Solution 12.1

The procedure will follow that used in the solution to problem 12.2.

As the cooling water flow-rate will be around half that of the caustic solution, it will be best to put the cooling water through the tubes and the solution through the annular jacket.

The jacket heat transfer coefficient can be estimated by using the hydraulic mean diameter in equation 12.11.

Solution 12.2

Heat balance

$$Q + m\,C_p\,(T_{out} - T_{in})$$

$$Q = (6000/3600) \times 4.93 \times (65 - 15) = 411 \text{ kW}$$

Cross-section of pipe = $(\Pi/4)(50 \times 10^{-3})^2 = 1.963 \times 10^{-3}$ m

Fluid velocity, $u = \dfrac{6000}{3600} \times \dfrac{1}{866} \times \dfrac{1}{1.963 \times 10^{-3}} = 0.98$ m/s

$$Re = \frac{866 \times 0.98 \times (50 \times 10^{-3})}{0.44 \times 10^{-3}} = 96{,}441$$

$$Pr = \frac{4.3 \times 10^{-3} \times 0.44 \times 10^{-3}}{0.3895} = 4.86$$

Liquid is not viscous and flow is turbulent, so use eqn 12.11, with C = 0.023 and neglect the viscosity correction factor.

$$Nu = 0.023(96441)^{0.8}(4.86)^{0.33} = 376$$

$$h = (0.385/50 \times 10^{-3}) \times 376 = 2895 \text{ Wm}^{-2}\,{}^{\circ}\text{C}^{-1}$$

Take the steam coefficient as 8000 $\text{Wm}^{-2}\,{}^{\circ}\text{C}^{-1}$

$$1/U_o = \frac{1}{8000} + \frac{60 \times 10^{-3}(60/50)}{2 \times 480} + \frac{60}{50} \times \frac{1}{2895} \qquad (12.2)$$

$$U_o = 1627 \text{ Wm}^{-2}\,{}^{\circ}\text{C}^{-1}$$

$$\Delta T_{lm} = (85 - 35)/\text{Ln}\,(85/35) = 56.4\,{}^{\circ}\text{C} \qquad (12.14)$$

$$A_o = (411 \times 10^3)/(1627 \times 56.4) = 4.5 \text{ m}^2 \qquad (12.1)$$

$A_o = \Pi \times d_o \times L$, $L = 4.5 /(\Pi \times 60 \times 10^{-3}) = 23.87$ m

Number of lengths = 23.87/ 3 = 8 (rounded up)

Check on viscosity correction

Heat flux, $q = 411/4.5 = 91.3$ kW/m^2

ΔT across boundary layer = q/h = 91,300/2895 = 32 °C

Mean wall temperature = (15 + 65)/2 + 32 = 72 °C

From table, $\mu_w \cong 300$ mN m^{-2} s

$\mu/\mu_w = (0.44/0.3)^{0.14} = 1.055$, so correction would increase the coefficient and reduce the area required.

Leave estimate at 8 lengths to allow for fouling.

Solution 12.3

Physical properties. from tables

Steam temperature at 2.7 bar = 130 °C

Mean water temperature = (10 + 70)/2 = 40 °C

Density = 992.2 kg/m^3, specific heat = 4.179 kJ kg^{-1} °C^{-1}, viscosity = 651 x 10^{-3} N m^{-2} s, Thermal conductivity = 0.632x 10^{-3} W m^{-1} °C^{-1}, Pr = 4.30.

Take the material of construction as carbon steel, which would be suitable for uncontaminated water and steam, thermal conductivity 50 W m^{-1} °C^{-1}.

Try water on the tube side.

Cross–sectional area = $124 (\Pi /4 \times (15 \times 10^{-3})^2) = 0.0219$ m^2

$$\text{Velocity} = \frac{50000}{3600} \times \frac{1}{992.2} \times \frac{1}{0.0219} = 0.64 \text{ m/s}$$

$$Re = \frac{992.2 \times 0.64 \times 15\times10^{-3}}{0.651 \times 10^{-3}} = 14{,}632 \qquad (1.5 \times 10^{-4})$$

From Fig 12.23, $j_h = 4 \times 10^{-3}$

$Nu = 4 \times 10^{-3} \times 14632 \times 4.0^{-0.33} = 92.5$

$h_i = 92.5 \times (632 \times 10^{-3}) / 15 \times 10^{-3} = 3897\ Wm^{-2}\ °C^{-1}$

Allow a fouling factor of 0.0003 on the waterside and take the condensing steam coefficient as 8000 $Wm^{-2}\ °C^{-1}$; see section 12.4 and 12.10.5.

$$1/U_o = (1/3897 + 0.0003)(19/15) + \frac{19 \times 10^{-3} Ln(19/15)}{2 \times 50} + 1/8000 = .000875$$

$U_o = 1143\ Wm^{-2}\ °C^{-1}$

$$\Delta T_{lm} = \frac{(130 - 70) - (130 - 10)}{Ln(60/120)} = 86.6\ °C \qquad (12.4)$$

The temperature correction factor, F_t , is not needed as the steam is at a constant temperature.

Duty, $Q = (50{,}000/3600) \times 4.179(70 - 10) = 3482.5$ kW

$$\text{Area required, } A_o = \frac{3482.5 \times 10^3}{1143 \times 86.6} = 35.2\ m^2$$

Area available = $124(\Pi \times 19 \times 10^{-3} \times 4094 \times 10^{-3}) = 30.3\ m^2$

So the exchanger would not meet the duty, with the water in the tubes.

Try putting the water in the shell.

$$\text{Flow area, } A_s = \frac{(24 - 19)}{24} 337 \times 10^{-3} \times 106 \times 10^{-3} = 7.44 \times 10^{-3}\ m^2 \qquad (12.21)$$

Hydraulic mean diameter, $d_e = (1.10/19)(24^2 - 0.917 \times 19^2) = 14.2$ mm (12.2)

Velocity, $u_s = 50000/3600 \times 1/992.2 \times 1/7.44 \times 10^{-3} = 1.88$ m/s

$$Re = \frac{992.2 \times 1.88 \times 14.2 \times 10^{-3}}{0.65 \times 10^{-3}} = 40{,}750 \qquad (4.1 \times 10^4)$$

From Fig 12.29 for 25% baffle cut, $j_h = 3.0 \times 10^{-3}$

$Nu = 3.0 \times 10^{-3} \times 40750 \times 4.3^{0.33} = 198$

$h_s = 198 \times 632 \times 10^{-3} / 14.2 \times 10^{-3} = 8812\ Wm^{-2}\ °C^{-1}$

A considerable improvement on the coefficient with the water in the tubes.

$$1/U_o = (1/8000)(19/15) + \frac{19 \times 10^{-3} Ln(19/15)}{2 \times 50} + (1/8812 + 0.0003)$$

U_o = 1621 $Wm^{-2}\,°C^{-1}$

$$A_o = \frac{3482.5 \times 10^3}{1621 \times 86.6} = 24.80 \text{ m}^2$$

So the exchanger should be capable of fulfilling the duty required, providing the water in put through the shell.

Note; the viscosity correction factor has been neglected when estimating the heat transfer coefficients. Water is not a viscous liquid, sot he correction would be small.

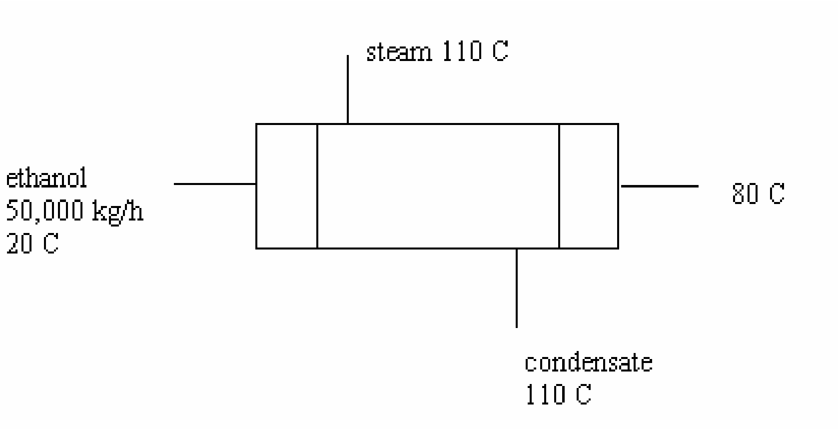
In practice, it would be necessary to check that the pressure drop on the water-side could be met by the supply pressure

Solution 12.4

There is no unique solution to a design problem. The possible solutions for this design have been constrained by specifying the tube dimensions and the disposition of the fluid streams. Specifying steam as the heating medium and putting in the shell simplifies the calculations. It avoids the need to make tedious, and uncertain, calculations to estimate the shell-side coefficient.

The heat exchanger design procedure set out in Fig. A, page 680, will be followed.

Step 1 Specification



Flow-rate of ethanol = 50000/3600 = 13.89 kg/s

Ethanol mean temperature = (20 + 80)/2 = 50 °C

Mean specific heat = 2.68 kJ kg^{-1} °C $^{-1}$ (see table step 2)

Duty = m C_p ($T_1 - T_2$) = 13.89 x 2.68 x (80 – 20) = 2236 kW

Step 2 Physical properties

Saturation temperature steam at 1.5 bar, from steam tables, = 111.4 °C

Thermal conductivity of carbon steel = 50 W m^{-1} $°C^{-1}$

Properties of ethanol

Temp °C	C_p, kJ kg^{-1} $°C^{-1}$	k, W m^{-1} $°C^{-1}$	ρ, kg/m^3	μ, N m^{-2} s x 10^3
20	2.39	0.164	789.0	1.200
30	2.48	0.162	780.7	0.983
40	2.58	0.160	772.1	0.815
50 (mean)	2.68	0.158	763.2	0.684
60	2.80	0.155	754.1	0.578
70	2.92	0.153	744.6	0.495
80	3.04	0.151	734.7	0.427
90	3.17	0.149	724.5	0.371
100	3.31	0.147	719.7	0.324
110	3.44	0.145	702.4	0.284

Step 3 Overall coefficient

Ethanol is not a viscous fluid, viscosity similar to water, so take a initial value for U of 1000 Wm^{-2} $°C^{-1}$, based on the values given in Table 12.1 and Fig. 12.1.

Step 4 Passes and LMTD

A typical value for the tube velocity will be 1 to 2 m/s; see section 12.7.2.

Use 1 m/s to avoid the possibility of exceeding the pressure drop specification.

Fixing the tube-side velocity will fix the number of passes; see step 7.

$$\Delta T_{lm} = \frac{(111.4 - 80) - (111.4 - 20)}{Ln((111.4 - 20)/(111.4 - 20))} = 56.16 \text{ °C} \qquad (12.4)$$

Step 5 Area

Trial area, A = (2236 x 10^3)/(1000 x 56.16)   =   39.8 m^2 (12..1)

Step 6 Type

As the mean temperature difference between the shell and tubes is less than 80 °C, a fixed tube sheet exchanger can be used.

Ethanol in the tubes, as specified.

Step 7 Number of tubes

Surface area of one tube = $\Pi \times (29 \times 10^{-3}) \times 4 = 0.364\ m^2$ (based on the o.d.)

Number of tubes needed = 39.8/0.364 = 109.3, say 110

Cross-sectional area of one tube = $\Pi/4 \times (25 \times 10^{-3})^2 = 4.91 \times 10^{-4}\ m^2$

Volumetric flow-rate of ethanol = 13.89/763.2 = $0.0182\ m^3/s$

Tube-side velocity = volumetric flow/cross-sectional area per pass

So, cross-sectional area per pass = 0.0182/1 = $0.0182\ m^2$

Number of passes = total cross-sectional area/ cross-sectional area per pass

$$= (110 \times 4.91 \times 10^{-4})/0.0182 = 2.9$$

Take as 4 passes. This will increase the tube-side velocity to above the chosen value. So, increase the number of tubes to 120, giving a uniform 30 tubes per pass. Use E type shell.

Step 8 Shell diameter

The shell diameter is not needed at this point as the shell-side coefficient is not dependent on the diameter. Leave till after checking the overall coefficient and tube-side pressure drop.

Step 9 Tube-side coefficient

Velocity, u_t = volumetric flow-rate/cross-sectional area per pass

$$= (0.0182)/(30 \times 4.91 \times 10^{-4}) = 1.24\ m/s$$

$$Re = \frac{763.2 \times 1.24 \times 25 \times 10^{-3}}{0.684} = 34{,}589 \quad (3.6 \times 10^4)$$

From Fig. 12.23, $j_h = 3.4 \times 10^{-3}$

$$Pr = \frac{2.68 \times 10^3 \times 0.684 \times 10^{-3}}{0.158} = 11.6$$

$$Nu = 3.4 \times 10^{-3}(34589)(11.6)^{0.33} = 264 \qquad (12.5)$$

$$h_i = (264 \times 0.158)/(25 \times 10^{-3}) = 1668\ Wm^{-2}\,°C^{-1}$$

The viscosity correction factor has been neglected as ethanol is not viscous.

Step 10 Shell-side coefficient

Take the shell-side coefficient for condensing steam as 8000 $Wm^{-2}\,°C^{-1}$; section 12.10.5
This includes the fouling factor.

Step 11 Overall coefficient

Ethanol should not foul the tubes, so take the fouling factor for the tube-side as 0.0002, that for light organics in Table 12.2.

$$1/U_o = (1/1668 + 0.0002)(29/25) + \frac{29 \times 10^{-3}(29/25)}{2 \times 50} + 1/8000 = 0.001389$$

$$U_o = 720 \text{ Wm}^{-2}\,°\text{C}^{-1}$$

Too low, so back to Step 3. Put the overall coefficient = 720 $Wm^{-2}\,°C^{-1}$

Area required = $(2236 \times 10^3)/(720 \times 56.16) = 52.3 \text{ m}^2$

Number of tubes = 52.3/0.364 = 143.7 (144)

Try 144 tubes with 4 passes.

New tube velocity = 1.24 x 120/144 = 1.03 m/s

New Re = 34589 x 1.03/1.24 = 28,731 (2.9×10^4)

From Fig 12.23, $j_h = 3.8 \times 10^{-3}$

$$Nu = 3.8 \times 10^{-3}(28731)(11.6)^{0.33} = 245$$

$$h_i = (245 \times 0.158)/(25 \times 10^{-3}) = 1548 \text{ Wm}^{-2}\,°\text{C}^{-1}$$

$$1/U_o = (1/1548 + 0.0002)(29/25) + \frac{29 \times 10^{-3}(29/25)}{2 \times 50} + 1/8000 = 0.001443$$

$$U_o = 693 \text{ Wm}^{-2}\,°\text{C}^{-1}$$

Still too low but check pressure drop with this arrangement to see if the number of passes could be increased, rather than the number tubes.

Step 12 Pressure drops

$$\Delta P_i = 4(8 \times 3.7 \times 10^{-3} \times (4/25 \times 10^{-3}) + 2..5)(763.2 \times 1.03^2/2) = 11{,}718 \text{ N/m}^2 \qquad (12..30)$$

$$= 0.12 \text{ bar}$$

Well below the allowable drop of 0.7 bar. So, try six passes, 24 tubes per pass.

New tube-side velocity = 1.03 x 6/4 = 1.54 m/s

New Re = 28731 x 1.54/1.03 = 42,957 (4.3×10^4)

From Fig. 12.24 $j_f = 3.3 \times 10^{-3}$

$$\Delta P_i = 4(8 \times 3.3 \times 10^{-3} \times (4/25 \times 10^{-3}) + 2..5)(763.2 \times 1.54^2/2) = 24{,}341 \text{ N/m}^2$$

$= 0.24$ bar

Check on nozzle pressure drops.

Take nozzle / pipe velocity to be 2 m/s; see chapter 5, section 5.6.

Area of nozzle = volumetric flow-rate/velocity = $0.0182/2 = 0.0091\ m^2$

Nozzle diameter = $\sqrt{(4 \times 0.0091/\Pi)} = 0.108$ m

Select standard pipe size, 100 mm

Nozzle velocity = $2 \times (108/100)^2 = 2.33$ m/s

Velocity head = $u^2/2g = 2.33^2 / 2 \times 9.8 = 0.277$ m

Allow one velocity head for inlet nozzle and a half for the outlet; see section 12.8.2.

Pressure drop over nozzles = $\rho gh = 763.2 \times 9.8 \times (1.5 \times 0.277) = 3{,}108\ N/m^2$

$= 0.03$ bar

Total tube-side pressure drop = 0.24 + 0.0 3 = 0.27 bar, well below the 0.7 bar allowed .

No limiting pressure drop is specified for the shell-side.

Back to steps 9 to 11

From Fig 12.3, j_h , at Re = 4.3×10^4, = 3.2×10^{-3}

$Nu = 3.2 \times 10^{-3}(42957)(11.6)^{0.33} = 309$

$h_i = (309 \times 0.158)/(25 \times 10^{-3}) = 1{,}953\ Wm^{-2}\,°C^{-1}$

$$1/U_o = (1/1953 + 0.0002)(29/25) + \frac{29 \times 10^{-3}(29/25)}{2 \times 50} + 1/8000 = 0.001287$$

$U_o = 777\ Wm^{-2}\,°C^{-1}$

Greater than the assumed value of 720 $Wm^{-2}\,°C^{-1}$, so the design is satisfactory.

Shell-side design (Step 10)

Use a square pitch as high shell-side velocities are not rquired with a condensing vapour.

Take the tube pitch = 1.25 x tube o.d. = $29 \times 10^{-3} \times 1.25 = 36.25 \times 10^{-3}$ m

Bundle diameter, from Table 12.4, for 6 passes , square pitch , $K_1 = 0.0402$, $n_1 = 2.617$.

$D_b = 29 (144/0.0402)^{1/2.617} = 661.4$ mm

Bundle to shell clearance, from Fig 12.10, for a fixed tube sheet = 14 mm

So, Shell inside diameter = 661.4 + 14 = 675.4, round to 680 mm

A close baffle spacing is not needed for a condensing vapour. All that is needed is sufficient baffles to support the tubes. Take the baffle spacing as equal to the shell diameter, 680 mm.

Number of baffles = $(4 \times 10^3 / 680) - 1 = 5$

Step 13 Cost

From chapter 6, Fig 6.3, basic cost for carbon steel exchanger = £10,000

Type factor for fixed tube sheet = 0.8. Pressure factor 1.0.

So, cost = 10000 x 0.8 x 1.0 = £8000 at mid-1998 prices.

Step 14 Optimisation

The design could be improved, to make use of the full pressure drop allowance on the tube-side. If the number of tubes were reduced to, say 120, the tube-side velocity would be increased. This would increase the tube-side heat transfer, which would compensate for the smaller surface area.

The heat transfer coefficient is roughly proportional to the velocity raised to the power of 0.8.

$h_i \cong 1953 (144/120)^{0.8} = 2344$ $Wm^{-2}\,°C^{-1}$, giving $U_o = 1046$ $Wm^{-2}\,°C^{-1}$

So the number of tubes required = 144 x (720/1046) = 99

Pressure drop is roughly proportional to the velocity squared.

$\Delta P_i = 0.24 \times (144/120)^2 = 0.35$ bar, still well below that allowed.

To just meet the pressure drop allowance = 0.7 - 0.03 = 0.67 bar, allowing for the drop across the nozzles, the number of tubes could be reduced to $144/(0.657/0.24)^{1/2} = 87$.

So it would be worth trying a six-pass design with 15 tubes per pass.

Solution 12.5

This is a rating problem, similar to problem 12.3. The simplest way to check if the exchanger is suitable for the thermal duty is to estimate the area required and compare it with the area available. Then check the pressure drops.

Procedure

1. Carry out a heat balance to determine the rate of heat transfer required and the water flow-rate

2. Estimate the tube-side coefficient using equation 12.15.

3. Evaluate the shell-side coefficient using Kern's method, given in section 12.9 .3.

4. Determine the overall coefficient using equation 12.2.

5. Calculate the mean temperature difference; section 12.6

6. Determine the area required, equation 12.1.

7. Calculate the surface area available
= number of tubes x (Π x tube o.d. x tube- length).

If area available exceeds that required by a sufficient margin to allow for the uncertainties in the design methods, particularly Kern's method, say +20%, accept that the exchanger will satisfy the thermal duty.

If there is not sufficient margin, more sophisticated methods should be used to check the shell-side coefficient; such a, Bell's method (using standard clearances) or a CAD method

8. Check the tube-side pressure drop, equation 12.20. Add the pressure drop over the nozzles, section 12.8.2.
.
9. Check the shell-side pressure drop, including the nozzles; use Kern's method section 12.9.3.

If the pressure drops are within limits, accept the exchanger.
If the shell-side limit is critical, a reasonable margin is needed to cover the approximate nature of the method used

Notes

1. The density of the ammonia stream will vary for the inlet to outlet due to the change in temperature. Use the mean density in the calculations.

2. The viscosity correction factor can be neglected for both streams.

Solution 12.6

First check that the critical flux will not be exceeded. Then check that the exchanger has sufficient area for the duty specified.

By interpolation, saturation temperature = 57 °C.

From steam tables, steam temp = 115.2 °C.

Duty, including sensible heat, $Q = (10{,}000/3600)(322 + 2.6(57 - 20)) = 1162$ kW

Surface area of exchanger $= (\Pi \times 30\text{x}10^{-3} \times 4.8)50 = 22.6\ m^2$

Flux $= 1162 \times 10^3 / 22.6 = 51{,}416\ W/m^2$

Critical flux, modified Zuber equation, 12.74

$$q = 0.44(45/30) \times \frac{322 \times 10^3}{\sqrt{(2 \text{ X } 50)}}(0.85 \times 9.8(535 - 14.4)14.4^2)^{0.5} = 654{,}438\ W/m^2$$

Apply the recommended factor of safety. 0.7

Critical flux for the bundle $= 0.7 \times 654438 = 458{,}107\ W/m^2$

So, the operating flux will be well below the critical flux.

Use the Foster-Zuber equation, 12.62, to estimate the boiling coefficient.

Tube surface temperature = steam temperature - temperature drop across the tube wall and condensate..

Tube wall resistance $= \dfrac{d_o \ln(d_o/d_I)}{2\ k_w} = \dfrac{30 \times 10^{-3} \ln(30/25)}{2 \times 50} = 0.000055\ °C\ m^2 W^{-1}$ (12.2)

Take the steam coefficient as 8000 $Wm^{-2}\,°C^{-1}$; section 12.10.5.

Condensate resistance $= 1/8000 = 0.000125\ °C\ m^2W^{-1}$

Temperature drop = q x resistance $= 51416 \times (0.000055 + 0.000125) = 9.3$ °C

$T_s = 115.5 - 9.3 = 106.2$ °C, $P_s = 17.3$ bar

$$h_{nb} = 0.00122 \left[\frac{0.094^{0.79}\ (2.6 \times 10^3)^{0.45}\ 535^{0.49}}{0.85^{0.5}\ (0.12 \times 10^{-3})^{0.29}\ (322 \times 10^3)^{0.24} \times 14.4^{0.24}}\right]$$

$$\times (106.2 - 57)^{0.24} \{(17.3 - 6) \times 10^5\}^{0.75} = 4647\ Wm^{-2}\,°C^{-1} \quad (12..62)$$

$$1/U_o = (1/5460)(30/25) + 0.000055 + 0.00125 \quad (12..2)$$

$$U_o = 2282\ Wm^{-2}\,°C^{-1}$$

As the predominant mode of heat transfer will be pool boiling, take the driving force to be the straight difference between steam and fluid saturation temperatures.

$$\Delta T_m = 112.5 - 57 = 55.5 \text{ °C}$$

Area required = $(1163 \times 10^3)/(2282 \times 55.5) = 9.2 \text{ m}^2$ (12.1)

Area available = 22.6 m^2. So there is adequate area to fulfil the duty required; with a good margin to cover fouling and the uncertainty in the prediction of the boiling coefficient.

Solution 12.7

This a design problem, so there will be no unique solution. The solution outlined below is my first trial design . It illustrates the design procedure and methods to be used.

The physical properties of propanol were taken from Perry's Chemical Engineering Handbook and appendix D. Those for steam and water were taken from steam tables.

Propanol, heat of vaporisation = 695.2 kJ/kg , specific heat = $2.2 \text{ kJ kg}^{-1} \text{ °C}^{-1}$.

Mass flow-rate = 30000/3600 = 8.33 kg/s

Q, condensation = 8.33 x 695.2 = 5791 kW

Q, sub-cooling = 8.33 x 22(118 – 45) = 1338 kW

For condensation, take the initial estimate of the overall coefficient as $850 \text{ Wm}^{-2} \text{ °C}^{-1}$; Table 12.1. For sub-cooling take the coefficient as $200 \text{ Wm}^{-2} \text{ °C}^{-1}$, section 12.10.7.

From a heat balance, using the full temperature rise. cooling water flow-rate =

$$(5791 + 1338)/(4.2(60 - 30)) = 56.6 \text{ kg/s}$$

Temperature rise from sub-cooling = 1388/(4.2 x 56.6) = 5.8 °C

Cooling water temperature after sub-cooling = 30 + 5.8 = 35.8 °C

Condensation

118 - - - - → - - - - - 118 °C

60 - - - - - ← - - - - - 35.8 °C

$\Delta T_M = \Delta T_{LM}$ = (118 – 60) – (118 – 35.8)]/[Ln (58/82.2)] = 69.4 °C (12.4)

Area required, A = $5791 \times 10^3/(850 \times 69.4) = 98 \text{ m}^2$

Sub-cooling

118 - - - - - → - - - - - 45 °C

35.8 - - - - ← - - - - - - 30 °C

ΔT_{LM} = [(118 – 35.8) - (45 - 30)]/[Ln (82.2/15) = 39.5 °C

R = (118 – 45)/(35.8 – 30) = 12.6, S = (35.8 – 30)/(118 – 30) = 0.07 (12.5)(12/6)

F_t = 1.0, Fig 12.19, one shell pass even tube passes. So, ΔT_M = 39.5 °C

Area required = 1338 x 10^3/(200 x 39.5) = 169 m^2

Best to use a separate sub-cooler

Condenser design

ΔT_M = ΔT_{LM} = (118 – 60) – (118 – 30)]/[Ln (58/88)] = 72 °C (12.4)

Area required = 5791 x 10^3/(850 x 72) = 95 m^2

Surface area of one tube = Π x 19 x 10^{-3} x 2.5 = 0.149 m^2

Number of tubes = 95/0.149 = 638

Put the condensing vapour in the shell.

Tube cross-sectional area = Π/4(16 x $10^{-3})^2$ = 2.01 x 10^{-4} m^2

Water velocity with one pass = (56.6/990.2)/(638 x 2.01 x 10^{-4}) = 0.45 m/s

Low, try 4 passes, 160 tubes per pass, 640 tubes

u_t = (56.6/990.2)/(160 x 2.01 x 10^{-4}) = 1.8 m/s

Looks reasonable, pressure drop should be within limit.

Outside coefficient

Use square pitch, p_t = 1.25d_o = 12.5 x 19 = 23.75 mm

Bundle diameter, D_b = 19(640 /0.158)$^{1/2.263}$ = 746 mm

Number of tubes in centre row = D_b/p_t = 746/23.75 = 32

Take N_r = 2/3 x 32 = 21

Mol mass propanol = 60.1

Density of vapour = (60.1/22.4) x (273/391) x (2.1/1) = 3.93 kg/m^3

Γ_h = W_c/LN_t = 8.33/(2.5 x 640) = 0.0052 kg/m

$$(h_c)_b = 0.95 \times 0.16 \left[\frac{740(740 - 3.93)\ 9.8}{447 \times 10^{-6} \times 0.0052}\right]^{1/3} \times 21^{-1/6} = 1207\ Wm^{-2}\,°C^{-1} \qquad (12.50)$$

Inside coefficient

$Re = (990.2 \times 1.8 \times 16 \times 10^3)/(594 \times 10^{-6}) = 48010$, $Pr = 3.89$

From Fig 12.24, $j_h = 3.3 \times 10^{-3}$. Neglect viscosity correction

$$Nu = 3.3 \times 10^{-3} (48010)(3.89)^{0.33} = 248 \qquad (12.15)$$

$h_i = 248 \times 638 \times 10^{-3}/16 \times 10^{-3} = 9889\ Wm^{-2}\ °C^{-1}$

$$1/U_o = (1/9889)(19/16) + \frac{19 \times 10^{-3}(Ln(19/16)}{2 \times 50} + 1/1207 \qquad (12.2)$$

$U_o = 1019\ Wm^{-2}\ °C^{-1}$. Greater than the initial value, with sufficient margin to allow for fouling.

Check pressure drops

Tube-side: $u_t = 1.8$ m/s, $Re = 48010$, $j_f = 3.1 \times 10^{-3}$ Fig 12.24, neglect viscosity correction factor.

$$\Delta P = 4[8 \times 3.1 \times 10^{-3} (2.5/16 \times 10^{-3}) + 2.5](990.2 \times 1.8^2/2) = 62160\ N/m^2 = 62\ kN/m^2 \qquad (12.20)$$

A bit high, only 8 kN/m^2 available to for losses in nozzles.

Could try increasing the number of tubes or reducing the number of passes, or both.

Overall coefficient is tight, so could try, say, 800 tubes with two tube passes.

Shell-side: shell clearance, for split-ring floating head exchanger = 65 mm, Fig 12.10.

So, $D_s = 746 + 65 = 811$ mm

Take baffle spacing = D_s = 811, close spacing not needed for a condenser.

$$A_s = \frac{(23.75 - 19)}{23.75}(811 \times 10^{-3} \times 811 \times 10^{-3}) = 0.132\ m^2 \qquad (12.21)$$

$u_s = (8.33/3.93)/0.132 = 16.1$ m/s

$$de = 12.7(23.75^2 - 0.785 \times 19^2)/19 = 18\ mm \qquad (12.22)$$

$Re = (16.1 \times 3.93 \times 18 \times 10^{-3})/(0.01 \times 10^{-3}) = 113891$

$j_f = 3.5 \times 10^{-2}$, Fig 12.30. Neglect viscosity correction

$\Delta P_s = 8 \times 3.5 \times 10^{-2} [(811 \times 10^{-3}/18 \times 10^{-3}) (2.5/0.811)](3.93 \times 16.1^2/2) = 19808 \text{ N/m}^2$ (12.26)

So pressure drop based on the inlet vapour flow-rate = 19.8 kN/m^2.

This is well below the limit for the total pressure drop so there is no need to refine the estimate.

Sub-cooler design

Put propanol in shell.

118 - - - -→ - - - - 45 °C

60 - - - -← - - - - 30 °C

$$\Delta T_{LM} = \frac{(118-60)-(45-30)}{\text{Ln}(58/15)} = 31.8 \text{ °C}$$

R = (118 – 45)/(60 – 30) = 2.4, S = (60 – 30)/(118 – 30) = 0.34

Correction factor F_t is indeterminant for a single shell pass exchanger, Fig 12.19.

Try two shell passes, Fig 12.20, F_t = 0.9

ΔT_m = 0.9 x 31.8 = 28.6 °C

Could use two single shell-pass exchangers to avoid the use of a shell baffle.
I will design a two shell-pass exchanger to illustrate the method.

From table 12.1, U = 250 to 750 Wm^{-2} °C^{-1} . Try 500 Wm^{-2} °C^{-1}

$$A_s = \frac{1338 \times 10^3}{500 \times 28.6} = 94 \text{ m}^2$$

Number of tubes = 94/0.149 = 631

Tube-side coefficient

Cooling water flow-rate = 1338/(4.2 x 30) = 10.62 kg/s

$$\text{Tube side velocity, single pass} = \frac{10.62 / 990.2}{631 \times 2.56 \times 10^{-4}} = 0.066 \text{ m/s}$$

Far too low, try 8 passes, 83 tubes per pass, 664 tubes.

$$u_t = \frac{10.62 / 990.2}{83 \times 2.56 \times 10^{-4}} = 0.505 \text{ m/s}$$

$$Re = \frac{990.2 \times 0.505 \times 16 \times 10^{-3}}{594 \times 10^{-6}} = 13469$$

$j_h \cong 4.0 \times 10^{-3}$, Fig 12.23

$Nu = 4.0 \times 10^{-3} \times 13469 \times (3.89)^{0.33} = 84.3$

$h_i = 84.3 \times (638 \times 10^{-3} / 16 \times 10^{-3}) = 3361 \text{ Wm}^{-2}\,°\text{C}^{-1}$

Shell-side coefficient

$D_b = 19(664/0.0365)^{1/2.675} = 743$ mm

Use 25% cut baffles, spacing 0.5 D_s = 372 mm.

Triangular pitch, $p_t = 1.25\ d_o$

$$A_s = \frac{23.75 - 19}{23.75}(743 \times 10^{-3} \times 372 \times 10^{-3}) = 0.055 \text{ m}^2 \qquad (12.21)$$

For two shell passes, the cross-flow area is taken as half that given by equation 12.21, as the shell baffle divides the shell cross-section into two equal halves.

So, $u_s = (8.33/752) / (0.055/2) = 0.403$ m/s

$d_e = (1.10/19)/(23.75^2 – 0.917 \times 19^2) = 13.5$ mm

$$Re = \frac{752 \times 0.403 \times 13.5 \times 10^{-3}}{508 \times 10^{-6}} = 8054$$

From Fig 12.29, $j_h = 6.3 \times 10^{-3}$. Neglect viscosity correction

$Pr = (2.2 \times 10^3 \times 508 \times 10^{-6} / 0.164) = 6.2$

$Nu = 6.3 \times 10^{-3} \times 8054 \times (6.2)^{0.33} = 92.6$

$h_s = 92.6 \times (0.164/13.5 \times 10^{-3}) = 1125 \text{ Wm}^{-2}\,°\text{C}^{-1}$

$$1/U_o = (1/3316)(19/16) + \frac{19 \times 10^{-3} \text{ Ln } (19/16)}{2 \times 50} + 1/1125$$

$U_o = 781 \text{ Wm}^{-2}\,°\text{C}^{-1}$ well above the trial value of 500 $\text{Wm}^{-2}\,°\text{C}^{-1}$.

Reasonable margin to allow for fouling; accept design but check pressure drops.

Tube-side

For Re = 13469, j_f = 4.5×10^{-3}, Fig 12.24

ΔP_t = $8 [8 \times 4.5 \times 10^{-3} (2.5/16 \times 10^{-3}) + 2.5)] (990.2 \times 0.505^2) /2$ = 8207 N/m^2

Well below the limit set for the cooling water.

Shell-side

From Fig 12,10 clearance for split-ring floating head exchanger = 65 mm
D_s = 743 + 65 = 808 mm

For Re = 8054, j_f = 5.0×10^{-2} , Fig 12.30

For a two pass-shell, the number of tube crosses will be double that given by the term L/l_b in equation 12.26. There will be set of cross-baffles above the shell baffle and a set below, which doubles the path length.

So, ΔP_s = $8 \times 5.0 \times 10^{-2} (808 \times 10^{-3} / 12 \times 10^{-3}) \times 2(2.5/0.372) (752 \times 0.403^2)/2$

= 19650 N/m^2 = 19.7 kN/m^2

Looks reasonable. The condensate would be pumped through the sub-cooler.

Solution 12.8

The design method will follow that used in problem 12.6, except that the condensing coefficient will be estimated for vertical tubes; section 12.10.3.

Put he condensate in the shell.

The condensing coefficient will be lower for vertical tubes, so the number of tubes will need to be increased. It would be better to increase the tube length to obtain the increased area required but the tube length is fixed.

The sub-cooler design will be the same as that determined in the solution to problem 12.7.

Solution 12.9

The design procedure will follow that illustrated in the solution to 12.7.

As the vapour is only partially condensed, from a non-condensable gas, the approximate methods given in section 12.10.8 need to be used to estimate the condensing coefficient.

Solution 12.10

As the process fluid is a pure liquid, Frank and Pricket's method can be used to give a conservative estimate of the number of tubes required. See example 12.9.

Solution 12.11

This a design problem, so there will be no unique solution. The solution outlined below is my first trial design . It illustrates the design procedure and methods to be used.

Mass flow-rate = 10000/3600 = 2.78 kg/s

Duty = 2.78 [0.99(10 – 10) + 260] = 722.8 kW

The water outlet temperature is not fixed. The most economic flow will depend on how the water is heated. The simplest method would be by the injection of live steam. The heated water would be recirculated. As a trial value, take the water outlet temperature as 40 °C.

Water flow-rate = 722.8/[4.18(50 – 40)] = 17.3 kg/s

10 °C - - - - - →- - - - - - 10 °C

50 °C - - - - - -← - - - - - 40 °C

$\Delta T_M = \Delta T_{LM} = (40 - 30)/\text{Ln}(40/30) = 34.8\ °C$ (12.4)

The coefficient for vaporisation will be high, around 5000 $Wm^{-2}\ °C^{-1}$. That for the hot water will be lower, around 2000 $Wm^{-2}\ °C^{-1}$. So, take the overall coefficient as 1500 $Wm^{-2}\ °C^{-1}$.

Area required = $(722.8 \times 10^3)/(1500 \times 34.8) = 13.8\ m^2$ (12.1)

Surface area of one U-tube = $\Pi \times 25 \times 10^{-3} \times 6 = 0.47\ m^2$

Number of U-tubes required = 12.8/0.47 = 30

Shell-side coefficient

Heat flux, q = $722.8 \times 10^3/(30 \times 0.47) = 51262\ W/m^2$

Taking k_w for stainless steel = 16 $W\ m^{-1}\ °C^{-1}$

Resistance of tube wall, R = $\dfrac{25 \times 10^{-3} \text{Ln}(25/21)}{2 \times 16} = 0.000136\ (Wm^{-2}\ °C^{-1})^{-1}$

ΔT cross tube wall = q x R = 51262 x 0.000136 = 7 °C

So mean tube wall surface temperature, T_w = 45 - 7 = 38 °C

$Ln(P_w) = 9.34 - 1978/(38 + 246)$, $P_w = 10.75$ bar

$$h_{nb} = 0.0012\left[\frac{0.13^{0.79} \times 990^{0.45} \times 1440^{0.49}}{0.013^{0.5}\,(0.3 \times 10^{-3})^{0.29}\,(260 \times 10^{3})^{0.24}\,16.4^{0.24}}\right](38 - 10)^{0.24}\,[(10.75 - 5)10^{5}]^{0.75}$$

$$= 21043 \text{ Wm}^{-2}\,°\text{C}^{-1} \qquad (12.62)$$

Tube side coefficient

Properties of water at 45 °C, from steam tables: $\rho = 990.2$ kg/m^3,
$\mu = 594 \times 10^{-6}$ N m^{-2} s, $k = 638 \times 10^{-3}$ W m^{-1} °C^{-1}, Cp = 4.18 kJ kg^{-1}°C^{-1}, Pr = 3.89

Cross-sectional area of one tube = $\Pi/4 \times (21 \times 10^{-3})^2 = 3.46 \times 10^{-4}$

$u_t = (17.3/990.2) /(30 \times 3.46 \times 10^{-4}) = 1.68$ m/s

$Re = 990.2 \times 1.68 \times 21 \times 10^{-3})/594 \times 10^{-6} = 58{,}812$

$j_h = 3.2 \times 10^{-3}$, Fig 12.23. Neglect the viscosity correction

$$Nu = 3.2 \times 10^{-3} \times 58812 \times 3.89^{0.33} = 294.6 \qquad (12.15)$$

$h_i = 294.6 \times (638 \times 10^{-3}/21 \times 10^{-3}) = 8950$ Wm^{-2} °C^{-1}

$1/U = 1/21043 + 0.000136 + 1/8950$

$U = 3387$ Wm^{-2} °C^{-1} Well above the assumed value.

Check maximum heat flux

Take the tube pitch to be 1.5 x tube o.d., on a square pitch, to allow for vapour flow.

$p_t = 25 \times 1.5 = 37.5$ mm

$N_t = 30 \times 2 = 60$ (U-tubes)

$$q_{ch} = 0.44 \times (37.5/25)(260 \times 10^{3})[0.013 \times 9.8(1440 - 16.3)16.3^{2}]^{0.25}$$

$$= 2{,}542{,}483 \text{ W/m}^2 \qquad (12.74)$$

Apply a 0.7 factor of safety, = 1,779,738 W/m^2

Actual flux = 51,262 W/m^2, well below the maximum.

Check tube-side pressure drop

For Re = 58,812, from Fig 12.24, $j_f = 3.2 \times 10^{-3}$

L in equation 12.20 = half U-tube length = 3m

$\Delta P_t = 2[8 \times 3.2 \times 10^{-3} (3/21 \times 10^{-3}) = 2.5] 990.2 \times 1.68^2/2$

$=17208$ N/m^2 = 0.17 bar, well within the limit specified

Shell design

A shell similar to that designed in example 12.11 could be used. Or, the bundle could be inserted in a simple, vertical, pressure vessel, with sufficient height to provide adequate disengagment of the liquid drops; see section 10.9.2.

Solution 12.12

The properties of the solutions to be taken as for water. As there is little difference in the mean temperatures of the two streams, use the properties at 45 °C.
From steam tables: $\rho = 990.2$ kg/m^3, $\mu = 594 \times 10^{-6}$ N m^{-2} s, $k = 638 \times 10^{-3}$ W m^{-1} °C^{-1}, Cp = 4.18 kJ kg $^{-1}$°C^{-1}, Pr = 3.89.

The temperature change of the cooling water is the same as that of the solution, so the flow-rates will be the same.

Flow-rate = 200000/3600 = 55.6 kg/s

There are 329 plates which gives 329 – 1 flow channels.

The flow arrangement is 2:2, giving 4 passes

So, the number of channels per pass = (329 – 1)/4 = 82

Cross-sectional area of a channel = $0.5 \times 3 \times 10^{-3} = 1.5 \times 10^{-3}$ m^2

The velocity through a channel = $(55.6/990.2)/(82 \times 1.5 \times 10^{-3}) = 0.46$ m/s

Equivalent diameter, $d_e = 2 \times 3 = 6$ mm

$Re = (990.2 \times 0.46 \times 6 \times 10^{-3})/594 \times 10^{-6} = 4601$

$Nu = 0.26 (4601)^{0.65} \times (3.89)^{0.4} = 107.6$ (12.77)

Neglecting the viscosity correction factor

$h_p = 107.6 \times (638 \times 10^{-3} / 6 \times 10^{-3}) = 11441$ Wm^{-2} °C^{-1}

As the flow-rates and physical properties are the same for both streams the coefficients can be taken as the same.

The plate material is not given, stainless steel would be suitable and as it has a relatively low thermal conductivity will give a conservative estimate of the overall coefficient.

Take thermal conductivity of plate = 16 W m^{-1} $°C^{-1}$

$1/U = 1/11441 + 0.75 \times 10^{-3}/16 + 1/11441$

$U = 4511$ Wm^{-2} $°C^{-1}$

70 - - - → - - - 30 °C

60 - - - ← - - - 20 °C

As the terminal temperature differences are the same, $\Delta T_{LM} = \Delta T = 10$ °C

NTU = (70 – 30)/10 = 4

F_t from Fig 12.62 = 0.87

$\Delta T_M = 10 \times 0.87 = 8.7$ °C

Duty, Q = 55.6 x 4.18(70 –30) = 9296.3 kW

Area required = $(9296.3 \times 10^3)/(4511 \times 8.7) = 236.9$ m^2

Number of thermal plates = total – 2 end plates = 329 – 2 = 327

Area available = 327(1.5 x 0.5) = 245 m^2

So exchanger should be satisfactory. but there is little margin for fouling.

Pressure drop

The pressure drop will be the same for each stream

$j_f = 0.6 \times (4601)^{-0.3} = 4.8 \times 10^{-2}$

Lp, two passes = 2 x 1.5 = 3 m

$\Delta P_p = 8 \times 4.8 \times 10^{-2} (3/6\times10^{-3})\ 990.2 \times 0.46^2/2 = 20115$ N/m^2 (12.78)

Port area = $\Pi \times (0.15^2)/4 = 17.7 \times 10^{-3}$ m^2

Velocity $u_{pt} = (55.6/990.2)/(17.7 \times 10^{-3}) = 3.17$ m/s

$\Delta P_{pt} = 1.3 \times 990.2 \times (3.17^2/2) \times 2 = 12936$ N/m^2 (12.79)

Total pressure drop for each stream = 20115 + 12936 = 33052 N/m^2

= 0.33 bar