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Design and Experimental Analysis Study to Improve the Performance of Shell n Tube Heat Exchangers Using Helical Baffles

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Abstract

In industries and many of the research fronts, the most commonly used Heat exchangers are the Shell n Tube Heat Exchangers. Selection of baffle in Shell n Tube Heat Exchangers is most important as it influences the thermo-hydraulic performance of heat exchangers. Baffle selection facilitates to control and improve the thermo-hydraulic performance of heat exchanger The main roles of a baffle in a shell and tube heat exchanger is to hold the tubes in position both in production and operation as well as prevent the effects of vibration, which is increased with both the length and fluid velocity of the exchanger .The Fluid velocity and the effective heat transfer co-efficient of the exchanger enhance due to use of baffles.There are some limitations of using segmental baffles in shell side flow path as there is less heat transfer and excessive pressure drop. These segmental types of arrangements of baffle limits maximum thermal effectiveness. Helical baffle is novel type of baffle, which provides further improvement. The rotational and helical flow pattern is obtained using helical baffles. This is owing to its helical geometry. Thus helical baffles have a contrary effect compared to segmental baffles resulting in an increase in heat transfer coefficient per unit pressure drop.

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Keywords*: Shell-and-tube heat exchangers, Baffle,Heat transfer co-efficient, Segmental baffles, Helical baffles.*

1. Introduction

Heat exchangers are one of the most used equipment in process industries. Heat exchanger is a device specially designed for the efficient transformation of heat from one fluid to another fluid over solid surface. In Heat exchangers the fluid temperature changes as it passes through the exchangers, and thus the temperature of dividing wall between the fluids also changes along with the length of exchanger. In this particular project we are studying a Shell n Tube HEX with Helical Baffles.

S. Lalot et al. [1]reffering the study of the gross flow mal-distribution of experimental electrical heater, showed it is seen that the effect of flow nonuniformity on the performance of heat exchangers.It is shown that it is very more important to understand mal-distributions for electrical heaters compared to two-fluid heat exchangers. The study of the flow distribution in a particular heater shows that reverse flows may occur for poor inlet header design.

Suggested is a simple approach to homogenize the Flow distribution and a simple law to calculate, with good accuracy, the velocity ratio (ratio of the highest velocity in the tubes to the lowest velocity). The original fluid distribution is applied to heat exchangers (condensers, counterflow and crossflow heat exchangers), and it is seen that gross flow mal-distribution leads to a loss of effectiveness of about 7% for condensers and counterflow heat exchangers, and up to 25% for crossflow exchangers, for velocity ratios up to 15.

Wang Shuli [2] the shell-and-tube heat exchangers flow field with helical baffles was measured by laser Doppler anemometry (LDA). This influenced on velocity distribution, impulsive velocity by helix inclination angle, and flow rate was observed. Six distinct inclination angles were designed in double-helix style: 30±, 35±, 40±, 42±, 45±, and 50±.

In general, it was found that the optimum helix angle inclination depends on the Reynolds number of the working fluid on the shell side of heat exchanger. Ender Ozden, Ilker Tari [3] the shell side design of a shell-and-tube heat exchanger; for a particular baffle spacing, baffle cut and shell diameter dependencies of the HT coefficient and the pressure drop are investigated by numerically modeling a small heat exchanger.

The flow and temperature fields inside the shell are solved by using a commercial CFD package. A set of CFD simulations is performed for a one shell pass and one tube pass heat exchanger with a variable number of baffles and turbulent flow. The results are viewed as to be sensitive to the model which is turbulent.

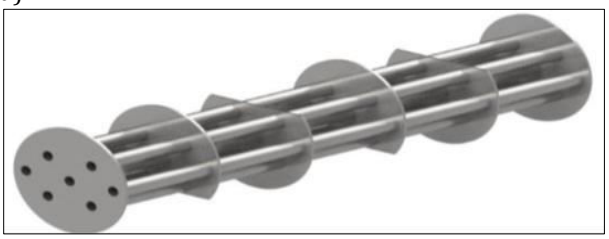
The best turbulence model among the ones considered is determined by comparing the CFD results of heat transfer coefficient, the pressure drop, the outlet temperature and the pressure drop with the Bell–Delaware method results. For two different baffle cut values, the effect of the baffle spacing to shell diameter ratio on the heat exchanger performance is investigated by changing the flow rate.

Zhang Zhnegguo et al. [4] the Heat transfer and pressure drop of helically baffled heat exchanger combined with petal-shaped finned tubes for oil (ISO VG-32) cooling with water as coolant was experimentally studied.

The preliminary heat transfer enhancement mechanisms were discussed for water flow helical In this work, the commercial software star CCM is used to conduct the numerical study for the similar STHX with

two different baffle types, to evaluate their performance. First the STHX parts such as tubes, shell, baffles and nozzles are modeled for two cases with segmental and helical baffles.

a)



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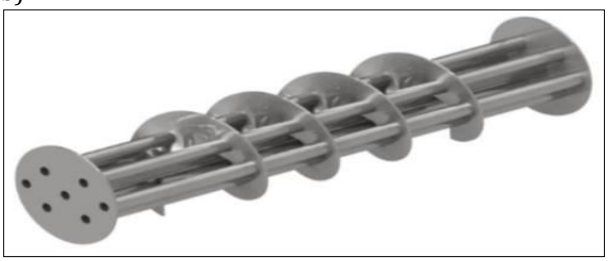


Fig1. Model with different baffles type: a) Segmental baffles b) Helical baffles [7].

2.Objectives

- We increase the number of helical coil for 500mm heat exchanger.
- We do thermal analysis of new designed baffles and check the results.
- We do CFD analysis of changed new helical baffles.
- We compare the performance of helical baffle heat exchanger with segmental baffle heat exchanger

3.Methodology

- Firstly we perform experiment and validate the analytical and experimental results.
- Then we increase the number of helical coil for one meter heat exchanger and change the helical pitch angle of helical coil and do the CFD analysis of that new heat exchanger with new design we find out the optimum performance of the heat exchanger from the analysis and manufacture the Heat Exchanger.
- Finally we find out the optimum performance of the heat exchanger from the analysis of the Heat Exchanger for a particular Pitch and Helix angle.

4. Design

2.1 Parameters considered:-

Table 1 Experimental procedure Dependent Parameters

$T_{h,o}$	Hot water outlet temperature
$T_{c,o}$	Cold water outlet temperature

Table 2 Experimental procedure Independent Parameters

$T_{h,i}$	Hot water outlet temperature
$T_{c,i}$	Cold water outlet temperature
C_h	Heat capacity of hot fluid
C_o	Heat capacity of hot fluid
U	Overall heat transfer coefficient
A	Heat transfer area
Flow Arrangement	Flow direction of the two fluids

2.2 Preliminary Estimation of Unit Size:
Selected shell n tube heat exchanger should satisfy the process requirements with allowable pressure drops until the next scheduled cleaning of the plant. The basic logical structure of Shell n tube heat exchanger is shown in Fig. 2
The size of heat exchanger can be obtained from Eq. 1:

$$A_o = \frac{Q}{U_o \Delta T_{lm,cf}} \quad (1)$$

where A_O is Outside heat transfer surface area based on outside diameter of tube, Q is heat duty of the exchanger and U_O is the overall Heat Transfer coef
For a purely counter-current heat exchanger correction factor F= 1.00. For a preliminary design shell with even number of tube side passes, F may be estimated as 0.9. Heat load can be estimated from the heat balance as

$$Q=(\dot{m}c_p)\,(T_2-T_1) \quad (2)$$

where $\dot{m}$ is the mass flow rate of water T_1 and T_2 are inlet and outlet temperatures of water.
We need to calculate the LMTD from the counter-current flow from the fourth given inlet/outlet temperatures. If three temperatures are known, the fourth one can be found from heat balance :

$$\Delta T_{lm,cf} = \frac{(T_{h1}-T_{h2})-(T_{h2}-T_{c1})}{ln\frac{T_{h1}-T_{c2}}{T_{h1}-T_{c1}}} \quad (3)$$

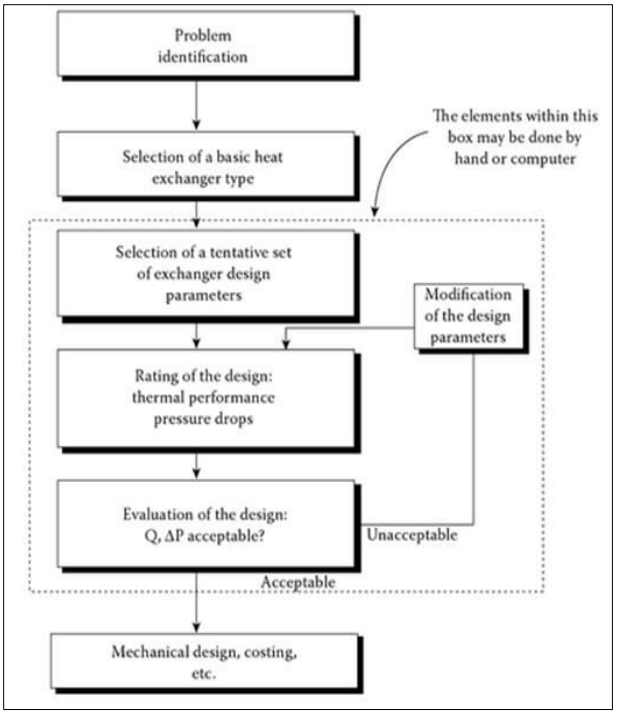


Fig.2. Basic logic structure for process heat exchanger design [3]

The problem now is to convert the area calculated from Eq. (1) into reasonable dimensions of tube diameter d_o and shell diameter, D_s,which would contain the right number of tubes, N_t with given tube length, L.

$$A_0 = \pi d_o N_t L \qquad (4)$$

The total number of tubes, N_t, can be predicted in fair approximation as a function of shell diameter by taking the shell circle and dividing it by projected area of tube layout pertaining to single tube A_1:

$$N_t = (CTP)\frac{\pi {D_s}^2}{4A_1} \qquad (5)$$

where CTP is the tube count calculation constant which accounts for the incomplete coverage of the shell diameter by the tubes due to necessary clearances between the shell and the outer tube circle and tube omissions due to tube pass lanes for multiple pass design.

Based on a fixed tube sheet, the following values are suggested:

one tube pass: CTP = 0.93
two tube passes: CTP = 0.9
three tube passes: CTP = 0.85
A_1= (CL)${P_T}^2$
where, CL is the tube layout constant:
CL= 1.0 for 90° and 45°
CL= 0.87 for 30° and 60°

$$N_t = 0.785\,\frac{CTP}{CL}\frac{{D_s}^2}{(PR)^2{d_o}^2} \qquad (6)$$

where PR is the tube pitch ratio (P_T/d_o) .
Substituting N_t from equation (4) and (6), an expression for the shell diameter in terms of main constructional diameters can be obtained as:

$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left(\frac{A_0}{L}PR^2 d_o\right)^{\frac{1}{2}} \qquad (7)$$

2.2 Prototype Design:-
Design a BEM shell n tube heat exchanger to maintain the temperature of a swimming pool at 30°C flowing through the shell side having inlet temperature 17°C by using hot water coming from boiler flowing at 82°C flowing through the low carbon steel tube having Tube OD of ¾ inch (OD=19mm ID=16mm inch). Tubes are laid in 30° Triangular Pitch with a pith ratio of 1.33 .The tube length of 0.5m is required because of space limitations. The maximum velocity of tube side fluid is 1.5 m/s to avoid fouling. The Reynolds Number on Shell side should be more than 10000 to keep the flow on shell side to be turbulent. The baffle spacing is approximated to be in between $0.4D_s$ and 0.6 $D_s$permissible maximum pressure drop on the shell side is 5 psi. Compare the performance of this shell n tube heat exchanger with segmental baffles with that of helical baffles.

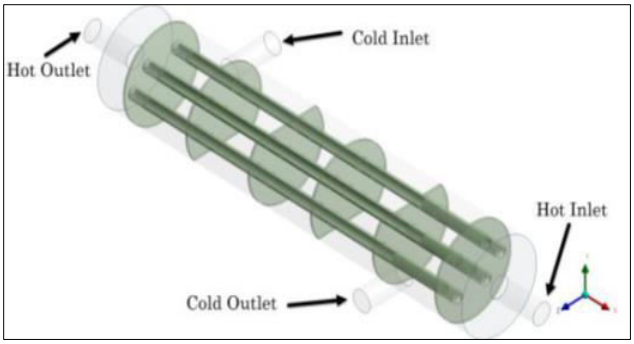
$\dot{m}_{tube}$ =1.5* π * ${d_i}^2/4$
= 1.5*3.1428*$0.016^2/4$
= 1.203 kg/sec.

Q =($\dot{m}c_p$) (T_2-T_1)
= 1.203*4181.816*(30-17)
=1.203*4181.816*13
=66884.3 W
=66.8843 kW

Applying the design formulae mentioned in section 4, the preliminary estimation of unit size is:
Shell Diameter D_s =0.18m
Tube length L = 0.5m
Tube diameter OD = 19mm, ID = 16mm
Number of tubes N_t= 29
Baffle spacing B = 0.064m, baffle cut 25%
Pitch ratio P_T/d_o = 1.33, 30° triangular pitch .

5. Physical Models
The configurations of the STHX with segmental and helical baffles are shown respectively in Fig.3. Another one point that need to be pondered is that baffle spacing of the two modelled heat exchangers is kept the same, which ensures that all of the values of geometry parameters are consistent except the baffle type. It could be considered that under the same conditions, comparisons between the two different baffles type are more convincing. The working fluid in both the shell and tube side of the heat exchanger is water

a)



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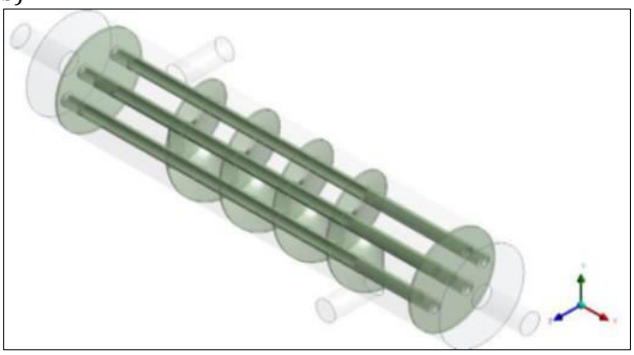

Fig.3. Isometric view representing computational domains for STHXs with: a) Segmental baffles b) Helical baffles [7]

6. Potential of Helical baffles to enhance the performance of shell n tube heat exchanger

The design of a shell-and-tube heat exchanger is a compromise between higher heat transfer coefficients and lower pressure drop in the fluids, since the two parameters are highly dependent on each other.

In order to improve the heat transfer coefficient, it is necessary to increase the fluid velocities. This always produces an increase in frictional pressure drops. Heat transfer coefficient per unit pressure drop at the shell side h/Δp, is adopted to evaluate the optimal ranges for both parameters. In practical application there is a limit to pressure drop of the heat exchangers, the aim in this project is to find the design parameter combination that results in the highest heat transfer coefficient within the pressure drop limitations. Thus, the ratio h/Δp should be a more reasonable comparison quantity.

On average, the performance ratio h/Δp for STHX with helical baffles is 39% higher on average than STHXs with segmental baffles [6]. Within the tested range of volume flow rate in the present study, the h/Δpof the STHXsHB with 30° helix angle grows up by 35–45% over that with 20° helix angle, while the h/Δp of the STHXsHB with 40° helix angle is 100–105% higher than that with 30° helix angle [7].

7. CFD Analysis:

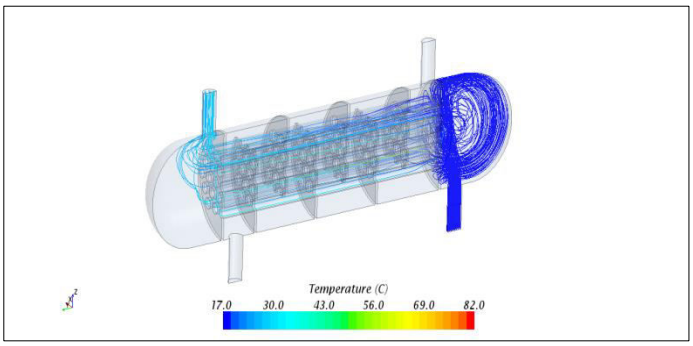

Fig. 4. Stream lines for cold fluid for Segmental baffle Heat Exchanger

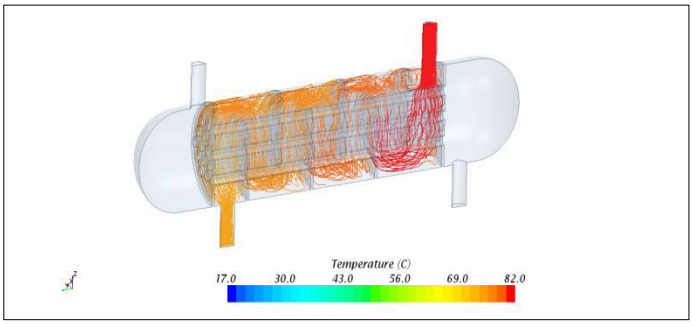

Fig.5. Stream lines for cold fluid for Segmental baffle Heat Exchanger.

8.Results

The results in this paper are compared for numerical calculation and CFD analysis for the designed heat exchanger in order to validate the CFD software for the further analysis of Shell n Tube Heat Exchanger with Helical Baffles. The Results are tabulated below

Table 3 Comparison of calculated and CFD results for segmental baffle shell n tube heat exchanger

Sr.No.	Parameter	By Calculation	By CFD Analysis
1.	$T_{c,o}$	30°C	27°C
2.	$T_{h,o}$	74°C	75°C
3.	Q	69.8 kW	62.04 kW
4.	Δp	9189 Pa	8430 Pa

Conclusions

After designing the heat exchanger for the given application and carrying out CFD analysis for shell n tube heat exchanger with segmental baffles and comparing both results we observe that the calculated result and the CFD results have maximum deviation of 10%.Thus, it could be concluded that present numerical model give good prediction for heat transfer characteristics.

In further study we are going to analyze the continuous helical baffle heat exchanger for 3 helix angles;

1. 0° helix angle.
2. 10° helix angle.
3. 20° helix angle.

After carrying out simulations for the above 3 cases of continuous helical baffle heat exchanger, we compare

the results with designed segmental baffle shell n tube heat exchanger and plot the results.

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