{45°C} = 1.333$$

Hence,

$$q_3 = q/1.33 = 3000\,W/1.333 = 2250\,W \qquad <$$

Having determined the ratio of heat rates, it follows that

$$\frac{q}{q_3} = \frac{C_h\left(T_{h,i} - T_{h,o}\right)}{C_h\left(T_{h,i} - T_{h,o}\right)_3} = \frac{20°C}{160°C - T_{h,o(3)}} = 1.333$$

Hence,

$$T_{h,o(3)} = 160°C - 20°C/1.333 = 145°C \qquad <$$

With $\Delta T_{lm,3} = (125 - 95)/\ln(125/95) = 109.3°C$,

$$U_3 = \frac{q_3}{(\pi D L)\Delta T_{lm,3}} = \frac{2250\,W}{\pi(0.02\,m)0.968\,m\left(109.3°C\right)} = 338\,W/m^2 \cdot K \qquad <$$

With $U = \left[(1/h_i) + (1/h_o)\right]^{-1}$ and $U_3 = \left[(1/h_i) + (1/h_o) + R''_{f,c}\right]^{-1}$,

$$R''_{f,c} = \frac{1}{U_3} - \frac{1}{U} = \left(\frac{1}{338} - \frac{1}{500}\right) m^2 \cdot K/W = 9.59 \times 10^{-4} m^2 \cdot K/W \quad <$$

COMMENTS: Over time fouling will always contribute to a degradation of heat exchanger performance. In practice it is desirable to remove fluid contaminants and to implement a regular maintenance (cleaning) procedure.

C. The only thing that changes from part A is the dTLM.
For co-current flow the inlets for both fluids are on the same side...
dTLM = (160 - 20) - (140 - 80) / ln (160-20)/(140-80) = 94.4 C

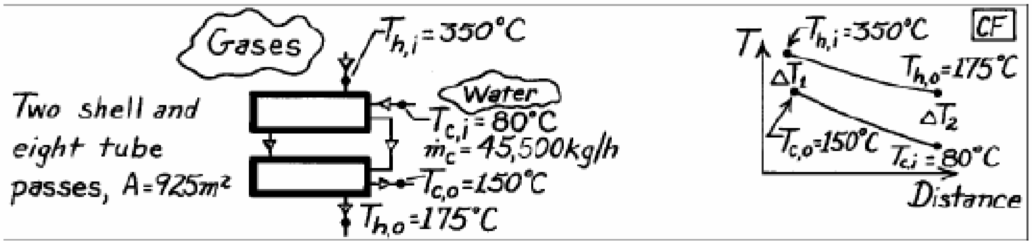
Following from above, L = 1.01 m (it's not incredibly different- just an extra 5 cm, but counter flow still provides better heat transfer).

4. Water enters a shell and tube heat exchanger at 45,500 kg/h and an inlet temperature of 80 °C. The shell and tube exchanger has 2 shell passes and 8 tube passes with a total outer surface area of 925 m^2. The water is heated to 150 °C using heated air at 350 °C. The air exits the heat exchanger with a temperature of 175 °C.
A. Determine the flow rate of air in the heat exchanger, assuming perfect insulation.
B. Determine the overall heat transfer coefficient, U_o, for this process. Use the ε-NTU method and Tables 11.3 and/or 11.4.

KNOWN: Heat exchanger with two shell passes and eight tube passes having an area 925m^2; 45,500 kg/h water is heated from 80°C to 150°C; hot exhaust gases enter at 350°C and exit at 175°C.

FIND: Overall heat transfer coefficient.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible losses to surroundings, (2) Constant properties, (3) Exhaust gas properties are approximated as those of atmospheric air.

PROPERTIES: *Table A-6*, Water $\left(\overline{T}_c = (80+150)°C/2 = 388K\right)$: $c_c = c_{p,f} = 4236$ J/kg·K.

ANALYSIS: Since this is a shell-and-tube heat exchanger, we will use the ε - NTU method, for which

$$C_c = \dot{m}_c c_c = \frac{45{,}500 \text{ kg/h}}{3600 \text{ s/h}} \times 4236 \text{ J/kg}\cdot\text{K} = 5.35 \times 10^4 \text{ W/K}$$

$$q = C_c(T_{c,o} - T_{c,i}) = 5.35 \times 10^4 \text{ W/K}\,(150-80)°C = 3.75 \times 10^6 \text{ W}$$

Then we can find C_h from an energy balance on the hot stream,

$$C_h = q/(T_{h,i} - T_{h,o}) = 3.75 \times 10^6 \text{ W}/(350-175)°C = 2.14 \times 10^4 \text{ W/K}$$

Thus

$$C_r = C_{min}/C_{max} = 0.40$$

$$\varepsilon = q/C_{min}(T_{h,i} - T_{c,i}) = 3.75 \times 10^6 \text{ W}/2.14 \times 10^4 \text{ W/K}\,(350-80)°C = 0.648$$

From Eqs. 11.31b and c, with n = 2,

$$F = \left(\frac{\varepsilon C_r - 1}{\varepsilon - 1}\right)^{1/n} = 1.45, \quad \varepsilon_1 = \frac{F-1}{F-C_r} = 0.429$$

From Eqs. 11.30c and 11.30b,

$$E = \frac{2/\varepsilon_1 - (1+C_r)}{(1+C_r^2)^{1/2}} = 3.0$$

$$(NTU)_1 = -(1+C_r^2)^{-1/2}\ln\left[\frac{E-1}{E+1}\right] = 0.637$$

and from Eq. 11.31d,

$$NTU = n(NTU)_1 = 1.27$$

Therefore,

$$U = NTU \times C_{min}/A = 1.27 \times 2.14 \times 10^4 \text{ W/K}/(925 \text{ m}^2) = 29.5 \text{ W/m}^2\cdot\text{K}$$ <

COMMENTS: Compare the above result with representative values for air-water exchangers, as given in Table 11.2.

5. A shell and tube heat exchanger with 2 tube passes and 1 shell passes is used to heat water (shell-side) with engine oil (tube-side). There are 100 tubes per pass. The tubes are made of copper and have an inside diameter of 0.95 cm and outside diameter of 1.15 cm. The tubes are perfectly clean on the inside, but have a fouling factor on the outside of $R_{f,o}$" = 0.002 m^2-K/W. Water enters the shell at 1.2 kg/s at 31 ^oC and exits at 67 ^oC. Oil enters the tube-side at 112 ^oC and a flow rate of 1.5 kg/s. h_o for the water = 134 W/m^2-K and h_i = 46 W/m^2-K.
A. What is the rate of change of internal energy of the water?
B. What is the outlet temperature of the oil?
C. Determine UA in terms of L, an as-yet undefined length of the tubes.
D. How long must the tubes be per pass to achieve the required heat transfer? Use the LMTD method with the fudge factor, F. (see class handout for graphs needed!)

A. dU = m cp (Tout - Tin) = 1.2 kg/s * c_p @ T_{mean} = 49 oC * (67-31 C)
From Table A.6, cp at 49 C (or 322 K) is 4.181 kJ/kg-K or 4,181 J/kg-K
thus, dU = 1.2 kg/s * 4181 J/kg-K * 36 K = 180,600 W.

B. matching dU1 = - dU2
180,600 W = m cp,oil (Tin - Tout) = 1.5 kg/s * $c_{p,oil}$ at mean T J/kg-K* (112 - T_o K)
Slight problem here. Since we don't know T_o yet, we'll need to guess a To, determine Tmean, look up cpoil, and then solve for To. Hopefully we'll be close, otherwise the problem might require iteration.
Let's guess To = 92 C, so Tmean = 102 C (375 K).
By interpolation in Table A.5, cp = 2222 J/kg-K.

plugging in, 112 - T_o = 180600 W / (1.5 kg/s * 2222 J/kg-K)

T_o = 57.8 C.
This is a bit off of our guess, so let's iterate one time.
T_{mean} = 112 + 58 / 2 = 85 C (358 K) cp = 2152 J/kg-K
Now, T_o = 56.0 C This is satisfactory.

C. 1/UA = 1/hoAo + R_{fo}/A_o + ln (r_2/r_1)/2 pi L k_{copper} + R_{fi}/A_i + $1/h_iA_i$ (term 4 = 0, no Rfi)
factor out L from each of the terms & flip.
1/UA = 1/L * {(1/134 W/m2-K*pi*0.0115 m) + (0.002/pi*.0115) + (ln (0.0115/0.0095)/2*pi*401) + 1/(46W/m2-K*pi*.0095m)}
1/UA = 1/L * (.206 + 0.055 + 7.5 * 10-5 + 0.728) = 0.99 / L
or UA = 1.01 L
(or if you included the number of tubes and number of passes as part of the area, then UA = 2020 L)

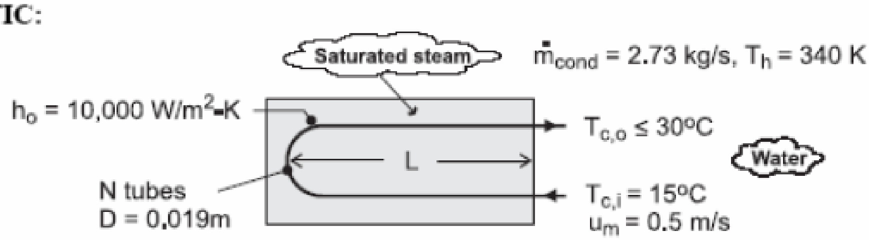
D. q = UAN_TN_PF dTLM
= 1.01 * L * 100 * 2 * "F" * {(112-67) - (56-31)}/ ln ((112-67)/(56-31))
Using the Figure from the handout for 1 shell, 2 tube passes,
to fing F (F_G), need Z and eta. Z = (112 - 57.8) / (67 - 31) = 1.51
eta = (67 - 31) / (112 - 31) = 0.44
For 1 shell & 2 tube passes, F is approximately 0.6
q = dU = 180,600 W = 1.01 * L * 100 * 2 * 0.6 * 34.0 ^oC (dT_{LM})
Thus, L = 43.8 m
This is pretty long.... probably needs a redesign, perhaps change to 4 or 8 tube passes, or increase the number of tubes.

6. Work Incropera and DeWitt 11.38. Use the ε-NTU method.

KNOWN: Temperature, convection coefficient and condensation rate of saturated steam. Tube diameter for shell-and-tube heat exchanger with one shell pass and two tube passes. Velocity and inlet and maximum allowable exit temperatures of cooling water.

FIND: (a) Minimum number of tubes and tube length per pass, (b) Effect of tube-side heat transfer enhancement on tube length.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat exchange with surroundings, (2) Negligible tube wall conduction and fouling resistance, (3) Constant properties, (4) Fully developed internal flow throughout.

PROPERTIES: *Table A-6*, Sat. water (340 K): $h_{fg} = 2.342 \times 10^6$ J/kg; Sat. water $\left(\overline{T}_c = 22.5°C \approx 295\ K\right)$: ρ=998 kg/m^3, c_p = 4181 J/kg·K, $\mu = 959 \times 10^{-6}$ N·s/m^2, k = 0.606 W/m·K, Pr = 6.62.

ANALYSIS: (a) The required heat rate and the maximum allowable temperature rise of the water determine the minimum allowable flow rate. That is, with

$$q = q_{cond} = \dot{m}_{cond} h_{fg} = 2.73\ kg/s \times 2.342 \times 10^6\ J/kg = 6.39 \times 10^6\ W$$

$$\dot{m}_{c,min} = \frac{q}{c_{p,c}\left(T_{c,o} - T_{c,i}\right)} = \frac{6.39 \times 10^6\ W}{4181\ J/kg \cdot K(15°C)} = 101.9\ kg/s$$

With a specified flow rate per tube of $\dot{m}_{c,1} = \rho u_m \pi D^2/4$ = 998 kg/m^3 × 0.5 m/s × π (0.019m)^2/4 = 0.141 kg/s, the minimum number of tubes is

$$N_{min} = \frac{\dot{m}_{c,min}}{\dot{m}_{c,1}} = \frac{101.9\ kg/s}{0.141\ kg/s} = 720$$ <

To determine the corresponding tube length, we must first find the required heat transfer surface area. With $Re_D = \rho u_m D/\mu$ = 998 kg/m^3 (0.5 m/s) 0.019m/959 × 10^{-6} N·s/m^2 = 9,886, the Dittus-Boelter equation yields

$$\overline{h}_i = (k/D)0.023\ Re_D^{4/5}\ Pr^{0.4} = (0.606\ W/m \cdot K / 0.019m)0.023(9886)^{4/5}(6.62)^{0.4} = 2454\ W/m^2 \cdot K$$

Hence, $U = \left[\bar{h}_i^{-1} + h_o^{-1}\right]^{-1} = 1970\ W/m^2 \cdot K$

With $C_r = 0$, $C_{min} = \dot{m}_c\ c_{p,c} = 101.9$ kg/s × 4181 J/kg·K = 4.26 × 10^5 W/K, $q_{max} = C_{min}\,(T_{h,i} - T_{c,i})$ = 4.26 × 10^5 W/K (340 – 288) K = 2.215 × 10^7 W and $\varepsilon = q/q_{max} = 0.289$, Eq. 11.35b yields NTU = – ln (1 – ε) = – ln (1 – 0.289) = 0.341. Hence the tube length per pass is

$$L = \frac{A}{2N\pi D} = \frac{NTU \times C_{min}}{2N\pi DU} = \frac{0.341 \times 4.26 \times 10^5\ W/K}{2 \times 720 \times \pi (0.019m) 1970\ W/m^2 \cdot K} = 0.858m \qquad <$$

(b) If the tube-side convection coefficient is doubled, $\bar{h}_i = 4908\ W/m^2 \cdot K$ and U = 3292 W/m^2·K. Since q, C_r, C_{min}, q_{max} and hence ε are unchanged, the number of transfer units is still NTU = 0.341. Hence, the tube length per pass is now

$$L = \frac{NTU \times C_{min}}{2N\pi DU} = \frac{0.341 \times 4.26 \times 10^5\ W/K}{2 \times 720 \times \pi (0.019m) 3292\ W/m^2 \cdot K} = 0.513m \qquad <$$

COMMENTS: Heat transfer enhancement for the flow with the smallest convection coefficient significantly reduces the size of the heat exchanger.

7. Open heart surgery is normally done in hypothermic conditions, where the patient's blood is cooled before the surgery and re-warmed afterward. A proposed concentric tube counterflow heat exchanger is designed to have a length of 0.5 m and inside tube diameter of 55 mm (thin-walled), and has an overall heat transfer coefficient, U, of 500 W/m^2-K. Given that the blood enters the heat exchanger at 18 ^{o}C and 0.05 kg/s, and water enters the annulus at 60 ^{o}C at 0.10 kg/s,

(A) What is the temperature of the blood exiting the exchanger?

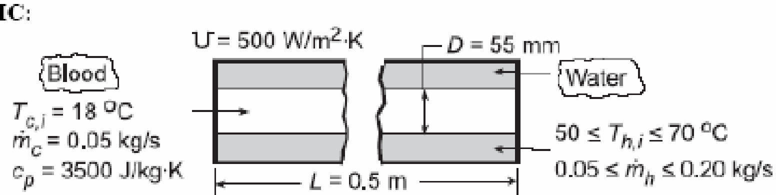
(B) How long would the exchanger need to be to achieve a blood outlet temperature of 37 ^{o}C?

You may assume the mean temperature of the water to be 55 ^{o}C, and the heat capacity of the blood to be 3500 J/kg-K.

KNOWN: Dimensions, fluid flow rates, and fluid temperatures for a counterflow heat exchanger used to heat blood.

FIND: (a) Outlet temperature of the blood, (b) Effect of water flowrate and inlet temperature on heat rate and blood outlet temperature.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat loss to surroundings, (2) Constant properties.

PROPERTIES: *Table A.6*, Water ($\overline{T}_m \approx 55°C$): c_p = 4183 J/kg·K.

ANALYSIS: (a) Using the ε - NTU method, we first obtain $C_h = (\dot{m}_h c_{p,h}) = (0.10 \text{ kg/s} \times 4183 \text{ J/kg·K})$ = 418.3 W/K and $C_c = (\dot{m}_c c_{p,c}) = (0.05 \text{ kg/s} \times 3500 \text{ J/kg·K}) = 175 \text{ W/K} = C_{min}$. Hence, (C_{min}/C_{max}) = 0.418 and

$$NTU = \frac{UA}{C_{min}} = \frac{\left(500\ W/m^2\cdot K\right)\pi(0.055\,m)(0.5\,m)}{175\,W/K} = 0.247.$$

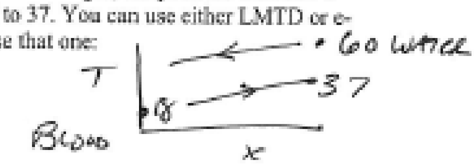
From Eq. 11.29a, ε = 0.21. Hence, from Eq. 11.22

$$q = \varepsilon C_{min}\left(T_{h,i} - T_{c,i}\right) = 0.21(175\,W/K)(60-18)^\circ C = 1544\,W.$$

From Eq. 11.7,

$$T_{c,o} = T_{c,i} + \frac{q}{C_c} = 18^\circ C + \frac{1544\,W}{175\,W/K} = 26.8^\circ C$$

(B) Here, since we are given a desired outlet temperature and want to find a length, the problem becomes a design problem. Obviously, the tubes must be longer to heat the blood to 37. You can use either LMTD or ε-NTU method at this point. Since LMTD is generally shorter, I'll choose that one:



ΔT_{lm} can be found.

$$\Delta U_{Blood} = \dot{m}\, c_p (T_o - T_i)$$

$$= (.05\ kg/s)(3500\ J/kg\text{-}K)(37-18) = 3325\ W$$

$$\Delta U_{water}: -3325\ W = (0.10\ kg/s)(4183\ J/kg\text{-}K)(T_o - 60^\circ C)$$

$$T_o = 52.1^\circ C$$

$$\Delta T_{lm,CF} = \frac{(60-37) - (52.1-18)}{\ln\left(\frac{60-37}{52.1-18}\right)} = 28.2^\circ C$$

7 p.2

$$q = UA\,\Delta T_{LM,CF}$$

$$= U \pi D L\,\Delta T_{LM,CF}$$

$$L = \frac{q}{U \pi D\,\Delta T_{LM,CF}}$$

$$= \frac{3325\ W}{(500\ W/m^2K)\,\pi\,(.055\ m)(28.2\ ^\circ C)}$$

$$= \underline{1.365\ m}$$

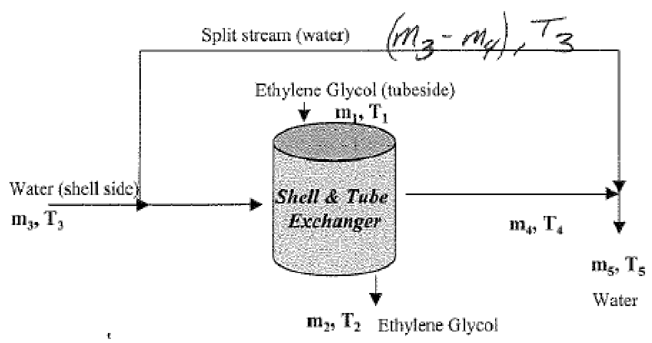
8. The shell and tube heat exchanger below is being retrofit to heat a feed stream of ethylene glycol, using supplied plant water that is available in excess.

For the specifications given, determine:

(a) the required mass flow rate of the plant water through the exchanger, $\mathbf{m_4}$.

(b) the number of tubes per pass in the exchanger, $\mathbf{N_T}$.

(c) the temperature of the recombined plant water streams, $\mathbf{T_5}$. *(you may assume c_p is constant so that $m_4 (T_5 - T_4) = -(m_3 - m_4)(T_5 - T_3)$).*



ETHYLENE GLYCOL FEED (TUBE)
Inlet Temperature (T_1) = 37 °C
Outlet Temperature (T_2)= 57 °C
Flow Rate = 24 kg/s (m_1)
$c_p = 2.505$ KJ/kg-K

PLANT WATER (UTILITY-SHELL)
Available supply: 30 kg/s;92 °C (m_3, T_3)
Exchanger Outlet Temp (T_4) = 52 °C
$c_p = 4.191$ KJ/kg-K

EXCHANGER DESIGN:
Overall Heat Transfer Coefficient (based on outside area) $U_o = 732\ W/m^2$-K
Length = 5 m
Tube side: 2-pass, 1.4 cm outer diameter copper tubes, w/ wall thickness 0.25 cm
Shell side: 1-pass; baffle spacing of 20 cm

(1) $q_c = \dot{m}_c c_{p_c} (T_{c_o} - T_{c_i})$

(2) $q_H = \dot{m}_H c_{pH} (T_{H_i} - T_{H_o})$

(3) $q = UAF\, \Delta T_{lm,CF}$

(A)

(1) $q = (24 \frac{kg}{s})(2.505 \frac{KJ}{kg\text{-}K})(57\text{-}37°C)$
$= 1202$ KW

(2) $\dot{m}_H = \dfrac{q_H}{c_{pH}(T_{H_i} - T_{H_o})} = \dfrac{1202\ KW}{(4.191 \frac{KJ}{kg\text{-}K})(92\text{-}52°c)}$

$\boxed{\dot{m}_H = 7.17\ kg/s}$

(B) use eqn 3.

$\Delta T_{lm,CF} = \dfrac{(92\text{-}57) - (52\text{-}37)}{\ln \frac{(92\text{-}57)}{(52\text{-}37)}} = 23.6°c$

F from fig. 11.10

$P = \dfrac{57\text{-}37}{92\text{-}37} = .36 \quad R = \dfrac{92\text{-}52}{57\text{-}37} = 2.$

$\therefore F \approx 0.75$

$A_o = \dfrac{q}{U_o F \Delta T_{lm}} = \dfrac{1202000\ W}{(732 \frac{W}{m^2\text{-}K})(.75)(23.6°c)}$

$A_o = 92.77 m^2 = \pi D_o L N_T N_p$

$N_T = \dfrac{92.77\ m^2}{\pi (.014m)(5m)(2)} = 210.9$

$\boxed{N_T = 211}$

(C) Assuming C_p is constant

$\dot{m}_5 T_5 = (\dot{m}_3 - \dot{m}_4) T_3 + \dot{m}_4 T_4$

($\dot{m}_5 = \dot{m}_3$)

$(30 \frac{kg}{s})(T_5) = (30 - 7.17)(92°c) + 7.17(52°c)$

$\boxed{T_5 = 82.4°C}$