CFD Analysis to Study the Effects of Inclined Baffles on Fluid Flow in a Shell and Tube Heat Exchanger

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Abstract- In this paper, an effort has made for Computational Fluid Dynamic (CFD) analysis of a single pass parallel flow Shell and Tube Heat Exchanger (STHX) with different baffle inclinations. STHX is the most common and widely used type of heat exchangers in oil refineries, condensers and other large chemical processing plants. To analyze the performance of STHX, hot fluid has made to flow through tubes and cold fluid is allowed to flow through the shell. The main objective is to design the STHX with segmental and helical baffles and to study the flow and temperature field inside the shell side. Also, attempts were made to investigate the effects and heat transfer characteristics of a STHX for three different baffle inclinations namely 0°, 10°, and 20° for a given baffle cut of 36%. The basic geometry of shell and tube heat exchanger has made by CATIA V5 and meshing has completed by using HYPER MESH (11.0). The flow and temperature fields inside the shell have studied using ANSYS-FLUENT (6.3). The heat exchanger contains 7 tubes with 600 mm length and shell diameter 90 mm. The study indicated that flow pattern in the shell side of the heat exchanger with continuous helical baffle was forced to rotate the fluid, which results in significant increase in heat transfer rate and heat transfer coefficient per unit pressure drop than segmental baffle STHX. From the CFD simulation results, the shell side outlet temperature, pressure drop, optimum baffle inclination and optimal mass flow rate were determined.

Index Terms- STHX, segmental and helical baffles, Helixchanger, CFD, heat transfer rate, pressure drop, baffle inclination angle…etc.

1. INTRODUCTION

Heat exchangers are used to exchange heat between two process streams. Typically one fluid is cooled while the other is heated. Process fluids, usually are heated or cooled before the process or undergo a phase change. Heat exchangers are used in cooling, heating, condensation, boiling or evaporation purposes. They are widely used in petroleum refineries, chemical plants, petrochemical plants, natural gas processing, air-conditioning, refrigeration, and automotive applications. One common example of a heat exchanger is the radiator in a car, in which it transfers heat from the water (hot engine-cooling fluid) in the radiator to the air passing through the radiator [1] [2] [3].

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, and
- Indirect contact heat exchanger, where both media are separated by a wall through which heat is transferred so that they never mix.

1.1 History of the study

In addition to the basic need for transferring heat there are certain additional requirements which tend to be specific to the industry, in which they are employed. For example, the exchanger used in automotive and aviation industry need to be lightweight. These exchangers as well as those used in commercial and domestic refrigeration tend to use the same types of fluid in many applications. The exchangers used in chemical process industry tend to be used for a wide variety of fluid types with different degree of cleanliness. In contrast, the exchangers used in cryogenic applications invariable handle relatively clean fluids. These and other similar industry specific requirements have resulted in development of different types of exchanger ranging from the conventional shell and tube heat exchanger to other tubular and non-tubular exchangers of varying degree of compactness [3].

Shell and tube heat exchanger (STHX) is a type of heat exchanger used for efficient heat transfer from one fluid to other fluid that consist of series of finned tubes in which one of the fluid runs in the tube and the other fluid run over the tube to be heated or cooled during the heat exchange process [4]. Over the years, significant research and development efforts
are devoted to better understand the shell side geometry. A variety of different strategies are available to process and equipment designers to improve industrial heat transfer [3].

Shell and Tube heat exchanger has larger surface area with a small shape. This type of heat exchanger is good mechanical layout and good for pressurized operation. Shell and tube heat exchanger is easy to clean. The shell and tube heat exchanger is made up of different type of materials in which selected materials is used for operating pressure and temperature [4].

The major problem of shell and tube heat exchanger that it causes a larger pressure drop and results in dead zone between adjacent baffles. It leads to an increase of fouling resistance and the dramatic zigzag flow causes a high risk of vibration failure on tube bundle [4].

1.2 Desirable features of heat exchangers

To obtain maximum heat exchanger performance at the lowest possible operating and capital costs without comprising the reliability, the following features are required of an Exchanger:

(1) Higher heat transfer coefficient and larger heat transfer area.
(2) Lower pressure drop.

2. SELECTION FOR SHELL AND TUBE HEAT EXCHANGERS (STHXS)

More than 35-40 % of heat exchangers are of shell and tube type due to their robust geometrical construction, easy maintenance and possible upgrades. Rugged safe construction, availability in a wide range of materials, mechanical reliability in service, availability of standards for specifications and designs, long collective operating experience and familiarity with the designs are some of the reasons for its wide usage in industry. Fig. 1 shows a STHX.

3. HELIXCHANGER AND SEGMENTAL HEAT EXCHANGER

Baffle is an important shell side component of STHXs. It supports the tube bundles, and form flow passages for the shell side fluid in conjunction with the shell. The most commonly used baffle is the segmental baffle, which forces the fluid in a zigzag manner, thus improving the heat transfer but with a large pressure drop penalty. This type of heat exchanger has been well developed and probably is still the most commonly used type of the shell and tube heat exchanger.

The major drawbacks of the conventional shell and tube heat exchangers with segmental baffles are threefold: firstly it causes a large side pressure drop; secondly it results in a dead zone in each component between two adjacent segmental baffles, leading to an increase of fouling resistance; thirdly the dramatic zigzag flow pattern also causes high risk of vibration failure on tube bundle. [9].

A new type of baffle, called helical baffle, provides further improvement. This type of baffle was first proposed by D.Kral and J. Nemcansky, (1993) [12], where they investigated the flow field patterns produced by such helical baffle geometry with different helix angles. They found that these flow patterns were much close to plug flow conditions, which was expected to reduce shell side pressure drop and to improve heat transfer performance. Fig. 2 shows an isometric view of baffles and tubes in STHX with 20° baffle inclination.

3.1 Segmental-baffle shell and tube heat exchanger

Baffle spacing (B) is the center line distance between two adjacent baffles, Baffle is provided with a cut (Bc) which is expressed as the percentage of the segment height to shell inside diameter. Baffle cut can vary between 15% and 45% of the shell inside diameter. In the present study 36% baffle cut (Bc) is used.

3.2 Helical-baffle shell and tube heat exchanger

The effectiveness and cost are two important parameters in heat exchanger design. So, in order to improve the thermal performance at a reasonable cost of the Shell and tube heat exchanger, baffles in the present study are provided with some inclination in order to maintain a reasonable pressure drop in heat exchanger [1].
The concept of helical baffle heat exchangers was developed for the first time in Czechoslovakia. The Helical baffle heat exchanger, also known as ‘Helixchanger’, is a superior shell and tube exchanger solution that removes many of the inherent deficiencies of conventional segmental baffle exchangers. Helical baffle heat exchangers have shown very effective performance especially for the cases in which the heat transfer coefficient in shell side is controlled; or less pressure drop and less fouling are expected [3].

The Helixchanger design provides:
- Enhanced (Heat transfer performance/ Shell-side pressure drop) ratio.
- Reduced fouling characteristics.
- Effective protection from flow-induced tube vibrations.
- It results in lower capital costs, reduced operating costs, lower maintenance costs and consequently, significant lower total life cycle costs.
- For existing plants, the Helixchanger design helps to increase the capacity while lowering maintenance cost, plot space and energy costs.

It is better to consider the Helixchanger option when investigating the following:
- Plant upgrade with replacement tube bundles.
- Capacity expansion with limited plot space.
- Reduce fouling problems and frequent downtime.

4. COMPUTATIONAL MODELING AND SIMULATION

Computational Fluid Dynamics (CFD) provides a qualitative (and sometimes even quantitative) prediction of fluid flows by means of:
- Mathematical modeling (partial differential equations)
- Numerical methods (discretization and solution techniques)
- Software tools (solvers, pre- and post-processing utilities) [10].

To simplify numerical simulation, following assumptions have made:
- The shell side fluid processes constant thermal properties.
- The fluid flow and heat transfer processes are turbulent and in steady state.
- The leak flows between tube and baffle and that between baffles and shell are neglected.
- The natural convection induced by the fluid density variation is neglected.
- The tube wall temperature kept constant in the whole shell side.
- The heat exchanger is well insulated hence the heat loss to the environment is totally neglected.

4.1 Geometry modeling

The model is designed according to TEMA (Tubular Exchanger Manufacturers Association) Standards Gaddis [13], using CATIA V5 software as shown in Fig.5. Design parameters and fixed geometric parameters have been taken similar to Ozden et al. [14], as indicated in Table 1.

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Description</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Heat exchanger length, L</td>
<td>mm</td>
<td>600</td>
</tr>
<tr>
<td>2</td>
<td>Shell inner diameter, Di</td>
<td>mm</td>
<td>90</td>
</tr>
<tr>
<td>3</td>
<td>Tube outer diameter, do</td>
<td>mm</td>
<td>20</td>
</tr>
<tr>
<td>4</td>
<td>Tube bundle geometry and pitch triangular</td>
<td>mm</td>
<td>30</td>
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</tbody>
</table>
4.2 Grid generation

Mesh generation is performed using HYPER MESH (11.0). The surfaces of the model are meshed using Triangular elements. The fluid volume is meshed using tetrahedral elements. Mesh size selected are fine with node number around 50,000 and element number around 2,000,000. In order to capture both the thermal and velocity boundary layers the entire model is discretized using 2D triangular mesh elements for all the surfaces and tetrahedral mesh elements for 3D volume fluid elements, which are accurate and involve less computation effort. The entire geometry is divided into 6 components, namely, inlet, outlet, tube, shell, baffle and fluid.

5. TURBULENCE MODELING

Turbulent flows have some characteristic properties which distinguish them from laminar flows. In this study, realizable k-epsilon (k-ε 2 eqn.) turbulence model has tried.

5.1 k-ε turbulence model

The k-epsilon (k-ε 2 eqn.) model has been implemented in most general purpose CFD codes and is considered the industry standard model. It has proven to be stable and numerically robust and has a well-established regime of predictive capability. For general purpose simulation, the model offers a good compromise in term of accuracy and robustness.

5.2 Realizable k - ε model

This model differs from the standard k - ε model in that it features a realizability constraint on the predicted stress tensor, thereby giving the name of realizable k - ε model. The difference comes from correction of the k-equation where the normal stress can become negative in the standard k - ε model for flows with large strain rate.

6. GOVERNING EQUATIONS

The governing equations of the flow are modified according to the conditions of the simulated case. Since the problem is assumed to be steady, time dependent parameters are dropped from the equations. The resulting equations are:

- Conservation of mass: \( \nabla . (\rho V_r) = 0 \).
- X-momentum: \( \nabla . (\rho u V_r) = - (\partial p/\partial x) + (\partial \tau_{xx}/\partial x) + (\partial \tau_{xy}/\partial y) + (\partial \tau_{xz}/\partial z) \).
- Y-momentum: \( \nabla . (\rho v V_r) = - (\partial p/\partial y) + (\partial \tau_{yx}/\partial x) + (\partial \tau_{yy}/\partial y) + (\partial \tau_{yz}/\partial z) \).
- Z-momentum: \( \nabla . (\rho w V_r) = - (\partial p/\partial z) + (\partial \tau_{zx}/\partial x) + (\partial \tau_{zy}/\partial y) + (\partial \tau_{zz}/\partial z) \).
- Energy: \( \nabla . (\rho e V_r) = - \rho \nabla V_r \cdot \nabla \cdot (k \nabla T) + q \varphi \).

In Eq. (5), \( \varphi \) is the dissipation function that can be calculated from: \( \varphi = \mu [2[(\partial u/\partial x)^2 + (\partial v/\partial y)^2 + (\partial w/\partial z)^2] + (\partial u/\partial y + \partial v/\partial x)^2 + (\partial u/\partial z + \partial w/\partial x)^2 + (\partial v/\partial z + \partial w/\partial y)^2 + \lambda \nabla \cdot V_r]^2 \).

7. BOUNDARY CONDITIONS

The working fluid in the shell side, shell inlet temperature, constant wall temperature assigned to the tube walls, gauge pressure assigned to the outlet nozzle, inlet velocity profile assumed, slip condition assigned to all surfaces, heat flux boundary condition assigned to the shell outer wall (excluding the baffle shell interfaces), assuming the shell is perfectly insulated, are used in the CFD analysis are given in the following table 2. [11][2].

<p>| | | |</p>
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<tr>
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<tbody>
<tr>
<td>5</td>
<td>Number of tubes, Nt</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>Number of baffles, Nb</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>Central baffle spacing, B</td>
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<tr>
<td>8</td>
<td>Baffle inclination angle, θ</td>
<td>degree</td>
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Fig.5. Discretized fluid domain for segmental-baffle STHX with uniform volume 3D tetrahedral mesh

Fig.6. Discretized solid domain for helical-baffle STHX with uniform surface 2D triangular mesh

The heat exchanger is discretized into solid and fluid domains to have better control over the number of nodes. The fluid mesh is made finer than solid mesh for simulating conjugate heat transfer phenomenon. The entire 3D fluid domains are as shown in Fig.5 and 2D surface meshing is shown in Fig. 6.
### Table 2. Boundary conditions used in CFD analysis

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<th>Sl.No.</th>
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<th>Condition/value</th>
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<td>Working fluid</td>
<td>Water</td>
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<tr>
<td>2</td>
<td>Shell inlet temperature</td>
<td>300K</td>
</tr>
<tr>
<td>3</td>
<td>Tube wall temperature</td>
<td>450K</td>
</tr>
<tr>
<td>4</td>
<td>Gauge pressure</td>
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</tr>
<tr>
<td>5</td>
<td>Inlet velocity profile</td>
<td>Uniform m/sec</td>
</tr>
<tr>
<td>6</td>
<td>Slip</td>
<td>No slip</td>
</tr>
<tr>
<td>7</td>
<td>Heat flux</td>
<td>Zero W/m²</td>
</tr>
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</table>

## 8. RESULTS AND DISCUSSION

Fig. 7 to 12 shows the temperature contours and Fig. 13 to 18 shows the pressure contours obtained for different baffle inclinations such as such as 0 degree, 10 degree and 20 degree for segmental as well as helical baffle shell and tube heat exchangers.

### 8.1 Variation of temperature

Fig. 7 to 12 shows the temperature contour plots for both segmental and helical baffle shell and tube heat exchangers across the cross section at different inclinations of baffle along the length of heat exchanger.

#### 8.1.1 For segmental baffle (m = 1kg/s)

![Temperature distribution for 0 degree baffle inclination](image1)

![Temperature distribution for 10 degree baffle inclination](image2)
Fig. 9. Temperature distribution for 20 degree baffle inclination

For Helical Baffle (m = 1kg/s)

Fig. 10. Temperature distribution for 0 degree baffle inclination

Fig. 11. Temperature distribution for 10 degree baffle inclination
5.3 Variation of Pressure

Fig.13 to 18 shows pressure distribution across the shell and tube heat exchanger. With the increase in Baffle inclination, pressure drop inside the shell decreases and then increases. The contours of static pressure are shown in Fig. 13 to 18 gives detail idea.

For Segmental Baffle (m = 1kg/s)

Fig.13. Pressure distribution across shell at 0 degree baffle inclination

Fig.14. Pressure distribution across shell at 10 degree baffle inclination
Fig. 15. Pressure distribution across shell at 20 degree baffle inclination for Helical Baffle ($m = 1\text{ kg/s}$)

Fig. 16. Pressure distribution across shell at 0 degree baffle inclination

Fig. 17. Pressure distribution across shell at 10 degree baffle inclination
Table 3. Overall calculations and comparison of segmental and helical baffle STHX

<table>
<thead>
<tr>
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<tbody>
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<td>0</td>
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Table 3. shows the overall calculations and comparison of CFD analysis results for both circular segmental-baffle shell and tube heat exchanger and helical-baffle shell and tube heat exchanger. The study focused on the variation of shell side pressure drop, heat transfer coefficient and heat transfer rate with respect to different mass flow rates and under different baffle orientations.

It has been seen that for segmental baffle shell and tube heat exchanger, shell side pressure drop decreased and then slightly increased with an increase in baffle inclination angle at a given mass flow rate. Corresponding variations in heat transfer coefficient and heat transfer rate were occurred. When the pressure drop decreased the heat transfer coefficient and rate of heat transfer were increased. But in the case of helical heat exchanger the value of shell side pressure drop increased continuously and thus heat transfer coefficient and heat transfer rate reduced with increased baffle inclinations at a given mass flow rate.

8.3 Effect of baffle inclination on outlet temperature and pressure drop

From the CFD simulation results, for fixed tube wall and shell inlet temperatures, shell side outlet temperature and pressure drop values for varying fluid flow rates are provided in table 3. In this work, mass flow rate values are changed from 0.5kg/s to 2.0kg/s., Table 3 shows that the corresponding values of shell outlet temperature are decreased slightly and the pressure drop, heat transfer coefficient and heat transfer rate values are increased. The Outlet temperature variation with baffle inclination angle also given in the Table 5, which shows that there is no
much variation in shell outlet temperature. The average outlet temperature varies in between 325K to 328K while calculating from constant shell inlet temperature 300K. Compared to segmental baffle shell and tube heat exchanger, helical baffle shell and tube heat exchanger has slight higher values of outlet temperatures.

8.3.1 Segmental baffle STHX

Fig.19 Effect of baffle inclination on outlet temperature

It is found that for segmental baffle STHX the shell outlet temperature decreases with increasing mass flow rates as expected even the variation is minimal as shown in Fig. 19. It is found that for three mass flow rates 0.5 kg/s, 1 kg/s, and 2 kg/s there is no much effect on outlet temperature of the shell even though the baffle inclination is increased from 0° to 20°.

Fig.20 Effect of baffle inclination on pressure drop

However the shell-side pressure drop is slightly decreased with increase in baffle inclination angle i. e., as the inclination angle is increased from 0° to 20°. For the given geometry the mass flow rate must be below 2 kg/s, if it is increased beyond 2 kg/s the pressure drop increases rapidly with little variation in outlet temperature as shown in Fig. 20. It can also be observed that shell and tube heat exchanger with 10° baffle inclination angle results in a reasonable pressure drop with maximum shell outlet temperature. Hence it can be concluded shell and tube heat exchanger with 10° baffle inclination angle results in better performance compared to 20° and 0° inclination angles. The values of heat transfer rate and pressure drop at 10° (m = 1.0kg/s) are 1, 01,166.33W and 308.32Pa respectively.

8.3.2 Helical baffle STHX

Fig.21 Effect of baffle inclination on outlet temperature

In helical baffle STHX, the shell outlet temperature first decreased and then increased with increase in baffle inclination for a constant mass flow rate as shown in fig.5.21. It is found that for three mass flow rates 0.5 kg/s, 1 kg/s, and 2 kg/s there is no much effect on outlet temperature of the shell as baffle inclination is increased from 0° to 20°, even though maximum value is at small mass flow rate.

Fig.22 Effect of baffle inclination on pressure drop

It is found that shell-side pressure drop is slightly increased with increase in baffle inclination angle i. e., as the inclination angle is increased from 0° to 20°. For the given geometry the mass flow rate must be below 2 kg/s, if it is increased beyond 2 kg/s the pressure drop increases rapidly with little variation in outlet temperature as shown in fig.5.22. So it can be concluded that in helical baffle shell and tube heat exchanger with 0° baffle inclination angle results in better performance compared to 10° and 0° inclination angles. The values of heat transfer rate and pressure drop at 0° (m = 1.0kg/s) are 1, 12,506 W and 295.29 Pa respectively.

8.4 Comparison of pressure drop and the rate of heat transfer for segmental and helical baffle STHXS (m = 2.0kg/s)

Fig.23 shows the comparison of shell side pressure drop in segmental and helical baffle shell and
tube heat exchanger for the mass flow rate of 2.0kg/s. It has been seen that for segmental baffle shell and tube heat exchanger, pressure drop decreased first and then increased. But in the case of helical baffle shell and tube heat exchanger the value of shell side pressure drop continuously increases. Thus the optimum value of shell side pressure drop for segmental baffle heat exchanger can be taken under 10 degree baffle angle. In the case of helical baffle heat exchanger pressure drop increases with increase in baffle angle, so we can take 0 degree as the optimum baffle inclination with lowest pressure drop value.

![Fig.23 Comparison of shell side pressure drop](image)

Fig.23 Comparison of shell side pressure drop

Fig.24 shows the comparison of heat transfer rate for segmental and helical baffle shell and tube heat exchangers under the mass flow rate of 2.0kg/s. For segmental baffle arrangement heat transfer rate gradually increases. But in helical baffle shell and tube heat exchanger the value of heat transfer rate is almost uniform, but slightly maximum at zero degree baffle orientation.

![Fig.24 Comparison of heat transfer rate](image)

Fig.24 Comparison of heat transfer rate

By comparing the fig.24 and fig.25 the optimum baffle inclination angle for segmental baffle shell and tube heat exchanger is 10°, which shows the minimum pressure drop with a better heat transfer value. In the case of helical baffle shell and tube heat exchanger the optimum baffle inclination angle can be taken as 0°, since above zero degree baffle inclination there is a rapid rise in shell side pressure drop with less variation in heat transfer rate.

9. CONCLUSION

Considering the heat transfer rate for 1.0kg/s mass flow rate in the case of segmental baffle heat exchanger under 10° baffle inclination is 101.1663kW. Then by taking the heat transfer rate for helical baffle shell and tube heat exchanger for the same mass flow rate (m= 1.0kg/s), under 0° baffle inclination is 112.506 kW. So from this study the better option for a shell and tube heat exchanger is a helical baffle at zero degree than a segmental baffle with 10 degree baffle inclination. So it can be summarized as follows:

- The shell side of a small shell and tube heat exchanger is modeled with sufficient detail to resolve the flow and temperature fields.
- For the given geometry the mass flow rate must be below 2.0kg/s, if it is increased beyond 2.0kg/s the pressure drop increases rapidly with little variation in outlet temperature.
- In segmental baffle STHX it is observed that 10° baffle inclination angle results in a reasonable pressure drop with maximum shell outlet temperature and higher heat transfer rate.
- It is observed that at 0° baffle inclination angle helical baffle STHX results in better performance compared to 10° and 0° inclination angles, i.e., with less pressure drop, maximum shell outlet temperature and higher heat transfer rate.
- By comparing both segmental and helical baffle STHXs, the helical baffle with 0° inclination results in better performance than segmental baffle with 10° baffle inclination, i.e., minimum pressure drop with maximum heat transfer rate.

REFERENCES


