Solution 13.1

See section 13.3.4, equations 13.7 to 13.18

Solution 13.2

Use equation 13.34.

(a) rigid constant $C = 0.43$

(b) free to rotate, $C = 0.56$

Solution 13.3

See section 13.5.1

Use equation 13.39 for the cylindrical section and equation 13.40 or 13.41 for the ends.

Solution 13.4

Specification

Shell 387 mm id, tubes 14.83 mm id, 19.05 mm od, length 6096 mm

Kerosene in the shell, operating pressure 5 bar.

Crude in the tubes, operating pressure 6.5 bar

Material of construction, semi-killed or silicon killed carbon steel.

(a) Design pressures: take as 10% greater than operating pressures; section 13.4.1.

Shell = $(5 - 1) \times 1.1 = 4.4 \text{ bar} = 4.4 \times 10^5 \text{ N/m}^2$

Tubes = $(6.5 - 1) \times 1.1 = 6.05 \text{ bar} = 6.5 \times 10^5 \text{ N/m}^2$

Design temperature: maximum operating temperature = 200 °C. Take this as the design temperature for both the shell and tubes. The tubes could reach the kerosene temperature if there was no flow of crude oil; section 13.4.2.

(b) Corrosion allowance: no information is given on the purity of the kerosene or the composition of the crude. If sulphur free the kerosene should not corrode. If wet the crude could be corrosive.

Take the kerosene allowance as 2 mm and the crude as 4 mm; section 13.4.6
(c) End covers: shell and floating head use torispherical, header- cover flat plate.; see figure, example 12.2.

(d) Stressing: take design stress as 105 N/mm$^2$ at 200 °C; , Table 13.2.

Shell: $e = \frac{4.4 \times 10^5 \times 0.387}{2 \times 105 \times 10^6 - 4.4 \times 10^5} = 0.0008 \text{ m} = 0.8 \text{ mm}$ \hfill (13.39)

add corrosion allowance = $0.8 + 2 = 2.8 \text{ mm}$

This is less than the minimum recommended thickness, section 13.4.8, so round up to 5 mm.

Header: $e = \frac{6.05 \times 10^5 \times 0.387}{2 \times 105 \times 10^6 - 6.05 \times 10^5} = 0.0011 \text{ m} = 1.1 \text{ mm}$ \hfill (13.39)

add corrosion allowance = $1.1 + 4 = 5.1 \text{ mm}$

Shell end-cover, torispherical, take $R_c = 0.3, R_k/R_c = 0.1$; section 13.5.4

$C_s = \frac{1}{4}(3 + \sqrt{10}) = 2.37$. Take joint factor as = 1.0, formed head.

$e = \frac{4.4 \times 10^5 \times 0.3 \times 2.37}{2 \times 105 \times 10^6 \times 1 + 4.4 \times 10^5(2.37 - 0.2)} = 0.00148 \text{ m} = 1.5 \text{ mm}$ \hfill (13.44)

add corrosion allowance = $1.5 + 2 = 3.5 \text{ mm}$

Floating-head cover, torispherical:

Bundle to shell clearance, Fig 12.10 $\cong 53 \text{ mm}$, take as split ring.

$D_b = 0.387 - 0.334 = 0.334 \text{ mm}$

Take $R_c$ as 0.3, $R_k/R_c = 0.1$

$C_s = 2.37$

$e = \frac{6.05 \times 10^5 \times 0.3 \times 2.37}{2 \times 105 \times 10^6 \times 1 + 6.05 \times 10^5(2.37 - 0.2)} = 0.00206 \text{ m} = 2.1 \text{ mm}$ \hfill (13.44)

add corrosion allowance = $2.1 + 4 = 6.1 \text{ mm}$

Flat plate (header cover):

Type (e) Fig 13.9, $C_p = 0.55$. $D_i = 387 \text{ mm}$, so $D_e \cong 0.4 \text{ m}$

$e = 0.55 \times 0.4 \sqrt{(6.05 \times 10^5 / 105 \times 10^6)} = 0.167 \text{ m}$ \hfill (13.42)
add corrosion allowance = 16.7 + 4 = 20.7 mm

All thicknesses would be rounded to nearest standard size.

(e) Tube rating

Tube id = 14.83 mm, od = 19.05 mm, design stress 105 \times 10^6 \text{N/m}^2, design pressure 6.05 \text{N/m}^2.

Thickness required, \( e = \frac{6.05 \times 10^5 \times 1.83 \times 10^3}{2 \times 105 \times 10^6 - 6.05 \times 10^5} = 0.0000053 \text{m} \) \hspace{1cm} (13.39)

= 0.005 \text{mm}

Actual wall thickness = (19.05 - 14.83)/2 = 2.1 mm.

So ample margin for corrosion.

(f) Tube-sheet thickness should not be less than tube diameter; section 12.5.8.

So take thickness as = 20 mm

(g) Would use weld neck flanges; Appendix F.

Shell od = 387 + (2 \times 5) = 397 mm, say, 400 mm

Design pressure = 4.05 \times 10^5 \text{N/m}^2, design temperature = 200 °C

6 bar rating would be satisfactory, table 13.5.

Floating head od ≃ 350 mm, design pressure = 6.05 \times 10^5 \text{N/m}^2, design temperature 200 °C.

Use a 10 bar rated flange, table 13.5.

(h) Supports

Use saddle supports, section 13.9.1, Fig 13.26d.

Smallest size given in Fig 13.26d is 600 mm diameter. So, scale all dimensions to 400 mm and make all plate 5 mm.

Rough check on weight

Diameter ≃ 0.4 m, length ≃ 10 m

Shell and header, volume of steel = Π \times 0.4 \times 10 \times 10^{-3} = 0.063 \text{m}^3
Volume of shell head, take as flat, \( \cong \pi/4 \times 0.4^2 \times 3.5 \times 10^{-3} \) = 0.0004 m\(^3\)

Volume of floating head, take as flat \( \cong \pi/4 \times 0.334^2 \times 6 \times 10^{-3} \) = 0.0005 m\(^3\)

Volume of flat plate end cover = \( \pi/4 \times 0.4^2 \times 21 \times 10^{-3} \) = 0.0026 m\(^3\)

Volume of tube-sheet = 0.0026 m\(^3\), ignoring the holes

Volume of tubes = 168 \[ \pi/4 \times (19.05^2 - 14.83^2) \times 10^{-6} \times 6.09 \] = 0.115 m\(^3\)

Number of baffles = 6090/77.9 - 1 = 77

Taking baffles as 3 mm thick and ignore the baffle cut,

\[ \text{volume} = 77 \left( \frac{\pi}{4} \times 0.387^2 \times 3 \times 10^{-3} \right) = 0.027 \text{ m}^3 \]

Total volume of steel

\[
\begin{array}{|l|c|}
\hline
\text{Component} & \text{Volume} \\
\hline
\text{Shell} & 0.063 \\
\text{Shell head} & 0.0004 \\
\text{Floating head} & 0.0005 \\
\text{End-cover} & 0.0026 \\
\text{Tube-sheet} & 0.0026 \\
\text{Tubes} & 0.115 \\
\text{Baffles} & 0.027 \\
\hline
\text{Total} & 0.21 \text{ m}^3 \\
\hline
\end{array}
\]

Taking density of steel as 7800 kg/m\(^3\), mass of exchanger = 0.21 x 7800 = 1638 kg

Weight = 1638 x 9.8 = 16,052 N = 16 kN

Mass of water, ignore volume of tubes, = 1000(\pi/4 \times 0.4^2 \times 10) = 1257 m\(^3\)

Weight = 1257 x 9.8 = 12319 N = 12 kN

Maximum load on supports = 16 + 12 = 28 kN

Load given in Table 13.26a for a 600 mm diameter vessel = 35 kN, so design should be satisfactory.

**Solution 13.5**

The design procedure will follow that set out in solution 13.4.

The exchanger will be hung from brackets, see section 13.9.3.
**Solution 13.6**

The procedure for solving this problem follows that used in examples 13.3 and 13.4.

1. Determine the minimum plate thickness to resist the internal pressure, equation 13.39.

2. Select and size the vessel ends, use torispherical or ellipsoidal heads; section 13.5.4

3. Increase the basic plate thickness to allow for the bending stress induced by the wind loading at the base of the vessel, and the small increase in stress due to the dead weight of the vessel.

4. Check that the maximum combined stresses at the base are within the design stress and that the critical buckling stress is not exceeded.

5. Decide which openings need compensation. The 50 mm nozzles are unlikely to need compensation but the vapour outlet and access ports probably will. Use the equal area method for determining the compensation required; section 13.6.

6. Use standard flanges; section 13.10.5 and appendix F.

7. Design the skirt support. A straight skirt should be satisfactory. Consider the wind load, the weight of the vessel, and the weight of the vessel full of water. Though the vessel is not likely to be pressure tested during a storm, a fault condition could occur during operation and the vessel fill with process fluid. The process fluid is unlikely to be more dense than water.

8. Design the base ring following the method given in section 13.9.2.

**Solution 13.7**

Only the jacketed section need be considered, the vessel operates at atmospheric pressure.

The jacketed section of the vessel will be subjected to an external pressure equal to the steam pressure (gauge).

The jacket will be under the internal pressure of the steam.

Operating pressure = \(20 - 1 = 19\) barg = \(19 \times 10^5\) N/m\(^2\) = 1.9 N/mm\(^2\)

\[
\text{o.d. of vessel} = 2 + 2 \times 25 \times 10^{-3} = 2.05\ m
\]

\[
\text{i.d. of jacket} = 2.05 + 2 \times 75 \times 10^{-3} = 2.2\ m
\]

Jacket, required thickness, \(e = \frac{1.9 \times 2.2}{2 \times 100 - 1.9} = 0.021\ m = 21\ mm\) \ (13.39)
So the specified thickness of 25 mm should be OK, with adequate margin of safety.

Vessel section:

Take Poisson’s ratio, $\nu$, for carbon steel as 0.3. 
E is given as $180,000 \text{ N/mm}^2 = 1.8 \times 10^{11} \text{ N/m}^2$

Check collapse pressure without any consideration of stiffening

$$P_c = 2.2 \times 1.8 \times 10^{11} \left(25 \times 10^{-3}/2.05\right)^3 = 718,214 \text{ N/m}^2 = 7.2 \text{ bar} \quad (13.51)$$

So vessel thickness is adequate to resist the steam pressure.

**Solution 13.8**

The pipe is a thick cylinder, see section 13.15.1 and the solution to problem 13.7.

**Solution 13.9**

Tank diameter = 6 m, height of liquid, $H_L = 16$ m, density of liquid, $\rho_L = 1520 \text{ kg/m}^3$, $g = 9.81$, design stress, $f_t = 90 \text{ N/mm}$. Take joint factor, $J$, as 0.7, a safe value.

$$e_s = \frac{1520 \times 16 \times 9.81 \times 6}{2 \times 90 \times 10^6 \times 0.7} = 0.0114 \text{ m} \quad (13.130)$$

Say 12 mm