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DESIGN & OPTIMIZATION OF SHELL & TUBE HEAT EXCHANGER USING HTRI SOFTWARE

KEYWORDS: Design Of Shell and Tube Heat Exchanger, HTRI Software, Kern Method and Bell – Delaware Method Kern Method and Bell-Delaware Method

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ABSTRACT Shell-and-tube heat exchangers are probably the most common type of heat exchangers applicable for a wide range of operating temperatures and pressures. They have larger ratios of heat transfer surface to volume than double-pipe heat exchangers, and they are easy to manufacture in a large variety of sizes and flow configurations. They can operate at high pressures, and their construction facilitates disassembly for periodic maintenance and cleaning. Shell-and-tube heat exchangers find widespread use in refrigeration, power generation, heating and air conditioning, chemical processes.

The main design objectives here are to accommodate thermal expansion, to furnish ease of cleaning, or to provide the least expensive construction if other features are of no importance.

In this present work we focused on design and optimization of shell and tube heat exchanger using HTRI software. To design shell and tube heat exchanger we were use Design methods,

1) Kern method
2) Bell-Delaware method

I. INTRODUCTION

A shell and tube heat exchanger is a class of heat exchanger designs. It is the most common type of heat exchanger in oil refineries and other large chemical processes, and it is suite for high pressure applications. As its name implies, this type of heat exchanger consists of a shell (a large pressure vessel) with a bundle of tubes inside the shell.

The basic principle of operation is very simple as flows of two fluids with different temperature brought into close contact but prevented from mixing by a physical barrier. Then the temperature between two fluids tends to equalize by transfer of heat through the tube wall. The fluids can be either liquids or gases on either the shell or the tube side. In order to transfer heat efficiently, a large heat transfer area should be used, leading to the use of many tubes. In this way, waste heat can be put to use. This is an efficient way to conserve energy.

The basic construction & Components of shell and tube heat exchanger are tubes (or tube bundle), shell, front-end head, rear-end head, baffles, and tubes heats are shown in fig.1 (a).

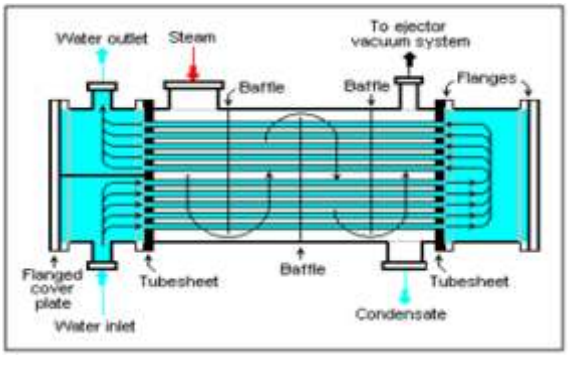


Fig. 1(a) –Components of Shell and Tube Heat Exchanger

Essentially, a shell and tube exchanger consists of a bundle of tubes enclosed in a cylindrical shell. The ends of the tubes are fitted into tube sheets, which separate the shell-side and tube-side fluids. Baffles are provided in the shell to direct the fluid flow and support the tubes. The assembly of baffles and tubes is held together by support rods and spacers as shown in fig.1 (b).

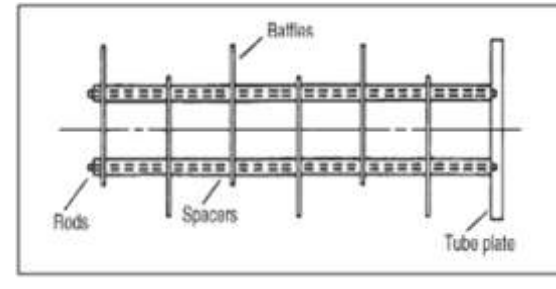


Fig. 1(b) – Baffles Spacer and Tie Rod

II. HEAT-EXCHANGER STANDARDS AND CODES

The mechanical design features, fabrication, materials of construction, and testing of shell and tube exchangers is covered by British Standard, BS 3274. The standards of the American Tubular Heat Exchanger Manufacturers Association, the TEMA standards, are also universally used. The TEMA standards cover three classes of exchanger: class R covers exchangers for the generally severe duties of the petroleum and related industries; class C covers exchangers for moderate duties in commercial and general process applications; and class B covers exchangers for use in the chemical process industries. The British and American standards should be consulted for full details of the mechanical design features of shell and tube exchangers; only brief details will be given in this chapter. The standards give the preferred shell and tube dimensions; the design and manufacturing tolerances; corrosion allowances; and the

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recommended design stresses for materials of construction. The shell of an exchanger is a pressure vessel and will be designed in accordance with the appropriate national pressure vessel code or standard.

III. SELECTION OF HEAT EXCHANGER

The basic criteria for heat exchanger selection from various types available are:

1) The heat exchanger must satisfy the process specifications; it must continue to the next scheduled shut down of the plant for maintenance.
2) The heat exchanger must withstand the service conditions of the plant environment. It must also resist corrosion by the process and service streams as well as the environment. The heat exchanger should also resist fouling.
3) The exchanger must be maintainable. The usually implies choosing a configuration that permits cleaning; and replacement of any components that may be especially vulnerable to corrosion, erosion or vibration. This requirement will dictate the positioning of exchanger, and the space requirement around it.
4) The heat exchanger should be cost effective. The installed operating, maintenance costs including the loss of production due to exchanger unavailability must be calculated and exchanger should cost as little as possible.
5) There may be limitations on exchanger diameter, length, weight, and tube configuration due to site requirements, lifting and servicing, capabilities, or inventory considerations.

In this present work we focused on BEM type shell and tube heat exchanger. This TEMA types the simplest design and is constructed without packed or gasketed joints on the shell side. The tube sheet is welded to the shell and the heads are bolted to the tube sheet.

ADVANTAGES:

1) Economical design.
2) No possibility of contamination compared to designs with floating head gaskets.
3) High pressure shell side designs more easily accomplished than in floating head types.
4) Extremely efficient shell side heat transfer due to small annulus between outer tube limit (O.T.I.) and shell I.D.
5) Shell design can be modified for large expansion area where partial vaporization of liquid occurs.
6) No gasketed joints on shell side.

IV. DESIGN METHODOLOGY

1) Bell (1960, 1963) developed a semi-analytical method based on work done in the cooperative research programmed on shell and tube exchangers at the University of Delaware. His method accounts for the major bypass and leakage streams and is suitable for a manual calculation.
2) Though Kern's method does not take account of the bypass and leakage streams, it is simple to apply and is accurate enough for preliminary design calculations and for designs where uncertainty in other design parameters is such that the use of more elaborate methods is not justified.

In this present work we focused on Kern's design Method .In this design our aim to get required overall heat transfer coefficient, as per the thermal as well as mechanical data required to client.

KERN'S METHOD

This method was based on experimental work on commercial exchangers with standard tolerances and will give a reasonably satisfactory prediction of the heat-transfer coefficient for standard designs. The prediction of pressure drop is less satisfactory, as pressure drop is more affected by leakage and bypassing than heat transfer. The shell-side heat transfer and friction factors are correlated in a similar manner to those for tube-side flow by using a hypothetical shell velocity and shell diameter. As the cross-sectional area for flow will vary across the shell diameter, the linear and mass velocities are based on the maximum area for cross-flow: that at the shell equator. The shell equivalent diameter is calculated using the flow area between the tubes taken in the axial direction (parallel to the tubes) and the wetted perimeter of the tubes; see Figure 3(b)

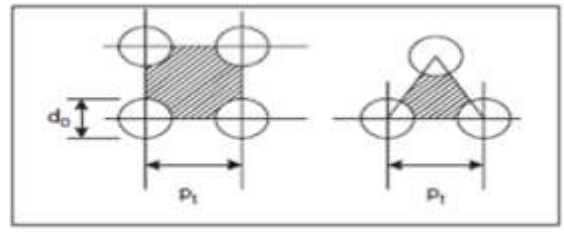


Fig.2 (a) Equivalent Diameter, cross sectional area and wetted perimeters

The procedure for calculating the shell-side heat-transfer coefficient and pressure drop for a single shell pass exchanger is given below:

PROCEDURE

1) Calculate the area for cross-flow as for the hypothetical row of tubes at the shell equator, given by:

$$A_s = \frac{(P_t - d_o) D_s l_b}{p_t} \quad \ldots\ldots\ldots (1)$$

Where,

P_t = tube pitch,
d_o = tube outside diameter,
D_s = shell inside diameter, m,
l_b = baffle spacing, m.

The term $\frac{(P_t - d_o)}{p_t}$ is the ratio of the clearance between tubes and the total distance between tube centers.

2) Calculate the shell-side mass velocity Gs and the linear velocit $_s$:

$$G_s = \frac{W_s}{A_s} \qquad u_s = \frac{G_s}{\rho}$$

Where,
W_s = fluid flow-rate on the shell-side, kg/s,
ρ = shell-side fluid density, kg/m3.

3) Calculate the shell-side equivalent diameter (hydraulic diameter), Figure 2(a) for a square pitch arrangement

$$d_e = \frac{4\left(\frac{p_t^2 - \pi d_o^2}{4}\right)}{\pi d_o} = \frac{1.27}{d_o}\,(p_t^2 - 0.785\, d_o^2) \ldots\ldots (2)$$

For an equilateral triangular pitch arrangement:

$$d_e = \frac{\left(\frac{p_t}{2} \times 0.87\, p_t - \frac{1}{2}\pi \frac{d_o^2}{4}\right)}{\frac{\pi d_o}{2}}$$

$$= \frac{1.10}{d_o}\,(p_t^2 - 0.917\, d_o^2) \quad \ldots\ldots\ldots (3)$$

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4) Calculate the shell-side Reynolds number, given by:

$$R_e = \frac{G_s d_e}{\mu} = \frac{u_s d_e \rho}{\mu} \quad \ldots\ldots\ldots (4)$$

5) For the calculated Reynolds number, read the value of j_h from Figure 2(b) for the selected baffle cut and tube arrangement, and calculate the shell-side heat transfer coefficient h_s from:

$$N_u = \frac{h_s d_e}{k_f} = j_h R_e p_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad \ldots\ldots (5)$$

6) For the calculated shell-side Reynolds number, read the friction factor from Figure 2© and calculate the shell-side pressure drop from:

$$\Delta P_s = 8 j_f \left(\frac{D_s}{d_e}\right).\left(\frac{L}{l_b}\right)\frac{\rho \mu_s^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \quad \ldots\ldots (6)$$

Where,

L = tube length,

l_b = Baffle spacing.

The term (L/l_b) is the number of times the flow crosses the tube bundle = $(N_b + 10)$, Where N_b is the number of baffles.

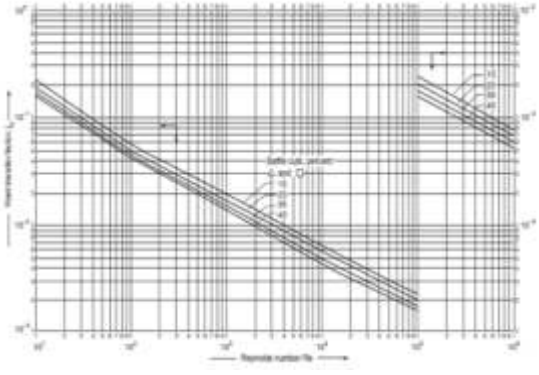


Fig.2 (b) Shell-Side Heat-Transfer Factors, Segmental Baffles

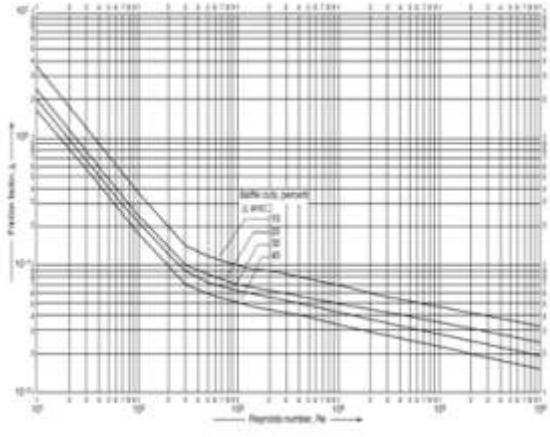


Fig. 2 (c) Shell-Side Friction Factors, Segmental Baffles

SHELL NOZZLE-PRESSURE DROP

The pressure loss in the shell nozzles will normally only be significant with gases. The nozzle pressure drop can be taken as equivalent to $1\frac{1}{2}$ velocity heads for the inlet and 1/2 for the outlet, based on the nozzle area or the free area between the tubes in the row immediately adjacent to the nozzle, whichever is the least.

V. RESULT & DISCUSSION

In this our present work we used the HTRI software for thermal design and analysis of shell and tube "BEM" type heat exchanger. And for theoretical results, Kern's method is used. In this design our aim to get required overall heat transfer co-efficient, as per the thermal as well as mechanical data required to client.

During the work we found that there are some variation in between theoretical and software calculation.

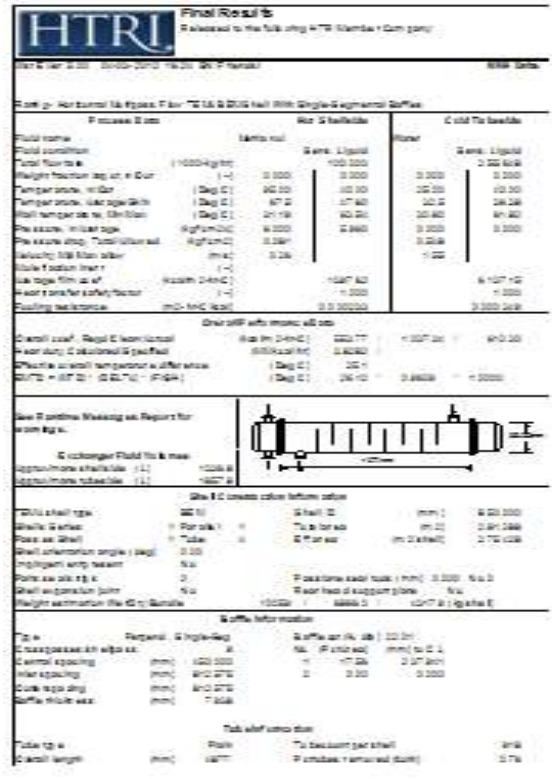
After choosing lengths of tubes are 4000 mm.4877 mm and 5000 mm. the overall heat transfer co-efficient is calculated. The result analysis is discussed in below table.

Sr no	Length of tubes(mm)	Required over all heat transfer co-efficient ($W/m^2.^0C$)	Obtained overall heat transfer co-efficient ($W/m^2.^0C$)
1	4000	600	628
2	4877	600	619
3	5000	600	633

Table.1 overall heat transfer co-efficient with length

From above table it is clear that the required co-efficient is 600 W/m2 .0C, now by choosing the tubes-length of 4000 mm we got 628 W/m2 .0C, which is higher than that of required. In the next step we choose length 4877 mm and got co-efficient 619 W/m2 .0, which is quite nearer to 600 W/m2 .0C. At last we choose the length 5000 mm and got 633 W/m2 .0C , which is much higher than that of required. So we use finally the length of tube is 4877 mm. and then made design of tube and tube pitch, design of shell, baffle cut and baffle design, tube sheet design and all over design of "BEM" type shell and tube heat exchanger under all operating conditions.

In the HTRI software analysis we got the heat transfer co-efficient 553 W/m2 .0C. This variation in software analysis and theoretical analysis is due to fouling, leakage and vibration in practical work.



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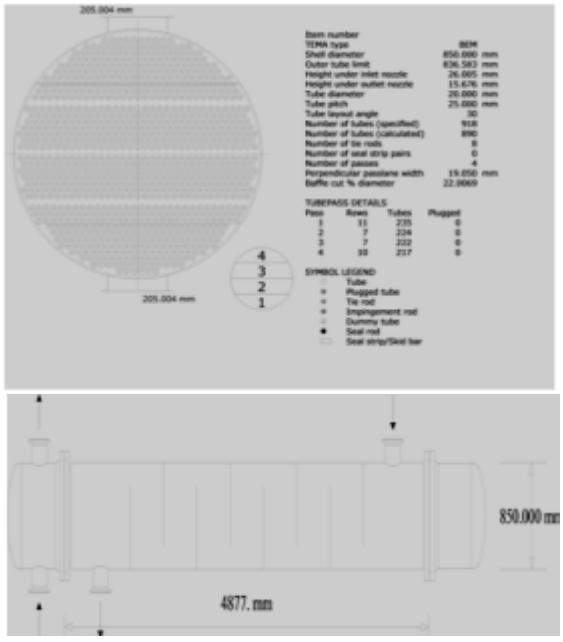


Fig.3 (a) Tube Layout

The three dimensional model which are generated in the software, given below,

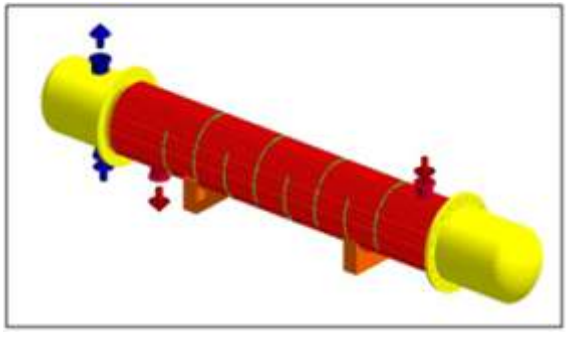


Fig. 3(b) Tubes and Baffles Layout

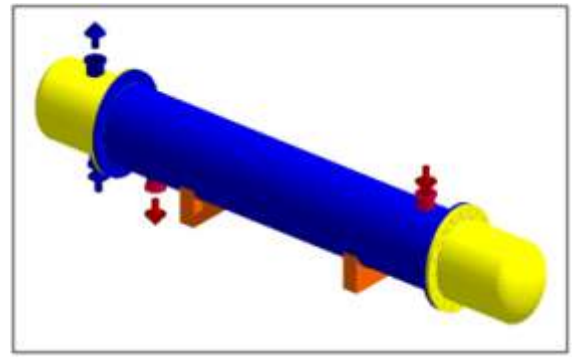


Fig. 3 (c) Shell Layout

VI. CONCLUSION:-

In this Present work we use HTRI software for thermal design. we choose tube length as a variable parameter and different mechanical parameters by mathematical calculation. Which are required in design and construction of shell and tube heat exchanger.

Optimum design of shell and tube heat exchanger by choosing minimum length and corresponding maximum nearer heat transfer co-efficient.

We got software result which are compared to theoretical results, discussed below,

- Software result :

Length = 4877 mm

Heat transfer co-efficient = 553 W/m2 .0C.

- Theoretical result

Length = 4877 mm

Heat transfer co-efficient = 619 W/m2 .0C.

Finally we got optimum thermal design of shell and tube heat exchanger.

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