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Review on Shell and Tube Heat Exchangers using Helical Baffles and Nanofluids

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ABSTRACT

A shell and tube heat exchanger is a class of heat exchanger. It is most commonly used in oil refineries, high pressure applications and other large scale chemical processes. So there is a lot of need for cooling as well as for dissipating heat. For cooling purpose, we generally use cooling medium like water, ethylene glycol etc. Flow in this type of heat exchanger can be both parallel and counter flow. At present performance is the major factor, which plays a vital role in the selection of any heat exchanger. So in order to increase the performance of the shell and tube heat exchanger it is designed by Kern's method using helical baffles. Baffle cut of 25% is improving pressure drop and increase the velocity. To enhance the heat transfer rate in parallel and counter flow we are adding copper oxide (CuO), Magnesium Oxide (MgO), nano particles mixed with fluid such that it becomes nano fluids and trying to obtain maximum heat transfer rate improving pressure drop and increasing the velocity, heat transfer coefficient efficiency in shell and tube heat exchangers.

Keywords: Shell and Tube Heat Exchanger, Kern Method, Pressure Drop, Helical Baffles.

I. INTRODUCTION

Heat exchangers are always been important part to the lifecycle and operations of many systems. Over the past quarter century, the importance of heat exchangers has increased from the viewpoint of energy conversion and performance recovery. Much more attention is paid to heat exchangers because of environmental concerns such as thermal, air and water pollution, as well as waste heat recoveries. It can be considered as a key equipment in the chemical process industry. Heat exchanger is a device of finite volume used to transfer heat between a solid and a fluid or between two or more fluids. These two fluids are separated by solid wall to prevent mixing and also to prevent direct contact between them. Typically one system is been cooled while the other is heated. More than 30-40% of heat exchangers used in various industries are of this type due to their robust geometry construction, easy maintenance and possible upgrades. One common example of heat exchanger is the radiation in the car, in which it transfers heat from the water (hot engine-cooling fluid) in the radiator to the air passing through the radiator.There are two main types of heat exchangers.

- Direct contact heat exchanger, where both media between which heat is exchanged are in direct contact with each other.
- Indirect contact heat exchanger, where both the media are separated by a wall through which heat is transferred so that they never mix.
- Heat exchangers are also classified based on different parameters like flow direction, compactness of the body, transfer type and construction.
- Parallel flow heat exchangers: Parallel flow heat exchangers are the one in which two fluids flow in parallel to each other.
- Counter flow heat exchangers: In counter flow heat exchangers the fluid flows in opposite direction.
- Cross flow heat exchangers: It is a combination of both parallel and counter flow.

Heat exchangers are globally assumed to be operating under adiabatic conditions. It therefore means that heat losses or gains between the heat exchangers and the environment can be assumed. The thermal inertia for heat exchangers is negligible and therefore mostly assumed therefore the general balance equation of energy is reduced toWhere the total energy h_t is a value that can be approximated by enthalpy and stands for the difference between the output and the input.A primary objective in the heat exchanger Design is the estimation of the minimum heat transfer area required for a given heavy duty.

II. METHODS AND MATERIAL

A. Shell and Tube Heat Excanger

This type of heat exchanger is said to have originated from the jacketed coil distiller. Shell and tube heat exchanger is an indirect contact type heat exchanger. In this we make use of both parallel and counter flow . Shell and tube heat exchangers in various sizes are widely used in industrial operations and energy conversion systems. As the name suggests this type of heat exchangers consists of a shell (a large pressure vessel) with a bundle of pipes inside it. The shell is a container for the shell fluid. Usually, it is cylindrical in shape with a circular cross section, although shells of different shapes are used in specific applications. For this particular study E shell is considered, which is generally a one pass shell. E shell is the most commonly used due to its low cost and simplicity, and has the highest long-mean temperature-difference (LMTD) correction factor. One fluid runs through the tube and the other fluid flows over the tubes (through the shell) to transfer heat between the two fluids. The set of tubes is called “tube bundle” pipes composed in it can be plain ,longitudinally finned etc. Although the tubes may have single or multiple passes, there is one pass on the shell side, while the other fluid flows within the shell over the tubes to be heated or cooled. Tubular Exchanger Manufacturers Association (TEMA) regularly publishes standards and design recommendations.

Shell and tube heat exchangers are designed normally by using Kern’s method o Bell-Delware method .Kern’s method is mostly used for the preliminary design and provide conservative results whereas; the Bell – Delaware method is more accurate method and can provide detailed results. It can predict and estimate pressure drop and heat transfer coefficient with better accuracy.

B. Basic components of a shell and tube heat exchanger

The major components of a shell and tube heat exchangers are tubes (tube bundles), tube sheets, shell, impingement plates, channel covers, baffles.

1. **TUBES :**The tubesare the basic components of the shell and tube heat exchanger, providing the heat transfer surface between one fluid flowing inside the tube and the other fluid flowing across the outside of the tubes. It therefore recommended that the tubes material should be highly thermal conductive otherwise proper heat transfer will not occur. The tubes may be seamless or welded and most commonly made of copper or steel alloys.
2. **TUBE SHEETS:** The tubes are held in place by being inserted into holes in the tubes sheets and there either expanded into grooves cut into the holes or welded to the tube sheet. The tube sheet is usually a single round plate of metal that has been suitably drilled and grooved to take the tubes however where the mixing between two fluids must be avoided, a double tube sheet may be provided. The space between the tube sheets is open to the atmosphere so any leakage of either fluid should be quickly detected. The tube sheet must withstand to corrosion .The tube sheets are made from low carbon steel with a thin layer of corrosion resisting alloy metallurgic ally bounded to one side.
3. **SHELL:** The shell is simply the container for the shell side fluid, and the nozzles are the inlet and exit ports. The shell normally has a circular cross section and is commonly made by rolling the metal plate of appropriate dimensions in to cylinder and welding the longitudinal joint. In large heat exchanger , the shell is made out of low carbon steel wherever possible for the reason of the economy, though other alloys can be and are used when corrosion or to high temperature strength demand must be made.
4. **IMPINGEMENT PLATES:** When the fluid under high pressure enters the shell there are high chances that if the fluid will directly impinge over the tubes then their breakage or deformation may occur. To avoid the same impingement plates are installed to waste the kinetic energy of fluid upto some extent

so that fluid may impact the tubes with lower velocity.

5. **CHANNEL COVERS:** The channel covers are round plates to bolt to the channel flanges and can be removed for the tube inspection without disturbing the tube side piping. In smaller heat exchangers, bonnets with flanged nozzles or threaded connections for the tube side piping are often used instead of channel and channel covers.
6. **BAFFLES:** Baffles serve two functions: Most importantly, they support the tubes in the proper position during assembly and operation and prevent vibration of the tubes caused by flow induced eddies, and secondly, they guide the shell side flow back and forth across the field, increasing the velocity and heat transfer coefficient.

C. Kern's Method

Kern provided a simple method for calculating shell-side pressure drop and heat transfer coefficient. However, this method is restricted to a fixed baffle cut (25%) and cannot adequately account for baffle-to-shell and tube-to-baffle leakage. Kern method is not applicable in laminar flow region where shell-side Reynolds number is less than 2000. Although the Kern equation is not particularly accurate, it does allow a very simple and rapid calculation of shell-side heat transfer coefficient and pressure drop to be carried out.

D. Nano Fluids

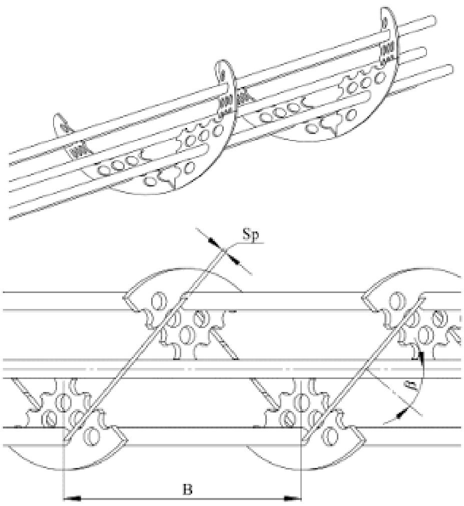
Any metal will have most enriched properties as a power in Nano (10^{-9}) size than in its solid form. Basically heat exchangers use water ethylene glycol etc. But here we are using Nano fluids for better rate of heat transfer. Nanofluids are maiden used by Choi ,then later experimented by L.B Mapa, Sana Mazhar, “Heat transfer in mini heat exchanger using nano fluids”, Thus we are opting for nanoparticles of copper oxide and magnesium oxide with water thus making it a nanofluid as a medium in heat exchanger. This nano fluid is used in heat exchanger in the cold side.

E. Literature Review

A lot has been written about designing heat exchangers, and specifically shell-and-tube heat exchangers. For example, the book by Kern [1] published in 1950 details basic design procedures for a variety of heat exchangers. In the majority of published papers as well as in industrial applications, heat transfer coefficients are estimated, based, generally on literature tables. These values have always a large degree of uncertainty. So, more realistic values can be obtained if these coefficients are not estimated, but calculated during the design task. A few numbersof papers present shell and tube heat exchanger design including overall heat transfer coefficient calculations (Polley et al., 1990, Polley and Panjeh Shah, 1991, Jegede and Polley, 1992, and Panjeh Shah, 1992, Ravagnani, 1994, Ravagnani et al. (2003), Mizutani et al., 2003, Serna and Jimenez, 2004, Ravagnani and Caballero, 2007a, and Ravagnani et al., 2009) Gang yong Lei et al [1] have showed the effects of baffle inclination angle on flow and heat transfer of a heat exchanger with helical baffles, where the helical baffles are separated into inner and outer parts along the radial direction of the shell. While both the inner and outer helical baffles baffle the flow consistently, smoothly and gently, and direct flow in a helical fashion so as to increase heat transfer rate and decrease pressure drop and impact vibrations, the outer helical baffle becomes easier to manufacture due toits relatively large diameter of inner edge. Lutcha J et al [2]have done experiments to the improvement of tubularheat exchangers with helical baffles for investigation of the flow field patterns generated by various helix angles which is expected to decline pressure at shell side and increase heat transfer process significantly.

F. Helical Baffles

The Helical Baffle heat Exchanger is otherwise known as a Helix changer solution that removes many of the deficiencies of Segmental Baffle Heat Exchanger. Itis very effective where heat exchanger are predicted to be faced with vibration condition Quadrant shaped baffle segment are arranged right angle to the tube axis in a sequential pattern that guide the shell side flow in a helical path over the tube bundle. The Helical flow provides the necessary characteristics to reduce flow dispersion and generate near plug flow conditions. The shell side flow configuration offers a very high conversion of pressure drop to heat transfer. Advantages over segmental STHE are increased heat transfer rate, reduced bypass effects, reduced Shell Fouling Factor, Prevention of flow induced vibration & Reduces Pumping cost. For the convenience of manufacturing, up to now all helical baffles actually used in Shell and tube heat exchangers arenon-continuous approximate helicoids. The non-continuous helical bafflesareusuallymadebyfourelliptical sector-shaped plates joined in succession. The elliptical sectorshaped plates are arranged in a pseudohelical (non continuous) manner, with each baffle occupying one-quarter of thecross section of the heat exchanger and being angled to the axis ofthe heat exchanger.The two adjacent baffles may be joined end toend at the perimeter of each sector, forming acontinuous helix atthe outer periphery (Fig. 1(a)); this structure of connecting bafflestogether is called a single helix manner. Another connection be-tween two adjacent sectors is the middle-over lappedconnection, as shown in Fig. 1(a) where the helix angle, designated by, helical pitch, B, and baffle thickness, Sp, are presented. As shown in Fig. 1(c), the helix angle is referred to as the angle between the normal line of the elliptical sector-shaped plates and the heat ex-changer axis. The research results of experimental measurements and numerical simulations provide the bases of engineering design method, for which the primary objects are to determine the required heat transfer surfaces and the fluid pressure drops of shell-and-tube sides. In the design method, the input data are flow rates and at least three of the inlet and outlet temperatures of both sides in heat exchanger. After primary guessing for the heat exchanger structure, the over-all heat transfer coefficient and the pressure drop can be determined by adopting correlations obtained from tests or simulations. If the calculated heat transfer rate and pressure drops cannot satisfy the design requirements, the heat exchanger is re-constructed, and the calculation is repeated again until the calculated heat transfer rate and the pressure drops can satisfy the pre-conditions. It can be seen that the heat transfer and pressure drop correlations are the basis for the design method.



G. Boundary Conditions

- The working fluid of the shell side is water
- The shell inlet temperature is set to 300k
- The constant wall temperature of 450k is assigned to the tube walls
- Zero gauge pressure is assigned to that outlet nozzle
- The inlet velocity profile is assumed to be uniform
- No slip condition is assigned to all surfaces

The zero heat flux boundary conditions are assigned to the shell outer wall (excluding the baffle shell interfaces), assuming the shell is perfectly insulated.

H. Geometry Modeling

Heat exchanger length L	250mm
Shell inner diameter, Di	83mm
Tube outer diameter, d_0	7mm
Tube bundle geometry and pitch triangular	30mm
Number of tubes, Nt	12
Number of baffles, Nb	4
Central baffle spacing, B	78mm
Baffle inclination angle ,	0^0

III. RESULTS AND DISCUSSION

Formulae

1) Correlations for Flow and Heat Transfer Characteristics in Shell Side of Shell and tube heat exchangers:

Based on Ref [.2–5, 6] the correlations for flow and heat transfer in shell side of Shell and tube heat exchangers are proposed and collected in this section. Most of the symbols used the following presentation are the same as what were used in Ref. [6] for the sake of convenience.

1.1) Correlations for Heat Transfer Coefficient in Shell Side of Shell and tube heat exchanger

The average Nu number for the shell side of shell and tube heat exchanger [6] is determined by:

$$Nu_s = 0.62 \text{ X } (0.3 + \sqrt{Nu_{lam}^2 + Nu_{turb}^2}\ ) \times Y_2 \times Y_3 \times Y_4 \times Y_5 \times Y_6 \times Y_7 \times Y_9 \times Y_{10} \quad (1)$$

Where

$$Nu_{lam} = 0.664 \text{ Re}^{0.5} \text{ Pr}^{0.33} \quad (2)$$

$$Nu_{turb} = \frac{0.037\, Re^{0.7}\ Pr}{1+2.433Re^{-0.1}\ (Pr^{0.67}-1)} \quad (3)$$

In Eq. (1) coefficientsYiare the correction factors. Their physical meanings are defined as follows 6.Y2accounts for the thermal-physics properties effects; Y3accounts for the scale-up from a single tube row to a bundle of tubes; Y4accounts for the adverse temperature gradient; Y7accounts for the bundle-shell bypass streams; Y8accounts for the baffle spacing in inlet and outlet sections;Y9accounts for the change in the cross-flow characteristics in heat exchanger; andY10accounts for the turbulent enhancement.

Average heat transfer coefficient for shell side of Shell and tube heat exchanger [6] is

$$H_s = \frac{Nu_s}{l} \times \lambda_s \quad (4)$$

Where

$$l = \frac{\pi d_0}{2} \quad (5)$$

Wheredois the outside diameter of the tube; and is thermal conductivity of shell-side fluid. The application ranges of Eqs. (1)–(5) are $10<Re<10^6$,$10<Pr<10^3$,nrc10, and 5≤45 deg, where

$$n_{rc} = n_{rp}(n_p\text{-}1)$$

$n_{rp}$is the number of tube rows in the cross section of heat ex-changer; andnpis the number of baffles.

1.2) Correlations for Pressure Drop in Shell Side of Shell and tube heat exchangers:

According to Stehlik et al.[6], the pressure dropcross the bundle per unit cycle without bypass flow can be determined by

$$\Delta p_{t0}^1 = 2\lambda_{22} n_r^1 \rho_2 u_2^2 Z_2 Z_6 Z_7 \quad (7)$$

The pressure drop cross the whole bundle zone with bypass flows [6]

$$\Delta p_{t0} = \Delta p_{t0}^1 \frac{l_{t0}}{B} Z_3 \quad (8)$$

The pressure drop in the inlet and outlet zones [6]

$$\Delta p_{tn} = \Delta p_{t0}^1 Z_5 \quad (9)$$

Where $n_r^1$is the number of tube rows on the centre stream line within one cycle.22is the friction factor of ideal cross-flowthrough tube bundle, which can be determined by referring to [5,6.] $l_{to}$is the baffled length of tube bundle.

In Eqs. (7)–(9) correction factors are defined as[6] follows.Z_2accounts for the thermal-physics properties effects;Z_3accountsfor the bundle-shell bypass streams;Z_5accounts for the baffles pacing in inlet and outlet sections;Z_6accounts for the change in the cross-flow characteristics in heat exchanger; andZ_7accountsfor the turbulent enhancement.

The pressure drop in the inlet and outlet nozzles can be calculated by [8,9]

$$\Delta p_{nozzle} = \xi \text{ x } 0.5 \text{ x } \rho v_{s,nozzle}^2 \quad (10)$$

Where is taken as 1.5 or 2.0 by referring to Refs. [8, 9].The over-all pressure drop of the shell-side fluid

$$\Delta p_{s,all} = \Delta p_{tn} + \Delta p_{nozzle} + \Delta p_{t0} \quad (11)$$

(6)

From above presentation, it can be seen that the determination of factorsYiandZiis the key issue to obtain the shell-side fluid heat transfer coefficient and pressure drop. Section 2.3 is for this purpose.

1.3) Determination of Factors

$Y_2$and Z_2 [5,6]

$$Y_2 = \left(\frac{\eta_s}{\eta_{s,w}}\right)^{0.14} \quad (12)$$

$$Z_2 = \left(\frac{\eta_s}{\eta_{s,w}}\right)^{-0.14} \quad (13)$$

Whereηs, w is the dynamic viscosity at average temperature of tube wall.

The determination of average temperature of tube wall is con-ducted by [5]

$$t_w = t_{t,avg} + \left(\frac{t_{s,avg} - t_{t,avg}}{1+\frac{h_t}{h_s}}\right) \quad (14)$$

Where $t_{t,avg}$ and $t_{t,avg}$ are the averaged inlet and outlet temperatures of tube side and shell side in the heat exchanger, respectively. h_t and h_s are heat transfer coefficients for tube side and shell side, respectively.

Y3[6,7]. For in-line arrangement,

$$Y_3 = 1+\frac{0.7}{\epsilon^{1.5}}(b / a)/(b / (a\ \ ) + 0.7))^2 \quad (15)$$

For staggered arrangement,

$$Y_3 = 1+\frac{2}{3b} \quad (16)$$

Whereas the ratio of distance between the tube normal to the flow direction and the central tube pitch, is the ratio of distance between tube in the flow direction and the central tube pitch, as shown in Fig. 2, and the parameters determined by

If b≥1 ; $\varepsilon = 1-\frac{\pi}{4a}$ (17)

If b<1 : $\varepsilon = 1-\frac{\pi}{4ab}$ (18)

Y $_7$and Z_3 [6].Y_7andZ_3 are functions of t_t.n $_{pt}$ /D $_1$ and S $_{22}$ /S $_{2z}$, as shown in graphs presented by Stehlik et al.[6]. These curves have been fitted to the following equations (usingxandy, respectively, to substitutet $_t$ n $_{pt}$ /D $_1$andS$_{22}$ /S $_{2z}$for simplicity:

$$Y_7 = \exp[-1.343x(1-(2y)^{0.338})] \quad (19)$$

$$Z_3 = \exp[-3.56x(1-(2y)^{0.363})] \quad (20)$$

$$S_{ss} = 0.5(B-{}^{S_p}/_{cos\beta})[D_1 - D_s - S_{tt}]$$

$$S_{2z} = 0.5(B-{}^{S_p}/_{cos\beta})[D_i - D_1 + \frac{D_1-d_0}{t_1}(t_t - d_0)]$$

In the above equations, t_t is the tube pitch, D $_1$is the inner diameter of shell,Spis the thickness of baffle, S_{tt} is distance between the two tubes' outside surfaces, nptis the number of stealing strip pairs, and Dsis the diameter of tube bundle. It should be emphasized that for the shell and tube heat exchanger because the shell-side flow pattern resulted from the helical-type structure is close to helical flow, the cross section area,S_{2z} , is actually only half of the entire cross section at the shell centreline of the heat exchanger.

Y_8 and Z_5 [6].Y_8 and Z_5 are functions of (ltc−lto/ltc) and B /D1, as shown in graphs presented by Stehlik et al.[6]. Again, the present authors have made curve-fitting for the convenience of design as follow susingxandy ,respectively, to substitute (ltc−lto/ltc) and B/D1 for simplicity:

$$Y_8 = 1.079y^{0.0487} - 0.445y^{-0.301}x^{1.2} \quad (23)$$

$$Z_5 = (-0.0172 + 0.0899y)x^{-1.2} \quad (24)$$

Where l_{tc} is the effective length of the tube bundle, andl_{t0} is the baffled length of tube bundle.

Figure 3 illustrates the definitions of ltoandltc. The helical pitch can be calculated withD1andat hand(see Eq.(25)) [9], and then the maximum number of baffle numbers can be determined with specified value of ltc The baffle number is an integral. Then ltoand the distances between inlet and outlet baffles to tube sheet,ltn1andltn2, can be determined with ease

$$B = \alpha n.D_1 \sin\frac{\pi}{n}.\tan\beta, \qquad n \geq 2, \qquad 0<\alpha \leq 1 \quad (25)$$

Where α is the dimensionless radius of the contacting point of the two successive helical baffles see Fig. (4). Y9and Z6[6]. From the graphs presented by Stehlik et al[.13],Y9andZ6are only influenced by helical angle.

The curves in [16]can be fitted to the following equations:

$$Y_9 = 0.977+0.00455x - 0.0001821x^2 \quad (18 \text{ deg} \le x \le 45 \text{ deg}) \qquad (26)$$

$$Y_9 = 1 \quad (x< 18 \text{ deg})$$

$$Z_6 = 0.289 - (5.06 \times 10^{-4})x - (4.53 \times 10^{-5})x^2$$

Where x represents the helical angleβ

Y10and Z7[6] .Y10andZ7are also only influenced by the helical angle, as shown in the graphs presented by Stehlik et al.[6]. The following curve-fitted equations are obtained by thepresent authors:

$$Y_{10} = -56.39 + 8.28x - 0.46x^2 + 0.012x^3 - (1.64 \times 10^{-4})x^4 + (8.19 \times 10^{-7})x^5 \quad (25 \text{ deg} < x < 45 \text{ deg}) \qquad (29)$$

$$Y_{10} = 1 \quad (x<25 \text{ deg})$$

$$Z_7 = -5.411 + 0.379x - 0.00402x^2 \quad (22 \text{ deg} < x < 45 \text{ deg})$$

$$Z_7 = 1 \quad (x<22 \text{ deg})$$

Where x represents the helical angleβ

2) **Correlations for the Flow and Heat Transfer in Tube**

Side of Shell and tube heat exchanger

2.1 Correlations for Heat Transfer Coefficient in Tube Side of Shell and tube heat exchanger

The average heat transfer coefficient of tubeside is calculated by the Gnielinski equation in turbulence condition or the Sieder–Tate equation in laminar condition [7–9]

2.2 Correlations for Pressure Drop in Tube Side of Shell and tube heat exchangers [8,9]:

Where√is taken as 1.5 or 2.0 [8,9]; kcandkeare friction factorsfor the sudden contraction and expansion effects, respectively, when the tube side fluid flows into and out of the tubes; Npis the number of tube passes; and if there is only one tube pass in the heat exchanger, the number "4" in Eq.33should be omitted. The friction factor ftcan be determined by referring to Refs.[8,9].

$$\Delta P_{t,all} = \frac{1}{2}\xi\rho v_{t,nozzle}^2 + \frac{1}{2}\rho u_t^2 \left(\frac{f_t L_{tc}}{d_i}\frac{1}{(\varphi_t^r)} + k_c + k_e + 4\right) N_p \qquad (33)$$

(27)

IV. CONCLUSION

(28)

From results obtained as per experimental analysis, the overall heat transfer and effectiveness increases along with the increase in the pressure drop with decrease of LMTD(logarithmic mean temperature difference). Highest effectiveness is obtained at low flow rate directions of fluid. Upto 46% of increment is seen in effectiveness and heat transfer rate by using helical baffles and by addition of 0.2% of copper oxide and 0.4% of magnesium oxide nano particles to the fluid.Thus our assumption of increment of heat transfer rate can be obtained.

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