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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 andsulphur-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 shows 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 miniscule in comparison to that of convection and conduction. Conduction 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.

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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℃)

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 tubepasses. 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 utilizesa rigorous reiterative 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 exchangersby calculating optimum shell-side, tube–side pressure drops from the equation provided in his work. The six optimized dimensional parameters are number of tubes, tube length, shell diameter, number of baffles, baffle cut, and baffle spacing. The proposed model also carries out also cost analysis.

Lam and Lo et al in his article presented the flow patterns and corresponding analysis of four circular cylinders subjected to a cross flow. Experiments werecarried out at sub critical Reynolds number of 2100. Square arrangement of the cylinders with varying spacingratios and angles of incidents were examined.

Motos et al explained in their study a two dimensional heat transfer analysis in circular and elliptical tube heat exchangers. A finite element method is used to fluid flow and heat transfer equations and Two dimensional Isoperimetric, Four nodded, linear element was implemented for the FEA progress. An and ***Choi et al*** developed procedure for the detailed phenomena in shell and tube heat exchangers to predict the heat and mass transfer characteristics of shell and tube heat exchangers.

Moghadassi and Hosseini et al had given the importance of heat exchangers in chemical and petrochemical industries, Heat exchangers analysis and heat translate calculations are provided. The conventional and prevalent methods (such as KERN method) are presented.

Liljana Markovska and Vera Mesko et al gave optimization of shell-and-tube-heat exchanger is accomplished by use of the optimizer software package. The objective function is defined together with the implicit constraint. The simultaneous equation solving method is used to solve the equations that describe the process.

Lebele-Alawa and Victor Egwanwo et al presented a numerical method of solution, capable of accounting for temperature dependent variation of fluid properties and heat transfer. Field data were collected for three different industrial heat exchangers and basic governing equations were applied. The parameters analyzed include: the outlet temperatures, the heat transfer coefficients and the heat exchanger effectiveness.

Naik and Matawala et al studied design and assessment of counter flow shell and tube heat exchangerby entropy generation minimization method. ***McAdams*** was one of the earliest workers to quantitatively demonstrate this. His analysis was simple and based on tubular heat exchanger. By taking into account the cost of power and fixed cost of the exchanger, per unit heat transferred, simple expressions for estimating the optimum mass velocities for both inside tubes and outside tube fluids are developed .

Peter and Timmerhaus et al recognized the importance of optimizing tube side pressure drop; shell side pressure drop and heat transfer area simultaneously. Consequently, they produced the most detailed and useful work to date on a single shell-and-tube heat exchanger optimization. The problem with their method however, is that it is restricted to shell-and-tube heat exchangers fitted with plain tube. Extension to other exchanger types requires new equations. No guidance is given on how to generate these equations.

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.

$$\varepsilon = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer} = \frac{Q}{Qmax} \quad (1)$$

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

$$Q = \dot{m}_h C_{ph}(t_{h1} - t_{h2}) = \dot{m}_c C_{pc}(t_{c2} - t_{c1}) \quad (2)$$

The fluid capacity rate C:

$$\dot{m}_h C_{ph} = C_h = hot\ fluid\ capacity\ rate$$

$$\dot{m}_c C_{pc} = C_c = cold\ fluid\ capacity\ rate$$

$$C_{min} = (C_h or\ C_h)\ minimum\ fluid\ capacity\ rate$$

$$C_{max} = (C_h or\ C_h)\ maximum\ fluid\ capacity\ rate$$

$$\text{number of transfer units (NTU)} = \frac{UA}{C_{min}} \quad (3)$$

Where,
U = overall heat transfer coefficient in W/m^2
A = surface area in m^2

$$\varepsilon = \frac{C_h(t_{hi} - t_{ho})}{C_{min}(t_{hi} - t_{ci})} = \frac{C_c(t_{co} - t_{ci})}{C_{min}(t_{hi} - t_{ci})} \quad (4)$$

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

$$\text{Heat rate } Q = C_h(t_{hi} - t_{ho}) = C_c(t_{\infty} - t_{ci}) \quad (5)$$

Where,
Q = heat duty of heat exchanger, W
C_h= specific heat of the hot fluid, J/kgK
Cc = specific heat of the cold fluid, J/kgK
T_{hi}= temperature of the hot fluid inside, K
T_{ho}= temperature of the hot fluid outside, K
T_{ci}= temperature of the cold fluid inside, K
T_{∞}= temperature of the cold fluid outside, K
Where heat capacity rate for hot or cold fluid
C = $\mathring{m}C_p$
Where,
$\mathring{m}$= mass flow rate, kg/sec
C= heat capacity
Log mean temperature difference for pure counter flow

$$\Delta T_{lm,cf} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln[(T_{hi} - T_{co})/(T_{ho} - T_{ci})]} \quad (6)$$

The effective mean temperature difference for cross flow

ΔT_m = F ΔTlm,cf
Where F=correction factor
Shell-side area is calculated by $A_s = \frac{(D_s CB)}{P_t}$ (7)
Where,
Ds=shell diameter, m
C= clearance, m
B= baffle spacing, m

$$Gs = \frac{\dot{m}}{A_s} \quad (8)$$

Where, $\mathring{m}$= mass flow rate of shell side, kg/s
As= area of the shell, m^2

$$D_e = 2\cos\frac{4\left(P_t^2 - \frac{(\pi D_o^2)}{4}\right)}{\pi D_o} \qquad (9)$$

Where P_t =tube pitch, m
Do= outer Diameter of the tube, m

$$Re_s = \frac{D_e G_s}{\mu} \qquad (10)$$

Where De= equivalent diameter, m
μ= dynamic viscosity, Ns/m^2

Shell side Nusselt number is given by Kern

$$Nu = 0.36\left[\frac{D_e G_s}{\mu_b}\right]^{0.55}\left[\frac{C_p \mu_b}{k}\right]^{0.33}\left[\frac{\mu_b}{\mu_w}\right]^{0.14} \qquad (11)$$

Where
k= thermal conductivity, W/mK
μ_w= dynamic viscosity of water fluid, Ns/m^2
μ_b= shell fluid dynamic viscosity at average temperature, Ns/m^2
G_s= mass velocity of shell side, kg/m^2s
De= equivalent diameter of shell side, m

The shell-side heat transfer coefficient, ho, is then calculated as:

$$h_o = \frac{N_u k}{D_e} \qquad (12)$$

where
ho= heat transfer coefficient, W/m^2k
k= thermal conductivity, W/mK

Tube-side heat transfer coefficient by:

$$A_t = \frac{(\pi d_i^2)}{4} \qquad (13)$$

Where Di= tube inner diameter, m

$$A_{tp} = \frac{N_t A_t}{\text{no. of passes}}$$

Where Nt= number of tubes

$$G_t = \frac{\dot{m}_t}{A_{tp}} \qquad (14)$$

Where G_t= mass velocity of tube, kg/m^2s
A_{tp}= heat transfer area based on tube surface, m^2

$$U_t = \frac{G_t}{\rho} \qquad (15)$$

Where
ρ= density of fluid at average temperature, kg/m^3

$$Re_t = \frac{u_t \rho d_i}{\mu} \qquad (16)$$

Where
di= inner diameter of tube, m

Using the Petukhov and kirillov correlation:

$$Nu = \frac{\left(\frac{f}{2}\right)RePr}{1.07 + 12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left[(Pr)^{\frac{2}{3}} - 1\right]} \qquad (17)$$

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, hi, is then found as:

$$h_i = \frac{Nu.k}{d_i} \qquad (19)$$

The shell-side pressure drop can be calculated from equations

$$\Delta P = \frac{f G_s^2 (N_b + 1) D_s}{2 \rho D_e \emptyset_s} \qquad (20)$$

ΔP= pressure drop for shell side, Pa
Nb= number of baffles
Ǿs= viscosity correction factor for shell side fluid
Nb = L/B
f = exp(0.576 – 0.19 ln Rs) (21)

The tube-side pressure drop can be calculated from Equation:

$$\Delta P_t = \left(4f\frac{LN_p}{d_i} + 4N_p\right)\frac{\rho \mu_m^2}{2} \qquad (22)$$

where Np= number of passes
f= friction factor of tube side

B. Step Wise Procedure for calculation:

The following steps are adopted for the calculation of parameters of shell and tube heat exchanger

1. The outlet temperatures of shell and tube heat exchanger are computed by equations (4) and (5)
2. The log mean temperature difference to the shell and tube are computed using equations (6)
3. Reynolds numbers on shell side using the following equations (10) is calculated
4. Nusselt number on shell side using the equation (11) by using Macadam's correlation I computed.
5. Heat transfer coefficient on shell side using the equation (12) is calculated.
6. Pressure drop on shell side using the following equation (20) is calculated.
7. Tube side pressure drop by using equations (16), (18), (22) is calculated.

III. RESULT AND DISCUSSION:

The analysis is compared with the equations mentioned and drawn in Figs (1-3)

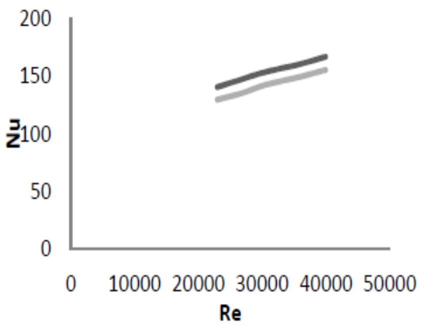

Fig. 1 Variation of Nu vs. Re

Fig.1 shows the comparison between Nusselt number on tube side using Petukhov – kirrilov correlation in Kern method by well known Dittus - Boelter equation. As perthe 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.

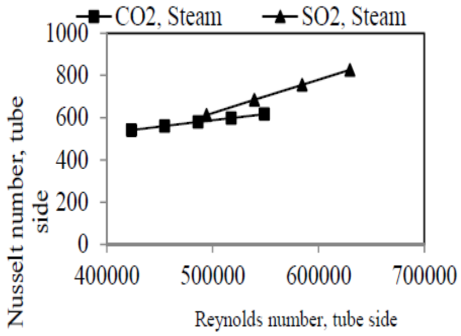

Fig. 2 Variation of Nut Vs Ret

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.

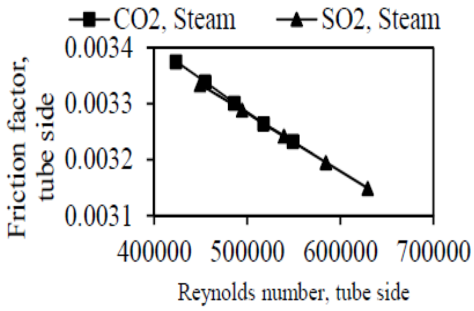

Fig. 3 Variations of f and Re

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. It was found that Nusselt numbers shows steep increase for SO2 steam combination than CO2 steam combination. However taking above two combinations the friction factor almost remains same.
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.

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