Heat Exchanger Design

The purpose of heat exchanger after gasifier is to lower the high temperature of output gas to an acceptable range. The syngas produced is still not cleaned, removal of particulates and undesirable gasses is still to be done. The gas cleaning processes requires temperatures lower the gasifier itself due to which there is a necessity to cool the syngas produced. In the present case a typical Shell and Tube heat exchanger is being used for the cooling service.

→ According to Process Heat Transfer (Kern) the calculation for the heating or cooling of a gas differs in only minor respects from the calculation for liquid-liquid systems.

→ *Viscosity:*

For gases it ranges from 0.015 centipoise to 0.025

→ *Thermal Conductivities:*

About 1/5th of the values obtained for liquids

→ *Specific Heat:*

Slightly lower than those of organic liquids

However, the overall design procedure and equations would be fairly same.

→ There are numerous design procedures and steps available in different literatures, the subject is nearly a century old. *Process Heat Transfer* by D.Q Kern is one of the pioneer work done on heat transfer equipment. A quite detailed design theory and procedure are presented by Kern. But the problem is that the work done is now decades old and nowadays several new features, simplifications and changes have been adopted. Although the basis of calculations will be from the Kern but the design procedure adopted will be from *Coulson Richardson Volume 6*.

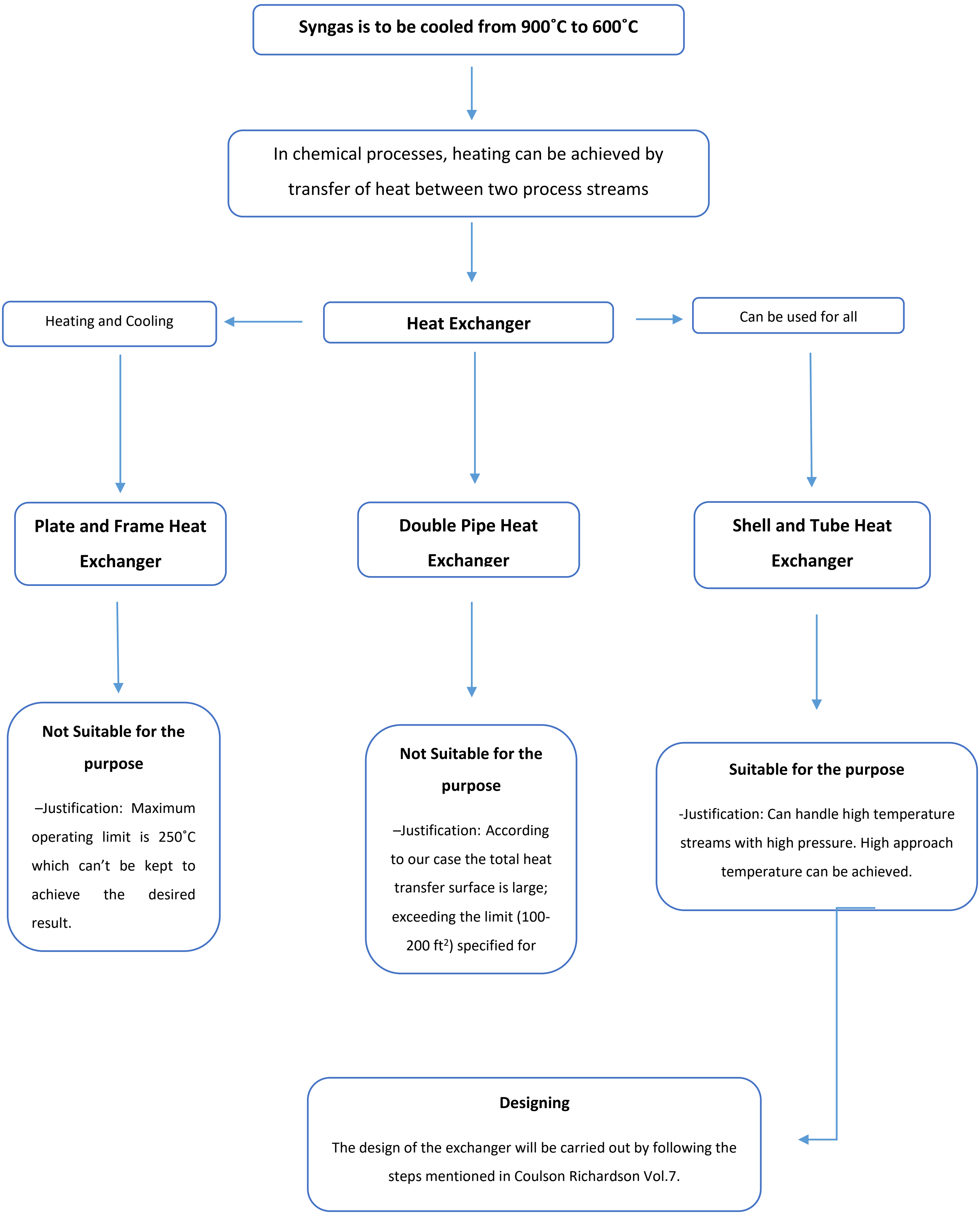
→ There are three main types of exchangers available for the required duty:

- Double Pipe Heat Exchanger
- Plate and Frame Heat Exchanger
- Shell and Tube Heat Exchanger

→ The selection will be done on following basis:

- Heat Transfer area
- Application on the basis of approach temperature
- Cost
- Ease of availability

A flow diagram representing the methodology adopted for the Exchanger selection



Shell and Tube Heat Exchangers:

Some significant advantages of Shell and tube in our particular case:

- Good mechanical layout: a good configuration for high pressure operation. As use of steam usually requires high pressure according to its temperature, which is good in case of high temperature steam.

- Well established design procedure: a good and extensive design conditions and procedure is available in literature which can be advantageous in the case where assumptions are to be taken.

- The greater temperature difference should be at least (20°C) or more than 20°C.

Design Procedure for Shell and Tube Heat Exchangers

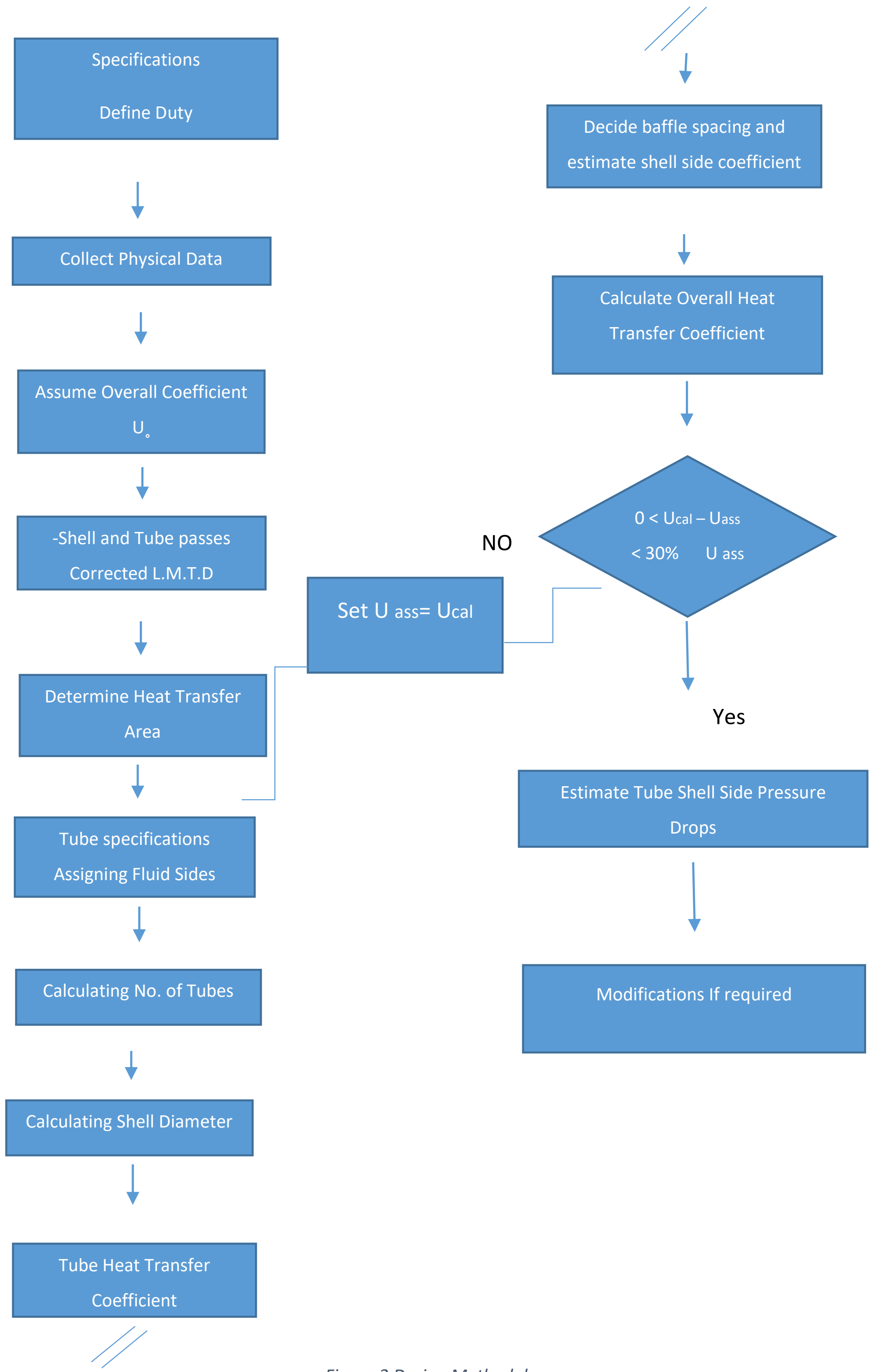


Figure 2 Design Methodology

Calculations:

Hot Fluid - Syngas

Table** Error! No text of specified style in document.**.1 LMTD

Inlet Temperature	900˚C
Outlet Temperature	600˚C
Heat Capacity	2.5 KJ/kg˚C
Mass Rate	17125.95 kg/hr
Heat Duty-Q	m*Cp*ΔT
	3567.9 KW

Cold Fluid – Water

Inlet Temperature	25˚C
Outlet Temperature	350˚C
Heat Capacity	4.18 KJ/kg˚C
Mass Rate	9454.88 kg/hr
Heat Duty-Q	m*Cp*ΔT
	3575.2 KW

Qin ˜ Qout

10.1.1. Physical Properties

Table ***Error! No text of specified style in document.****.2 Physical Properties*

Syn Gas	
Thermal Conductivity	0.250 W/m˚C
Density at Average Temperature	2.68 kg/m³
Viscosity centipoise/mNs/m² - Given range for gasses in Kern is 0.015-0.025 centipoise	Taking 0.025 mNs/m²
Water	
Thermal Conductivity	0.615 W/m˚C
Density	995 kg/m³
Viscosity from kern	0.89 mNs/m²

→ Syngas will be employed in tubes due higher temperature and pressure.

→ **Assuming Overall Coefficient:**

From table 12.1 – C.R Vol 6 the given range is:

20-300 W/m²˚C

It always better to assume a higher value for reduction in error, taking **200 W/ m²˚C**

10.1.2. L.M.T.D:

Assuming Co-Current flow arrangement:

Hot Fluid			Cold Fluid	
900 ˚C	T_1	-	t_1	25˚C = ΔTH = 875 ˚C
600 ˚C	T_2	-	t_2	350˚C = ΔTC = 250 ˚C

$$\Delta TLM = \frac{\Delta TH - \Delta TC}{\ln \frac{\Delta TH}{\Delta TC}}$$

$= 498^\circ C$

10.1.3. Temperature Correction:

For trial design purposes it is customary to consider even no. of passes, in this position the inlet and outlet nozzles are at the same end of exchanger which simplifies the pipework.

Taking one shell pass and two tube passes:

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$= 0.92$

$$S = \frac{(t_2 - t_1)}{T_1 - t_1}$$

$= 0.37$

From figure 12.19 C. R Vol 6

Ft = 0.96

$\Delta TLM = \Delta TLM * Ft$

$\Delta TLM = 478\ ^\circ C$

10.1.4. Heat Transfer Area:

$$\mathbf{A} = \frac{\mathbf{Q}}{\mathbf{Ud * \Delta TLM}}$$

$$= \frac{3567.9 * 10^3}{150*478\ ^\circ C}$$

$$= 49.76\ m^2$$

10.1.5. Layout and Tube sizing:

As the temperature difference is high in this case a provision must be made always for the ease of cleaning. Considering this, using a *Split-Ring Floating Head Exchanger*.

As the gas temperature is high, there is a bright possibility of corrosion inside the tubes and water can also corrode the outside. Operating pressure is also slightly higher. Aluminum can be used for layout due to following advantages:

→ High heat transfer efficiency

→ Low cost

→ Increase corrosion resistance when alloyed with bronze and nickel.

Taking a *¾ inch*, *19mm* tube outside diameter (A good trial estimate C.R Vol 6) and *5m* long tubes (A popular size). Using a *square pitch* for the ease of cleaning.

→ No. of tubes:

Area = $0.0019 * 5 * \pi$

$= 0.29\ m^2$

Number of tubes: $\frac{49.76}{0.29}$

= 172 tubes

$$\text{Tube per pass} = \frac{172}{2} = 86 \text{ tubes}$$

10.1.5.1. Checking for tube side Velocity:

Tube cross-sectional area = $\frac{\pi\,(\text{Tube Inner dia})^2}{4}$ $\therefore$ 15.74 mm at 16BWG -(table 10 –Kern)

$= 0.00972 \text{ m}^2$

Area per tube $= 86 \times 0.00097 = 0.08086 \text{ m}^2$

$$\text{Volumetric flowrate: } \frac{17125.95 \text{ kg}}{\text{hr}} \left| \frac{\text{hr}}{3600 \text{ sec}} \right| \frac{\text{m}^3}{2.38 \text{ kg}}$$

$= 1.99 \text{ m}^3/\text{sec}$

$$\text{Tube side velocity: } u_t = \frac{1.99}{0.08086}$$

$= 24.61 \text{ m/sec}$

→ For high tube side velocities, it will give high heat transfer co-efficient but there will be excessive chance of erosion. Plastic inserts can be used to prevent this.

→ In order to decrease the velocity no. tubes can be increased.

10.1.5.2. Bundle and Shell Diameter:

$$Db = do(\frac{Nt}{K1})^{\frac{1}{n1}}$$

From table 12.4 C.R Vol 6 at square pitch ($pt = 1.25do$)

For two tube passes:

$K1$ = 0.156 , n_1 = 2.207

$$Db = 19\ (\frac{100}{0.156})^{\frac{1}{2.207}}$$

$$Db = 355.2\ mm$$

For Split-Ring Floating Head exchanger typical Shell clearance from figure (12.10) C.R Vol 6

= 55mm

Shell inside diameter = $Ds = 355.2 + 55$

= 410.2 mm (16.1 inch)

10.1.5.3. Tube side Heat Transfer Coefficient:

$$Re = \frac{\rho ud}{\mu}$$

$$= \frac{2.348 * 24.61 * 15.74 * 10^{-3}}{0.025 * 10^{-3}}$$

= 36876.8

$$Pr = \frac{cp\mu}{k}$$

$$= \frac{2.5*10^3*0.025*10^{-3}}{0.250}$$

= 0.25

$$\frac{L}{di} = \frac{5000}{15.74} = 317.66$$

From figure 12.23 C. R Vol 6:

$$jh = 3.6 * 10^{-3}$$

$$Nu = jh\ Re\ Pr^{0.33}$$

$$= 3.6 * 10^{-3} * 36876.8 * 0.25^{0.33}$$

$$= 84$$

$$hi = Nu\ [\frac{k}{di}]$$

$$hi = 84[\frac{0.250}{15.74 * 10^{-3}}]$$

$$hi = 1334.18\ W/m^2 °C$$

10.1.5.4. Shell Side Heat Transfer Coefficient:

Taking 20mm Shell outside diameter (A good trial estimate – C.R Vol6)

Baffle spacing, for trial purpose taking 5 mm

$$Baffle\ Spacing = \frac{Ds}{5} = \frac{410}{5} = 82mm$$

$$Area\ of\ Shell = \frac{(pt - do)}{pt} baffle\ spacing * Ds$$

$$pitch = 1.25 * do = 25mm$$

$$Area = 0.006724\ m^2$$

For square pitch arrangement equivalent diameter:

$$\frac{1.27}{do}(pt^2 - 0.785\ do^2)$$

$de = 19.74\ mm$

Volumetric flowrate on shell side = $\frac{9454.88}{3600*995} = 2.6 * 10^{-3}\ \frac{m}{sec}^{3}$

Shell side velocity = $\frac{2.6*10^{-3}}{0.006724} = 0.39\ \frac{m}{sec}$

$$Re = \frac{995*0.39*19.74*10^{-3}}{0.8904*10^{-3}} = 8606.86$$

$$\text{Pr} = \frac{cp\mu}{k} = \frac{4.183 * 10^{3} * 0.89 * 10^{-3}}{0.611} = 6.08$$

For reduce pressure drop and reasonable heat transfer coefficient using 25% cut baffles:

From figure 12.29 C. R Vol 6

$$jh = 6.8 * 10^{-3}$$

Neglecting viscosity factor:

$$Nu = \frac{hs\ de}{kf} \quad hs = Nu\ \left[\frac{kf}{de}\right] = 6.8 * 10^{-3} * 8606.86\ (\ 6.08)^{0.33}\ [\ 30.95]$$

hs= 3286.26 W/m² ˚C

10.1.6. Overall Coefficient:

$$\frac{1}{Uo} = \frac{1}{ho} + \frac{1}{hod} + \frac{do \ln(\frac{do}{di})}{2Kw} + \frac{do}{di} * \frac{1}{hid} + \frac{do}{di} * \frac{1}{hi}$$

- Outside dirt factor $hod = 0.0003$ - Table 12.2 C.R Vol 6
- Inside dirt factor $hid = 0.0002$
- Thermal conductivity of Copper - Aluminum Bronze (95% Cu, 5% Al) = 83 W/m ˚C
- $do = Tube\ outside\ diameter$
- $di = Tube\ inside\ diameter$

$$\frac{1}{Uo} = 0.01255$$

$$Uo = 80\ W/m\ ^{\circ}\mathrm{C}$$

- Well below the assumed value of 150 W/m ˚C. Proceeding further

10.1.7. Pressure Drop:

10.1.7.1. Tube Side:

From Figure 12.24, Re = 36876.8

$$jf = 3.5 * 10^{-3}$$

Neglecting Viscosity Correction term,

$$\Delta Pt = Np\left[8jf\left(\frac{L}{di}\right) + 2.5\right]\frac{\rho * ut^2}{2}$$

$$\Delta Pt = 2\left[8(3.5 * 10^{-3}\left(\frac{5000}{15.74}\right) + 2.5\right]\frac{2.38 * (24.61)^2}{2}$$

$$\Delta Pt = 57.78 * 720$$

$$\Delta Pt = 41643.54\ N/m2$$

$$\Delta Pt = 41\ Kpa$$

$$\Delta Pt = 6\ psi$$

$$\Delta Pt = 0.5\ bar$$

(Under the limit of 10 Psi)

10.1.7.2. Shell Side:

Linear Velocity:

$$\frac{Gs}{\rho} = \ 1742.52\ Kg/sm2$$

From figure 12.30 at Re = 8606.86

$$jf = 4.1 * 10^{-2}$$

Neglecting Viscosity Correction;

$$\Delta P = 8jf\ \left[\frac{Ds}{de}\right]\left[\frac{L}{lB} * \frac{\rho * us^2}{2}\right]$$

$$\Delta P = 8(\ 5 * 10^{-2}\ )\ \left[\frac{410\ mm}{19.74}\right]\left[\frac{5000}{82} * \frac{995 * (0.39)^2}{2}\right]$$

$$= 0.4\ (\ 20.77\ )(\ 60.97\ )(\ 75.66\ )$$

$$\Delta P = 38332.8\ N/m2$$

$$\Delta P = 38.33\ Kpa$$

$$\Delta P = 5.55\ \text{Psi}$$

$$\Delta P = 0.38\ \ bar$$

(Under the limit of 10 Psi)

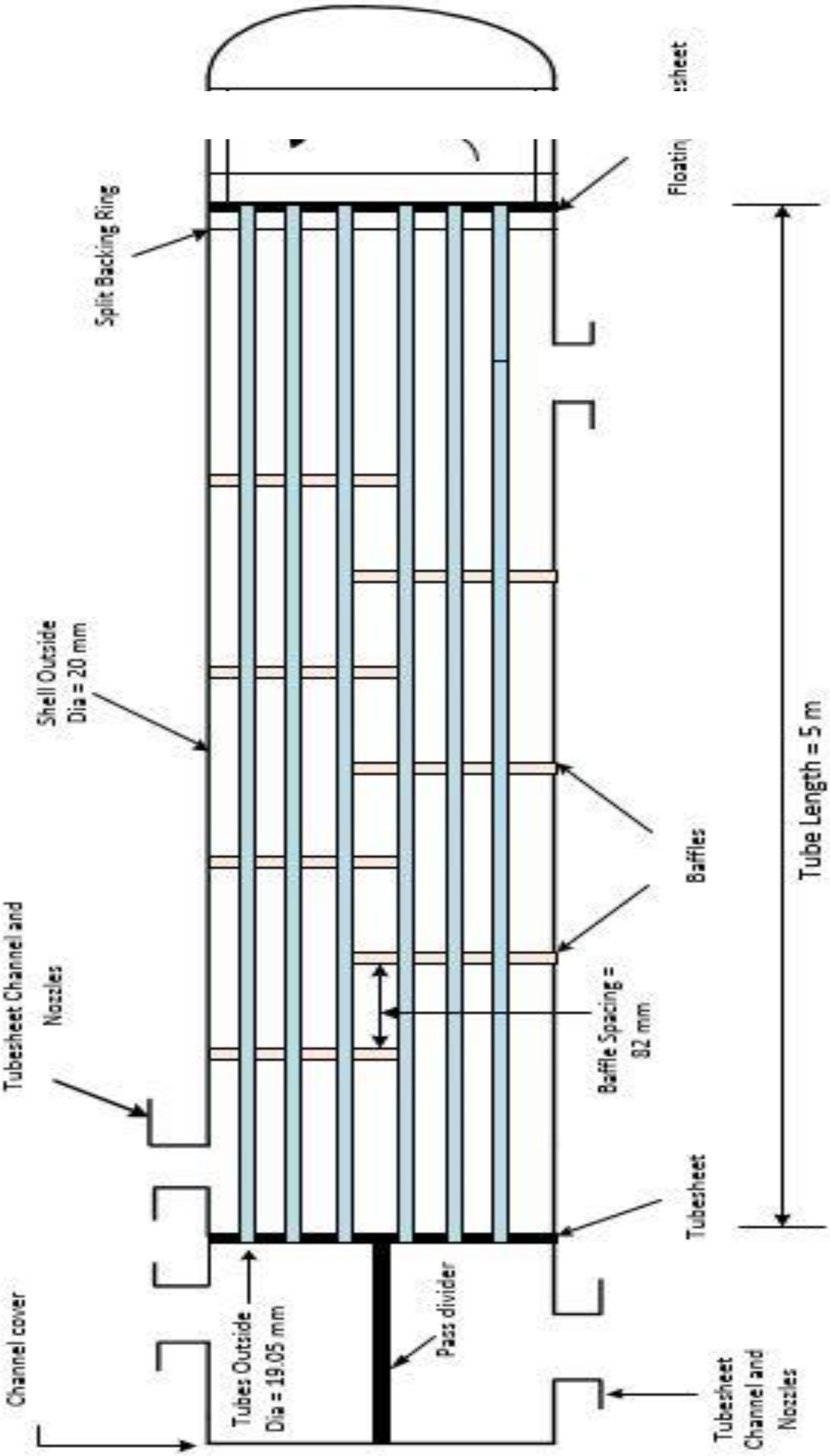


*Figure **Error! No text of specified style in document.**.3 Diagram of Shell & Tube Heat Exchanger*