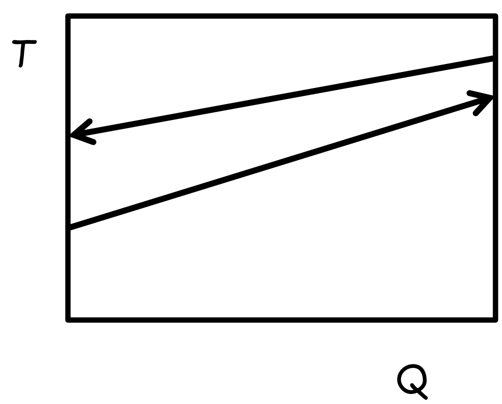
Chapter 2

Short Answer Problems

1. The LMTD factor is required for heat exchangers when the two streams do not exhibit pure counter-current flow. Most exchangers such as S&T and other arrangements fall into this category.
2. This statement is not generally true. In fact, condensers often have high heat transfer coefficients on the condensing side. One exception is for partial condensers where the majority of the condenser-side of the exchanger comprises of non-condensing gas – in this case the heat transfer coefficient on the condensing side may be limiting.
3. Turbulent flow on both sides of the heat exchanger – see Table 2.9
   a. Shell-side $\Rightarrow h \propto \mathrm{Re}^{0.6} \therefore \frac{h_2}{h_{basecase}} = \left(\frac{1.5}{1}\right)^{0.6} = 1.275$ or and increase of 27.5%
   b. Tube-side $\Rightarrow h \propto \mathrm{Re}^{0.8} \therefore \frac{h_2}{h_{basecase}} = \left(\frac{1.5}{1}\right)^{0.8} = 1.383$ or and increase of 38.3%
4. Laminar flow on both sides of the heat exchanger – – see Table 2.9
   a. Shell-side $\Rightarrow h \propto \mathrm{Re}^{0.45} \therefore \frac{h_2}{h_{basecase}} = \left(\frac{1.5}{1}\right)^{0.45} = 1.20$ or and increase of 20%
   b. Tube-side $\Rightarrow h \propto \mathrm{Re}^{1/3} \therefore \frac{h_2}{h_{basecase}} = \left(\frac{1.5}{1}\right)^{1/3} = 1.145$ or and increase of 14.5%
5. All else considered, the heat transfer coefficient for turbulent flow will be (much) higher than for laminar flow. For this reason when possible flow should be in the turbulent regime.
6. Refer to Section 2.2.1.7 – three reasons for placing a fluid on the tube side of a S&T exchanger
   2- If the fluid is corrosive
   2- If the fluid causes severe fouling or scaling
   2- If the fluid is at a much higher pressure than the other fluid
   2- If the fluid has a much higher film coefficient
   2- If the material in contact with the fluid must be an expensive alloy

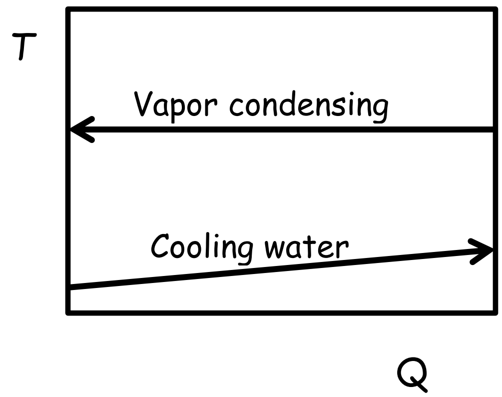
7. For placing the fluid on the shell side – the opposite reasons given in Problem 6 apply. In addition, if the fluid requires a very low pressure drop through the exchanger.
8. The reason that fins are used in some heat exchangers is to increase the surface area in contact with a fluid that has a low heat transfer coefficient. Most often, fins are used in contact with gas flows or high viscosity liquids.
9. Definitions for
   a. Baffle cut – this is the maximum distance between the edge of the baffle and the shell wall – usually expressed as a fraction or % of the shell diameter.
   b. Number of tube passes – this refers to the number of times the tube side fluid flows through the shell
   c. Tube pitch – this is the distance between the centers of adjacent tubes in the tube bundle
   d. Tube arrangement – this refers to the arrangement of the tubes in the tube bundle, such as square pitch, triangular pitch, and rotated square or triangular pitch.
10. Basic design equation for a S&T exchanger with no phase change

$$Q = UA\Delta T_{lm}F$$

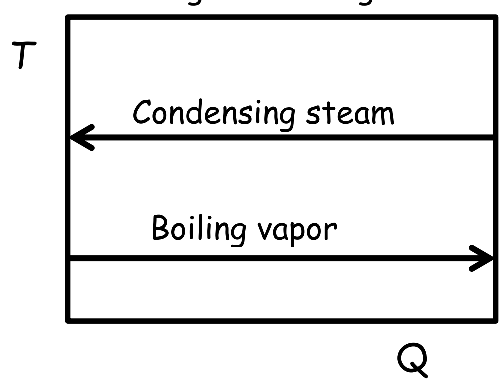


11. Draw T-Q diagrams for

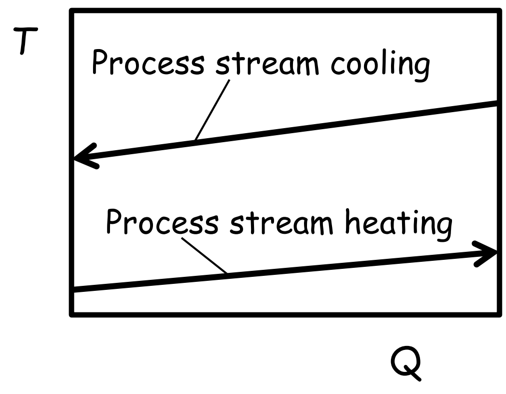
a. Condensing (pure) vapor using cooling water



b. Distillation reboiler using condensing steam as the heating media



c. Process liquid stream cooled by a another process stream



Problems to Solve

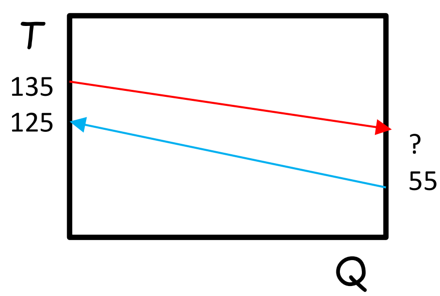
12. A process fluid (Stream 1) (C_p = 2100 J/kgK) enters a heat exchanger at a rate of 3.4 kg/s and at a temperature of 135°C. This stream is to be cooled with another process stream (Stream 2) (c_p = 2450 J/kgK) flowing at a rate of 2.65 kg/s and entering the heat exchanger at a temperature of 55°C. Determine the following (you may assume that the heat capacities of both streams are constant):

a. The exit temperature of Stream 1 if pure countercurrent flow occurs in the exchanger and the minimum approach temperature between the streams anywhere in the heat exchanger is 10°C.

$$\dot{m}_1 c_{p,1} = (3.4)(2100)/(1000) = 7.14 \text{ kW/K and } T_{1,in} = 135^{\circ}\text{C}$$

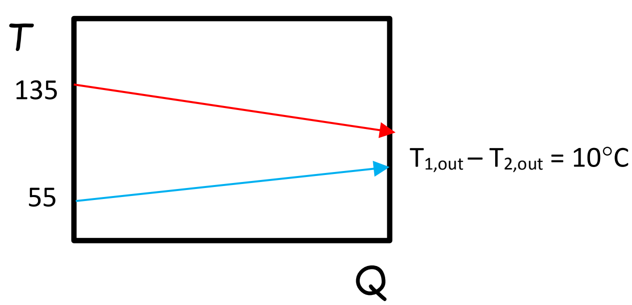
$$\dot{m}_2 c_{p,2} = (2.65)(2450)/(1000) = 6.4925 \text{ kW/K and } T_{2,in} = 55^{\circ}\text{C}$$

Because $\dot{m}_1 c_{p,1} > \dot{m}_2 c_{p,2}$ $\Delta T_1 < \Delta T_2$ and the 10°C approach will occur at the left hand side of the diagram



Energy balance gives $(135 - T_{1,out}) = \dfrac{(6.4925)}{(7.1400)}(125 - 55) \Rightarrow T_{1,out} = 71.35^{\circ}\text{C}$

b. The exit temperature of Stream 1 if pure co-current flow occurs in the exchanger and the minimum approach temperature between the streams anywhere in the heat exchanger is 10°C. For this case the outlet temperatures for each stream are unknown but they differ by 10°C.



Let the exit temperature for Stream 1 = x

And energy balance gives

$$(135 - x) = \frac{(6.4925)}{(7.1400)}((x - 10) - 55) \Rightarrow x = 101.66^{o}\,\mathrm{C}$$

and the corresponding exit temperature for Stream 2 = x – 10 = 91.66°C.

13. Repeat problem 12 except that

Stream 1: C_p = 2000 + 3(T-100) J/kgK

Stream 2: c_p = 2425 + 5(T-50) J/kgK

a. Based on the Cp values, it is not clear if the 10°C approach will be at the right hand or left hand side of the T-Q diagram. However, assume 10°C approach at the inlet of stream 1 and check the assumption. The energy balance should be written in integral form as follows:

$$\dot{m}_1 \int_{T_{1,out}}^{135} c_{p,1} dt = \dot{m}_2 \int_{55}^{125} c_{p,2} dt$$

$$3.4 \int_{T_{1,out}}^{135} (2000 + 3(T - 100)dt = 2.65 \int_{55}^{125} (2425 + 5(T - 50)dt$$

$$3.4[1700(135 - T_{1,out}) + 3(135^2 - T_{1,out}^2)/2] = 2.65[2175(125 - 55) + 5(125^2 - 55^2)/2]$$

$$\Rightarrow T_{1,out} = 63.30$$

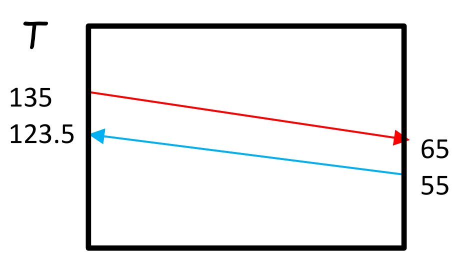
So our initial assumption was incorrect and the limiting end of the exchanger is now the right hand side – so the exit temperature for stream 1 is 65°C – reformulating the energy balance we can find the exit temperature for Stream 2 by

$$\dot{m}_1 \int_{65}^{135} c_{p,1} dt = \dot{m}_2 \int_{55}^{T_{2,out}} c_{p,2} dt$$

$$3.4 \int_{65}^{135} (2000 + 3(T - 100)dt = 2.65 \int_{55}^{T_{2,out}} (2425 + 5(T - 50)dt$$

$$3.4[1700(135 - 65) + 3(135^2 - 65^2)/2] = 2.65[2175(T_{2,out} - 55) + 5(T_{2,out}^2 - 55^2)/2]$$

$$\Rightarrow T_{2,out} = 123.5^{o}\text{C}$$



Q

b. For co-current flow, the approach of 10°C must be at the exit. Let x = $T_{1,out}$, therefore

$$\dot{m}_1 \int_{x}^{135} c_{p,1} dt = \dot{m}_2 \int_{55}^{x-10} c_{p,2} dt$$

$$3.4 \int_{x}^{135} (2000 + 3(T - 100) dt = 2.65 \int_{55}^{x-10} (2425 + 5(T - 50) dt$$

$$3.4[1700(135 - x) + 3(135^2 - x^2)/2] = 2.65[2175(x - 10 - 55) + 5((x - 10)^2 - 55^2)/2]$$

$$\Rightarrow T_{1,out} = 100.65^{o}\mathrm{C}$$

14. Find the number of shells for each case

a. Calculate R and P

$$P=\frac{(t_2-t_1)}{(T_1-t_1)}=\frac{(75-40)}{(130-40)}=\frac{35}{90}=0.3889 \quad \text{and } R=\frac{(T_1-T_2)}{(t_2-t_1)}=\frac{\dot{m}c_p}{\dot{M}C_P}=\frac{(130-90)}{(75-40)}=1.1429$$

Now use Equation 2.16 to determine the number of shell passes

$$N_{shells}=\frac{\ln\left[\frac{1-PR}{1-P}\right]}{\ln\left[\frac{1}{R}\right]}=\frac{\ln\left[\frac{1-(0.3889)(1.1429)}{1-0.3889}\right]}{\ln\left[\frac{1}{1.1429}\right]}=0.7138$$

Therefore a 1-2 S&T exchanger is needed

b. Calculate R and P

$$P=\frac{(t_2-t_1)}{(T_1-t_1)}=\frac{(73-40)}{(130-40)}=\frac{33}{90}=0.3667 \quad \text{and } R=\frac{(T_1-T_2)}{(t_2-t_1)}=\frac{\dot{m}c_p}{\dot{M}C_P}=\frac{(130-50)}{(73-40)}=2.4242$$

Now use Equation 2.16 to determine the number of shell passes

$$N_{shells}=\frac{\ln\left[\frac{1-PR}{1-P}\right]}{\ln\left[\frac{1}{R}\right]}=\frac{\ln\left[\frac{1-(0.3667)(2.4242)}{1-0.3667}\right]}{\ln\left[\frac{1}{2.4242}\right]}=1.9655$$

Rounding up to give N_{shells} = 2, therefore a 2-4 S&T exchanger is needed

c. Calculate R and P

$$P=\frac{(t_2-t_1)}{(T_1-t_1)}=\frac{(114-80)}{(130-80)}=\frac{34}{50}=0.68 \quad \text{and } R=\frac{(T_1-T_2)}{(t_2-t_1)}=\frac{\dot{m}c_p}{\dot{M}C_P}=\frac{(130-90)}{(114-80)}=1.1765$$

Now use Equation 2.16 to determine the number of shell passes

$$N_{shells}=\frac{\ln\left[\frac{1-PR}{1-P}\right]}{\ln\left[\frac{1}{R}\right]}=\frac{\ln\left[\frac{1-(0..68)(1.1765)}{1-0.68}\right]}{\ln\left[\frac{1}{1.1765}\right]}=2.8920$$

Rounding up to give N_{shells} = 3, therefore a 3-6 S&T exchanger is needed

15. At a given point in a thin walled heat exchanger, the shell-side fluid is at 100°C and the tube side fluid is at 20°C, ignoring any fouling resistances and the resistance of the wall, determine the wall temperature for the following cases:

a. The inside and outside heat transfer coefficients are equal
With the thin wall assumption we are justified in ignoring the radius effect when combining resistances and thus Equation 2.23 reduces to

$$U = \left[\frac{1}{h_o} + \frac{1}{h_i}\right]^{-1} \text{ and for this case } h_o = h_i = h \;\; \therefore U = \left[\frac{1}{h} + \frac{1}{h}\right]^{-1} = \frac{h}{2}$$

The flux of energy through the inner and out films are related by

$$\frac{Q}{A} = h(100 - T_w) = h(T_w - 20) \Rightarrow T_W = 120/2 = 60^{\circ}\text{C}$$

b. The inside coefficient has a value 3 times that of the outside coefficient

$$h_i = 3h_o \text{ and } \frac{Q}{A} = h_o(100 - T_w) = h_i(T_w - 20) \Rightarrow h_o(100 - T_w) = 3h_o(T_w - 20)$$

$$T_W = \frac{100 + (3)(20)}{4} = 40^{\circ}\text{C}$$

c. The inside coefficient has a value 1/3 that of the outside coefficient

$$h_i = \frac{h_o}{3} \text{ and } \frac{Q}{A} = h_o(100 - T_w) = h_i(T_w - 20) \Rightarrow h_o(100 - T_w) = \frac{h_o}{3}(T_w - 20)$$

$$T_W = \frac{100 + (1/3)(20)}{4/3} = 80^{\circ}\text{C}$$

d. The inside coefficient is limiting $\Rightarrow T_W = 100^{\circ}\text{C}$ or $h_i = \frac{h_o}{3}$ and

$$\frac{Q}{A} = h_o(100 - T_w) = h_i(T_w - 20) \Rightarrow h_o(100 - T_w) = \frac{h_o}{n}(T_w - 20) \text{ where } n >> 1$$

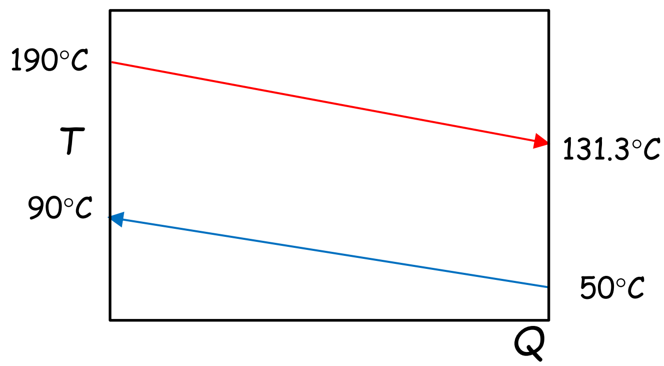
$$nh_o(100 - T_w) = h_o(T_w - 20) \;\; \Rightarrow T_w = \frac{n100 + 20}{(n+1)} \underset{n \to \infty}{\to} 100^{\circ}\text{C}$$

16. In a heat exchanger, water (c_p = 4200 J/kgK) flows at a rate of 1.5 kg/s and toluene (c_p = 1953 J/kgK) flows at a rate of 2.2 kg/s. The water enters at 50°C and leaves at 90°C and the toluene enters at 190°C. For this situation, do the following:

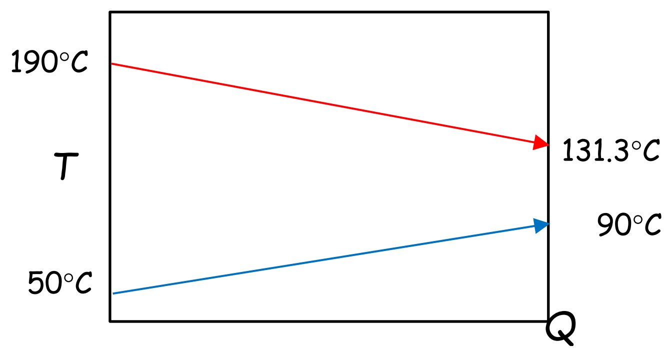
$$Q = \dot{m}c_p\Delta T \Rightarrow (1.5)(4200)(90-50) = 252\times10^3 = (2.2)(1953)(190-T_{tol,out})$$

$$T_{tol,out} = 190 - \frac{252,000}{(2.2)(1953)} = 131.3^{\circ}\mathrm{C}$$

a. Sketch the T vs. Q diagram if the flows are countercurrent



b. Sketch the T vs. Q diagram if the flows are concurrent



17. At a given location in a double pipe heat exchanger, the bulk temperature of the fluid in the annulus is 100°C and the bulk temperature of the fluid in the inner pipe is 20°C. The tube wall is very thin and the resistance due to the metal wall may be ignored. Likewise fluid fouling resistances may also be ignored. If the individual heat transfer coefficients at this point in the heat exchanger are:

h_i = 100 W/m²K and h_o = 500 W/m²K

a. What is the temperature of the wall at this location?

$$\frac{Q}{A} = h_o(100 - T_w) = h_i(T_w - 20) \Rightarrow 500(100 - T_w) = 100(T_w - 20)$$

$$\therefore \mathrm{T}_w = \frac{(500)(100) + (20)(100)}{500 + 100} = 86.67^{\mathrm{o}}\mathrm{C}$$

b. What is the heat flux across the wall at this location?

$$\frac{Q}{A} = h_o(100 - T_w) = 500(100 - 86.67) = 6.67\,\mathrm{kW/m^2}$$

c. If there was a fouling resistance of 0.001 m²K/W on the inside surface of the inner pipe, what would the temperature of the wall be at this location? From Equation 2.23 (or 2.24) and assuming $D_o \cong D_i$

$$U_o = \left[\frac{1}{h_o} + R_{fo} + \frac{D_o \ln(D_o / D_i)}{2k_w} + \frac{D_o}{D_i} R_{fi} + \frac{D_o}{D_i}\frac{1}{h_i}\right]^{-1} = \left[\frac{1}{h_o} + R_{fi} + \frac{1}{h_i}\right]^{-1}$$

$$U_o = \left[\frac{1}{500} + 0.001 + \frac{1}{100}\right]^{-1} = 76.92$$

$$\frac{Q}{A} = U\Delta T = (76.92)(100 - 20) = 6.153\ \mathrm{kW/m^2}$$

$$\frac{Q}{A} = 6153 = h_o(100 - T_w) \Rightarrow T_W = 100 - \frac{6153}{500} = 87.69^{\mathrm{o}}\mathrm{C}$$

18. A single phase fluid $(\dot{m} = 2.5 \text{ kg/s and } c_p = 1800 \text{ J/kgK})$ is to be cooled from 175°C to 75°C by exchanging heat with another single phase fluid $(\dot{m} = 2.0 \text{ kg/s and } c_p = 2250 \text{ J/kgK})$ which is to be heated from 50°C to 150°C. You have the choice of one of the following five heat exchangers:

	Shell Passes	Tube Passes
Exchanger 1	1	2
Exchanger 2	2	4
Exchanger 3	3	6
Exchanger 4	4	8
Exchanger 5	5	10

Which exchanger do you recommend using for this service – carefully explain your answer.

$$P = \frac{(t_2 - t_1)}{(T_1 - t_1)} = \frac{(150-50)}{(175-50)} = \frac{100}{125} = 0.80 \quad \text{and } R = \frac{(T_1 - T_2)}{(t_2 - t_1)} = \frac{\dot{m}c_p}{\dot{M}C_P} = \frac{(175-75)}{(150-50)} = 1.0$$

Now use Equation 2.16 for R = 1 to determine the number of shell passes

$$N_{shells} = \frac{P}{1-P} = \frac{0.8}{1-0.8} = 4$$

So both Exchangers 4 and 5 are acceptable. This jibes with the results from Figures B.1.4 and 1.5 that give the F factors of 0.8 and 0.89 for Exchangers 4 and 5, respectively. Probably want to go with Exchanger 4 as this will be a little cheaper to construct.

19. What is the critical nucleate boiling flux for water at 20 atm?

The critical pressure for water, Pc = 22.06×10^6 Pa, using the correlation of Chichelli-Bonilla, Equation 2.46 gives

$$\left[\frac{Q}{A}\right]_{\max} = 0.3673 P_c \left(\frac{P}{P_c}\right)^{0.35}\left(1-\frac{P}{P_c}\right)^{0.9}$$

$$\left[\frac{Q}{A}\right]_{\max} = (0.3673)(22.06\times10^6)\left(\frac{(20)(1.013\times10^5)}{(22.06\times10^6)}\right)^{0.35}\left(1-\frac{(20)(1.013\times10^5)}{(22.06\times10^6)}\right)^{0.9} = 3.22\,\text{MW/m}^2$$

20. Water flows inside a long ¾-in 16 BWG tube at an average temperature of 110°F. Determine the inside heat transfer coefficient for the following cases:

Properties of water at 110°F are

ρ = 61.9 lb_m/ft^3, c_p = 1.00 Btu/lb_m°F, μ = 0.424×10^{-3} lb_m/ft sec, k = 0.366 Btu/hr ft °F, Pr = $c_p\,\mu/k$ = $(1.00)(0.424\times10^{-3})(3600)/(0.366)$ = 4.17

D_i for ¾" 16BWG tube (Table 2.4) = 0.620 inch

a. Velocity = 0.1 ft/s

Re = $\rho Du/\mu$ = $(61.9)(0.620/12)(0.1)/(0.424\times10^{-3})$ = 754.3 – laminar flow

Without any further information we will assume that the limiting condition Nu_∞ = 3.66 exists.

H = (3.66)k/D = (3.66)(0.366)/(0.62/12) = 25.9 Btu/hr ft^2 °F

b. Velocity = 3 ft/s

Re = $\rho Du/\mu$ = $(61.9)(0.620/12)(3)/(0.424\times10^{-3})$ = 22,630 – turbulent flow

Use Equation 2.26 and do not account for the viscosity correction as this will be small for water

$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right]\text{Re}^{0.8}\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} = (0.023)(22630)^{0.8}(4.17)^{1/3} = 112.76$$

$$h_i = \text{Nu}\frac{k}{D_i} = \frac{(112.76)(0.366)}{(0.62/12)} = 798.8\ \text{Btu/hr ft}^2\ {}^\circ\text{F}$$

21. Use Equations 2.47 and 2.48 to estimate the heat flux and heat transfer coefficient for boiling acetone at 1 atm pressure for a temperature driving force of 10°C.

For boiling acetone at 1 atm

ρ_l = 748.6 kg/m^3, ρ_v = 4.3592 kg/m^3, c_{pl} = 2282 J/kgK, λ = 538.4 kJ/kg, μ_l = 2.36×10^{-4} Ns/m^2, k_l = 0.1522 W/mK , σ = 0.0193 N/m , and

Pr_l =$c_{pl}\mu_\lambda/k_l$ = (2282)(2.36×10^{-4})/(0.1522) = 3.54

Use a value of C_f = 0.01 and s = 1.7 (for substances other than water)

$$\frac{Q}{A} = \mu_l \lambda \left(\frac{g(\rho_l - \rho_v)}{\sigma} \right)^{1/2} \left(\frac{c_{pl} \Delta T_s}{C_f \lambda \Pr_l^s} \right)^3$$

$$= (2.36 \times 10^{-4})(538.4 \times 10^3) \left[\frac{(9.81)(748.6 - 4.3592)}{(0.0193)} \right]^{1/2} \left(\frac{(2282)(10)}{(0.01)(538.4 \times 10^3)(3.54)^{1.7}} \right)^3$$

$$\frac{Q}{A} = 9432.9 \text{ W/m}^2$$

and

$$h = \mu_l \lambda \left(\frac{g(\rho_l - \rho_v)}{\sigma} \right)^{1/2} \left(\frac{c_{pl}}{C_f \lambda \Pr_l^s} \right)^3 \Delta T_s^2 = 943.3 \text{W/m}^2\text{K}$$

22. Use the Sieder-Tate equation to determine the inside heat transfer coefficient for a fluid flowing inside a 16 ft long, 1-in tube (14 BWG) at a velocity of 2.5 ft/s that is being cooled and has an average temperature of 176°F and an average wall temperature of 104°F.

Di = 0.834 inch

Consider the following fluids:

a. Acetone (liquid)

ρ = 44.80 lb_m/ft^3, c_p = 0.5706 Btu/ lb_m°F, μ_{176} = 1.339×10^{-4} lb_m /ft s, μ_{104} = 1.806×10^{-4} lb_m /ft s, k = 0.0838 Btu/hr ft°F

$$\Pr = \frac{c_p\mu}{k} = \frac{(0.5706)(1.339\times10^{-4})}{(0.0838/3600)} = 3.28$$

$$\mathrm{Re} = \frac{\rho D u}{\mu} = \frac{(44.80)(0.834/12)(2.5)}{(1.339\times10^{-4})} = 58{,}130$$

$$\mathrm{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right]\mathrm{Re}^{0.8}\,\Pr^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$\mathrm{Nu} = 0.023\left[1+\left(\frac{(0.834)}{(12)(16)}\right)^{0.7}\right](58{,}130)^{0.8}(3.28)^{1/3}\left(\frac{1.339}{1.806}\right)^{0.14} = 217.1$$

$$h_i = \mathrm{Nu}\frac{k}{D_i} = \frac{(217.1)(0.0838)}{(0.834/12)} = 261.8\ \text{Btu/hr ft}^{2\circ}\text{F}$$

b. Iso-propanol (liquid)

ρ = 45.27 lb_m/ft^3, c_p = 0.8037 Btu/ lb_m °F, μ_{176} = 3.550×10^{-4} lb_m /ft s, μ_{104} = 9.123×10^{-4} lb_m /ft s, k = 0.0708 Btu/hr ft°F

$$\Pr = \frac{c_p\mu}{k} = \frac{(0.8037)(3.550\times10^{-4})}{(0.0708/3600)} = 14.51$$

$$\mathrm{Re} = \frac{\rho D u}{\mu} = \frac{(45.27)(0.834/12)(2.5)}{(3.55\times10^{-4})} = 22{,}160$$

$$\mathrm{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right]\mathrm{Re}^{0.8}\,\Pr^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$\mathrm{Nu} = 0.023\left[1+\left(\frac{(0.834)}{(12)(16)}\right)^{0.7}\right](22{,}160)^{0.8}(14.51)^{1/3}\left(\frac{3.550}{9.123}\right)^{0.14} = 150.5$$

$$h_i = \mathrm{Nu}\frac{k}{D_i} = \frac{(150.5)(0.0708)}{(0.834/12)} = 153.3\ \text{Btu/hr ft}^{2\circ}\text{F}$$

c. Methane at 1 atm pressure – use a velocity of 50 ft/s

ρ = 0.0346 lb_m/ft^3, c_p = 0.5676 Btu/ lb_m°F, μ_{176} = 8.619×10^{-6} lb_m /ft s, μ_{104} = 7.802×10^{-6} lb_m /ft s, k = 0.0245 Btu/hr ft°F

$$\text{Pr} = \frac{c_p \mu}{k} = \frac{(0.5676)(8.619\times10^{-6})}{(0.0245/3600)} = 0.719$$

$$\text{Re} = \frac{\rho D u}{\mu} = \frac{(0.0356)(0.834/12)(50)}{(8.619\times10^{-6})} = 13,950$$

$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right]\text{Re}^{0.8}\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$\text{Nu} = 0.023\left[1+\left(\frac{(0.834)}{(12)(16)}\right)^{0.7}\right](13,950)^{0.8}(0.719)^{1/3}\left(\frac{8.619}{7.802}\right)^{0.14} = 44.18$$

$$h_i = \text{Nu}\frac{k}{D_i} = \frac{(44.18)(0.0245)}{(0.834/12)} = 15.57 \text{ Btu/hr ft}^{2\text{o}}\text{F}$$

d. Compare the results using the Dittus-Boelter equation

For a.

$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(58,130)^{0.8}(3.28)^{0.3} = 212.9$$

$$h_i = \text{Nu}\frac{k}{D_i} = (212.9)\frac{(0.0838)}{(0.834/12)} = 256.7\,\text{Btu/hrft}^{2\text{o}}\text{F} \sim \text{ 2\% difference}$$

For b.

$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(22,157)^{0.8}(14.51)^{0.3} = 153.7$$

$$h_i = \text{Nu}\frac{k}{D_i} = (153.7)\frac{(0.0708)}{(0.834/12)} = 156.6\,\text{Btu/hrft}^{2\text{o}}\text{F} \sim \text{ -2\% difference}$$

For c.

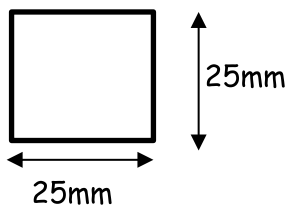
$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(13,950)^{0.8}(0.72)^{0.3} = 43.09$$

$$h_i = \text{Nu}\frac{k}{D_i} = (43.09)\frac{(0.0245)}{(0.834/12)} = 15.19\,\text{Btu/hrft}^{2\text{o}}\text{F} \sim \text{ 2\% difference}$$

Therefore the corrections for tube entrance and wall viscosity are v. small for these cases.

23. Water at 5 atm pressure and 30°C flows at an inlet velocity of 1 m/s into a square duct 2.5 m in length that has a cross section of 25mm by 25mm. The wall of the duct is maintained by condensing steam at a temperature of 120°C. What will the exit temperature of the water be when it exits the duct?

ρ = 996 kg/m^3, μ_{30} = 8.291×10^{-4} Pa.s, μ_{120} = 2.19×10^{-4} Pa.s, k = 0.6130 W/m°C, c_p = 4187 J/kg°C



$$D_H = 4\frac{\text{flow area}}{\text{wetted perimeter}} = 4\frac{(25)(25)}{4(25)} = 25\text{ mm}$$

Assume flow is turbulent – check this when Re is found

$$\text{Re}_H = \frac{\rho u D_H}{\mu} = \frac{(996)(1)(0.025)}{(8.291\times10^{-4})} = 30,030$$ - turbulent use Seider-Tate Eqn with Re_H

$$\text{Pr} = \frac{c_p\mu}{k} = \frac{(4187)(8.291\times10^{-4})}{(0.6130)} = 5.66$$

$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_H}{L}\right)^{0.7}\right]\text{Re}_H^{0.8}\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$\text{Nu} = 0.023\left[1+\left(\frac{0.025}{2.5}\right)^{0.7}\right](30,030)^{0.8}(5.66)^{1/3}\left(\frac{8.291\times10^{-4}}{2.19\times10^{-4}}\right)^{0.14} = 196.2$$

$$h_i = \text{Nu}\frac{k}{D_i} = 195.2\frac{(0.613)}{(0.025)} = 4811\,\text{W/m}^{2\text{o}}\text{C}$$

Since the wall temperature is constant, the outside heat transfer coefficient can be considered to be ∞ and $U = h_i$. The following analysis assumes that the physical properties of water are constant over the length of the channel.

$$Q = \dot{m} c_p (T_{out} - 30) = UA \frac{(T_{out} - 30)}{\ln \dfrac{(120 - 30)}{(120 - T_{out})}}$$

$$Q = (0.025)^2 (1)(996)(4187)(T_{out} - 30) = (4811)(4)(0.025)(2.5) \frac{(T_{out} - 30)}{\ln \dfrac{(90)}{(120 - T_{out})}}$$

$$\Rightarrow \ln \frac{(90)}{(120 - T_{out})} = \frac{(4811)(4)(0.025)(2.5)}{(0.025)^2 (1)(996)(4187)} = 0.4615 \Rightarrow T_{out} = 63.3^{\circ}\text{C}$$

24. A shell-and-tube condenser contains six rows of five copper tubes per row on a square pitch. The tubes are ¾-in, 14 BWG and 3 m long. Cooling water flows through the tubes such that h_i = 2000 W/m^{2}K. The water flow is high so that the temperature on the tube side may be assumed to be constant at 35°C. Pure saturated steam at 2 bar is condensing on the shell side. Determine the capacity of the condenser (Q in kW) if the condenser tubes are oriented (a) vertically, and (b) horizontally.

a. Vertical Arrangement – assume $Re_\delta < 1800$ and use Equation 2.64
D_i = 0.584 inch = 0.01483 m, D_o = 3/4 inch = 0.01905 m, h_i = 2000 W/m^{2}K, k_{cu} = 377 W/mK (assume T_w = 100°C). For saturated steam at 2 bar, T_{sat} = 120.2°C
We need T_w to be able to apply Eqn 264 or 2.66 – so follow the algorithm shown in Example 2.14:
i) guess h_o
ii) calculate film temperature and film properties
iii) apply equation for h_o and iterate.
Assume h_o = 2000 W/m^{2}K

$$U_o = \left[\frac{1}{h_o} + \cancel{R_{fo}} + \frac{D_o \ln(D_o/D_i)}{2k_w} + \frac{D_o}{D_i}\cancel{R_{fi}} + \frac{D_o}{D_i}\frac{1}{h_i}\right]^{-1}$$ and

$$\frac{Q}{A_o} = h_o(T_{sat} - T_{wall}) = U_o(T_{sat} - T_{water})$$ rearranging gives

$$\frac{(120.2 - T_{wall})}{\frac{1}{2000}} = \frac{(120.2-35)}{\frac{1}{2000} + \frac{(0.01905)\ln(0.01905/0.01483)}{(2)(377)} + \frac{(0.01905)}{(0.01483)}\frac{1}{2000}}$$

$$(120.2 - T_{wall}) = \frac{(870.6)(120.2-35)}{(2000)} \Rightarrow T_{wall} = 83.1^{\circ}\mathrm{C}$$

$T_{film} = T_{sat} - 0.75(T_{sat} - T_{wall})$ = 120.2 – 0.75(120.2 – 83.1) = 92.4°C

For water at 92.4°C, k_l = 0.6721 W/mK, ρ_l = 963.2 kg/m^3, λ = 2201.6 kJ/kg (at 2 bar), $\mu_l = 3.062 \times 10^{-4}$ kgm/s, $c_{p,l}$ = 4215.8J/kgK, ρ_v = 1.129 kg/m^3

$$\lambda' = \lambda + 0.68 c_{p,l}(T_{sat} - T_{wall}) = 2201.6 + \frac{0.68(4215.8)(120.2-83.1)}{1000} = 2307.9 \text{ kJ/kg}$$

$$\text{Nu} = \frac{\bar{h}_c L}{k_l} = 1.13\left[\frac{\rho_l g(\rho_l - \rho_v)\lambda' L^3}{\mu_l k_l (T_{sat} - T_w)}\right]^{1/4}$$

$$\text{Nu} = 1.13\left[\frac{(963.2)(9.81)(963.2-1.129)(2307.9)(3)^3}{(3.062\times10^{-4})(0.6721)(120.3-83.1)}\right]^{1/4} = 3317$$

$$\bar{h}_c = \text{Nu}\frac{k_l}{L} = (3317)\frac{(0.6721)}{(3)} = 743.0\,\text{Wm}^2\text{K}$$

Use this new value of $h_o = \bar{h}_c$ and iterate – results shown in Table below.

Converged Solution gives:

h_o = 620.2 W/m^{2}K and U_o = 442.3 W/m^{2}K

$$Q = U_o A_o \Delta T = (442.3)(30)(\pi)(0.01905)(3)(120.2-35) = 203.0\,\text{kW}$$

Check for Re_δ

$$\text{Re}_\delta = \frac{4\dot{m}}{\mu_l W} = \frac{4(Q/\lambda')}{\mu_l(\pi D_o)n_{tubes}} = \frac{4(203.0/2375.13)}{(3.85\times10^{-4})(\pi)(0.01905)(30)} = 495$$ <1800 so

correct equation used – if this were not the case, you would not need to iterate using Eqn 2.64.

<u>Q = 203 kW</u>

<u>Results for Iterative Solution to Problem 24 – Vertical Condenser Tubes</u>

	Iteration 1		Iteration 2		Iteration 3		Iteration 4		Iteration 5	
Guess h_o	2000	W/m^{2}K	743.0	W/m^{2}K	634.342	W/m^{2}K	621.848	W/m^{2}K	620.416	W/m^{2}K
h_i	2000	W/m^{2}K	2000	W/m^{2}K	2000	W/m^{2}K	2000	W/m^{2}K	2000	W/m^{2}K
T_{sat}	120.2	C	120.2	C	120.2	C	120.2	C	120.2	C
T_{water}	35	C	35	C	35	C	35	C	35	C
k_{cu}	377	W/mK	377	W/mK	377	W/mK	377	W/mK	377	W/mK
D_o	0.01905	m	0.01905	m	0.01905	m	0.01905	m	0.01905	m
D_i	0.01483	m	0.01483	m	0.01483	m	0.01483	m	0.01483	m
U_o	870.621		501.4		449.429		443.122		442.394	
T_{wall}	83.1116		62.708		59.836		59.4875		59.4472	
T_{film}	92.3837		77.081		74.927		74.6656		74.6354	
λ	2201.6	kJ/kg	2201.6	kJ/kg	2201.6	kJ/kg	2201.6	kJ/kg	2201.6	kJ/kg
c_{pl}	4215.8	J/kgK	4201.9	J/kgK	4200.5	J/kgK	4200.4	J/kgK	4200.4	J/kgK
μ_l	3.06E-04	kgm/s	3.72E-04	kgm/s	3.83E-04	kgm/s	3.84E-04	kgm/s	3.85E-04	kgm/s
ρ_l	963.2	kg/m^3	973.2	kg/m^3	974.5	kg/m^3	974.7	kg/m^3	974.7	kg/m^3
ρ_v	1.129	kg/m^3	1.129	kg/m^3	1.129	kg/m^3	1.129	kg/m^3	1.129	kg/m^3
k_l	0.6721	W/mK	0.662	W/mK	0.6604	W/mK	0.6602	W/mK	0.6601	W/mK
λ'	2307.92	kJ/kg	2365.87	kJ/kg	2374.02	kJ/kg	2375.01	kJ/kg	2375.13	kJ/kg
Nu	3316.7		2874.7		2824.9		2819.2		2818.5	
h_c	743.0		634.3		621.8		620.4		620.2	

b. Horizontal Arrangement – use Equation 2.67

$$\mathrm{Nu} = \frac{\bar{h}_c D_o}{k_l} = 0.728\left[\frac{\rho_l g(\rho_l - \rho_v)\lambda' D_o^3}{\mu_l k_l (T_{sat} - T_w)}\right]^{1/4}$$ Following the same algorithm from part a and Example 2.14 – assume a value of h_o = 2000 W/m^{2}K

$$\frac{Q}{A_o} = h_o(T_{sat} - T_{wall}) = U_o(T_{sat} - T_{water})$$ rearranging gives

$$\frac{(120.2 - T_{wall})}{\dfrac{1}{2000}} = \frac{(120.2-35)}{\dfrac{1}{2000} + \dfrac{(0.01905)\ln(0.01905/0.01483)}{(2)(377)} + \dfrac{(0.01905)}{(0.01483)}\dfrac{1}{2000}}$$

$$(120.2 - T_{wall}) = \frac{(870.6)(120.2-35)}{(2000)} \Rightarrow T_{wall} = 83.1^{\mathrm{o}}\mathrm{C}$$

$T_{film} = T_{sat} - 0.75(T_{sat} - T_{wall}) = 120.2 - 0.75(120.2 - 83.1) = 92.4$°C

For water at 92.4°C, k_l = 0.6721 W/mK, ρ_l = 963.2 kg/m^3, λ = 2201.6 kJ/kg (at 2 bar), μ_l = 3.062×10^{-4} kgm/s, $c_{p,l}$ = 4215.8J/kgK, ρ_v = 1.129 kg/m^3

$$\lambda' = \lambda + 0.68 c_{p,l}(T_{sat} - T_{wall}) = 2201.6 + \frac{0.68(4215.8)(120.2-83.1)}{1000} = 2307.9 \text{ kJ/kg}$$

$$\text{Nu} = \frac{\bar{h}_c D_o}{k_l} = 0.728\left[\frac{\rho_l g(\rho_l - \rho_v)\lambda' D_o^3}{\mu_l k_l (T_{sat} - T_w)}\right]^{1/4}$$

$$\text{Nu} = 0.728\left[\frac{(963.2)(9.81)(963.2-1.129)(2307.9)(0.01905)^3}{(3.062\times10^{-4})(0.6721)(120.3-83.1)}\right]^{1/4} = 48.1$$

$$\bar{h}_c = \text{Nu}\frac{k_l}{D_o} = 48.1\frac{(0.6721)}{(0.01905)} = 1695.8\,\text{Wm}^2\text{K}$$

Starting the second iteration

$$\frac{(120.2 - T_{wall})}{\frac{1}{(1695.8)}} = \frac{(120.2-35)}{\frac{1}{(1695.8)} + \frac{(0.01905)\ln(0.01905/0.01483)}{(2)(377)} + \frac{(0.01905)}{(0.01483)}\frac{1}{2000}}$$

$$(120.2 - T_{wall}) = \frac{(807.6)(120.2-35)}{(1695.8)} \Rightarrow T_{wall} = 79.6^{o}\text{C}$$

T_{film} = T_{sat} - 0.75(T_{sat}-T_{wall}) = 120.2 – 0.75(120.2 – 79.6) = 89.8°C

For water at 89.8°C, k_l = 0.6706 W/mK, ρ_l = 965.0 kg/m^3, λ = 2201.6 kJ/kg (at 2 bar), μ_l = 3.158×10^{-4} kgm/s, $c_{p,l}$ = 4213.3J/kgK, ρ_v = 1.129 kg/m^3

$$\lambda' = \lambda + 0.68 c_{p,l}(T_{sat} - T_{wall}) = 2201.6 + \frac{0.68(4213.3)(120.2-79.6)}{1000} = 2317.8 \text{ kJ/kg}$$

$$\text{Nu} = \frac{\bar{h}_c D_o}{k_l} = 0.728\left[\frac{\rho_l g(\rho_l - \rho_v)\lambda' D_o^3}{\mu_l k_l (T_{sat} - T_w)}\right]^{1/4}$$

$$\text{Nu} = 0.728\left[\frac{(965.0)(9.81)(965.0-1.129)(2317.8)(0.01905)^3}{(3.158\times10^{-4})(0.6706)(120.3-79.6)}\right]^{1/4} = 46.8$$

$$\bar{h}_c = \text{Nu}\frac{k_l}{D_o} = 46.8\frac{(0.6706)}{(0.01905)} = 1646.0\,\text{Wm}^2\text{K}$$

Additional iterations are shown in the following table

Converged Solution gives:
h_o = 1635.3 W/m^{2}K and U_o = 793.6 W/m^{2}K

$$Q = U_o A_o \Delta T = (793.6)(30)(\pi)(0.01905)(3)(120.2-35) = 364.18\,\text{kW}$$

Check for Re$_\delta$

$$\mathrm{Re}_\delta = \frac{4\dot{m}}{\mu_l W} = \frac{4(Q/\lambda')}{\mu_l (2) L_{tube} n_{tubes}} = \frac{4(364.18/2320.03)}{(3.182\times10^{-4})(2)(3)(30)} = 439$$ <1800 so correct equation used

Q = 364.18 kW

Results for Iterative Solution to Problem 24 – Horizontal Condenser Tubes

	Iteration 1		Iteration 2		Iteration 3		Iteration 4	
Guess h_o	2000	W/m^2K	1695.8	W/m^2K	1646.0	W/m^2K	1637.0	W/m^2K
h_i	2000	W/m^2K	2000	W/m^2K	2000	W/m^2K	2000	W/m^2K
T_{sat}	120.2	C	120.2	C	120.2	C	120.2	C
T_{water}	35	C	35	C	35	C	35	C
k_{cu}	377	W/mK	377	W/mK	377	W/mK	377	W/mK
D_o	0.01905	m	0.01905	m	0.01905	m	0.01905	m
D_i	0.01483	m	0.01483	m	0.01483	m	0.01483	m
U_o	870.621		807.561		796.0789		793.983	
T_{wall}	83.1116		79.6268		78.9923		78.8765	
T_{film}	92.3837		89.7701		89.29422		89.2074	
λ	2201.6	kJ/kg	2201.6	kJ/kg	2201.6	kJ/kg	2201.6	kJ/kg
c_{pl}	4215.8	J/kgK	4213.3	J/kgK	4212.8	J/kgK	4212.4	J/kgK
μ_l	3.06E-04	kgm/s	3.16E-04	kgm/s	3.178E-04	kgm/s	3.18E-04	kgm/s
ρ_l	963.2	kg/m^3	965	kg/m^3	965.3	kg/m^3	965.4	kg/m^3
ρ_v	1.129	kg/m^3	1.129	kg/m^3	1.129	kg/m^3	1.129	kg/m^3
k_l	0.6721	W/mK	0.6706	W/mK	0.6703	W/mK	0.6702	W/mK
λ'	2307.92	kJ/kg	2317.84	kJ/kg	2319.648	kJ/kg	2319.97	kJ/kg
Nu	48.1		46.8		46.5		46.5	
h_c	1695.8		1646.0		1637.0		1635.3	

25. Liquid dimethyl ether (DME) flows across the outside of a bank of tubes. It enters at 100°C and leaves at 50°C. The DME enters at a flowrate of 20 kg/s, the shell diameter is 15-in, the baffle spacing is 6-in, the baffle cut (*BC*) is 15%, and 1-in OD tubes on a 1.25-in pitch are used. The fluid inside the tubes may be assumed to be at a constant temperature of 35°C (cooling water) and the inside coefficient is expected to be much higher than the shell-side coefficient and thus the wall temperature may be taken as 35°C.

Use Kern's method to determine the average heat transfer coefficient for the shell side for the following arrangements:

BC = 15%, D_o = 0.0254 m, p = 1.25 inch = 0.03175 m, D_{shell} = 15 inch = 0.381 m, L_b = 6 inch = 0.1524 m,

Properties of DME

DME @50°C	DME @100°C	DME – average property
ρ = 612.1 kg/m^3	ρ = 494.3 kg/m^3	ρ = 553.2 kg/m^3
c_p = 2460 J/kgK	c_p = 2803 J/kgK	c_p = 2631.5 J/kgK
k = 0.1296 W/mK	k = 0.1014W/mK	k = 0.1155W/mK
μ = 99.13×10^{-6} kg/ms	μ = 84.02×10^{-6} kg/ms	μ = 91.58×10^{-6} kg/ms
$\mu_{35°C}$ = 108.8×10^{-6} kg/ms		$\mathrm{Pr} = \frac{c_p \mu}{k} = 2.087$

a. square pitch

From Equations 2.37-2.43

$$D_{H,s} = \frac{1.273p^2 - D_o^2}{D_o} = \frac{(1.273)(0.03175)^2 - (0.0254)^2}{(0.0254)} = 0.02512 \text{ m}$$

$$A_s = \frac{D_s L_b (p - D_o)}{p} = \frac{(0.381)(0.1524)(0.03175 - 0.0254)}{(0.03175)} = 0.01161 \text{ m}^2$$

Shell-side superficial mass velocity, $G_s = \frac{\dot{m}_s}{A_s} = \frac{20}{0.01161} = 1722.2 \text{ kg/m}^2\text{s}$

Shell-side velocity, $u_s = \frac{\dot{m}_s}{\rho A_s} = \frac{G_s}{\rho} = \frac{(1722.2)}{(553.2)} = 3.1132 \text{ m/s}$

Shell-side Reynolds number,

$$\text{Re}_s = \frac{G_s D_{H,s}}{\mu} = \frac{u_s \rho D_{H,s}}{\mu} = \frac{(3.1132)(553.2)(0.02512)}{(91.58\times10^{-6})} = 472,440$$

The average heat transfer coefficient for the shell side of the exchanger is given by:

$$\text{Nu} = \frac{h_s D_{H,s}}{k_f} = j_h \,\text{Re}\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

where for $100 < \text{Re} < 1\times10^6$

$$j_h = 1.2492(BC)^{-0.329}\,\text{Re}^{-0.4696} = (1.2492)(15)^{-0.329}(472{,}530)^{-0.4696} = 0.001109$$

$$\therefore \text{Nu} = \frac{h_s D_{H,s}}{k_f} = j_h \,\text{Re}\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} = (0.001109)(472,530)(2.087)^{1/3}\left[\frac{(91.58\times10^{-6})}{108.8\times10^{-6}}\right]^{0.14}$$

$$\text{Nu} = 653.7 \Rightarrow h_s = \text{Nu}\frac{k_f}{D_{H,s}} = (653.7)\frac{(0.1155)}{(0.02512)} = 3005\ \text{W/m}^2\text{K}$$

h_s = 3005 W/m^{2}K

b. equilateral triangular pitch
Following same equations as in part a. except we use Equation 2.38

$$D_{H,s} = \frac{1.103p^2 - D_o^2}{D_o} = \frac{(1.103)(0.03175)^2 - (0.0254)^2}{(0.0254)} = 0.01838\,\text{m}$$

$$A_s = \frac{D_s L_b (p - D_o)}{p} = \frac{(0.381)(0.1524)(0.03175 - 0.0254)}{(0.03175)} = 0.01161\ \text{m}^2$$

Shell-side superficial mass velocity, $G_s = \frac{\dot{m}_s}{A_s} = \frac{20}{0.01161} = 1722.2\ \text{kg/m}^2\text{s}$

Shell-side velocity, $u_s = \frac{\dot{m}_s}{\rho A_s} = \frac{G_s}{\rho} = \frac{(1722.2)}{(553.2)} = 3.1132\,\text{m/s}$

Shell-side Reynolds number,

$$\text{Re}_s = \frac{G_s D_{H,s}}{\mu} = \frac{u_s \rho D_{H,s}}{\mu} = \frac{(3.1132)(553.2)(0.01838)}{(91.58\times10^{-6})} = 345,560$$

The average heat transfer coefficient for the shell side of the exchanger is given by:

$$\mathrm{Nu} = \frac{h_s D_{H,s}}{k_f} = j_h \,\mathrm{Re}\,\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

where for $100 < \mathrm{Re} < 1\times10^6$

$$j_h = 1.2492(BC)^{-0.329}\,\mathrm{Re}^{-0.4696} = (1.2492)(15)^{-0.329}(345{,}750)^{-0.4696} = 0.001285$$

$$\therefore \mathrm{Nu} = \frac{h_s D_{H,s}}{k_f} = j_h \,\mathrm{Re}\,\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} = (0.001285)(345{,}560)(2.087)^{1/3}\left[\frac{91.58\times10^{-6}}{108.8\times10^{-6}}\right]^{0.14}$$

$$\mathrm{Nu} = 553.8 \Rightarrow h_s = \mathrm{Nu}\frac{k_f}{D_{H,s}} = (553.8)\frac{(0.1155)}{(0.01838)} = 3481\ \mathrm{W/m^2K}$$

h_s = 3482 W/m^{2}K

Triangular pitch gives ~16% increase in h_s

26. A double pipe heat exchanger is comprised of a length of 1 in sch 40 pipe inside an equal length of 3 in sch 40 pipe. Water flows at a velocity of 1.393 m/s in the annular region between the pipes and enters the heat exchanger at 30°C. Oil flows in the inner pipe at an average velocity of 1 m/s and enters at 100°C. The water and oil flow counter currently.

Properties of Water and Oil

Water @30°C	Oil @100°C	Oil @50°C
ρ = 998 kg/m^3	ρ = 690 kg/m^3	ρ = 727 kg/m^3
c_p = 4216 J/kgK	c_p = 2421 J/kgK	c_p = 2201 J/kgK
k = 0.60 W/mK	k = 0.1179 W/mK	k = 0.1295 W/mK
μ = 700×10^{-6} kg/ms	μ = 511×10^{-6} kg/ms	μ = 945×10^{-6} kg/ms

Determine how long the pipe lengths must be in order for the oil to leave the exchanger at 50°C? You may assume that both fluids are clean and that there is no fouling resistance.

Approach – using Equations 2.27 and 2.28 – this makes the problem non-iterative

Inside HT Coefficient - oil

Average properties are ρ = 708.5 kg/m^3, c_p = 2311 J/kgK, k = 0.1237 W/mK, μ = 72.8×10^{-6} kg/ms, u = 1 m/s

For a 1" sch 40 pipe, D_i = 1.049 inch = 0.02665 m, D_o = 1.315 inch = 0.03340 m

For a 3" sch 40 pipe D_i = 3.068 inch = 0.07793 m

$$A_i = \frac{\pi D_i^2}{4} = \frac{\pi}{4}(0.02665)^2 = 5.576\times10^{-4}\ \text{m}^2$$
$$\dot{m}_{oil} = A_i\rho_{oil}u_i = (5.576\times10^{-4})(708.5)(1) = 0.3950\ \text{kg/s}$$

$$\mathrm{Re}_i = \frac{D_i u_i \rho_{oil}}{\mu_{oil}} = \frac{(0.02665)(1)(708.5)}{(728\times10^{-6})} = 25,930$$

$$\mathrm{Pr}_{oil} = \frac{c_{p,oil}\mu_{oil}}{k_{oil}} = \frac{(2311)(728\times10^{-6})}{(0.1237)} = 13.60$$

$$\mathrm{Nu}_i = \frac{h_i D_i}{k_{oil}} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.3} = 170.9$$

$$h_i = \mathrm{Nu}_i \frac{k_{oil}}{D_i} = 170.9\frac{(0.1237)}{(0.02665)} = 794\ \mathrm{W/m^2K}$$

Outside HT Coefficient - water

$$A_o = \frac{\pi}{4}(D_{i-o}^2 - D_{o-i}^2) = \frac{\pi}{4}(0.07793^2 - 0.03340)^2 = 3.8932\times10^{-3}\ \mathrm{m}^2$$

$$\dot{m}_{water} = A_o \rho_{water} u_o = (3.8932\times10^{-3})(998)(1.393) = 5.412\ \mathrm{kg/s}$$

Energy Balance on System

$$\dot{m}_{water} c_{p,water}(T_{w,out} - 30) = \dot{m}_{oil} c_{p,oil}(100-50)$$

$$\Rightarrow T_{w,out} = 30 + \frac{\dot{m}_{oil} c_{p,oil}}{\dot{m}_{water} c_{p,water}}(50) = 30 + \frac{(0.3950)(2311)}{(5.412)(4216)}(50) = 32.0^{\mathrm{o}}\mathrm{C}$$

So the water properties given at 30°C will be close enough

For an annulus – Equation 2.28

$$D_H = 4\frac{\text{flow area}}{\text{wetted perimeter}} = 4\frac{\pi(D_{i-o}^2 - D_{o-i}^2)}{4}\frac{1}{\pi(D_{i-o} + D_{o-i})}$$

$$D_H = D_{i-o} - D_{o-i} = (0.07793) - (0.03340) = 0.04453\ \mathrm{m}$$

$$\mathrm{Re}_o = \frac{D_H u_o \rho_{water}}{\mu_{water}} = \frac{(0.04453)(1.393)(998)}{(700\times10^{-6})} = 88,430$$

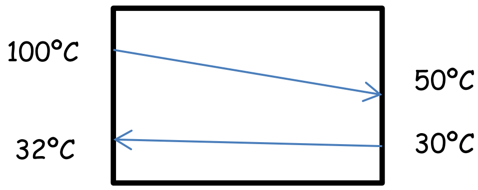
$$\mathrm{Pr}_{water} = \frac{c_{p,water}\mu_{water}}{k_{water}} = \frac{(4216)(700\times10^{-6})}{(0.60)} = 4.92$$

$$\mathrm{Nu}_o = \frac{h_o D_H}{k_{water}} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}\left(\frac{D_{i-o}}{D_{o-i}}\right)^{0.45} = (0.023)(88,430)^{0.8}(4.92)^{0.4}\left(\frac{0.07793}{0.03340}\right)^{0.45} = 577.2$$

$$h_o = \mathrm{Nu}_o \frac{k_{oil}}{D_H} = 577.2\frac{(0.60)}{(0.04453)} = 7778\ \mathrm{W/m^2K}$$

From Equation 2.23 eliminating terms that are zero gives

$$U_o = \left[\frac{1}{h_o} + \frac{D_o}{D_i}\frac{1}{h_i}\right]^{-1} = \left[\frac{1}{7778} + \frac{(0.07793)}{(0.03340)}\frac{1}{(794)}\right]^{-1} = 325.9 \text{ W/m}^2\text{K}$$



$$\Delta T_{lm} = \frac{(100-32)-(50-30)}{\ln\frac{(100-32)}{(50-30)}} = 39.22^{\circ}\text{C}$$

$$Q = \dot{m}c_p\Delta T = (0.3950)(2311)(100-50) = 45.64\times10^3 \text{ W}$$

$$Q = U_o A_o \Delta T_{lm} \Rightarrow A_o = \frac{Q}{U_o \Delta T_{lm}} = \frac{45.64\times10^3}{(325.9)(39.22)} = 3.57 \text{ m}^2$$

$$A_o = \pi L D_{o-i} \Rightarrow L = \frac{(3.57)}{\pi(0.03340)} = 34.0 \text{ m}$$

L = 34 m

27. A 3m long 1.25-in BWG 14 copper tube is used to condense ethanol at 3 bar pressure. Cooling water at 30°C flows through the inside of the tube at a high rate such that the wall temperature may be assumed to be at 30°C and the inside coefficient h_i>> h_o.

$P_{sat,\ ethanol}$ = 3 bar, T_{sat} = 108.7°C

Determine how much vapor will condense (kg/h) for clean (no fouling resistance) if,

a. The tube is oriented vertically
   assume Re_δ < 1800 and use Equation 2.64
   D_i = 1.084 inch = 0.02753 m, D_o = 1.25 inch = 0.03175 m, h_i = 2000 W/m^{2}K. We need T_w to be able to apply Eqn 264 or 2.66 – so follow the algorithm shown in Example 2.14:
   i) guess h_o = skip this step since from the problem statement, T_{wall} = 30°C
   ii) calculate film temperature and film properties

   T_f = T_{sat} - 0.75(T_{sat} – T_{wall}) = 108.7-0.75(108.7 – 30) = 49.7°C
   Properties of ethanol at 49.7°C: k_l = 0.1616 W/mK, $c_{p,l}$ = 2667 J/kgK, μ_l = 6.928×10^{-4}, ρ_l = 763.2 kg/m^3, ρ_v = 4.585 kg/m^3, λ = 782.5 kJ/kg

$$\lambda' = \lambda + 0.68 c_{p,l}(T_{sat} - T_w) = 782.5 + (0.68)(2.667)(108.7 - 30) = 925.2\ \text{kJ/kg}$$

$$\text{Nu} = \frac{\bar{h}_c L}{k_l} = 1.13\left[\frac{\rho_l g(\rho_l - \rho_v)\lambda' L^3}{\mu_l k_l (T_{sat} - T_w)}\right]^{1/4}$$

$$\text{Nu} = 1.13\left[\frac{(763.2)(9.81)(763.2 - 4.585)(925.2\times 10^3)(3)^3}{(6.928\times 10^{-4})(0.1616)(108.7 - 30)}\right]^{1/4} = 12,729$$

$$\bar{h}_c = \text{Nu}\frac{k_l}{L} = (12,729)\frac{(0.1616)}{(3)} = 685.7\ \text{W/m}^2\text{K}$$

A_o = $\pi D_o L$= π(0.03175)(3) = 0.2992 m^2

$$Q = U_o A_o (T_{sat} - T_{wall}) = (685.7)(0.2992)(108.7 - 30) = 16.15\ \text{kW}$$

$$\dot{m} = \frac{Q}{\lambda'} = \frac{16.15}{925.2}(3600) = 62.83\ \text{kg/h}$$

Check for Re_δ

$$\mathrm{Re}_\delta = \frac{4\dot{m}}{\mu_l W} = \frac{4\dot{m}}{\mu_l(\pi D_o)} = \frac{4(62.83/3600)}{(6.928\times10^{-4})(\pi)(0.03175)} = 1010 \ < 1800$$

<u>m = 62.83 kg/h</u>

b. The tube is oriented horizontally

$$Nu = \frac{\bar{h}_c D_o}{k_l} = 0.728\left[\frac{\rho_l g(\rho_l - \rho_v)\lambda' D_o^3}{\mu_l k_l (T_{sat} - T_w)}\right]^{1/4}$$

$$Nu = 0.728\left[\frac{(763.2)(9.81)(763.2-4.585)(925.2\times10^3)(0.03175)^3}{(6.928\times10^{-4})(0.1616)(108.7-30)}\right]^{1/4} = 289.9$$

$$\bar{h}_c = (289.9)(0.1616)/(0.03175) = 1475.7\ \mathrm{W/m^2K}$$

$A_o = \pi D_o L = \pi(0.03175)(3) = 0.2992\ m^2$

$$Q = U_o A_o (T_{sat} - T_{wall}) = (1475.7)(0.2992)(108.7-30) = 34.75\ \mathrm{kW}$$

$$\dot{m} = \frac{Q}{\lambda'} = \frac{34.75}{925.2}(3600) = 135.2\ \mathrm{kg/h}$$

Check for Re_δ

$$\mathrm{Re}_\delta = \frac{4\dot{m}}{\mu_l W} = \frac{4\dot{m}}{\mu_l(2L)} = \frac{4(135.2/3600)}{(6.928\times10^{-4})(2)(0.03175)} = 36.1 \ < 1800$$

<u>m = 135.2 kg/h</u>

28. Air (1 atm and 30°C) flows crosswise over a bare copper tube (1-in BWG 16). The approach velocity of the air is 20 m/s. Water enters the tube at 140°C and leaves the tube at an average temperature of 80°C. The average velocity of the water in the tubes is 1 m/s.

Determine the length of tube required to cool the water to the desired 80°C.

Properties of water and air

Water @140°C	Water @80°C	Air @30°C	Air @70°C
ρ = 925.6 kg/m^3	ρ = 971.4 kg/m^3	ρ = 1.1491 kg/m^3	ρ = 1.0148 kg/m^3
c_p = 4305 J/kgK	c_p = 4205 J/kgK	c_p = 1003 J/kgK	c_p = 1005 J/kgK
k = 0.6855 W/mK	k = 0.6641 W/mK	k = 0.0263 W/mK	k = 0.0291 W/mK
μ = 193.2×10^{-6} kg/ms	μ = 357.8×10^{-6} kg/ms	μ = 18.7×10^{-6} kg/ms	μ = 20.5×10^{-6}kg/ms

Using average properties of water: ρ = 948.5 kg/m^3 c_p = 4255 J/kgK k = 0.6748 W/mK, μ = 276×10^{-6} kg/ms

Assume that the heat transfer coefficient for the air is very low compared to the inside coefficient and hence the wall temperature of the tube will be the temperature of the water (this will be checked later). The average wall temperature is (140+80)/2 = 110°C. The film temperature = (T_{air} + T_{wall})/2= 70°C and properties are given above. For air at the average wall temperature (110°C), we have, ρ = 0.9088 kg/m^3, c_p = 1008 J/kgK, k = 0.0319 W/mK , μ = 22.3×10^{-6} kg/ms

For a 1-in BWG 16 tube D_i = 0.870 inch = 0.02210 m, D_o = 0.0254 m, k_{cu} = 380 W/mK,

Outside HT Coefficient

For flow across a single tube, we use the equation due to Zhukaukas, Equation 2.36

$$\mathrm{Nu}_f = \frac{hD_o}{k_f} = C\,\mathrm{Re}_f^m\,\mathrm{Pr}_f^{0.365}\left(\frac{\mathrm{Pr}_{Bulk}}{\mathrm{Pr}_w}\right)^{1/4}$$

Where

$$\mathrm{Pr}_{Bulk} = \left(\frac{c_p\mu}{k}\right)_{\mathrm{air,30^oC}} = \frac{(1003)(18.7\times10^{-6})}{(0.0263)} = 0.713$$

$$\mathrm{Pr}_{film} = \left(\frac{c_p\mu}{k}\right)_{\mathrm{air,70^oC}} = \frac{(1005)(20.5\times10^{-6})}{(0.0291)} = 0.708$$

$$\mathrm{Pr}_{Bulk} = \left(\frac{c_p\mu}{k}\right)_{\mathrm{air,110^oC}} = \frac{(1008)(22.3\times10^{-6})}{(0.0319)} = 0.705$$

Note that the PRs at all conditions are very similar so there should be little error in taking an average wall temperature

$$\mathrm{Re} = \frac{D_o u_{bulk}\rho_f}{\mu_f} = \frac{(0.0254)(20)(1.0148)}{(20.5\times10^{-6})} = 25{,}150$$

from Table 2.5 – C = 0.26 and m = 0.6

$$\mathrm{Nu}_f = \frac{hD_o}{k_f} = C\,\mathrm{Re}_f^m\,\mathrm{Pr}_f^{0.365}\left(\frac{\mathrm{Pr}_{Bulk}}{\mathrm{Pr}_w}\right)^{1/4} = 0.26(25150)^{0.6}(0.708)^{0.365}\left(\frac{0.713}{0.705}\right)^{1/4} = 100.4$$

$$h_o = (100.4)\frac{(0.0291)}{(0.0254)} = 115.1\ \mathrm{W/m^2K}$$

Inside HT Coefficient

Use Dittus-Boelter, Equation 2.27, since L is unknown and wall temp is essentially the same as the bulk temp of the water.

$$\mathrm{Nu} = \frac{h_i D_i}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^n \quad (n = 0.4 \text{ for heating fluid})$$

$$\mathrm{Re} = \frac{\rho_{water} u D_i}{\mu_{water}} = \frac{(948.5)(1)(0.0221)}{(276\times10^{-6})} = 75{,}950$$

$$\mathrm{Pr} = \left(\frac{c_p\mu}{k}\right)_{\mathrm{water,110^oC}} = \frac{(4255)(276\times10^{-6})}{(0.6748)} = 1.74$$

$$\mathrm{Nu} = \frac{h_i D_i}{k} = (0.023)(75{,}950)^{0.8}(1.74)^{0.4} = 230.5$$

$$h_i = (230.5)\frac{(0.6748)}{(0.0221)} = 7038\ \mathrm{W/m^2K}$$

Overall HT Coefficient

$$U_o = \left[\frac{1}{h_o} + \cancel{R_{fo}} + \frac{D_o \ln(D_o / D_i)}{2k_w} + \frac{D_o}{D_i} \cancel{R_{fi}} + \frac{D_o}{D_i} \frac{1}{h_i} \right]^{-1}$$

$$U_o = \left[\frac{1}{(115.1)} + \frac{(0.0254)\ln(0.0254/0.0221)}{2(380)} + \frac{(0.0254)}{(0.0221)} \frac{1}{(7038)} \right]^{-1} = 112.9 \text{ W/m}^2\text{K}$$

Assumption about wall temperature is correct all resistance is on the outside of the tube.

Heat Balance

$$Q = \dot{m} c_p \Delta T = \frac{\pi D_i^2}{4} u_i \rho_{water} c_{p,water} (T_{in} - T_{out})$$

$$Q = \frac{\pi (0.0221)^2}{4} (1.0)(948.5)(4255)(140-80) = 92,890 \text{ W}$$

$$Q = U_o A_o \Delta T_{lm} \Rightarrow A_o = \frac{Q}{U_o \Delta T_{lm}} = \frac{(92,890)}{(112.9) \frac{[(140-30)-(80-30)]}{\ln \frac{(140-30)}{(80-30)}}} = 10.81 \text{ m}^2$$

$$L = \frac{A_o}{\pi D_o} = \frac{(10.81)}{\pi (0.0254)} = 135.5 \text{ m}$$

Length of tube = 135.5 m

29. Oil flows inside a thin walled copper tube of diameter, D_i = 30 mm. Steam condenses on the outside of the tube and the tube wall temperature may be assumed to be constant at the temperature of the steam (150°C). The oil enters the tube at 30°C and a flow rate of 1.6 kg/s. The properties of the oil are as follows:

	30°C	50°C	150°C
Density, ρ (kg/m^3)	886	882	864
Thermal conductivity, k (W/mK)	0.2	0.2	0.18
Specific heat capacity, c_p (J/kgK)	2000	2000	1950
Viscosity, μ (kg/ms)	5×10^{-3}	4×10^{-3}	4×10^{-4}

a. Calculate the inside heat transfer coefficient h_i using the appropriate correlation. You should assume that the bulk oil temperature changes from 30 to 50°C along the tube.
Use average bulk temperature of oil of 40°C with the following properties:
ρ = 884 kg/m^3, k = 0.20 W/mK, c_p = 2000 J/kgK, μ = 4.5×10^{-3} kg/ms

Since there is a significant change in viscosity between the bulk and the wall, use the Sieder-Tate expression, Equation 2.26 with $L \gg D_i$ (check this at end of calculation)

Inside HT Coefficient

$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right]\text{Re}^{0.8}\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$u_{in} = \frac{4\dot{m}}{\pi D_i^2 \rho} = \frac{(4)(1.6)}{\pi(0.030)^2(886)} = 2.5548 \text{ m/s}$$

$$\text{Re}_{in} = \frac{\rho u D_i}{\mu} = \frac{(884)(2.5548)(0.030)}{(5\times10^{-3})} = 13,580$$

$$\text{Pr}_{bulk} = \frac{c_p \mu}{k} = \frac{(2000)(4.5\times10^{-3})}{(0.20)} = 45$$

$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right]\text{Re}^{0.8}\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$\text{Nu} = 0.023(1)(13580)^{0.8}(45)^{1/3}\left(\frac{4.5\times10^{-3}}{4.0\times10^{-4}}\right)^{0.14} = 232.4$$

$$h_i = \text{Nu}\frac{k}{D_i} = (232.4)\frac{(0.20)}{(0.030)} = 1549.6\ \text{W/m}^2\text{K}$$

***h_i* = 1550 W/m2K**

b. Using the result from part a., calculate the length of tube required to heat the oil from 30 to 50°C.

Energy Balance

$$Q = m_{oil}c_{p,oil}\Delta T_{oil} = (1.6)(2000)(50-30) = 64{,}000\ \text{W}$$

$$\Delta\text{T}_{lm} = \frac{(150-30)-(150-50)}{\ln\frac{150-30}{150-50}} = 109.7^{\circ}\text{C}$$

$$A_i = \frac{Q}{U_i\Delta\text{T}_{lm}} = \frac{(64{,}000)}{(1549.6)(109.7)} = 0.3765\ \text{m}$$

$$A_i = \pi D_i L \Rightarrow L = \frac{A_i}{\pi D_i} = \frac{(0.3765)}{\pi(0.030)} = 3.99\ \text{m}$$

Check for D_i/L correction term in Equation 2.26

$$\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right] = \left[1+\left(\frac{0.030}{3.99}\right)^{0.7}\right] = 1.0326$$

$$\Rightarrow h_{i,new} = (1.0326)(1549.6) = 1600.1$$

Iterating gives,

$$A_i = \frac{Q}{U_i\Delta\text{T}_{lm}} = \frac{(64{,}000)}{(1600.1)(109.7)} = 0.3646\ \text{m}$$

$$A_i = \pi D_i L \Rightarrow L = \frac{A_i}{\pi D_i} = \frac{(0.3646)}{\pi(0.030)} = 3.87\ \text{m}$$

Further iteration does not change the answer

L = 3.87 m

30. Follow the approach used in Example 2.12, solve the following problem. An organic liquid (acetic acid at 1 bar) is to be vaporized inside a set of vertical 3/4-in BWG 16 tubes using condensing steam on the outside of the tubes to provide the energy for vaporization. The major resistance to heat transfer is expected to be on the inside of the tubes and the wall temperature, as a first approximation, may be assumed to be at the temperature of the condensing steam, which for this case is 125°C. It may be assumed that the value of the vapor quality, x, varies from 0.05 to 0.95 in the tube. Determine the length of the tubes required to vaporize the acetic acid.

The physical parameters for acetic acid are:
D_i = 0.62 inch = 0.015748 m

ρ_v = 1.893 kg/m^3, ρ_l = 939.7 kg/m^3, μ_v = 11.32×10^{-6} kg/ms, μ_l =390.1×10^{-6} kg/ms, T_{sat} = 117.6°C , P_c = 57.9 bar, M = 60, $k_l = 0.1423$ W/mK, λ = 405.0 kJ/kg, $c_{p,l}$ = 2.434 kJ/kgK, $c_{p,v}$ = 1.319 kJ/kgK $\dot{m}$ = 0.04 kg/s/tube, T_w=125°C

$$\mathrm{Re}_l = \frac{\rho_l v D_i}{\mu_l} = \frac{4\dot{m}}{\pi \mu_l D_i} = \frac{(4)(0.04)}{\pi(390.1\times10^{-6})(0.015748)} = 8290$$

$$\mathrm{Pr} = \frac{c_{pl}\mu_l}{k_l} = \frac{(2434)(390.1\times10^{-6})}{(0.1423)} = 6.673$$

From Equation 2.49, the pool boiling coefficient is given by

$$h_{cb} = fh_l + sh_{pb}$$

Where from Equation 2.50 and 2.51 we have

$$f = 1 + 24,000 B_o^{1.16} + \frac{1.37}{X_{tp}^{0.86}}$$

$$X_{tp} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}$$

From Equation 2.53,

$$s = \frac{1}{1 + 1.15\times10^{-6} f^2 \mathrm{Re}_l^{1.17}}$$

and from Equation 2.55,

$$h_{pb}^{0.33} = 55P_r^{0.12}(-\log_{10} P_r)^{-0.55}(M)^{-0.5}(T_w - T_{sat})^{0.67}$$

$$h_{pb}^{0.33} = (55)\left[\frac{1}{57.9}\right]^{0.12}\left(-\log_{10}\left[\frac{1}{57.9}\right]\right)^{-0.55}(60)^{-0.5}(125-117.6)^{0.67} = 12.21$$

$$h_{pb} = (12.21)^{1/0.33} = 1964 \text{ W/m}^2\text{K}$$

The convective heat transfer coefficient, h_l, is given by the Sieder-Tate expression, Equation 2.26, assuming that $L \gg D$ and $\mu_w \cong \mu_l$ then,

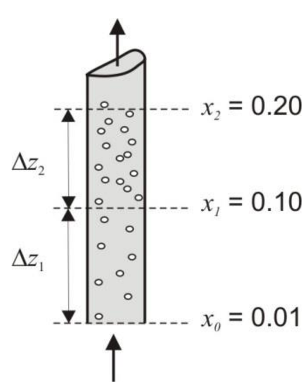
$$\text{Nu} = \frac{h_i D_i}{k} = 0.023\left[1+\left(\frac{D_i}{L}\right)^{0.7}\right]\text{Re}^{0.8}\,\text{Pr}^{1/3} = (0.023)(8290)^{0.8}(6.673)^{1/3} = 59.06$$

$$h_i = (59.06)(0.1423)/(0.015748) = 533.7 \text{ W/m}^2\text{K}$$

From Equation 2.45,

$$X_{tp} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{1.893}{939.7}\right)^{0.5}\left(\frac{390.1\times10^{-6}}{11.32\times10^{-6}}\right)^{0.1} = 0.06395\left(\frac{1-x}{x}\right)^{0.9}$$

The value of x in the above equation varies from 0.01 to 0.99 and hence X_{tp} will also vary over a wide range. In order to take account of the change in x, the problem will be discretized with respect to x and solved for each increment of $\Delta x = 0.1$. The calculations for $x = 0.01$ to 0.1 will be covered and then the results for the other increments will be summarized. In order to calculate f in Equation 2.44, the value of B_o must be found from Equation 2.46; however, the heat flux (q/A) is unknown. Using the discretization scheme shown in the Figure below the value of B_0 may be found in terms of Δz_i



Therefore, for the first increment

$$B_o = \frac{Q/A}{\lambda G} = \frac{\dot{m}(x_1 - x_0)\lambda}{\pi D_i \Delta z_1} \frac{1}{\lambda} \frac{\pi D_i^2}{4\dot{m}} = \frac{(x_1 - x_0)D_i}{4\Delta z_1} = \frac{(0.1-0.01)(0.015748)}{4\Delta z_1} = \frac{3.5433\times 10^{-4}}{\Delta z_1}$$

$\overline{x}_1 = \frac{(0.1+0.01)}{2} = 0.055$, and $X_{tp} = 0.06395\left(\frac{1-0.055}{0.055}\right)^{0.9} = 0.8268$

By guessing a value for Δz_1, the values of f and s from Equations 2.44 and 2.47 can be calculated and used in Equation 2.43 to calculate h_{cb}. Now an energy balance on the first increment of tube gives,

$$h_{cb}\pi D_i \Delta z_1 (T_w - T_{sat}) = \dot{m}\lambda(x_1 - x_0) \qquad \text{(a)}$$

and rearranging

$$\Delta z_1 = \frac{\dot{m}\lambda(x_1 - x_0)}{h_{cb}\pi D_i (T_w - T_{sat})} \qquad \text{(b)}$$

The solution for the first increment is found by iterating between Equations 2.43 and (b) until a constant value of Δz_1 is obtained.

For the first iteration, choose a value of $\Delta z_1 = 0.5$ m

$$B_o = \frac{(3.5433\times 10^{-4})}{(0.5)} = 7.0866\times 10^{-4}$$

$$f = 1 + 24{,}000(B_0)^{1.16} + \frac{1.37}{(X_{tp})^{0.86}} = 1 + 24{,}000(7.0866\times 10^{-4})^{1.16} + \frac{1.37}{(0.8268)^{0.86}} = 7.9434$$

and

$$s = \frac{1}{1 + 1.15\times 10^{-6} f^2 \mathrm{Re}_l^{1.17}} = \frac{1}{1 + 1.15\times 10^{-6}(7.9434)^2(8290)^{1.17}} = 0.2639$$

$$h_{cb} = fh_l + sh_{pb} = (7.9434)(533.7) + (0.2639)(1964) = 4758 \text{ W/m}^2\text{K}$$

$$h_{cb} = fh_l + sh_{pb}$$

Substitute into Equation (b) to get

$$\Delta z_1 = \frac{\dot{m}\lambda(x_1 - x_0)}{h_{cb}\pi D_i(T_w - T_{sat})} = \frac{(0.04)(405.0\times10^3)(0.1-0.01)}{(4758)\pi(0.015748)(125-117.6)} = 0.837 \text{ m}$$

Use this value of Δz_1 in Bo and continue to iterate until converged, this gives Δz_1 = 1.141 m. The results for the remaining increments are as follows,

$x_i - x_{i-1}$	$Bo_{0,i}$	$\overline{x_i}$	$X_{tp,i}$	f_i	s_i	$h_{cb,i}$ (W/m²K)	q/A (W/m²)	Δz_i (m)
0.01-0.1	0.000310	0.055	0.8268	4.660	0.5103	3489	25,820	1.141
0.1- 0.2	0.000412	0.150	0.3047	7.648	0.2789	4630	34,259	0.956
0.2- 0.3	0.000564	0.250	0.1719	11.315	0.1502	6334	46,870	0.699
0.3- 0.4	0.000769	0.350	0.1116	15.887	0.0823	8640	63,938	0.512
0.4- 0.5	0.001045	0.450	0.0766	21.849	0.0453	11,749	86,946	0.377
0.5- 0.6	0.001433	0.550	0.0534	30.094	0.0244	16,109	119,208	0.275
0.6- 0.7	0.002021	0.650	0.0366	42.515	0.0124	22,714	168,087	0.195
0.7- 0.8	0.003038	0.750	0.0238	63.965	0.0055	34,149	252,704	0.130
0.8- 0.9	0.005324	0.850	0.0134	112.108	0.0018	59,836	442,785	0.074
0.9-0.99	0.015101	0.945	0.0049	318.018	0.0002	169,727	1,395,534	0.023
						Total		**4.381**

The flux in the last cell is very high compared with the maximum heat flux for pool boiling but this will not alter the length of the tube very much, i.e., even if the flux is calculated using the relationship below.

$$\left[\frac{Q}{A}\right]_{max} = 0.3673 P_c \left(\frac{P}{P_c}\right)^{0.35}\left(1 - \frac{P}{P_c}\right)^{0.9} = 0.3673(57.9\times10^5)\left(\frac{1}{57.9}\right)^{0.35}\left(1 - \frac{1}{57.9}\right)^{0.9}$$

$$\left[\frac{Q}{A}\right]_{max} = 505{,}800 \text{ W/m}^2$$

<u>L_{tube} = 4.38 m or 14.4 ft</u>

31. Repeat Problem 28 to find the length of a tube with 2-in diameter, 1/16-in thick, annular fins spaced a distance of 3/16-in apart ().

r_{fin} = 1 inch = 0.0254 m, h = h_o = 115.1 W/m^{2}K, δ = 1/16 inch = 0.0015875 m, k_{cu} = 380 W/mK

From Figure 2.33:

$$b = r_{fin}\sqrt{\frac{2h}{k\delta}} = (0.0254)\sqrt{\frac{2(115.1)}{(380)(0.0015875)}} = 0.496$$

$$a = \frac{r_{tube}}{r_{fin}} = 0.5$$

From Figure 2.33, ε_{fin} ~ 0.97

The enhancement in heat transfer for the tube with fins is given by Equation 2.76:

$$\text{Enhancment in heat transfer} = \frac{A_{base} + \varepsilon_{fin}A_{fin}}{A_{bare}} = \frac{\pi D_o(L_{bare}) + \varepsilon_{fin}(2\pi(r_{fin}^2 - r_{tube}^2) + \pi r_{fin}L_{fin})}{\pi D_o(L_{bare} + L_{fin})}$$

where

L_{bare} = 3/16 inch, L_{fin} =1/16 inch , D_o =1 inch, r_{fin} =1 inch, r_{tube} =1/2 inch, $\varepsilon_{fin} = 0.97$

Enhancment in heat transfer = 6.8125

Therefore, the new tube length, *L* = 135.5/6.8125 = 19.89 m

32. An air heater consists of a shell and tube heat exchanger with 24 longitudinal fins on the outside of the tubes. The tubes are 1.5 in 14 BWG, and the fins are 0.75 mm thick and are 15 mm "long." There are a total of 8 fins spaced uniformly around the circumference of each tube. The tubes are made from carbon steel and their length is 3 m. Air at 0.8 kg/s is being heated from 30°C to 200°C in the shell, and the heat transfer coefficient may be taken as h_o = 25 W/m^{2}K (without fins). Steam is condensing at 254°C in the tubes, and at that temperature λ = 1700 kJ/kg. The tube side heat transfer coefficient may be taken as h_i = 6000 W/m^{2}K and no fouling occurs for this service. Because the outside heat transfer coefficient is limiting, the wall temperature will be close to 254°C and carbon steel, k_{SS} = 42.3 W/mK

For 1.5 inch 14 BWG, Do = 1.5 inch = 0.0381 m and Di = 1.334 inch = 0.033884 m

a. Calculate the overall heat transfer coefficient with and without fins.

For bare (finless) tubes

From Equation, 2.23

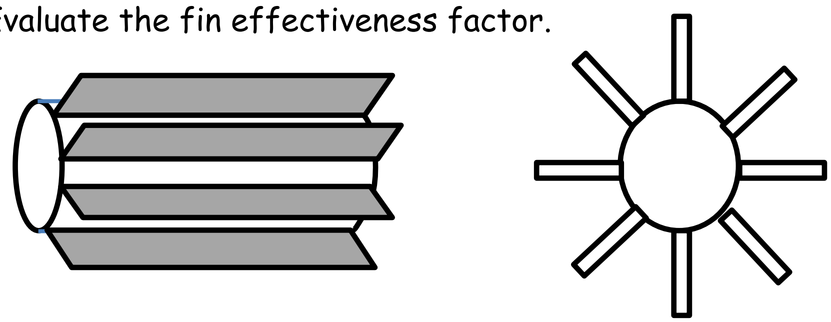
$$U_o = \left[\frac{1}{h_o} + \cancel{R_{fo}} + \frac{D_o \ln(D_o / D_i)}{2k_w} + \frac{D_o}{D_i} \cancel{R_{fi}} + \frac{D_o}{D_i} \frac{1}{h_i} \right]^{-1}$$

$$U_o = \left[\frac{1}{25} + \frac{(0.0381)\ln(0.0381/0.033884_i)}{2(42.3)} + + \frac{0.0381}{0.033884} \frac{1}{(6000)} \right]^{-1} = 24.85 \text{ W/m}^2\text{K}$$

$\underline{U_0 = 24.85 \text{ W/m}^2\text{K}}$

For finned tubes

Evaluate the fin effectiveness factor.



For each fin, δ = 0.00075m, L= 0.015 m, L_{tube} = 3 m, From Equation 2.79m

$L_C = L+\delta/2$=0.015+0.00075/2= 0.01538m

From Equation 2.69,

$$m = \sqrt{\frac{2h_o}{\delta k}} = \sqrt{\frac{2(25)}{(0.00075)(42.3)}} = 39.70$$

$$mL_c = (39.70)(0.01538) = 0.6104$$

From Equation 2.68, we have

$$\varepsilon_{fin} = \frac{\tanh(mL_c)}{mL_c} = \frac{\tanh(0.6104)}{(0.6104)} = 0.8919$$

The enhancement in heat transfer is given by Equation 2.76, so considering all 8 fins, we have

$$A_{fin} = (8)(2)L_C L_{tube} = (16)(0.01538)(3) = 0.738 \text{ m}^2$$

$$A_{base} = \pi D_o L_{tube} - 8\delta L_{tube} = \pi(0.0381)(3) - 8(0.00075)(3) = 0.34108 \text{ m}^2$$

$$A_{bare} = \pi D_o L_{tube} = \pi(0.0381)(3) = 0.35908 \text{ m}^2$$

$$\text{Enhancement in heat transfer} = \frac{A_{base} + \varepsilon_{fin} A_{fin}}{A_{bare}} = \frac{(0.34108) + (0.8919)(0.738)}{(0.35908)} = 2.78$$

$$h_{o,enhanced} = (25)(2.78) = 69.57 \text{ W/m}^2\text{K}$$

$$U_o = \left[\frac{1}{h_o} + \cancel{R}_{fo} + \frac{D_o \ln(D_o / D_i)}{2k_w} + \frac{D_o}{D_i}\cancel{R}_{fi} + \frac{D_o}{D_i}\frac{1}{h_i}\right]^{-1}$$

$$U_o = \left[\frac{1}{69.57} + \frac{(0.0381)\ln(0.0381/0.033884_i)}{2(42.3)} + + \frac{0.0381}{0.033884}\frac{1}{(6000)}\right]^{-1} = 68.43 \text{ W/m}^2\text{K}$$

U_0 68.43 W/m^{2}K

b. Calculate heat transfer area and the number of tubes needed with and without fins.
   For Air at 30°C – c_p = 1003.4 J/kgK and at 200°C c_p = 1020.4 J/kgK.
   Use an average c_p value of (1003.4+1020.4)/2 = 1011.9 J/kg

$$Q = \dot{m}_{air}\bar{c}_{p,air}\Delta T_{air} = (0.8)(1011.9)(200-30) = 137.6 \text{ kW}$$

$$Q = \dot{m}_{steam}\lambda \Rightarrow \dot{m}_{steam} = \frac{137.6}{1700} = 0.08095 \text{ kg/s}$$

$$\Delta T_{lm} = \frac{(254-30)-(254-200)}{\ln\frac{(254-30)}{(254-200)}} = 119.5^{o}\text{C}$$

   F = 1, since steam is condensing on one side

For bare (finless) tubes

$$A_o = \frac{Q}{U_o \Delta T_{lm}} = \frac{(137,600)}{(24.85)(119.5)} = 46.34\ \text{m}^2$$

$$A_o = \pi D_o L_{tube} n_{tubes} \Rightarrow n_{tubes} = \frac{A_o}{\pi D_o L_{tube}} = \frac{(46.34)}{\pi(0.0381)(3)} = 129$$

For finned tubes

$$A_o = \frac{Q}{U_o \Delta T_{lm}} = \frac{(137,600)}{(68.43)(119.5)} = 16.83\ \text{m}^2\ \text{(this is the bare surface area of tubes)}$$

$$A_o = \pi D_o L_{tube} n_{tubes} \Rightarrow n_{tubes} = \frac{A_o}{\pi D_o L_{tube}} = \frac{(46.34)}{\pi(0.0381)(3)} = 47$$

Design Algorithm and Worked Examples for Rating of Heat Exchangers

33. Your assignment is to design a replacement condenser for an existing distillation column. Space constraints dictate a vertical-tube condenser with a maximum height (equals tube length) of 3 m. An organic is condensed at a rate of 12,000 kg/h at a temperature of 75°C, and cooling water is used, entering at 30°C and exiting at 40°C. The person you are replacing has done some preliminary calculations suggesting that a 1-4 exchanger (water in tubes) using 1 inch 16 BWG copper tubes on 1.25 inch equilateral triangular pitch with a 37 inch shell diameter would be suitable. However, there are only partial calculations to support this claim, and the person who performed the original design is unavailable. Complete the detailed heat transfer calculations to evaluate the suitability of this heat exchanger design.

Data for condensing organic:

ρ_f = 800 kg/m^3, ρ_v = 5.3 kg/m^3, λ = 800 kJ/kg, k_f = 0.15 W/m K, μ_f = 400$\times$10^{-6} kg/m s, c_{pl} = 2600 J/kgK

heat transfer coefficient for tube side $\approx$ 6000 W/m^{2}K

typical fouling coefficient for plant cooling water = 1200 W/m^{2}K

assume no fouling on condensing side

Properties for water:

ρ_f = 994 kg/m^3, c_p = 4187 J/kg°C, k_f = 0.6195 W/m K, μ_f = 748$\times$10^{-6} kg/m s

Properties of tubes

D_o = 0.0254 m, D_i = 0.834 inch = 0.02118 m, k_{cu} = 382.9 W/mK (at 35°C)

Energy Balance

Shell-side: $Q = \dot{m}_{organic}\lambda_{organic} = \left(\dfrac{12000}{3600}\right)(800\times10^3) = 2,667 \text{ kW}$

Tube-side:

$$Q = 2,667\times10^3 = \dot{m}_{water}c_{p,water}\Delta T_{water} \Rightarrow \dot{m}_{water} = \frac{2,667\times10^3}{(4187)(40-30)} = 63.69 \text{ kg/s}$$

Estimate the Shell Side Coefficient, h_o

Using Equation 2.64 – assume T_W = (40+30)/2 = 35°C All resistance on shell side.

$$\lambda' = \lambda + 0.68c_{p,l}(T_{sat} - T_w) = 800\times10^3 + 0.68(2600)(75-35) = 870.7\times10^3 \text{ J/kg}$$

$$Nu = \frac{\bar{h}_c L}{k_l} = 1.13\left[\frac{\rho_l g(\rho_l - \rho_v)\lambda' L^3}{\mu_l k_l (T_{sat} - T_w)}\right]^{1/4} = 1.13\left[\frac{(800)(9.81)(800-5.3)(870.7\times10^3)(3)^3}{(400\times10^{-3})(0.15)(75-35)}\right]^{1/4}$$

$$Nu = 17,765 \Rightarrow \bar{h}_c = (17,765)\frac{(0.15)}{(3)} = 888.3 \text{ W/m}^2\text{K}$$

Estimate overall coefficient using given h_i

$$U_o = \left[\frac{1}{h_o} + \cancel{R_{fo}} + \frac{D_o \ln(D_o/D_i)}{2k_w} + \frac{D_o}{D_i}R_{fi} + \frac{D_o}{D_i}\frac{1}{h_i}\right]^{-1}$$

$$U_o = \left[\frac{1}{888.3} + \frac{(0..0254)\ln(0.0254/0.02118)}{(2)(382.9)} + \frac{(0.0254)}{(0.02118)}\frac{1}{1200} + \frac{(0.0254)}{(0.02118)}\frac{1}{6096}\right]^{-1} = 429.6 \text{ W/m}^2\text{K}$$

Estimate Number of tubes

$$\Delta T_{lm} = \frac{(75-30)-(75-40)}{\ln\frac{(75-30)}{(75-40)}} = \frac{10}{\ln\frac{45}{35}} = 39.79^{\circ}\text{C}$$

$$A_o = \frac{Q}{U_o \Delta T_{lm}} = \frac{(2667\times10^3)}{(429.6)(39.79)} = 156.0 \text{ m}^2$$ calculate the total number of tubes and the tubes per pass

$$A_o = n_{tube}\pi D_o L_{tube} \Rightarrow n_{tube} = \frac{A_o}{\pi D_o L_{tube}} = \frac{(156)}{\pi(0.0254)(3)} = 652$$

$$n_{pass} = \frac{n_{tube}}{4} = 163$$

Reevaluate h_i

$$u_i = \frac{4\dot{m}_{water}}{n_{pass}\pi D_i^2} = \frac{(4)(63.69)}{(163)(994)\pi(0.02118)^2} = 1.12 \text{ m/s}$$

$$\text{Re}_{tube} = \frac{\rho_{water} D_i u_i}{\mu_{water}} = \frac{(994)(0.02118)(1.12)}{(748\times10^{-6})} = 31,400$$

$$\text{Pr} = \frac{c_{p,water}\mu_{water}}{k_{water}} = \frac{(4187)(748\times10^{-6})}{(0.6195)} = 5.055$$

$$\text{Nu} = \frac{h_i D_i}{k_{water}} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{1/3} = (0.023)(31,400)^{0.8}(5.055)^{1/3} = 156.2$$

$$h_i = \text{Nu}\frac{k_{water}}{D_i} = (156.2)\frac{(0.6195)}{(0.02118)} = 4570$$

Estimate U_o and T_W and the recalculate h_o

$$U_o = \left[\frac{1}{888.3} + \frac{(0..0254)\ln(0.0254/0.02118)}{(2)(382.9)} + \frac{(0.0254)}{(0.02118)}\frac{1}{1200} + \frac{(0.0254)}{(0.02118)}\frac{1}{4570}\right]^{-1} = 418\ \text{W/m}^2\text{K}$$

$$U_o(T_{sat} - T_{water}) = h_o(T_{sat} - T_w)$$

$$\therefore T_W = T_{sat} - \frac{U_o}{h_o}(T_{sat} - T_{water}) = 75 - \frac{418}{888.3}(75-35) = 56.2$$

$$\lambda' = \lambda + 0.68c_{p,l}(T_{sat} - T_w) = 800\times10^3 + 0.68(2600)(75-56.2) = 870.7\times10^3\ \text{J/kg}$$

The iteration process continues until the following solution is reached:

h_i = 4807 W/m^{2}K, h_o = 1088.5 W/m^{2}K, U_o = 460.1 W/m^{2}K, A_o = 145.7 m2, n_{tubes} = 612 with 153 tubes per pass.

From Table 2.6, max number of tubes in a 37 inch diameter shell using 1 inch tubes on a triangular pitch of 1.25 inch is 638. Therefore, the suggested arrangement should work for this system.

34. A 1-2 shell and tube heat exchanger has the following dimensions:

Tube length =20 ft
Tube diameter = 1-in BWG 14 carbon steel (k_{cs} = 45 W/mK)
Number of tubes in shell = 608
Shell diameter = 35-in
Tube arrangement = triangular pitch, center-to-center = 1.25-in
Number of baffles = 19
Baffle spacing = 1 ft
Baffle = horizontal baffle with baffle cut = 25%

The fluids in the shell and tube sides of the exchanger have the following properties:

	Shell Side	**Tube Side**
Inlet temperature (°C)	120	30
Mass flowrate, $\dot{m}$ (kg/s)	120	180
Specific heat capacity, c_p (kJ/kgK)	2.0	4.2
Thermal conductivity, k (W/mK)	0.2	0.61
Density, ρ (kg/m^3)	850	1000
Viscosity, μ (kg/ms)	5.0×10^{-4}	0.72×10^{-3}

Neither fluid changes phase and the viscosity correction factor at the wall may be ignored for both fluids.

D_o = 0.0254 m, D_i = 0.834 inch = 0.021184m, p = 1.25 inch = 0.03175 m, D_S = 35 inch = 0.889 m, L_{tube} = 20 ft = 6.096 m, n_{baffle} = 19, L_b = $L_{tube}/(n_{baffle}+1)$ = 6.096/(19+1) = 0.3048 m, BC = 25%, n_{tube} = 608, n_{pass} = 608/2 = 304

a. The inside heat transfer coefficient – using the Seider-Tate relationship, Equation, 2.26)

$$u_i = \frac{4\dot{m}_i}{n_{pass}\rho\pi D_i^2} = \frac{(4)(180)}{(608/2)(1000)\pi(0.021184)^2} = 1.68 \text{ m/s}$$

$$\text{Re}_i = \frac{u_i D_i \rho_i}{\mu_i} = \frac{(1.68)(0.021184)(1000)}{(0.72\times10^{-3})} = 49,400$$

$$\text{Pr}_i = \frac{c_{p,i}\mu_i}{k_i} = \frac{(4200)(0.72\times10^{-3})}{(0.61)} = 4.96$$

$$\text{Nu}_i = \frac{h_i D_i}{k_i} = 0.023\left[1+\left(\frac{D_i}{L_{tube}}\right)^{0.7}\right]\text{Re}^{0.8}\,\text{Pr}^{1/3}\left(\frac{\mu_i}{\mu_w}\right)^{0.14}$$

$$\text{Nu}_i = 0.023\left[1+\left(\frac{(0.021184)}{6.096}\right)^{0.7}\right](49,400)^{0.8}(4.96)^{1/3}(1) = 227.4$$

$$h_i = \text{Nu}_i\frac{k_i}{D_i} = (227.4)\frac{(0.61)}{(0.021184)} = 6550 \text{ W/m}^2\text{K}$$

b. The shell side heat transfer coefficient – use Kern's method

$$D_{H,s}(\text{traingular pitch}) = \frac{1.103p^2 - D_o^2}{D_o} = \frac{1.103(0.03175)^2-(0.0254)^2}{(0.0254)} = 0.018375 \text{ m}$$

$$A_s = D_s L_b\frac{(p-D_o)}{p} = (0.889)(0.3048)\frac{(0.03175-0.0254)}{(0.03175)} = 0.05419 \text{ m}^2$$

$$G_s = \frac{\dot{m}_s}{A_s} = \frac{(120)}{(0.05419)} = 2215 \text{ kg/m}^2\text{s} \text{ and } u_s = \frac{G_s}{\rho_s} = \frac{2215}{850} = 2.61 \text{ m/s}$$

$$\text{Re}_s = \frac{G_s D_{H,s}}{\mu} = \frac{(2215)(0.018375)}{(5\times10^{-4})} = 81,400$$

$$j_h = 1.2492(BC)^{-0.329}\,\text{Re}_s^{-0.4696} = 1.2492(25)^{-0.329}(81,400)^{-0.4696} = 0.002141$$

$$\text{Pr} = \left[\frac{c_p\mu}{k}\right]_{shell} = \frac{(2000)(5\times10^{-4})}{(0.2)} = 5.0$$

$$\text{Nu} = \frac{h_s D_{H,s}}{k_f} = j_h\,\text{Re}_s\,\text{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} = (0.002141)(81400)(5.0)^{1/3}(1)^{0.14} = 298$$

$$h_s = \text{Nu}\frac{k_f}{D_{H,s}} = (298)\frac{(0.2)}{(0.018375)} = 3244 \text{ W/m}^2\text{K}$$

c. The overall heat transfer coefficient (assuming that fouling may be ignored)

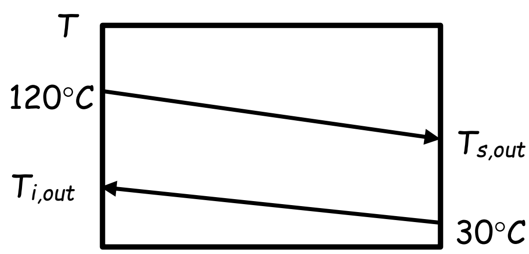
$$U_o = \left[\frac{1}{h_o} + \cancel{R}_{fo} + \frac{D_o \ln(D_o / D_i)}{2k_w} + \frac{D_o}{D_i} \cancel{R}_{fi} + \frac{D_o}{D_i} \frac{1}{h_i} \right]^{-1}$$

$$U_o = \left[\frac{1}{3244} + \frac{(0.0254)\ln(0.0254/0.021184)}{2(45)} + \frac{(0.0254)}{(0.021184)} \frac{1}{(6550)} \right]^{-1} = 1843 \text{ W/m}^2\text{K}$$

d. The exit temperatures of both fluids
Energy Balance gives

$$Q = m_i c_{p,i}(T_{i,out} - 30) = m_s c_{p,s}(120 - T_{s,out}) \qquad \text{(a)}$$

Design Equation



$$\Delta T_{lm} = \frac{(120 - T_{i,out}) - (T_{s,out} - 30)}{\ln \dfrac{(120 - T_{i,out})}{(T_{s,out} - 30)}} \qquad \text{(b)}$$

$$P = \frac{(T_{s,out} - 30)}{(120 - 30)} \quad \text{and } R = \frac{\dot{m}c_p}{\dot{M}C_p} = \frac{(180)(4200)}{(120)(2000)} = 3.15 \qquad \text{(c)}$$

$$F_{1-2} = \frac{\sqrt{(R^2+1)} \ln\left[\dfrac{1-P}{1-PR}\right]}{(R-1)\ln\left[\dfrac{2-P\left(R+1-\sqrt{(R^2+1)}\right)}{2-P\left(R+1+\sqrt{(R^2+1)}\right)}\right]} \qquad \text{(d)}$$

$$A_o = \pi D_o L_{tube} n_{tube} = \pi(0.0254)(6.096)(608) = 295.8 \text{ m}^2 \qquad \text{(e)}$$

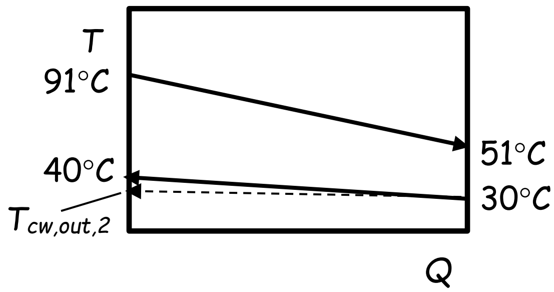
$$Q = U_o A_o \Delta T_{lm} F_{12} \qquad \text{(f)}$$

In Equations (a-f) there are three unknowns, Q, $T_{i,out}$, and $T_{s,\,out}$.
Substituting relationships (b – e) into (f) gives one equation and there are

two equations given in relationship (a). These all must be solved simultaneously. The results are as follows:

$T_{i,out}$ = 52.15°C, $T_{s,out}$ = 50.23°C, P = 0.2461, F_{12} = 0.781, Q = 16.7 MW

35. A 1-2 heat exchanger is used to cool oil in the tubes from 91°C to 51°C using cooling water at 30°C. In the design case the water exits at 40°C. The resistances on the water and oil sides are equal. What are the new cooling water outlet temperature and the required cooling water flowrate if the oil throughput must be increased by 25% but the outlet temperature must be maintained at 51°C?



$$\Delta T_{lm,1} = \frac{(91-40)-(51-30)}{\ln\frac{(91-40)}{(51-30)}} = 33.81^{\circ}\mathrm{C}$$

Using ratios, new case = 2, design case (base case) = 1

$$\frac{Q_2}{Q_1} = \frac{m_{cw,2}}{m_{cw,1}} \cancel{\frac{c_{p,cw,2}}{c_{p,cw,1}}} \frac{(T_{cw,out}-T_{cw,in})_2}{(T_{cw,out}-T_{cw,in})_1} = M_{cw}\frac{(T_{cw,out,2}-30)}{10} \qquad \text{(a)}$$

$$\frac{Q_2}{Q_1} = \frac{m_{oil,2}}{m_{oil,1}} \cancel{\frac{c_{p,oil,2}}{c_{p,oil,1}}} \cancel{\frac{(T_{oil,in}-T_{oil,out})_2}{(T_{oil,in}-T_{oil,out})_1}} = M_{oil} = 1.25 \qquad \text{(b)}$$

$$\frac{Q_2}{Q_1} = \frac{U_2 \cancel{A_2}}{U_1 \cancel{A_1}} \frac{\Delta T_{lm,2}}{\Delta T_{lm,1}} \cancel{\frac{F_2}{F_1}} = \frac{U_2}{U_1}\frac{(91-T_{cw,out,2})-(51-30)}{(33.81)\ln\frac{(91-T_{cw,out,2})}{(51-30)}} = U\frac{(70-T_{cw,out,2})}{(33.81)\ln\frac{(91-T_{cw,out,2})}{(21)}} \qquad \text{(c)}$$

Assume physical properties are constant, there are no change in the areas, and assume the LMTD correction factors, *F* do not change.
Let base case inside and outside HT coefficients = *h*

Therefore, $U = \frac{\left[\frac{1}{hM_{cw}^{0.6}}+\frac{1}{hM_{oil}^{0.8}}\right]^{-1}}{\left[\frac{1}{h}+\frac{1}{h}\right]^{-1}} = \frac{2}{\left[\frac{1}{M_{cw}^{0.6}}+\frac{1}{1.25^{0.8}}\right]}$ substitute in Eqn (c) to give:

$$\frac{Q_2}{Q_1} = \frac{1}{(33.81)} \frac{(70 - T_{cw,out,2})}{\ln \frac{(91 - T_{cw,out,2})}{(21)}} \frac{2}{\left[\frac{1}{M_{cw}^{0.6}} + \frac{1}{1.25^{0.8}} \right]} \quad \text{(d)}$$

Solving Eqns (a, b, and d) we get:

M_{cw} = 1.48, $T_{cw,out,2}$ = 38.5°C

Check assumption about F

Base case P = (91-51)/(91-30) = 0 .656, R =(40-30)/(91-51) = 0.25 ⇒ F_{12} = 0.9360

New case P =0.656, R = (38.46-30)/(91-51) = 0.2182 ⇒ F_{12} =0.9460

Change in F_{12} is only 1.1% so above answer is ok. If you do include the change in F_{12}, you should get the following answer

M_{cw} = 1.44, $T_{cw,out,2}$ = 38.7°C

36. Repeat Problem 35 if the oil side provides 80% of the total resistance to heat transfer. The approach is the same as in Problem 35, and Equations a and b remain the same. However, the estimation of the change in U changes:

Let base case inside (oil-side) coefficients = h therefore the water side coefficient = $4h \Rightarrow U = (1/h+1/4h)^{-1} = 5h/4$ and $(1/h)/(1/U) = 4/5 = 80\%$

Therefore, $U = \dfrac{\left[\dfrac{1}{4hM_{cw}^{0.6}} + \dfrac{1}{hM_{oil}^{0.8}}\right]^{-1}}{\left[\dfrac{1}{4h} + \dfrac{1}{h}\right]^{-1}} = \dfrac{1.25}{\left[\dfrac{1}{4M_{cw}^{0.6}} + \dfrac{1}{1.25^{0.8}}\right]}$ substitute in Eqn (c) to give:

$$\frac{Q_2}{Q_1} = \frac{1}{(33.81)} \frac{(70 - T_{cw,out,2})}{\ln\dfrac{(91 - T_{cw,out,2})}{(21)}} \frac{1.25}{\left[\dfrac{1}{4M_{cw}^{0.6}} + \dfrac{1}{1.25^{0.8}}\right]} \qquad \text{(d)}$$

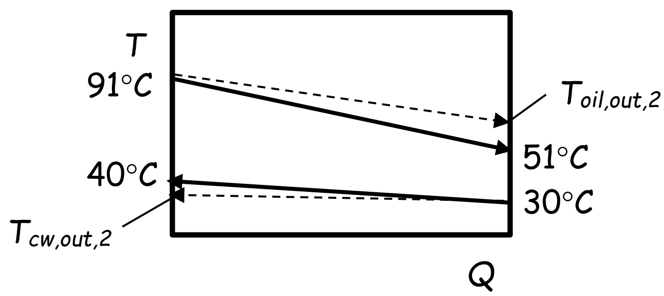
Solving Eqns (a, b, and d) we get:

M_{cw} = 1.61, $T_{cw,out,2}$ = 37.8°C

Taking into account the change in F_{12} gives

M_{cw} = 1.49, $T_{cw,out,2}$ = 38.4°C

37. Repeat Problem 35 for the case when the oil throughput must be increased by 25% but the cooling water flow rate remains unchanged from the base case. Determine the new outlet temperatures for both the process and cooling water streams?



$$\Delta T_{lm,1} = \frac{(91-40)-(51-30)}{\ln\dfrac{(91-40)}{(51-30)}} = 33.81^{\circ}\text{C}$$

Using ratios, new case = 2, design case (base case) = 1

$$\frac{Q_2}{Q_1} = \frac{\cancel{m_{cw,2}}}{\cancel{m_{cw,1}}}\frac{\cancel{c_{p,cw,2}}}{\cancel{c_{p,cw,1}}}\frac{(T_{cw,out}-T_{cw,in})_2}{(T_{cw,out}-T_{cw,in})_1} = \frac{(T_{cw,out,2}-30)}{10} \qquad \text{(a)}$$

$$\frac{Q_2}{Q_1} = \frac{m_{oil,2}}{m_{oil,1}}\frac{\cancel{c_{p,oil,2}}}{\cancel{c_{p,coil,1}}}\frac{(T_{oil,in}-T_{oil,out})_2}{(T_{oil,in}-T_{oil,out})_1} = M_{oil}\frac{91-T_{oil,out,2}}{91-51} = \left(\frac{1.25}{40}\right)(91-T_{oil,out,2}) \qquad \text{(b)}$$

$$\frac{Q_2}{Q_1} = \frac{U_2\cancel{A_2}}{U_1\cancel{A_1}}\frac{\Delta T_{lm,2}}{\Delta T_{lm,1}}\frac{\cancel{F_2}}{\cancel{F_1}} = \frac{U_2}{U_1}\frac{(91-T_{cw,out,2})-(T_{oil.out,2}-30)}{(33.81)\ln\dfrac{(91-T_{cw,out,2})}{(T_{oil.out,2}-30)}} = U\frac{(121-T_{cw,out,2}-T_{oil.out,2})}{(33.81)\ln\dfrac{(91-T_{cw,out,2})}{(T_{oil.out,2}-30)}} \qquad \text{(c)}$$

Assume physical properties are constant, there are no change in the areas, and assume the LMTD correction factors, *F* do not change.

Let base case inside and outside HT coefficients = *h*

Therefore, $U = \dfrac{\left[\dfrac{1}{hM_{cw}^{0.6}}+\dfrac{1}{hM_{oil}^{0.8}}\right]^{-1}}{\left[\dfrac{1}{h}+\dfrac{1}{h}\right]^{-1}} = \dfrac{2}{\left[\dfrac{1}{1}+\dfrac{1}{1.25^{0.8}}\right]} = 1.0890$ substitute in Eqn (c)

to give:

$$\frac{Q_2}{Q_1} = (1.0890)\frac{(121-T_{cw,out,2}-T_{oil.out,2})}{(33.81)\ln\dfrac{(91-T_{cw,out,2})}{(T_{oil.out,2}-30)}} \qquad \text{(d)}$$

Solving Eqns (a, b, and d) we get:

$T_{cw,out,2}$ = 41.4°C, $T_{oil,out,2}$ = 54.4°C

Check assumption about F

Base case P = (91-51)/(91-30) = 0 .656, R =(40-30)/(91-51) = 0.25 ⇒ F_{12} = 0.9287

New case P =0.656, R = (38.46-30)/(91-51) = 0.2115 ⇒ F_{12} =0.9423

Change in F_{12} is only 1.4% so above answer is ok. If you do include the change in F_{12}, you should get the following answer

$T_{cw,out,2}$ = 41.3°C, $T_{oil,out,2}$ = 54.7°C

38. For the situation in Problem 35, suppose that the process stream rate must be temporarily reduced while keeping the process exit temperature at 51°C. Therefore, it will be necessary to reduce the flow of the cooling water stream. Determine the maximum scale-down of the process fluid that can occur without the exit cooling water temperature exceeding the limit of 45°C (when excessive fouling is known to occur)? Plot the results as the ratio of the process stream from the base case (x-axis) vs. the cooling water exit temperature.

$$\frac{Q_2}{Q_1} = \frac{m_{cw,2}}{m_{cw,1}} \frac{c_{p,cw,2}}{c_{p,cw,1}} \frac{(T_{cw,out} - T_{cw,in})_2}{(T_{cw,out} - T_{cw,in})_1} = M_{cw} \frac{(T_{cw,out,2} - 30)}{10} \quad \text{(a)}$$

$$\frac{Q_2}{Q_1} = \frac{m_{oil,2}}{m_{oil,1}} \frac{c_{p,oil,2}}{c_{p,coil,1}} \frac{(T_{oil,in} - T_{oil,out})_2}{(T_{oil,in} - T_{oil,out})_1} = M_{oil} \quad \text{(b)}$$

$$\frac{Q_2}{Q_1} = \frac{U_2 A_2}{U_1 A_1} \frac{\Delta T_{lm,2}}{\Delta T_{lm,1}} \frac{F_2}{F_1} = \frac{U_2}{U_1} \frac{(91 - T_{cw,out,2}) - (51-30)}{(33.81)\ln\frac{(91 - T_{cw,out,2})}{(21)}} = U \frac{(70 - T_{cw,out,2})}{(33.81)\ln\frac{(91 - T_{cw,out,2})}{(21)}} \quad \text{(c)}$$

Since both the flows of the cooling water and process streams change, we have:

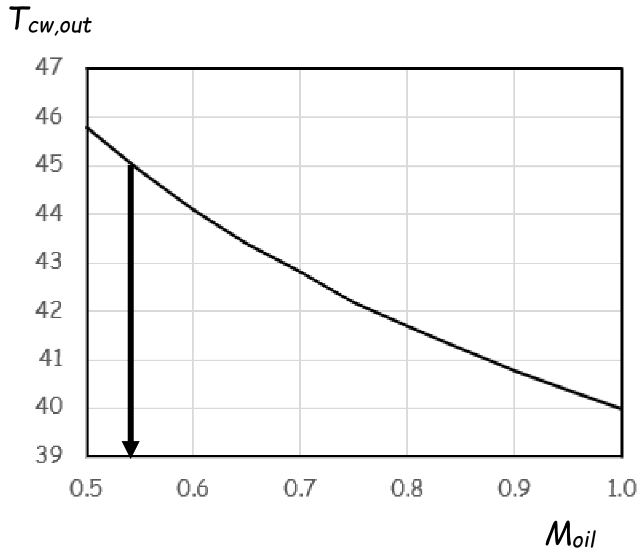
$$U = \frac{\left[\frac{1}{hM_{cw}^{0.6}} + \frac{1}{hM_{oil}^{0.8}}\right]^{-1}}{\left[\frac{1}{h} + \frac{1}{h}\right]^{-1}} = \frac{2}{\left[\frac{1}{M_{cw}^{0.6}} + \frac{1}{0.5^{0.8}}\right]}$$

Substitute this into Equation (c) to give:

$$\frac{Q_2}{Q_1} = \frac{2}{\left[\frac{1}{M_{cw}^{0.6}} + \frac{1}{0.5^{0.8}}\right]} \frac{(70 - T_{cw,out,2})}{(33.81)\ln\frac{(91 - T_{cw,out,2})}{(21)}} \quad \text{(d)}$$

Set M_{oil} = 0.95, 0.90, etc. and solve for M_{cw} and $T_{cw,out,2}$ and plot the results.

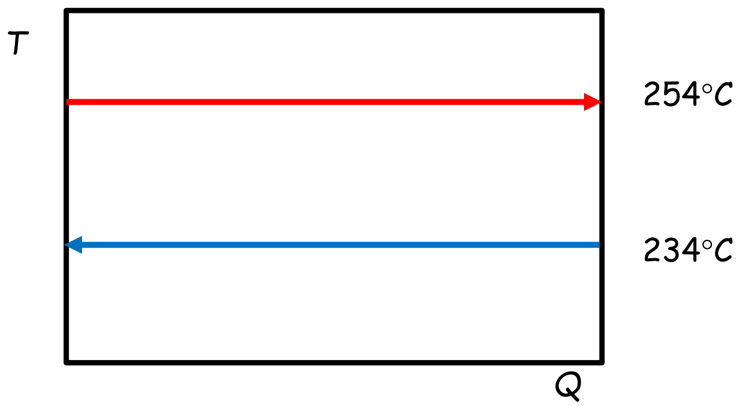
M_{oil}	M_{cw}	$T_{cw,out,2}$
1.00	1.00	40.00
0.95	0.92	40.38
0.90	0.83	40.79
0.85	0.76	41.22
0.80	0.68	41.70
0.75	0.61	42.20
0.70	0.55	42.80
0.65	0.48	43.40
0.60	0.43	44.10
0.55	0.37	44.90
0.50	0.32	45.80



Maximum scale-down = 0.54 or 54%

39. A reboiler is a heat exchanger used to add heat to a distillation column. In a typical reboiler, an almost pure material is vaporized at constant temperature, with the energy supplied by condensing steam at constant temperature. Suppose that steam is condensing at 254°C to vaporize an organic at 234°C. It is desired to scale up the throughput of the distillation column by 25%, meaning that 25% more organic (the process fluid) must be vaporized. What will be the new operating conditions in the reboiler (numerical value for temperature, qualitative answer for pressure)? Suggest at least two possible answers.

Consider the T-Q diagram for the reboiler



Without specifics about the heat transfer area, only approximate answers can be given. The performance equations are (1 – base case, 2- new case, *M* = process, *m* = steam)

$$Q = \frac{Q_2}{Q_1} = \frac{\dot{M}_2 \cancel{\lambda}_{s2}}{\dot{M}_1 \cancel{\lambda} s_{s1}} = M = 1.25$$

$$Q = \frac{Q_2}{Q_1} = \frac{\dot{m}_2 \cancel{\lambda}_{p2}}{\dot{m}_1 \cancel{\lambda}_{p1}} = m = 1.25$$

$$Q = \frac{Q_2}{Q_1} = \frac{U_2 \cancel{A_2} \Delta T_{lm,2} \cancel{F_2}}{U_1 \cancel{A_1} \Delta T_{lm,1} \cancel{F_1}} = \frac{U_2 \Delta T_2}{U_1 \Delta T_1} = \frac{U_2}{U_1} \frac{\Delta T_2}{(20)} = 1.25$$

If we assume that neither heat transfer coefficient is a function of the temperature driving force or flowrate then $U_2 = U_1$ and ΔT_2 = 25°C.

Two simple solutions are possible:

1. Decrease process temperature by 5°C to 239°C by lowering the column pressure
2. Increase the steam temperature by 5°C to 259°C by desuperheating the steam less or by using higher pressure steam.

40. In a shell-and-tube heat exchanger, initially h_o = 500 W/m^{2}K and h_i = 1500 W/m^{2}K. The fouling resistances and wall resistance may be assumed to be negligible. If the mass flowrate of the tube-side stream is increased by 30%, what change in the mass flowrate of the shell-side stream is required to keep the overall heat transfer coefficient constant?

For shell-side $h_o \propto \mathrm{Re}_o^{0.6}$, for tube-side $h_i \propto \mathrm{Re}_i^{0.8}$ and $U \cong \left[\frac{1}{h_o} + \frac{1}{h_i}\right]^{-1}$

Using subscripts 1 and 2 to denote base case and new case, respectively, and M to represent the ratios of mass flows, we have,

$$\frac{h_{o,2}}{h_{o,1}} \propto (M_{shell})^{0.6} \Rightarrow h_{o,2} = 500 M_{shell}^{0.6} \text{ W/m}^2\text{K}$$

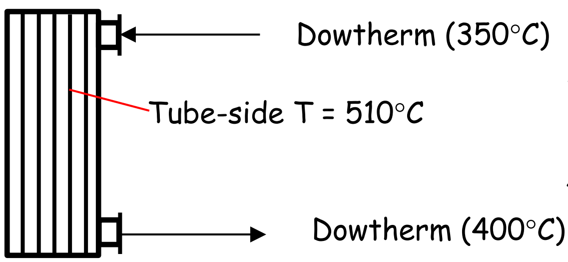
$$\frac{h_{i,2}}{h_{i,1}} \propto (M_{tube})^{0.8} \Rightarrow h_{o,2} = 1500(1.3)^{0.8} = 1850.3 \text{ W/m}^2\text{K}$$

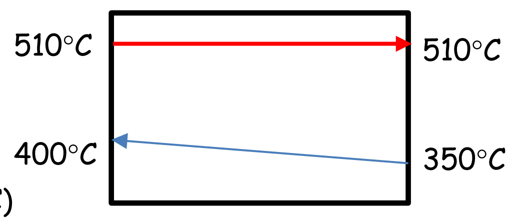
$$\frac{U_2}{U_1} \cong \frac{\left[\frac{1}{h_{o,2}} + \frac{1}{h_{i,2}}\right]^{-1}}{\left[\frac{1}{h_{o,1}} + \frac{1}{h_{i,1}}\right]^{-1}} = \frac{\left[\frac{1}{500(M_{shell})^{0.6}} + \frac{1}{1850.3}\right]^{-1}}{\left[\frac{1}{500} + \frac{1}{1500}\right]^{-1}} = 1 \Rightarrow M_{shell} = 0.9030$$

Solving for M_{shell} gives that the shell side flow rate must be reduced to 90.3% of its original value.

41. A reaction occurs in a shell-and-tube reactor. One type of shell-and-tube reactor is essentially a heat exchanger with catalyst packed in the tubes. For an exothermic reaction, heat is removed by circulating a heat transfer fluid through the shell. In this situation, the reaction occurs isothermally at 510°C. The Dowtherm always enters the shell coil at 350°C. In the design case, it exits at 400°C. In the base case, the heat transfer resistance on the reactor side is equal to that on the Dowtherm side. If it is required to increase throughput in the reactor by 25%, what is the required Dowtherm flowrate and the new Dowtherm exit temperature. You should assume that the reaction temperature remains at 510°C?

Design Case





$$\Delta T_{lm,1} = \frac{(510-350)-(510-400)}{\ln\frac{510-350}{510-400}} = 133.4^{\circ}\text{C}$$

$U \cong h_o$

Taking ratios between the new case (2) and the design case (1) gives

$$\frac{U_2}{U_1} = M_{DT}^{0.6}$$

$$\frac{Q_2}{Q_1} = 1.25 \text{ (reactor/tube side)} \qquad \text{(a)}$$

$$\frac{Q_2}{Q_1} = \frac{m_{DT,2}c_{p,DT,2}(T_{DT,out,2}-T_{DT,in,2})}{m_{DT,1}c_{p,DT,1}(T_{DT,out,1}-T_{DT,in,1})} = M_{DT}\frac{(T_{DT,out,2}-350)}{400-350} = M_{DT}\frac{(T_{DT,out,2}-350)}{50} \qquad \text{(b)}$$

$$\frac{Q_2}{Q_1} = \frac{U_2A_2\Delta T_{lm,2}}{U_1A_1\Delta T_{lm,1}} = \frac{U_2}{U_1}\frac{[(510-350)-(510-T_{DT,out,2})]}{(123.3)\ln\frac{(510-350)}{(510-T_{DT,out,2})}} = M_{DT}^{0.6}\frac{(T_{DT,out,2}-350)}{(133.4)\ln\frac{(160)}{(510-T_{DT,out,2})}} \qquad \text{(c)}$$

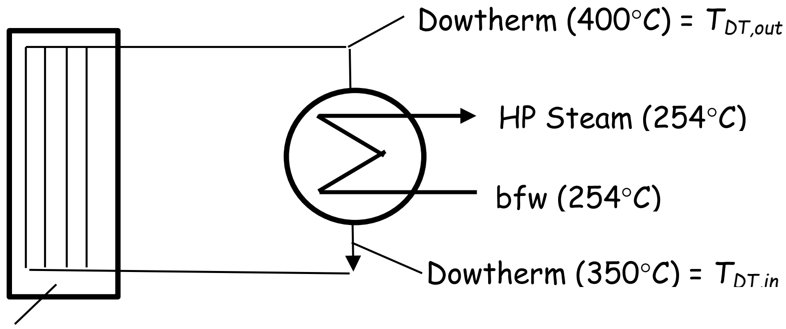
Solving Eqns. a-c for M_{DT} and $T_{DT,out,2}$ gives,

M_{DT} = 1.3974 or 139.7% of the design flow of Dowtherm and
$T_{DT,out,2}$ = 394.7°C

Note: In solving this problem, we have only considered the heat transfer aspects of the solution, i.e., how can we remove 25% more heat from the reactor. How to increase the production by 25% is another matter. For a shell and tube type reactor this would need a combination of increased *P* and/or *T* and/or a more active catalyst. To solve this problem from both the process and cooling aspects requires a more involved and complicated approach that is avoided here.

42. In Problem 41, the reactor is now a fluidized bed where Dowtherm circulates through tubes in the reactor with the reaction in the shell. The Dowtherm then flows to a heat exchanger in which boiler feed water (bfw) is vaporized on the shell side to high-pressure steam at 254°C. This removes the heat absorbed by the Dowtherm stream in the reactor so the Dowtherm can be recirculated to the reactor. So, the Dowtherm is in a closed loop. The resistance in the steam boiler is all on the Dowtherm side. In the base case of the reactor, the resistance on the reaction side is four times that on the Dowtherm side. The desired increase in production can be accomplished by adding 25% more catalyst to the bed and operating at the same temperature, that is what is assumed in this analysis.

Design Conditions



a. Write the six equations needed to model the performance of this system. Use subscripts 1 and 2 to represent design (base) case and the new case.

For Reactor (heat exchanger – Dowtherm in tubes)

$$\Delta T_{lm,1} = \frac{(510-350)-(510-400)}{\ln\frac{(510-350)}{(510-400)}} = \frac{50}{\ln\frac{160}{110}} = 133.4^{\circ}\text{C}$$

$$\frac{U_2}{U_1} = \frac{\left[\frac{1}{h}+\frac{1}{4hM_{DT}^{0.8}}\right]^{-1}}{\left[\frac{1}{h}+\frac{1}{4h}\right]^{-1}} = \frac{1.25}{\left[1+\frac{1}{4M_{DT}^{0.8}}\right]}$$

$$\frac{Q_2}{Q_1} = Q = \frac{m_{DT,2}c_{p,2}\Delta T_{DT,2}}{m_{DT,1}c_{p,1}\Delta T_{DT,1}} = M_{DT}\,\frac{(T_{DT,out,2}-T_{DT,in,1})}{(400-350)} = M_{DT}\,\frac{(T_{DT,out,2}-T_{DT,in,1})}{(50)} \qquad \text{(a)}$$

$$Q = \frac{m_{p,2}\xi_2 \Delta H_{R,2}}{m_{p,1}\xi_1 \Delta H_{R,1}} = 1.25 \qquad \text{(b)}$$

$$Q = \frac{U_2 A_2 \Delta T_{\text{lm},R,2}}{U_1 A_1 \Delta T_{\text{lm},R,1}} = \frac{U_2}{U_1} \frac{[(510 - T_{DT,in,2}) - (510 - T_{DT,out,2})]}{(133.4)\ln\frac{(510 - T_{DT,in,2})}{(510 - T_{DT,out,2})}}$$

$$Q = \frac{1.25}{\left[1 + \frac{1}{4M_{DT}^{0.8}}\right]} \frac{[(T_{DT,out,2} - T_{DT,out,1})]}{(133.4)\ln\frac{(510 - T_{DT,in,2})}{(510 - T_{DT,out,2})}} \qquad \text{(c)}$$

For Heat Exchanger (heat exchanger – steam boiler)

$$\Delta T_{lm,1} = \frac{(400-254)-(350-254)}{\ln\frac{(400-254)}{(350-254)}} = \frac{50}{\ln\frac{146}{96}} = 119.3^{\circ}\text{C}$$

$$\frac{U_2}{U_1} = \frac{\left[\frac{1}{hM_{DT}^{0.8}}\right]^{-1}}{\left[\frac{1}{h}\right]^{-1}} = M_{DT}^{0.8}$$

$$Q = \frac{m_{DT,2}c_{p,2}\Delta T_{DT,2}}{m_{DT,1}c_{p,1}\Delta T_{DT,1}} = M_{DT}\frac{(T_{DT,out,2} - T_{DT,in,1})}{(400-350)} = M_{DT}\frac{(T_{DT,out,2} - T_{DT,in,1})}{(50)} \qquad \text{(d)}$$

$$Q = \frac{\dot{m}_{water,2}\lambda_{water,2}}{\dot{m}_{water,1}\lambda_{water,1}} = M_{steam} = 1.25 \qquad \text{(e)}$$

$$Q = \frac{U_2 A_2 \Delta T_{\text{lm},Ex,2}}{U_1 A_1 \Delta T_{\text{lm},Ex,1}} = \frac{U_2}{U_1}\frac{[(T_{DT,out,2} - 254) - (T_{DT,in,2} - 254)]}{(119.3)\ln\frac{(T_{DT,out,2} - 254)}{(T_{DT,in,2} - 254)}}$$

$$Q = M_{DT}^{0.8}\frac{[(T_{DT,out,2} - T_{DT,in,2})]}{(119.3)\ln\frac{(T_{DT,out,2} - 254)}{(T_{DT,in,2} - 254)}} \qquad \text{(f)}$$

b. How many unknowns are there?

Q, M_{DT}, $T_{DT,out,2}$, $T_{DT,in,2}$, M_{steam}.

Note that although there are 6 equations (a-f) only 5 of them are independent as Eqns, a and d are the same, i.e., the heat balance on the Dowtherm in the reactor and steam generator are the same for steady state operations.

Moreover, Eqns. b and e are explicit, i.e., 25% more heat must be removed from the reactor and this heat will produce 25% more steam in the exchanger.
Solving Equations a,c, and f for M_{DT}, $T_{DT,out,2}$, $T_{DT,in,2}$, gives,

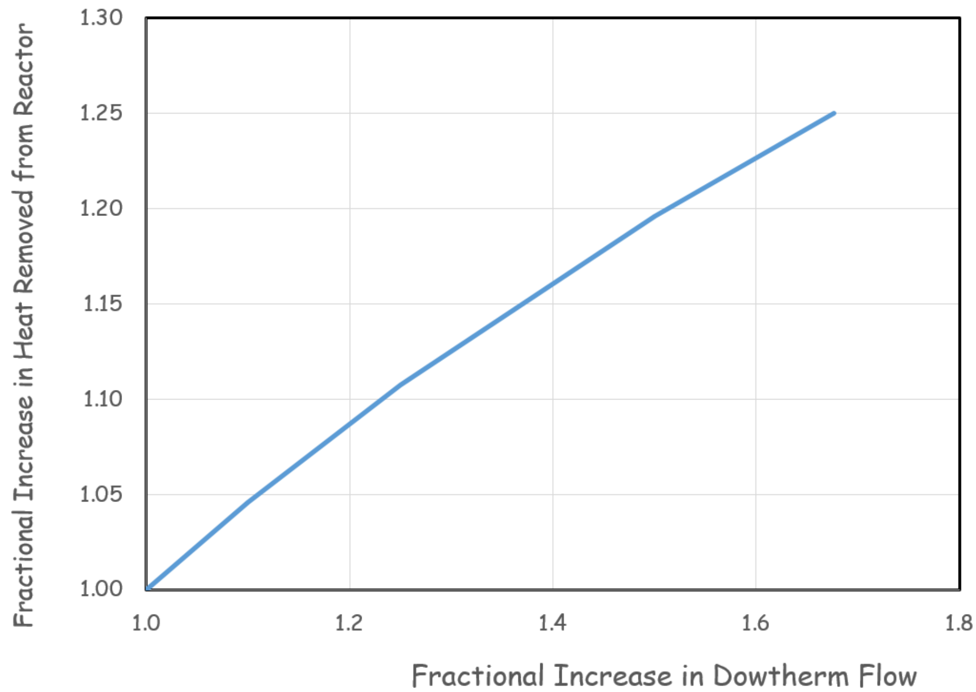
M_{DT} = 1.677
$T_{DT,out,2}$ = 372.4°C
$T_{DT,in,2}$ = 335.1°C

c. If the temperature of the reactor is to be maintained at 510°C, determine the amount of process scale-up and all other unknowns for the following cases:

i) 10% increase in Dowtherm flowrate
Set M_{DT} = 1.1 and solve for remaining variables to give,
Q = 1.046
$T_{DT,out,2}$ = 394.9°C
$T_{DT,in,2}$ = 347.4°C

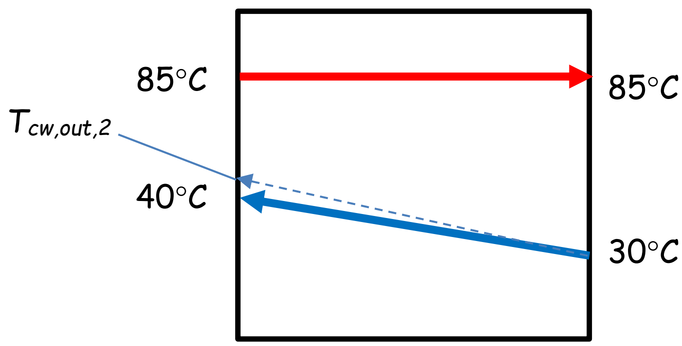
ii) 25% increase in Dowtherm flowrate
Set M_{DT} = 1.25 and solve for remaining variables to give,
Q = 1.107
$T_{DT,out,2}$ = 388.1°C
$T_{DT,in,2}$ = 343.8°C

iii) 50% increase in Dowtherm flowrate.
Set M_{DT} = 1.50 and solve for remaining variables to give,
Q = 1.196
$T_{DT,out,2}$ = 378.3°C
$T_{DT,in,2}$ = 338.6°C


Fractional Increase in Heat Removed from Reactor
1.30
1.25
1.20
1.15
1.10
1.05
1.00
1.0
1.2
1.4
1.6
1.8
Fractional Increase in Dowtherm Flow

43. It is necessary to decrease the capacity of an existing distillation column by 30%. As a consequence, the amount of liquid condensed in the shell of the overhead condenser must decrease by the same amount (30%). In this condenser, cooling water (in tubes) is available at 30°C, and, under present operating conditions, exits the condenser at 40°C. The maximum allowable return temperature without a financial penalty assessed to your process is 45°C. Condensation takes place at 85°C.

1 – design case, 2 – new (scaled-down)case



a. If the limiting resistance is on the cooling water side, what is the maximum scale-down possible based on the condenser conditions without incurring a financial penalty? What is the new outlet temperature of cooling water? By what factor must the cooling water flow change?

Design Case

$$\Delta T_{lm,1} = \frac{(85-30)-(85-40)}{\ln\frac{(85-30)}{(85-40)}} = \frac{10}{\ln\frac{55}{45}} = 49.83^{\text{o}}\text{C}$$

Energy Balances

$$\frac{Q_2}{Q_1} = Q = \frac{m_{cw,2}c_{p,2}\Delta T_{cw,2}}{m_{cw,1}c_{p,1}\Delta T_{cw,1}} = M_{cw}\frac{(T_{cw,out,2}-30)}{(40-30)} = M_{cw}\frac{(T_{DT,out,2}-30)}{(10)} \quad \text{(a)}$$

$$Q = \frac{m_{p,2}\lambda_2}{m_{p,1}\lambda_1} = M_p = 0.70 \quad \text{(b)}$$

Design Equation

$$\frac{U_2}{U_1} = \frac{\left[\frac{1}{hM_{cw}^{0.8}}\right]^{-1}}{\left[\frac{1}{h}\right]^{-1}} = M_{cw}^{0.8}$$

$$Q = \frac{U_2 A_2 \Delta T_{\mathrm{lm},R,2}}{U_1 A_1 \Delta T_{\mathrm{lm},R,1}} = \frac{U_2}{U_1} \frac{[(85-30)-(85-T_{cw,out,2})]}{(49.83)\ln\dfrac{(85-30)}{(85-T_{cw,out,2})}} = M_{cw}^{0.8} \frac{(T_{cw,out,2}-30)}{(49.83)\ln\dfrac{(55)}{(85-T_{cw,out,2})}} \qquad \text{(c)}$$

Solving Equations a-c for unknowns,

<u>***M_{cw}* = 0.6474**</u> (a decrease of ~36% from design case)
<u>***$T_{cw,out,2}$* = 40.81°C**</u>

b. Repeat part a. if the resistances are such that the cooling water heat transfer coefficient is three times the condensing heat transfer coefficient. You may assume that the value of the condensing heat transfer coefficient does not change appreciably from the design case. Does your solution exceed the maximum cooling water return temperature of 45°C? If so, can you suggest other options to decrease the condenser duty by 30% that would not violate the cooling water return temperature constraint?

Let $h_o = h$ then $h_i = 3h$. From Table 2.9, h_o is a function of $(T_{sat}-T_w)^{-1/4}$ but problem says ignore this change.

Energy Balances

$$\frac{Q_2}{Q_1} = Q = \frac{m_{cw,2} c_{p,2} \Delta T_{cw,2}}{m_{cw,1} c_{p,1} \Delta T_{cw,1}} = M_{cw} \frac{(T_{cw,out,2}-30)}{(40-30)} = M_{cw} \frac{(T_{DT,out,2}-30)}{(10)} \qquad \text{(a)}$$

$$Q = \frac{m_{p,2}\lambda_2}{m_{p,1}\lambda_1} = M_p = 0.70 \qquad \text{(b)}$$

Design Equation – follow procedure included in Example 2.20

$$\frac{U_2}{U_1} = \frac{\left[\dfrac{1}{3hM_{cw}^{0.8}} + \dfrac{1}{h}\right]^{-1}}{\left[\dfrac{1}{3h} + \dfrac{1}{h}\right]^{-1}} = \frac{(1.3333)}{\left[1 + \dfrac{1}{3M_{cw}^{0.8}}\right]}$$

$$Q = \frac{U_2 A_2 \Delta T_{\mathrm{lm},R,2}}{U_1 A_1 \Delta T_{\mathrm{lm},R,1}} = \frac{(1.3333)}{\left[1 + \dfrac{1}{3M_{cw}^{0.8}}\right]} \frac{(T_{cw,out,2}-30)}{(49.83)\ln\dfrac{(55)}{(85-T_{cw,out,2})}} \qquad \text{(c)}$$

Solving Equations a-c for unknowns,

<u>***M_{cw}* = 0.3787**</u> (a decrease of ~62% from design case)

$T_{cw,out,2}$ = 48.49°C (this exceeds the maximum cooling water return temperature of ~45°C)

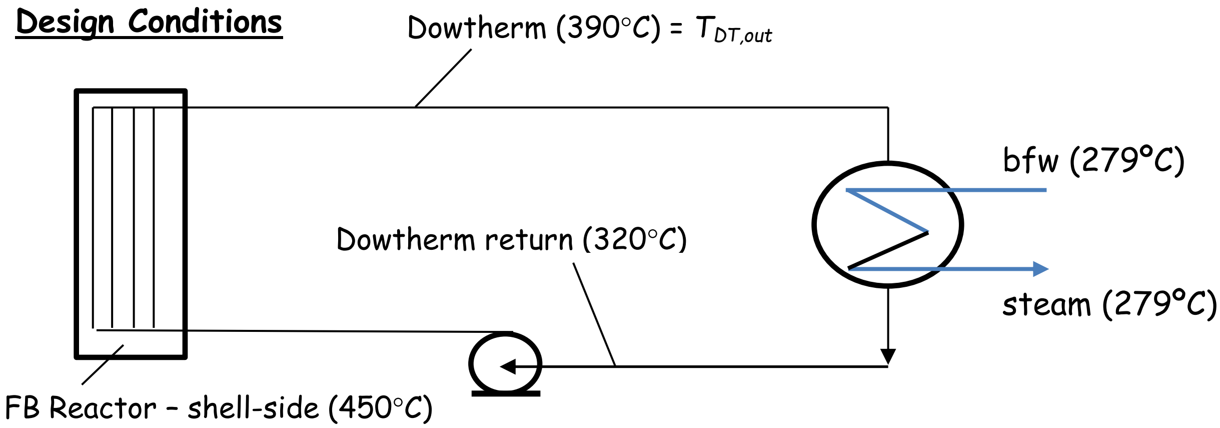
A possible alternative to avoid $T_{cw,out,2}$ >45°C is to decrease the pressure in the tower – this has the effect of reducing the condensation temperature.

44. A reaction occurs in a well-mixed fluidized bed reactor maintained at 450°C. Heat is removed by Dowtherm A™ circulating through a coil in the fluidized bed. In the design or base case, The Dowtherm enters and exits the reactor at 320 and 390°C, respectively. In the base case, the heat transfer resistance on the reactor side is two times that on the Dowtherm side. It may be assumed that for the fluidized bed, the heat transfer coefficient on the fluidized side is essentially constant and independent of the throughput.

The Dowtherm is cooled in an external exchanger that produces steam at 900 psig (T_{sat} = 279°C). The Dowtherm pump limits the maximum increase in Dowtherm flowrate, so the pump limits the heat removal rate based on its pump and system curves. For the current situation, the maximum increase in Dowtherm flowrate through the reactor and boiler is estimated to be 34%, by how much can the process be scaled-up.

You may assume that the limiting heat transfer coefficient on the steam boiler is Dowtherm that flows through the tube-side of the exchanger.

Design Conditions



Reactor

Energy Balances

$$\frac{Q_2}{Q_1} = Q = \frac{m_{DT,2}c_{p,2}\Delta T_{DT,2}}{m_{DT,1}c_{p,1}\Delta T_{DT,1}} = M_{DT}\frac{(T_{DT,out,2} - T_{DT,in,2})}{(390-320)} = M_{DT}\frac{(T_{DT,out,2} - T_{DT,in,2})}{(70)} \quad \text{(a)}$$

$$Q = \frac{m_{p,2}\xi_2\Delta H_{R,2}}{m_{p,1}\xi_1\Delta H_{R,1}} = \text{reactor scale-up factor} \quad \text{(b)}$$

Design Equation

For the base case, $h_o = h_i/2 = h$, and

$$\Delta T_{lm,1} = \frac{(450-320)-(450-390)}{\ln\frac{(450-320)}{(450-390)}} = \frac{70}{\ln\frac{130}{60}} = 90.53^{o}\text{C}$$

$$\frac{U_2}{U_1} = \frac{\left[\frac{1}{h_{o,2}}+\frac{1}{h_{i,2}}\right]^{-1}}{\left[\frac{1}{h_{o,1}}+\frac{1}{h_{i,1}}\right]^{-1}} = \frac{\left[\frac{1}{h}+\frac{1}{2hM_{DT}^{0.8}}\right]^{-1}}{\left[\frac{1}{h}+\frac{1}{2h}\right]^{-1}} = \frac{1.5}{\left[1+\frac{1}{2M_{DT}^{0.8}}\right]}$$

$$Q = \frac{U_2A_2\Delta T_{\text{lm},R,2}}{U_1A_1\Delta T_{\text{lm},R,1}} = \frac{U_2}{U_1}\frac{[(450-T_{DT,in,2})-(450-T_{DT,out,2})]}{(90.53)\ln\frac{(450-T_{DT,in,2})}{(450-T_{DT,out,2})}}$$

$$Q = \frac{1.5}{\left[1+\frac{1}{2M_{cw}^{0.8}}\right]}\frac{(T_{DT,out,2}-T_{DT,in,2})}{(90.53)\ln\frac{(450-T_{DT,in,2})}{(450-T_{DT,out,2})}} \quad \text{(c)}$$

Boiler

Energy Balances

$$\frac{Q_2}{Q_1} = Q = \frac{m_{DT,2}c_{p,2}\Delta T_{DT,2}}{m_{DT,1}c_{p,1}\Delta T_{DT,1}} = M_{DT}\frac{(T_{DT,out,2}-T_{DT,in,2})}{(390-320)} = M_{DT}\frac{(T_{DT,out,2}-T_{DT,in,2})}{(70)} \quad \text{(d)}$$

$$Q = \frac{m_{steam,2}\lambda_2}{m_{steam,1}\lambda_1} = M_{steam} \quad \text{(e)}$$

Design Equation

For the base case, $h_i = h$, $U = h$, and

$$\Delta T_{lm,1} = \frac{(390-279)-(320-279)}{\ln\frac{(390-279)}{(320-279)}} = \frac{70}{\ln\frac{111}{41}} = 70.28^{o}\text{C}$$

$$\frac{U_2}{U_1} = \frac{\left[\frac{1}{h_{o,2}}+\frac{1}{h_{i,2}}\right]^{-1}}{\left[\frac{1}{h_{o,1}}+\frac{1}{h_{i,1}}\right]^{-1}} = \frac{\left[\frac{1}{hM_{DT}^{0.8}}\right]^{-1}}{\left[\frac{1}{h}\right]^{-1}} = M_{DT}^{0.8}$$

$$Q = \frac{U_2A_2\Delta T_{\text{lm},R,2}}{U_1A_1\Delta T_{\text{lm},R,1}} = \frac{U_2}{U_1}\frac{[(T_{DT,out,2}-279)-(T_{DT,in,2}-279)]}{(70.28)\ln\frac{(T_{DT,out,2}-279)}{(T_{DT,in,2}-279)}}$$

$$Q = M_{cw}^{0.8}\frac{(T_{DT,out,2} - T_{DT,in,2})}{(70.28)\ln\frac{(T_{DT,out,2} - 279)}{(T_{DT,in,2} - 279)}} \tag{f}$$

Setting M_{DT} = 1.34 (maximum increase in flow) and solving Equations (a, c, and f) for 3 unknowns gives,

Q = 1.167
$T_{DT,out,2}$ = 379.06°C
$T_{DT,in,2}$ = 318.11°C
Therefore the maximum scale-up of the reactor from the base case is 1.167 or 16.7 %.