Design Procedure of Shell and Tube Heat Exchanger

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Abstract — In the current study inlet temperature of shell and tube side are taken as input parameters with a given bundle arrangement of rotated square pitch / Rotated triangle pitch. The heat transfer analysis is done by considering water inside the tube and steam on shell side. The design of shell and tube heat exchanger using Kern method for water and steam combination is validated by well-known Dittus-Boelter equation of turbulent flow inside tube. The analysis is extended using the above Kern method with different fluid combinations such as carbon-dioxide on the tube side steam on shell side and sulphur-dioxide side on tube side and steam on shell side. Parameters such as heat transfer coefficient, friction coefficient, length, area and pressure drop are determined. Graphs are shown behavior for different fluid combinations.

Index Terms — Shell and tube Heat Exchanger; Dittus - Boelter equation; Kern Method.

I. INTRODUCTION

A. Heat Exchanger

The purpose of a heat exchanger is just that-to exchange heat. Most processes require the heating or cooling of streams to produce a desired temperature before the stream can be fed to operations. In any heat exchanger there must be a fluid that requires a change in energy (heating or cooling) and a fluid that can provide that energy change. One fluid is sent through a pipe on the inside of the heat exchanger while the other fluid is sent through a pipe on the outside. In this configuration, no mixing of the hot and cold fluids needs to take place. This is very convenient for many processes, especially when product purity needs to be ensured. This arrangement also allows for large quantities of heat to be transferred quickly, and it is relatively easy to maintain consistent operating conditions.

There are three principle means of achieving heat transfer, conduction, convection, and radiation. Heat exchangers run on the principles of convective and conductive heat transfer. Radiation does occur in any process. However, in most heat exchangers the amount of contribution from radiation is minuscule in comparison to that of convection and conduction. Convection occurs as the heat from the hot fluid passes through the inner pipe wall. To maximize the heat transfer, the inner-pipe wall should be thin and very conductive. However, the biggest contribution to heat transfer is made through convection.

There are two forms of convection; these are natural and forced convection. Natural convection is based on the driving force of density, which is a slight function of temperature. As the temperature of most fluids is increased, the density decreases slightly. Hot fluids therefore have a tendency to rise, displacing the colder fluid surrounding it. This creates the natural “convection currents” which drive everything from the weather to boiling water on the stove. Forced convection uses a driving force based on an outside source such as gravity, pumps, or fans. Forced convection is much more efficient, as forced convection flows are often turbulent. Turbulent flows undergo a great deal of mixing which allow the heat to be transferred more quickly.

The heat exchangers are classified upon the following factors: a. Construction b. Flow arrangement c. Number of shells d. Contact between the processing streams e. Compactness f. Heat transfer mechanism

Shell and tube heat exchangers are typically used for high pressure applications (with pressures greater than 30 bar and temperatures greater than 260°C)

B. Literature Review

In designing of shell and tube heat exchangers, we paid 10 to 15 times attention in analyzing this problem both experimentally and theoretically. Kern et al provides correction factor charts for different number of shells and even number of tube passes. He presented the correction factor F, as a function of two variables R and S, which depends on the inlet and exit temperatures of the heat exchanger of both the fluids.

Roetzel and Nicolehas et al recognized the potential usefulness of explicit representations of LMTD Correction factors in developing computerized packages for heat exchanger design.

Tinker et al has suggested a schematic flow pattern, which divided the shell–side flow into a number of individual streams. Tinkers model has been the basis of “Stream analysis method”, which utilises a rigorous iterative approach and it is particularly suitable for computer calculations rather than hand calculations.

Saunders et al in his book proposed practical method and simple design factors are provided and the method is used rapidly for fixed set of geometrical parameters. In his work the correction factors are for heat transfer and pressure drop correlations.

Wills and Johnston et al in his article developed a stream analysis method that is visible for hand calculations. They developed a new and accurate hand calculation method for shell and tube pressure drop and flow distributions.

Reppich and Zagermann et al in his paper offers a computer-based design model to determine the optimum dimensions of segmentally baffled shell-and-tube heat exchangers by calculating optimum shell-side, tube-side pressure drops.
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II. DESIGN PROCEDURE OF SHELL AND TUBE HEAT EXCHANGER

A. Step wise Procedure for Calculation:

A heat exchanger can be designed by the LMTD when inlet and out let conditions are specified. When the problem is to determine the inlet and outlet temperatures for a particular heat exchanger, the analysis is performed more easily by using a method based on effectiveness of the heat exchanger and number of transfer units (NTU).

The heat exchanger effectiveness is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

\[ e = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} = \frac{Q}{Q_{\text{max}}} \]  

The actual heat transfer rate Q can be determined by energy balance equation,

\[ Q = \dot{m}_h C_{ph}(t_{hi} - t_{ho}) = \dot{m}_c C_{pc}(t_{ci} - t_{co}) \]  

The fluid capacity rate C:

\[ \dot{m}_h C_{ph} = C_h = \text{hot fluid capacity rate} \]
\[ \dot{m}_c C_{pc} = C_c = \text{cold fluid capacity rate} \]

\[ C_{\text{min}} = (C_h + C_c) / 2 \]  

\[ C_{\text{max}} = (C_h + C_c) \]  

The number of transfer units (NTU) is given as follows:

\[ \text{Heat rate } Q = C_h(t_{hi} - t_{ho}) = C_c(t_{ci} - t_{co}) \]  

Where,

- \( U \) = overall heat transfer coefficient in W/m²K
- \( A \) = surface area in m²
- \( e = \frac{C_h(t_{hi} - t_{ho})}{C_c(t_{ci} - t_{co})} \) 

The governing equation s for design problem are usually given as follows:

\[ \Delta T_{\text{mcf}} = \frac{\dot{m}_c C_c}{\dot{m}_h C_h}(T_{hi} - T_{ho}) = \frac{\dot{m}_h C_h}{\dot{m}_c C_c}(T_{ci} - T_{co}) \]  

Where,

- \( \dot{m}_h \) = mass flow rate, kg/sec
- \( C = \) heat capacity

Log mean temperature difference for pure counter flow

\[ \Delta T_{\text{mcf}} = \frac{\ln(T_{hi} - T_{ho}) - \ln(T_{ci} - T_{co})}{\ln(T_{hi} - T_{ho})/\ln(T_{ci} - T_{co})} \]  

The effective mean temperature difference for cross flow

\[ \Delta T_{\text{m}} = F \Delta T_{\text{mcf}} \]

Where \( F \) = correction factor

Shell-side area is calculated by \( A_s = \frac{D_s C}{B} \) 

Where,

- \( D_s \) = shell diameter, m
- \( C \) = clearance, m
- \( B \) = baffle spacing, m

\[ G_s = \frac{m}{A} \]
Where, \( m \) = mass flow rate of shell side, kg/s
\( A_s \) = area of the shell, m²
\( D_e = 2 \cos \left( \frac{\pi D_o}{\bar{D}_e} \right) \) (9)
Where \( D_o \) = tube pitch, m
\( D_e \) = outer Diameter of the tube, m
\( \bar{D}_e \) = equivalent diameter, m
\( \mu \) = dynamic viscosity, Ns/m²

\[ \text{Shell side Nusselt number is given by Kern} \]
\[ Nu = 0.36 \left[ \frac{k}{\mu} \right]^{0.56} \left[ \frac{G_s}{\mu} \right]^{0.31} \left[ \frac{D_e}{\bar{D}_e} \right]^{0.14} \] (11)
Where
\( k \) = thermal conductivity, W/mK
\( \mu_w \) = dynamic viscosity of water fluid, Ns/m²
\( \mu_{sh} \) = shell fluid dynamic viscosity at average temperature, Ns/m²
\( G_s \) = mass velocity of shell side, kg/m²s
\( D_e \) = equivalent diameter of shell side, m

The shell-side heat transfer coefficient, \( h_o \), is then calculated as:
\[ h_o = \frac{Nu \cdot k}{D_e} \] (12)
where
\( h_o \) = heat transfer coefficient, W/m²K
\( k \) = thermal conductivity, W/mK

**Tube-side heat transfer coefficient by:**
\[ A_t = \frac{\pi D_t^2}{4} \] (13)
Where \( D_t \) = tube inner diameter, m
\[ A_{tp} = \frac{N_t A_t}{\text{no. of passes}} \] (14)
Where \( N_t \) = number of tubes
\( G_t \) = \( \frac{G_s}{A_{tp}} \)

Where
\( G_t \) = mass velocity of tube, kg/m²s
\( A_{tp} \) = heat transfer area based on tube surface, m²
\( U_t = \frac{G_t}{\rho} \) (15)
Where
\( \rho \) = density of fluid at average temperature, kg/m³
\( Re_t = \frac{\pi \rho D_t}{\mu} \) (16)
Where
\( D_t \) = inner diameter of tube, m

Using the Petukhov and kirkillov correlation:
\[ Nu = \left( \frac{f \cdot Re_t}{107 + 11.7 \left( \frac{Re_t}{PR} \right)^{1/2}} \right)^{0.75} \] (17)
Where
\( f \) = friction factor of flow
\( Re \) = Reynolds number
\( Pr \) = prandtl number
Where \( f = (1.58 \ln Re - 3.28)^2 \) (18)

The tube-side heat transfer coefficient, \( h_i \), is then found as:
\[ h_i = \frac{Nu \cdot k}{D_t} \] (19)

The shell-side pressure drop can be calculated from equations
\[ \Delta P = \frac{C_{c} \cdot (N_t + 1)D_o}{2pD_o \bar{D}_e} \] (20)
\( \Delta P \) = pressure drop for shell side, Pa
\( N_t \) = number of baffles
\( \bar{D}_e \) = viscosity correction factor for shell side fluid
\( N_t = L/B \)
\( f = \exp(0.576 - 0.19 \ln Rs) \) (21)

The tube-side pressure drop can be calculated from Equation:
\[ \Delta P_t = \left( \frac{f \cdot LN_t + 4N_t}{4L} \right) \frac{\rho D_t^2}{2} \] (22)
where \( N_t \) = number of passes
\( f \) = friction factor of tube side

**III. RESULT AND DISCUSSION:**
The analysis is compared with the equations mentioned and drawn in Figs (1-3)

Fig. 1 shows the comparison between Nusselt number on tube side using Petukhov – kirkillov correlation in Kern method by well known Dittus - Boeltzer equation. As per the above plot it was found that there is close agreement between two equations with a deviation of around 10 percent. Hence this equation in kern-method is used further to calculate the Nusselt number on tube side.
The above graph (Fig. 2.) shows the variation between the Reynolds number on tube side and Nusselt number on tube side. It was found that as Reynolds number increases the Nusselt number increases. It is found that there is a significant increase of Nusselt number for SO2 and Steam combination when compared to CO2 and Steam combination for the given inlet temperatures.

The graph (Fig 3) shows the variation between the Friction factor on tube side and Reynolds number on tube side. The graph is drawn with combinations of fluids which CO2 - steam and SO2- steam.

From the above it was found that the Reynolds number of fluid on the tube side increases the friction factor decreases. However it is found that more or less for the above two combinations of fluids the friction factor remains unaffected.

**CONCLUSIONS**

The following conclusions are arrived from the transfer analysis of shell and tube heat exchanger for three different fluid combinations (water-steam, CO2-steam and SO2-steam) using kern’s method.

1. Taking the input parameters the values for Nusselt number, Reynolds number, heat transfer coefficient, and pressure drop and friction factor are determined.

2. Validation for Nusselt number on tube side for water using (Petukho v equation) is compared with well-known Dittus-Boelter equation with a deviation of 10 percent.

3. From the data arrived and drawn it is found that as Reynolds number increases Nusselt number increases and friction factor decreases both tube and shell side fluids.

4. Taking steam on shell side and CO2 and SO2 on tube side.

5. Taking Area of heat exchanger fixed, the length of the heat exchanger is found to be 2.16 m and the maximum effectiveness of shell and tube heat exchanger, from the calculations is found to be 0.65.

**FUTURE SCOPE OF WORK**

1. This work can be extended for different bundle tube configurations such as triangular pitch, and for different tube layouts for heat transfer analysis on shell and tube heat exchanger for particular applications.

2. Apart from Kern method considered in this work, it can be carried out by different methods such as Bell-Delaware method, Taborek method in designing/analyzing the shell and tube heat exchanger.

**REFERENCES**

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