Process (Thermal) Design Procedure

Process (thermal) design procedure

Shell and tube heat exchanger is designed by trial and error calculations. The main steps of design following the **Kern method** are summarized as follows:

Step #1. Obtain the required thermophysical properties of hot and cold fluids at the **caloric temperature or arithmetic mean temperature**. Calculate these properties at the caloric temperature if the variation of viscosity with temperature is large. The detailed calculation procedure of caloric temperature available is in reference

Step #2. Perform energy balance and find out the heat duty (Q) of the exchanger.

Step #3. Assume a reasonable value of overall heat transfer coefficient ($U_{o,assm}$). The value of $U_{o,assm}$ with respect to the process hot and cold fluids can be taken from the **Table 8.2.)**

Step #4. Decide tentative number of shell and tube passes (n_p). Determine the LMTD and the correction factor F_T

F_T normally should be greater than 0.75 for the steady operation of the exchangers. Otherwise it is required to increase the number of passes to obtain higher F_T values.

Step #5. Calculate heat transfer area (A) required: $A = \frac{Q}{U_{o,assm} \cdot LMTD.F_T}$ (1.1)

Step #6. Select tube material, decide the tube diameter (ID= d_i, OD $= d_o$), its wall thickness (in terms of BWG or SWG) and tube length (L). Calculate the number of tubes (n_t) required to provide the heat transfer area (A): $n_t = \frac{A}{\pi d_o L}$ (1.2)

Calculate tube side fluid velocity, $u = \frac{4\dot{m}(n_p / n_t)}{\pi \rho d_i^2}$ (1.3)

If $u < 1$ m/s, fix n_p so that, $\mathrm{Re} = \frac{4\dot{m}(n_p / n_t)}{\pi d_i \mu} \geq 10^4$ (1.4)

Where, $\dot{m}$, ρ and μ are mass flow rate, density and viscosity of tube side fluid. However, this is subject to allowable pressure drop in the tube side of the heat exchanger.

Step #7. Decide type of shell and tube exchanger (fixed tubesheet, U-tube etc.). Select the tube pitch (P_T), determine inside shell diameter (D_s) that can accommodate the calculated number of tubes (n_t). Use the standard tube counts table for this purpose. Tube counts are available in standard text books

Table 8.3

Step #9. Assign fluid to shell side or tube side (a general guideline for placing the fluids is summarized in **Table 1.4**). Select the type of baffle (segmental, doughnut etc.), its size (i.e. percentage cut, 25% baffles are widely used), spacing (B) and number. The baffle spacing is usually chosen to be within 0.2 D_s to D_s .

Step #10. Determine the tube side film heat transfer coefficient (h_i) using the suitable form of Sieder-Tate equation in laminar and turbulent flow regimes.

Estimate the shell-side film heat transfer coefficient (h_o) from:

$$j_H = \frac{h_o D_e}{k}\left(\frac{c\mu}{k}\right)^{-\frac{1}{3}}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{1.5}$$

You may consider, $\frac{\mu}{\mu_w} = 1.0$

Select the outside tube (shell side) dirt factor (R_{do}) and inside tube (tube side) dirt factor (R_{di}) **Table 12**

Calculate overall heat transfer coefficient ($U_{o,cal}$) based on the outside tube area (you may neglect the tube-wall resistance) including dirt factors:

$$U_{o,cal} = \left[\frac{1}{h_o} + R_{do} + \frac{A_0}{A_i}\left(\frac{d_0 - d_i}{2k_w}\right) + \frac{A_0}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_0}{A_i}R_{di}\right]^{-1} \tag{1.6}$$

Step #11. If $0 < \frac{U_{o,cal} - U_{o,assm}}{U_{o,assm}} < 30\%$, go the next **step # 12**. Otherwise go to **step #5**, calculate heat transfer area (A) required using $U_{o,cal}$ and repeat the calculations starting from **step #5**.

If the calculated shell side heat transfer coefficient (h_o) is too low, assume closer baffle spacing (B) close to 0.2 D_s and recalculate shell side heat transfer coefficient. However, this is subject to allowable pressure drop across the heat exchanger.

Step #12. Calculate % overdesign. Overdesign represents extra surface area provided beyond that required to compensate for fouling. Typical value of 10% or less is acceptable.

$$\% \text{ Overdesign} = \frac{A - A_{reqd}}{A_{reqd}} \times 100 \tag{1.7}$$

A = design area of heat transfer in the exchanger; A_{reqd} = required heat transfer area.

Step #13. Calculate the tube-side pressure drop (ΔP_T): (i) pressure drop in the straight section of the tube (frictional loss) (ΔP_t) and (ii) return loss (ΔP_{rt}) due to change of direction of fluid in a 'multi-pass exchanger'.

Total tube side pressure drop: $\Delta P_T = \Delta P_t + \Delta P_{rt}$ (1.8)

Step #14. Calculate shell side pressure drop (ΔP_S): (i) pressure drop for flow across the tube bundle (frictional loss) (ΔP_s) and (ii) return loss (ΔP_{rs}) due to change of direction of fluid.

Total shell side pressure drop: $\Delta P_S = \Delta P_s + \Delta P_{rs}$ (1.9)

If the tube-side pressure drop exceeds the allowable pressure drop for the process system, decrease the number of tube passes or increase number of tubes per pass. Go back to **step #6** and repeat the calculations steps.

If the shell-side pressure drop exceeds the allowable pressure drop, go back to **step #7** and repeat the calculations steps.

Step #15. Upon fulfillment of pressure drop criteria, go mechanical design. Refer **module # 2** for the details of mechanical design.

Design problem

The above design procedure is elaborated through the calculation of the following example

Design Problem

Problem Statement:

150000 lb per hour of kerosene will be heated from 75 to 120°F by cooling a gasoline stream from 160 to 120°F. Inlet pressure will be 50 psia for each stream and the maximum pressure drop of 7 psi for gasoline and 10 psi for kerosene are permissible. Published fouling factors for oil refinery streams should be used for this application. Design a shell and tube heat exchanger for this service.

PART 1: THERMAL DESIGN:

(PART 2: Mechanical design provided in module #2)

Given data:

Hot fluid inlet temperature (T_1)= 160°F

Hot fluid outlet temperature (T_2) = 120°F

Cold fluid inlet temperature (t_1) = 75°F

Cold fluid outlet temperature (t_2) = 120°F

Fouling factor of hot fluid (R_{dg}) = 0.0005 (for gasoline)

Fouling factor of cold fluid (R_{dk}) = 0.001 (for kerosene)

P_{inlet} (for hot fluid) = 50 psia

P_{inlet} (for cold fluid) = 50 psia

Δp_{max} (for hot fluid) = 7 psi

Δp_{max} (for cold fluid) = 10 psia

Mass flow rate of cold fluid ($\dot{m}_k$) = 150000 $lb.h^{-1}$

(Subscripts '*k*' for kerosene and '*g*' for gasoline)

I. Calculation of caloric temperature

the calculation of caloric temperature

$$r = \frac{\Delta t_c}{\Delta t_h} = \frac{T_2 - t_1}{T_1 - t_2} = \frac{120 - 70}{160 - 120} = 1.25$$

°API of hot fluid=76°; Therefore $K_c = 1$; $F_c = 0.455$

Caloric temperature of the hot fluid, $T_{hc} = T_2 + F_C(T_1 - T_2)$

=120+0.455×(160-120)

=138.2°F

Caloric temperature of the cold fluid, $T_{cc} = t_1 + F_C(t_2 - t_1)$

=75+0.455×(120-75)

=95.475°F

II. Fluid properties at caloric temperature

Viscosity:

76°API gasoline, μ_g=0.2cp (0.484 lb.ft^{-1}.h^{-1})

46°API kerosene, μ_k =1.6 cp (3.872 lb.ft^{-1}.h^{-1})

Density:

ρ_g=685 kg.m^{-3} (42.7 lb.ft^{-3})

ρ_k=800 kg.m^{-3} (49.8 lb.ft^{-3})

Thermal conductivity:

k_g=0.075 Btu h^{-1}ft^{-1} °F^{-1}

k_k=0.083 Btu h^{-1}ft^{-1} °F^{-1}

Specific heat capacity:

C_g = 0.57 Btu lb^{-1}ft^{-1}

C_k = 0.48 Btu lb^{-1}ft^{-1}

Specific gravity:

S_g = 0.685

S_k = 0.80

III. Energy balance

Assume no heat loss to the surrounding.

$$Q_g = Q_k = \dot{m}_k\, C_k(t_2 - t_1) = \dot{m}_g\, C_g(T_1 - T_2) = 3240000 \text{ Btu/h}$$

$$\Rightarrow 150000\times0.48\times(120-75) = \dot{m}_g \times0.57\times(160-120)$$

$$\Rightarrow \dot{m}_g = 142105 \text{ lb h}^{-1}$$

IV. Calculation of heat transfer area and tube numbers

Iteration #1:

The first iteration is started assuming 1 shell pass and 2 tube pass shell and tube exchanger with following dimensions and considerations.

- Fixed tube plate
- 1´´ OD tubes (d_o) (14 BWG) on 1¼´´ square pitch (P_T)
- Outer diameter of tube= 1´´
- Tube length (L_t) =16´
- Tube ID (d_i) = 0.834´´
- Fluid arrangement: Kerosene is placed in tube side because it has the higher fouling tendency

The log mean temperature correction factor (F_T) for 1-2 shell and tube exchanger:

$$F_T = \frac{\dfrac{\sqrt{R^2+1}\,\ln(1-S)}{1-RS}}{(R-1)\ln\left(\dfrac{2-S\left(R+1-\sqrt{(R^2)+1}\right)}{2-S\left(R+1+\sqrt{(R^2)+1}\right)}\right)}$$

$$= \frac{\sqrt{0.29^2+1}\,\ln\left[\dfrac{(1-0.529)}{(1-0.89*0.529)}\right]}{(0.89-1)\ln\left[\dfrac{2-0.529\left(0.89+1-\sqrt{(0.89)+1}\right)}{2-0.529\left(0.89+1+\sqrt{(0.89^2)+1}\right)}\right]}$$

$$= 0.802$$

where, $R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{160-120}{120-75} = 0.889$; $S = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{120-75}{160-75} = 0.529$

$$LMTD = \frac{(T_2 - T_1) - (t_2 - t_1)}{\ln\left(\dfrac{T_2 - T_1}{t_2 - t_1}\right)}$$

$$= \frac{(160-120)-(120-75)}{\ln\left(\dfrac{160-120}{120-75}\right)}$$

= 42.75 °F

Determining the heat transfer area (‘A’):

The value of overall heat transfer coefficient ($U_{o,assm}$) of 45 Btu $h^{-1}ft^{-2}\ °F^{-1}$ is assumed to initiate the design calculation for the kerosene and gasoline heat exchanger. The approximate range of overall heat transfer coefficient depending on the hot and cold fluid can be found out from text books

$$A = \frac{Q}{U_{assm} LMTD \times F_T} \tag{1.1}$$

$$= \frac{\dot{m}_g\, C_g\,(T_1 - T_2)}{U_{assm} \times LMTD \times F_T}$$

$$= \frac{142105 \times 0.57 \times (160 - 120)}{45 \times 42.75 \times 0.802}$$

= 2100 ft^2

Calculating no. of tubes (n_t):

$$n_t = \frac{A}{\pi d_o L_t} \tag{1.2}$$

$$n_t = \frac{2100}{\pi \times \left(\frac{1}{12}\right) \times 16} = 502$$

n_t = 518 is taken corresponding to the closest standard shell ID of 35´´ for fixed tube sheet, 1-shell and 2-tube pass exchanger with 1´´ tube OD on 1¼´´ square pitch. You may refer to standard heat transfer books for the selection of suitable shell ID.

Check for fluid velocity:

$$\text{Re} = \frac{4\,\dot{m}_k\,(n_p / n_t)}{\pi d_i \mu} \tag{1.4}$$

$$\text{Re} = \frac{4 \times (150000) \times 2/518}{\pi \times 0.834/12 \times 3.872}$$

= 2740.2<10^4

As Re<<10^4, the design parameters and considerations needs to be revised to meet the Reynolds number criteria subject to allowable pressure drop in the tube side of the heat exchanger.

Iteration #2:

Assumptions:

- Fixed tube plate type
- 1′′ OD tubes (14 BWG) on 1¼′′ square pitch (P_T)
- Tube length (L_t) = 24′ (the tube length is increased from 16′)
- 1 shell pass-6 tube pass (tube passes is increased to 6 from 2)
- Tube ID=0.834′′
- Flow area per tube=0.546 inch^2

No. of tubes:

$$n_t = \frac{A}{\pi d_o L_t} \tag{1.2}$$

$$n_t = \frac{2100}{\pi \times \left(\frac{1}{12}\right) \times 24} = 335$$

n_t = 368 is taken corresponding to the closest standard shell ID of 31′′ for fixed tube sheet, 1-shell and 6-tube pass exchanger with 1′′ tube OD on 1¼′′ square pitch. The tube-counts are available in heat transfer text book

Fluid velocity:

$$\mathrm{Re} = \frac{4\dot{m}_k (n_p / n_t)}{\pi d_i \mu} \tag{1.4}$$

$$\mathrm{Re} = \frac{4 \times (150000) \times {}^{6}\!/_{368}}{\pi \times {}^{0.834}\!/_{12} \times 3.872}$$

= 11571.4>10^4 corresponding to n_p=6.

$$u = \frac{\mathrm{Re}\,\mu_k}{d_i \rho_k} \tag{1.3}$$

$$= \frac{11571.4 \times 3.872}{{}^{0.834}\!/_{12} \times 49.8}$$

= 12945.15 ft/h (3.59 ft/s)

= 1.04 m/s (so the design velocity is within the acceptable range).

V. Determination of heat transfer co-efficient

Tube side heat transfer co-efficient (h_i):

$$j_H = \frac{h_i d_i}{k}\left(\frac{\mu_k C_k}{k_k}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{1.5}$$

j_H=42 for the tube side fluid at Re=11571.4

(Let's consider $\phi_t = \frac{\mu}{\mu_w} = 1$, μ = viscosity of the tube side fluid; μ_w = viscosity of tube side fluid at wall temperature)

$$42 = \frac{h_i\left(0.834/12\right)}{0.083}\left(\frac{0.48\times 3.872}{0.083}\right)^{-1/3}$$

h_i= 141.3 Btu h^{-1}ft^{-1} ^{o}F^{-1}

Shell side heat transfer co-efficient (h_o):

Assumptions:

- 25% cut segmental baffles
- Baffles spacing, B= 0.5D_S=15.5′′ (half of the shell ID is selected)

Equivalent diameter for the shell side: $D_e = \dfrac{4\left(P_T{}^2 - \frac{\pi}{4}d_o{}^2\right)}{\pi d_o}$ for square pitch

=0.082 ft

$$\left[\text{For triangular pitch, } D_e = \frac{4\left(\frac{1}{2}P_T \times 0.86P_T - \frac{1}{2}\frac{\pi}{4}d_o{}^2\right)}{\frac{1}{2}\pi d_o}\right]$$

Shell side cross flow area, $a_s = \dfrac{CBD_S}{P_T}$ (please refer to **Figure 1.6**).

C= Tube clearance

=P_T - d_o

=1¼ -1=0.25″

$$a_s = \frac{\left(\frac{0.25}{12}\right)\left(\frac{15.5}{12}\right)\left(\frac{31}{12}\right)}{\left(\frac{1.25}{12}\right)} = 0.675\ \text{ft}^2$$

$$\text{Mass velocity}, G_S = \frac{\dot{m}_g}{a_s} = \frac{142105}{0.675}$$

$=210526$ lb. $h^{-1}.ft^{-2}$

$$\text{Re} = \frac{D_e G_S}{\mu_g}$$

$$= \frac{0.082 \times (210526)}{0.484}$$

$=35668$

$$\text{Now for the shell side, } j_H = \frac{h_o D_e}{k_g}\left(\frac{\mu_g C_g}{k_g}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \qquad (1.5)$$

j_H=110 for the shell side fluid at Re=35668 with 25% cut segmental baffles

$$110 = \frac{h_0(0.082)}{(0.075)}\left(\frac{0.57 \times 0.484}{0.075}\right)^{-1/3}$$

($\phi_S = \frac{\mu}{\mu_W} = 1$ is considered for the shell side fluid)

h_o=155.3 Btu h^{-1} ft^{-2} $°F^{-1}$

Overall heat transfer co-efficient ($U_{o,cal}$):

Fouling factor, R_{dk}=0.001 h ft^2 °F Btu^{-1} for kerosene and R_{dg}= 0.0005 h ft^2 °F Btu^{-1} for gasoline is taken for this service.

$$U_{o,cal} = \left[\frac{1}{h_o} + R_{dg} + \frac{A_0}{A_i}\left(\frac{d_0 - d_i}{2k_w}\right) + \frac{A_0}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_0}{A_i}R_{dk} \right]^{-1} \qquad (1.6)$$

Let select, Admirality brass as tube material with thermal conductivity, k_w=70 Btu h^{-1} ft^{-1} °F^{-1}.

$$U_{o,cal} = \left[\frac{1}{155.3} + 0.0005 + \frac{\pi(1)^2}{\pi(0.834)^2}\left(\frac{\frac{1}{12} - \frac{0.834}{12}}{2\times 70}\right) + \frac{\pi(1)^2}{\pi(0.834)^2}\left(\frac{1}{141.3}\right) + \frac{\pi(1)^2}{\pi(0.834)^2}\times 0.001 \right]^{-1}$$

$U_{o,cal} = 53.5$ Btu h^{-1} ft^{-2} °F^{-1}

Now, $\dfrac{U_{o,cal} - U_{o,assm}}{U_{o,assm}} = \dfrac{53.5-45}{45}\times 100$

$= 18.9\% < 30\%$

Therefore, the calculated overall heat transfer co-efficient is well within the design criteria.

VI. Pressure drop calculation

VI.1. Tube side pressure drop:

Friction factor $f = 0.00028\times 144 = 0.04032$ ft^2/ft^2 for Re=11571.4

a_t= (no. of tubes)×(flow area per tube)/(no. of passes)

$= \dfrac{368\times 0.546}{6\times 144}$ ft^2

=0.232 ft^2

Tube side mass velocity: $G_t = \dfrac{\dot{m}_k}{a_t} = \dfrac{150000}{0.232}$

=646552 lb. h^{-1}.ft^{-2}

Frictional pressure drop: $\Delta P_t = \dfrac{fG_t^{\,2}L_t n_p}{7.5\times10^{12}\times d_i S_k \phi_t}$

$$=\frac{0.04032\times646552^2\times24\times6}{7.5\times10^{12}\times\dfrac{0.834}{12}\times0.8\times1}$$

=5.81 psi

Return loss ΔP_{rt} **:** (due to change in flow direction of the tube side fluid)

$$\Delta p_{tr}=1.334\times10^{-13}\left(2n_p-1.5\right)\frac{G_t^{\,2}}{S_k}$$

$$=1.334\times10^{-13}\left(2\times6-1.5\right)\frac{\left(646552\right)^2}{\left(0.8\right)}$$

=0.73 psi

Total tube side drop neglecting nozzle loss:

$$\Delta P_T=\Delta P_t+\Delta P_{tr} \tag{1.8}$$

=5.81+0.73

=6.54 psi<10 psi

Therefore the tube side pressure drop is within the maximum allowable pressure drop of 10 psi.

VI.2. Shell side pressure drop calculation

Tube clearance, C=0.25″

Spacing, B=15.5″

$a_s=0.675\,\text{ft}^2$

Mass velocity, $G_S=210526$ lb. h^{-1}.ft^{-2}

Re=35668

No of baffles, $n_b=\dfrac{\text{tube length}}{\text{baffle spacing}}=\dfrac{24}{15.5/12}=18.6\approx19$

Friction factor, $f=0.0017\times144=0.2448$ ft^2/ft^2 with 25% cut segmental baffles

Shell side frictional pressure drop ΔP_s :

$$\Delta P_s = \frac{f G_s^{\,2} D_S \left(n_b + 1\right)}{7.5\times 10^{12} \times D_e S_k \phi_k}$$

$$= \frac{0.2376\times 210526^2 \times \left(19+1\right)\times \dfrac{31}{12}}{7.5\times 10^{12} \times 0.082\times 0.685\times 1}$$

=1.4 psi <7 psi

$\Delta P_{rs} = 0$ (in case of single shell pass flow)

Total shell side drop neglecting nozzle loss:

$\Delta P_S = \Delta P_s + \Delta P_{sr}$ =1.4 psi (1.9)

Therefore the shell side pressure drop is within the maximum allowable pressure drop of 7 psi.

VII. Over surface and over design

Over surface $= \dfrac{U_C - U_{o,cal}}{U_C}$

The clean overall heat fransfer co-efficient: $U_C = \dfrac{h_o \times h_{io}}{h_o + h_{io}}$

$h_{io} = h_i \times \dfrac{d_i}{d_o}$ =141.3×0.834=117.8 Btu h $^{-1}$ ft^{-2} °F^{-1}

U_C =66.98 Btu h $^{-1}$ ft^{-2} °F^{-1}

% Over surface $= \dfrac{66.98 - 53.5}{66.98}\times 100$

=20% (acceptable)

Over design:

% Overdesign $= \dfrac{A\ \text{-}\ A_{reqd}}{A_{reqd}}\times 100$ (1.7)

The design area of heat transfer in the exchanger (n_t =318):

$A = \pi d_o L_t n_t$ =π× $\dfrac{1}{12}$ ×24×368=2312 ft^2

The required heat transfer area (where, n_t=335):

$$A_{reqd} = \pi d_o L_t n_t = \pi \times \frac{1}{12} \times 24 \times 335 = 2105 \text{ ft}^2$$

% Overdesign =9.8% which is within the acceptable limit.

Refer module # 2 for the mechanical design of shell and tube heat exchanger.