Heat Exchangers

Estimation of heat transfer coefficients

In the shell and tube heat exchanger, the overall heat transfer coefficient includes the tube-side and the shell-side film coefficients

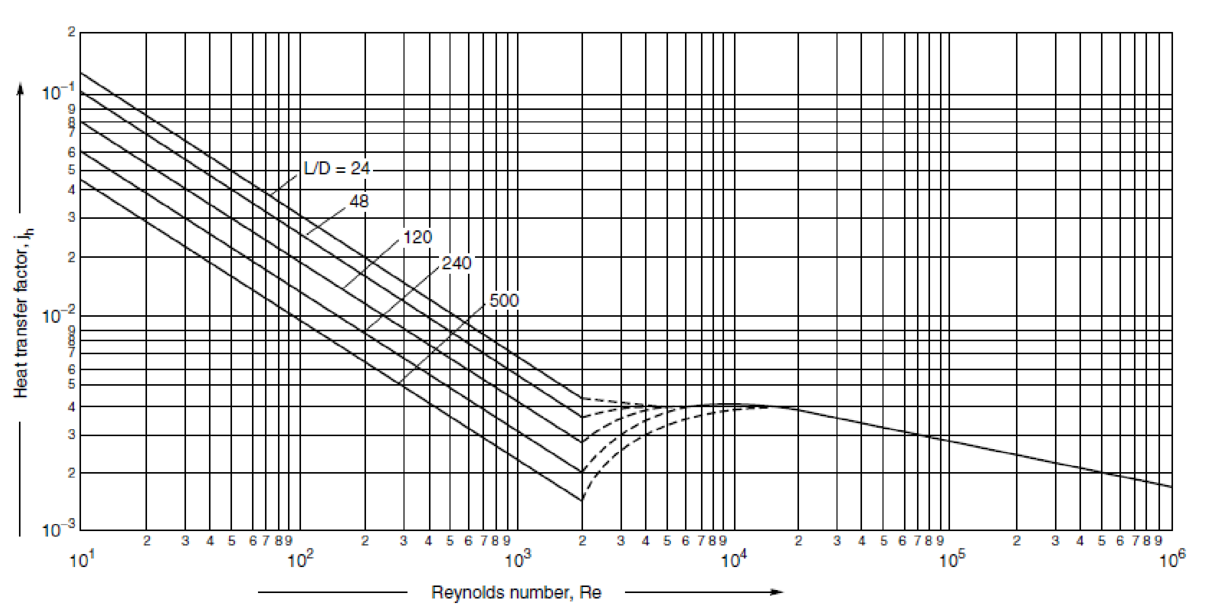
(a) Tube side heat transfer coefficient

- In case of the tubes, the fluid flows through pipes of circular geometry
- Heat transfer coefficients can be estimated using the Dittus-Boelter equation (for turbulent flow) or Sider Tate equation (for laminar flow)
- For ***turbulent flow***, $\boldsymbol{Nu = 0.023(Re)^{0.8}(Pr)^{n}}$ n = 0.4 for heating and = 0.3 for cooling

$$\boldsymbol{Nu = 0.027(Re)^{0.8}(Pr)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}}$$ when difference between T_w and T is substantial

- For ***laminar flow***, $\boldsymbol{Nu = 1.86\left[\frac{(Re)(Pr)}{L/d}\right]^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}}$
- A chart for the Colburn j_H factor may also be used to determine h_i
- This chart includes *Re* both in the laminar and the turbulent zone
- Colburn j-factor given in figure is

$$(j_H) = \left(\frac{h_i d_i}{k}\right)\left(\frac{C_p\mu}{k}\right)^{-0.33}\left(\frac{d_i v\rho}{\mu}\right)^{-1}\left(\frac{\mu}{\mu_w}\right)^{-0.44}$$



(b) Shell side heat transfer coefficient

- In case of the shell side, the fluid flows over the tubes in a tube bundle
- Due to the presence of baffles, the flow is partly along the axis and partly perpendicular to the axis
- The cross-sectional area and mass velocity of stream varies as the fluid crosses the tube bundle
- In a square pitch, the velocity of the fluid undergoes continuous fluctuation because of the constricted area between adjacent tubes compared with the flow area between successive rows
- In a triangular pitch, greater turbulence occurs because fluid flowing between adjacent tubes at high velocity impinges directly on successive rows

- **The heat transfer coefficient is affected by (a) tube diameter (b) tube pitch (c) clearance (d) baffle spacing (e) fluid flow characteristics**
- Kern suggested the following equation for calculating shell side heat transfer coefficient

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{D_e G_s}{\mu}\right)^{0.55} \left(\frac{C_p \mu}{k}\right)^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

where D_e = equivalent diameter of the shell side

k, μ, C_p = physical properties of the shell side fluid

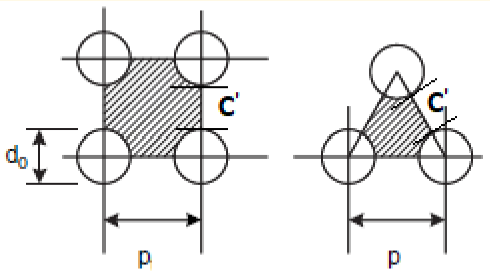
G_s = mass flux of the shell side fluid

$$G_s = \frac{W_s}{a_s}$$

W_s = mass flow rate of shell fluid, kg/s,
a_s = flow area of the shell-side fluid, m^2

$$a_s = \frac{C' D_s B}{p} = \frac{(p - d_o) D_s B}{p}$$

p = tube pitch, m D_s = shell inside diameter, m
B = baffle spacing, m d_o = outer tube diameter, m



For square pitch,

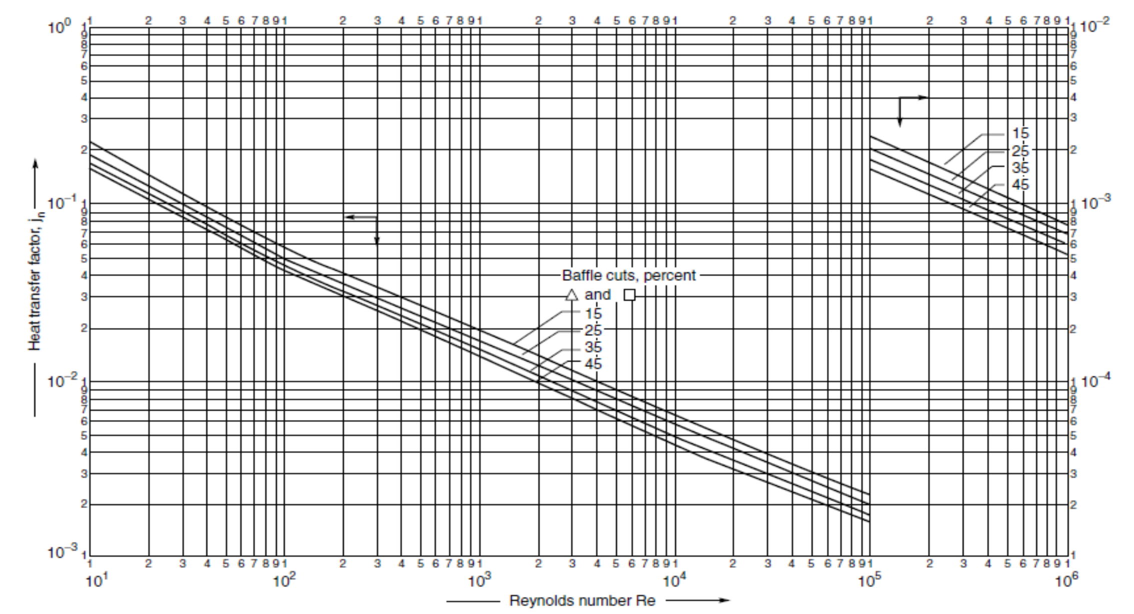
$$D_e = \frac{4\left(p^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o}$$

$$D_e = \frac{1.27}{d_o}(p^2 - 0.785\, d_o^2)$$

For triangular pitch,

$$D_e = \frac{4\left(\frac{1}{2} \times p \times 0.87p - \frac{1}{2} \times \frac{\pi d_o^2}{4}\right)}{\frac{\pi d_o}{2}}$$

$$D_e = \frac{1.10}{d_o}(p^2 - 0.917\, d_o^2)$$



- The Colburn j_H factor chart can also be conveniently used to estimate the shell-side heat transfer coefficient , h_o

- Colburn j-factor given in figure is

$$(j_H) = \left(\frac{h_o D_e}{k}\right)\left(\frac{C_p \mu}{k}\right)^{-0.33}\left(\frac{D_e G_s}{\mu}\right)^{-1}\left(\frac{\mu}{\mu_w}\right)^{-0.44}$$

Estimation of pressure drop

A criterion that needs to be satisfied in a heat exchanger design is that the *pressure drop should be within limits* for both the tube side and the shell side

(a) Tube side pressure drop

- The two major sources of pressure loss on the tube side of a shell and tube exchanger are:
  - Friction loss in the straight tubes
  - Losses due to change of direction in a multi-pass exchanger and contraction and expansion of flows
- The pressure drop in tubes is estimated as:

$$\Delta P_t = \frac{f G_t^2 L n}{2 g \rho_t d_i}\left(\frac{\mu}{\mu_w}\right)^{-m}$$

m = 0.14 for Re > 2100
= 0.25 for Re < 2100

where ΔP_t = pressure drop, kg/m^2

f = friction factor

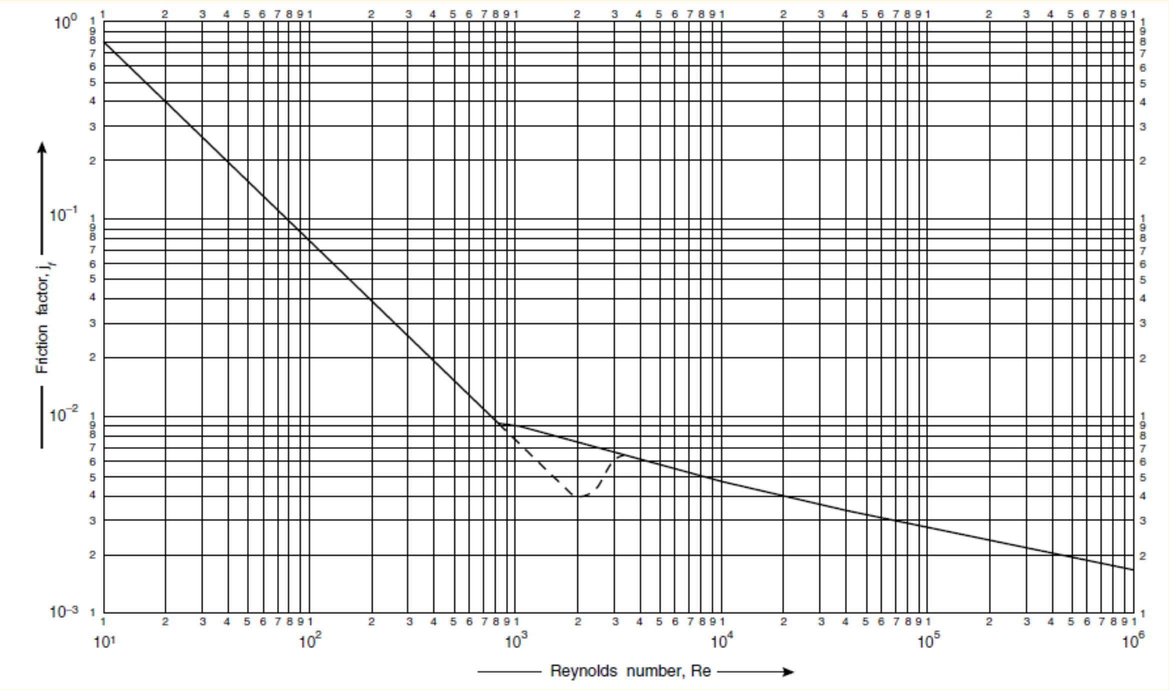
G_t = mass velocity of the tube fluid, kg/m^{2}s

L= tube length, m n = number of tube passes

g= acceleration due to gravity, 9.8 m/s^2

ρ_t = density of tube fluid, kg/m^3

d_i= inside diameter of the tube



- The pressure drop due to return losses:

$$\Delta P_r = 4n\left(\frac{v^2}{2g}\right)\rho_t$$

where ΔP_r = pressure drop, kg/m²
n = number of tube passes
v = linear velocity of the tube fluid, m/s

- Total tube side pressure drop :

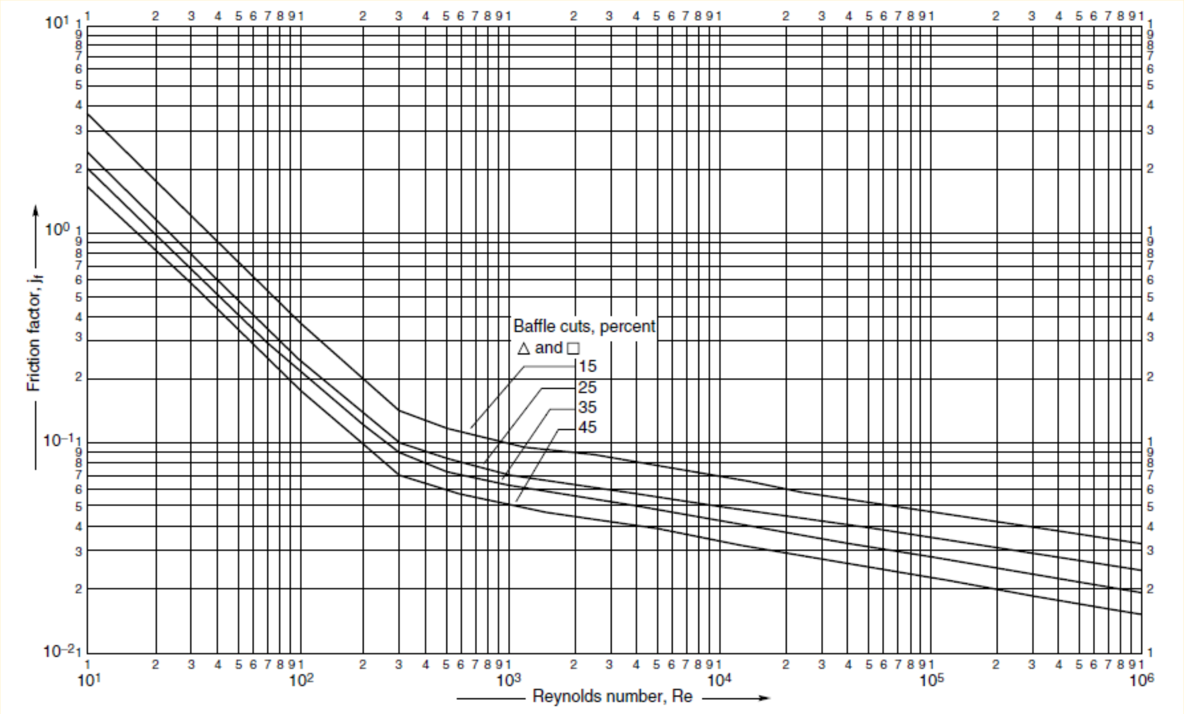
$$\Delta P_T = \Delta P_t + \Delta P_r$$

(b) Shell side pressure drop

- For an *unbaffled shell*, the pressure drop in the shell is:

$$\Delta P_s = \frac{f_s G_s^2 L N}{2g\rho_s D_H}\left(\frac{\mu}{\mu_w}\right)^{-1}$$

where ΔP_s = pressure drop, kg/m²
f_s= shell side friction factor
G_s = mass velocity of the shell side, kg/m²s
L= shell length, m N= number of shell passes
g= acceleration due to gravity, 9.8 m/s²
ρ_s = density of tube fluid, kg/m³
D_H= hydraulic diameter of the shell



$$D_H = 4 \times \frac{\pi/_4\,(D_s^2 - d_o^2 N_t)}{\pi(D_s + d_o N_t)}$$

D_s= shell inner diameter d_o= tube outer diameter N_t= number of tubes

(b) Shell side pressure drop

- For a *shell with segmental baffles*, the pressure drop in the shell is:

$$\Delta P_s = \frac{f_s G_s^2 D_s (N_b + 1)}{2 g \rho_s D_H} \left(\frac{\mu}{\mu_w}\right)^{-1}$$

where ΔP_s = pressure drop, kg/m²

f_s = shell side friction factor

G_s = mass velocity of the shell side, kg/m²s

D_S = inner diameter of the shell

g = acceleration due to gravity, 9.8 m/s²

N_b = number of baffles

ρ_s = density of tube fluid, kg/m³

D_H = hydraulic diameter of the shell

$$D_H = \frac{4\left(p^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o}$$ for square pitch

$$D_H = \frac{4\left(\frac{1}{2} \times p \times 0.87p - \frac{1}{2} \times \frac{\pi d_o^2}{4}\right)}{\frac{\pi d_o}{2}}$$ for triangular pitch

Design procedure for a double pipe heat exchanger

The steps in the design of a double pipe exchanger can be seen by means of an example:

Calculate the total length of a double pipe heat exchanger required to cool 5500 kg/h of ethylene glycol from 85°C to 68°C, using toluene as a cooling media which flows in the counter current fashion. Toluene enters at 30°C and leaves at 62°C. Ethylene glycol flows through the inner tube.

Data: Outer diameter of outer pipe (D_o) = 70 mm Outer diameter of inner pipe (d_o) = 43 mm

Wall thickness of both pipes = 3 mm

Mean properties of the two fluids:

Property	Ethylene glycol	Toluene
Density (kg/m^3)	1080	840
Specific heat (kJ/kgK)	2.68	1.80
Thermal conductivity (W/mK)	0.248	0.146
Viscosity (Pa.s)	3.4 x 10^{-3}	4.4 x 10^{-4}

Thermal conductivity of the metal pipe is 46.52 W/mK

1. Calculate heat duty

$$Q = (mC_p\Delta T)_{ethylene\ glycol} = (mC_p\Delta T)_{toluene}$$

$$Q = (mC_p\Delta T)_{ethylene\ glycol} = 5500 \times \frac{1}{3600} \times 2.68 \times 1000 \times (85 - 68) = 69605.56\, W$$

Flow rate of toluene, $m_{toluene} = \frac{69605.56}{1.80 \times 1000 \times (62-30)} = 1.2084 \frac{kg}{s} = 4350.35\ kg/h$

2. Inner tube calculations – ethylene glycol

$$\text{Flow area } = \frac{\pi d_i^2}{4} = \frac{\pi\times\left(37\times10^{-3}\right)^2}{4} = 1.0752\times10^{-3} m^2$$

(Inner diameter of inner pipe = 43 – 6 = 37 mm)

$$\text{Flow rate } = \frac{5500}{3600} = 1.5278\frac{kg}{s} = 1.4146\times10^{-3}\frac{m^3}{s}$$

$$\text{Velocity} = \frac{1.4146\times10^{-3}}{1.0752\times10^{-3}} = 1.3157\frac{m}{s}$$

$$\text{Reynolds No} = Re = \frac{d_i v\rho}{\mu} = \frac{37\times10^{-3}\times1.3157\times1080}{3.4\times10^{-3}} = 15463.34\ (turbulent)$$

$$\text{Prandtl No} = Pr = \frac{\mu C_p}{k} = \frac{3.4\times10^{-3}\times2.68\times1000}{0.248} = 36.74$$

Dittus Boelter equation for cooling $Nu = \frac{h_i d_i}{k} = 0.023(Re)^{0.8}(Pr)^{0.3}$

$$h_i = \frac{0.248}{37\times10^{-3}}\times0.023(15463.34)^{0.8}(36.74)^{0.3} = 1020.85\ W/m^2K$$

3. Outer tube calculations – toluene

Flow area $= \frac{\pi D_i^2}{4} - \frac{\pi d_o^2}{4} = \frac{\pi \times \left(64 \times 10^{-3}\right)^2}{4} - \frac{\pi \times \left(43 \times 10^{-3}\right)^2}{4} = 1.765 \times 10^{-3} m^2$

Equivalent diameter, $D_e = 4 \times r_H = 4 \times \frac{cross\text{–}sectional\ area\ of\ flow}{wetted\ perimeter} = 4 \times \frac{1.765 \times 10^{-3}}{\pi \times (43 \times 10^{-3})} = 0.0523\ m$

Flow rate $= 1.2084 \frac{kg}{s}$

Velocity $= 1.2084 \times \frac{1}{840} \times \frac{1}{1.765 \times 10^{-3}} = 0.8151 \frac{m}{s}$

Reynolds No $= Re = \frac{D_e v \rho}{\mu} = \frac{0.0523 \times 0.8151 \times 840}{4.4 \times 10^{-4}} = 81384.03\ (turbulent)$

Prandtl No $= Pr = \frac{\mu C_p}{k} = \frac{4.4 \times 10^{-4} \times 1.80 \times 1000}{0.146} = 5.425$

Dittus Boelter equation for heating $\quad Nu = \frac{h_o D_e}{k} = 0.023(Re)^{0.8}(Pr)^{0.4}$

$$h_i = \frac{0.146}{0.0523} \times 0.023(81384.03)^{0.8}(5.425)^{0.4} = 1070.95\ W/m^2K$$

4. Overall heat transfer coefficient

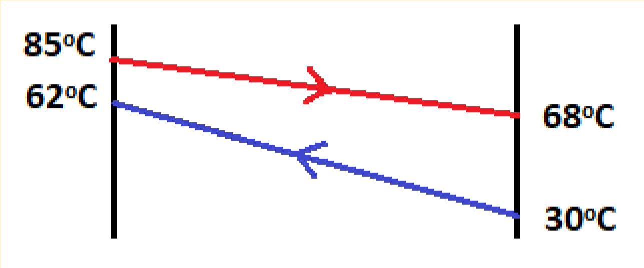
$$\frac{1}{U_o} = \frac{r_o}{r_i h_i} + \frac{r_o ln\left({}^{r_o}/_{r_i}\right)}{k_w} + \frac{1}{h_o}$$

$$\frac{1}{U_o} = \frac{43 \times 10^{-3}/2}{37 \times 10^{-3}/2 \times 1020.85} + \frac{\frac{43 \times 10^{-3}}{2} ln\left({}^{43 \times 10^{-3}}/_{37 \times 10^{-3}}\right)}{46.52} + \frac{1}{1070.95}$$

$$\boldsymbol{U_o = 466.9 \ W/m^2 °C}$$

5. Calculation of LMTD

$$\Delta T_m = \frac{\Delta T_2 - \Delta T_1}{ln\frac{\Delta T_2}{\Delta T_1}} = \frac{(85-62)-(68-30)}{ln\frac{(85-62)}{(68-30)}} = \frac{23-38}{ln\frac{23}{38}} = 29.87°C$$



6. Area and length calculation

$$Q = U_o A_o (LMTD)$$

$$69605.56 = 466.9 \times \pi \times 43 \times 10^{-3} \times L \times 29.87$$

$$L = 36.95 \ m$$

(If dirt factor is given, it needs to be included in the calculation of overall heat transfer coefficient)