Research Journal of Engineering Sciences ISSN 2278 – 9472
Vol. 3(7), 8-16, July (2014) Res. J. Engineering Sci.

Design of Shell and Tube Heat Exchanger Using Computational Fluid Dynamics Tools

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Available online at: www.isca.in, www.isca.me
Received 29th April 2014, revised 9th June 2014, accepted 11th July 2014

Abstract

When the helix angle was varied from 0^0 to 20^0 for the heat exchanger containing 7 tubes of outer diameter 20 mm and a 600 mm long shell of inner diameter 90 mm, the simulation shows how the pressure vary in shell due to different helix angle and flow rate. The heat transfer coefficient when recorded showed a very high value when the pressure inside the heat exchanger registered a decline value and this incremental hike was found to be highly significant in the present study. This might be due to the rotational and helical nature of flow pattern following the geometry change by the introduction of continuous helical baffles in the shell side of the heat exchanger. The simulation results obtained with Computational fluid dynamics tools for the baffle cut given to the modified heat exchanger are utilized for the calculation of various parameters like the pressure decline, desired baffle inclination angle and mass flow rate, outlet temperature at the shell side and recirculation at baffle side for the particular geometry of the heat exchanger. Small corners at variable angles of the liquid flow are the result of introduction of segmental baffles which improves heat transfer and huge decline in pressure thus increasing the fouling resistance. This recorded an effective heat transfer hike by the impact of helical baffle. The most desirable heat transfer coefficient of the highest order and pressure decline of the lowest order are the result generated in heat exchanger. Thus, the present study conclusively improved the performance of the heat exchanger by the use of helical baffle in place of segmental baffle from the numerical experimentation results.

Keywords: Shell and tube heat exchanger, computational fluid dynamics tools**,** helix Baffles, Heat Transfer Coefficient, Pressure Decline.

Introduction

Highest thermal performance is the key factor determining the efficiency of any Shell and Tube Heat Exchanger. Hence, in the design of Heat Exchanger, the foremost considerations should be given to the cost and effectiveness so as to improve its efficiency. The baffles of helical type instead of segmental type when used in conjunction with changes in its geometry by giving a full 360^0 CFD model inclination, a higher decline in pressure across the heat exchanger could be achieved using Ansys tools. The temperature distribution and flow structure are favorably obtained upon fine modelling the geometry accurately. The present study is taken up with the major objective of obtaining pressure decline at shell side and consequent achievement of hike in heat transfer by design and simulation of angle of helical baffles.

Lutcha and Nemcansky[1] upon investigation of the flow field patterns generated by various helix angles used in helical baffle geometry found that the flow patterns obtained in their study are similar to plug flow condition which is expected to decline pressure at shell side and increase heat transfer process significantly. Stehlik et al.[2] studied the effect of optimized segmental baffles and helical baffles in heat exchanger based on Bell-Delaware method and demonstrated the heat transfer and pressure decline correction factors for a heat exchanger. Oil-Water Shell and Tube Heat Exchangers with various baffle geometries of 5 continuous helical baffles and one segmental baffle and test results were compared for performance with respect to their heat transfer coefficient and pressure decline values at shell side by Kral et.al.[3]. When they have made comprehensive comparison on the most important geometric factor of helix angle, 40^0 helix angle outperformed the other angles with respect to the heat transfer per unit shell side fluid pumping power or unit shell side fluid pressure decline. The flow patterns in the shell side of the shell and tube heat exchanger with continuous helical baffles were found always rotational and helical due to the peculiar geometry of the continuous helical baffles which resulted in a significant increase in heat transfer coefficient per unit pressure decline in the shell and tube heat exchanger.

The continuous helical baffles when designed well can prevent the flow induced vibration and fouling in the shell side. Similar results on fouling were reported by Murugesan and Balasubramanian[4]. The experimental study of Shell and Tube Heat Exchangers with the use of continuous helical baffles can result approximately ten per cent hike in heat transfer coefficient in comparison with that of traditional segmental baffles for the same shell side pressure decline. Development of non-dimensional correlations for heat transfer coefficient and pressure decline based on the experimental data on proposed continuous helical baffle shell and tube heat exchangers with different shell configurations will be the another objective of the present study which might be useful for industrial applications and basic studies on this type of heat exchangers.

Material and Methods

The tube layout and arrangement with tube pitch and tube pitches parallel and normal to flow is typically shown for equilateral triangular arrangement in figure-1. A computational model of Shell and Tube Heat Exchanger with ten helix angle is shown in figure-2 after experimental test and the details of geometry parameters of the same are given in table 1. From figure- 2, it can be seen that the simulated Shell and Tube heat exchanger has total number of tubes seven and with six cycles of baffles in the shell side direction. The inlet and out let of the domain are connected with the corresponding tubes. The whole computation domain is bounded by the inner side of the shell and everything in the shell contained in the domain.

Following assumptions are made with respect to basic characteristics of the process to simplify numerical simulation: The shell side fluid is having constant thermal properties, The fluid flow and heat transfer processes are in steady state and turbulent, 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 and the heat exchanger is well insulated so that the heat loss to the environment is totally neglected. The following Navier Stokes Equation[5] solved in every mess shell and the simulation shows the result. Gaddis (2007)[6] method was used to design the model as per the Tubular Exchanger Manufacturers Association (TEMA) standards and the tube layout is provided in figure- 1.

$$\partial_t T+u\partial_x T+v\partial_y T=-\frac{1}{RePr}\left[\partial_x(K\partial_x T)+\partial_y\left(K\partial_y T\right)\right]$$

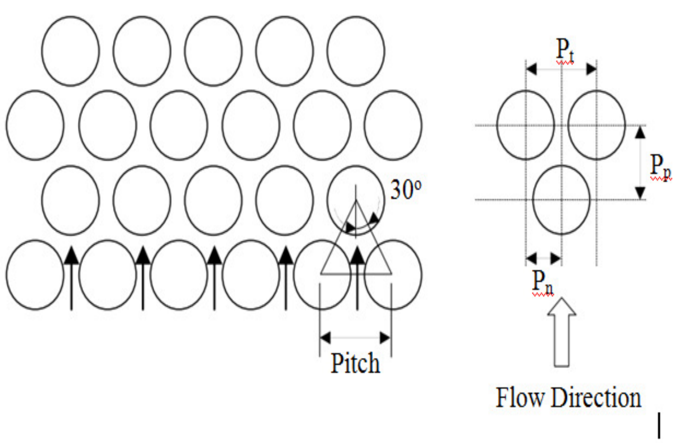

Figure-1
Tube layout and arrangement

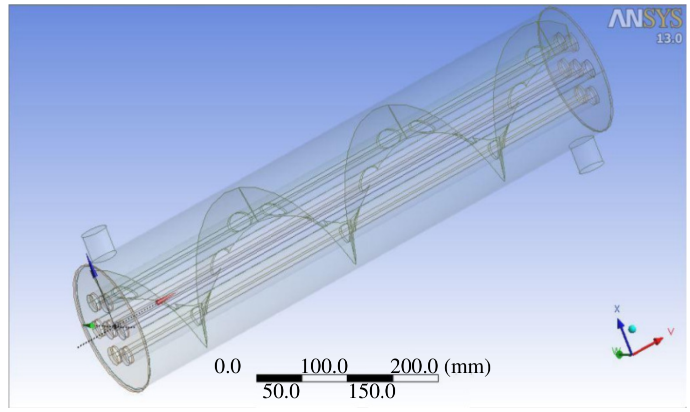

Figure-2
Computational model of Shell and tube heat exchanger with tubes and baffle inclination

Table-1
Geometric dimensions of shell and tube heat exchanger

Length of Heat exchanger, (L)	600 mm
Inner diameter of Shell (Di)	90 mm
Outer diameter of Tube (do)	20 mm
Tube bundle geometry and pitch (Triangular)	30 mm
Number of tubes (Nt)	7
Number of baffles (Nb)	6
Central baffle spacing B	86 mm
Baffle inclination angle (θ)	0 to 40

In order to get the correct velocity and thermal boundary layers with much less computation, the hexahedral mesh elements were utilized in discretizing the 3 D model in ICEM CFD. Fine control near the wall surface on hexahedral mesh was ensured so as to get the correct boundary layer gradient. In an effort to get correct number of nodes, best control was obtained by discretizing the shell and tube heat exchanger into solid and fluid domains. For six baffles of the entire geometry divisions, the fluid domains are distinguished as Fluid Inlet, Fluid Outlet and Fluid Shell and the solid domains are numbered from Solid Baffle 1 to Solid Baffle 6. Fluid mesh finer than solid mesh is used in the simulation of conjugate heat transfer process. In order to capture velocity and thermal boundary layers, a fluid domain which spans 100 microns is the height of the 1st cell from the tube surface. A minimum angle and minimum determinant of 18^0 and 4.12 respectively are found when the discretized model is checked for quality. The minimum required quality is met when the meshes are checked for errors and then it is exported to the pre-processor (ANSYS CFX).

Initially a relatively coarser mesh is generated with 1.8 Million cells. This mesh contains mixed cells (Tetra and Hexahedral cells) having both triangular and quadrilateral faces at the boundaries. Care is taken to use structured cells (Hexahedral) as much as possible, for this reason the geometry is divided into several parts for using automatic methods available in the ANSYS meshing client. It is meant to reduce numerical diffusion as much as possible by structuring the mesh in a well manner, particularly near the wall region. Later on, for the mesh independent model, a fine mesh is generated with 5.65 Million cells. For this fine mesh, the edges and regions of high temperature and pressure gradients are finely meshed. The details are provided in figure-3 and 4.

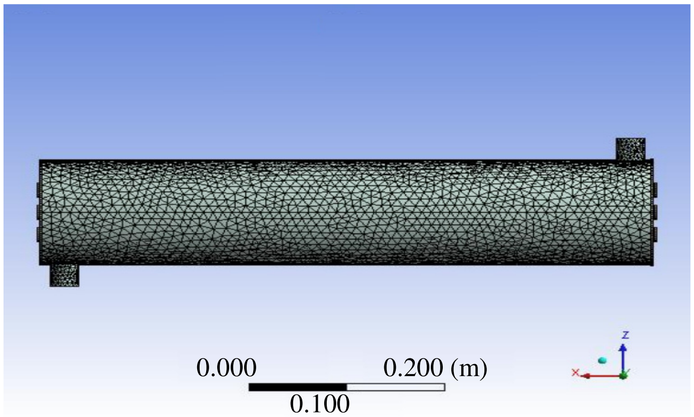

Figure-3
Meshing diagram of shell and tube heat exchanger

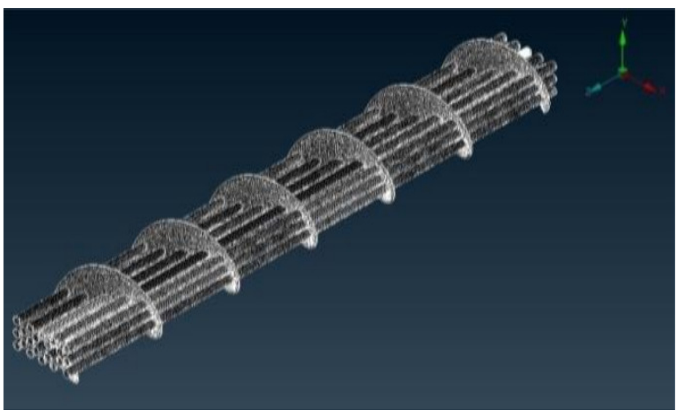

Figure-4
Surface mesh with Helical Baffle and tube bundle of shell and tube heat exchanger

ANSYS® FLUENT® v13 was used to carry out simulation. Steady time and absolute velocity formation were selected for the simulation after selecting Pressure Based type in the Fluent solver. Option energy calculation was on in the model, and the standard wall function was set as (k-epsilon 2 eqn.), the viscous was set as standard k-e. In cell zone fluid water-liquid was selected. Water-liquid and copper, aluminum was selected as materials for simulation. Boundary condition was selected for inlet, outlet. In inlet and outlet 1Kg/s velocity and temperature was set at 353K. Across each tube 0.05Kg/s velocity and 300K temperature was set. Mass flow was selected in each inlet. In reference Value Area set as $1m^2$, Density 998 Kg/m^3, enthalpy 229485 J/Kg, length 1m, temperature 353K, Velocity 1.44085 m/s, Ration of specific heat 1.4 was considered.

SIMPLEC was set as Pressure Velocity coupling and Skewness correction was set at zero. In Spatial Discretization zone, Least square cell based Gradient and standard Pressure were used. Momentum, Turbulent Kinetic Energy and Energy all were set as First order upwind. In Solution control, Pressure was 0.7, Density 1 , Body force 1, Momentum 0.2 , turbulent kinetic and turbulent dissipation rate was set at 1, energy and turbulent Viscosity was 1. Solution initialization was standard method and solution was initialized from inlet with 300K temperature. Under the Above boundary condition and solution initialize condition, simulation was set for 1000 iteration. For convergence of solution; the continuity, X-velocity, Y velocity, Z-velocity, k, epsilion should be less than 10^{-4} and the energy should be less than 10^{-7}. The heat transfer rate is calculated from the following formulae from which heat transfer rate is calculated across shell side. $Q = m * Cp * \Delta T$ Where m = mass flow rate, Cp = Speific Heat of Water, ΔT = Temperature difference between tube side

Results and Discussion

Baffle inclination: For Zero degree baffle inclination solution was converged at 160^{th} iteration. Simulation of 10^0 Baffle inclination is converged at 133^{th} iteration. Simulation of 20^0 baffle inclination is converged at 138^{th} iteration. Similar results were reported by Khairun et al.[7].

Variation of Temperature: Three different plots of temperature profile are taken in comparison with the baffle inclination at 0^0, 10^0, 20^0 and temperature distribution across tube outlet at 0^0 are given in figure-5, 6, 7 and 8.

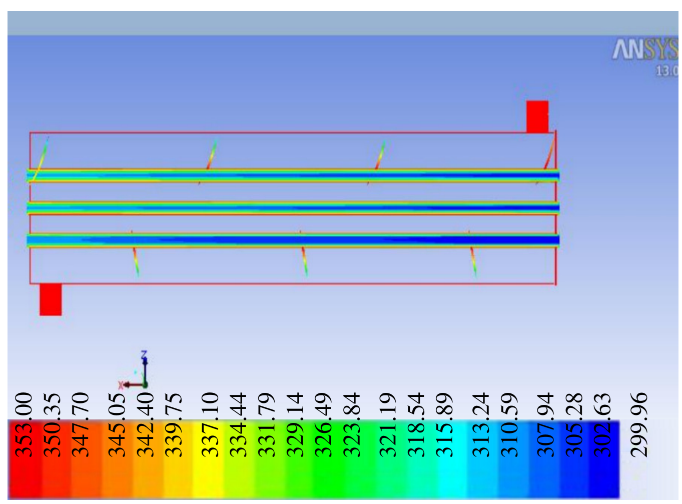

Figure-5
Temperature Distribution across the tube and shell

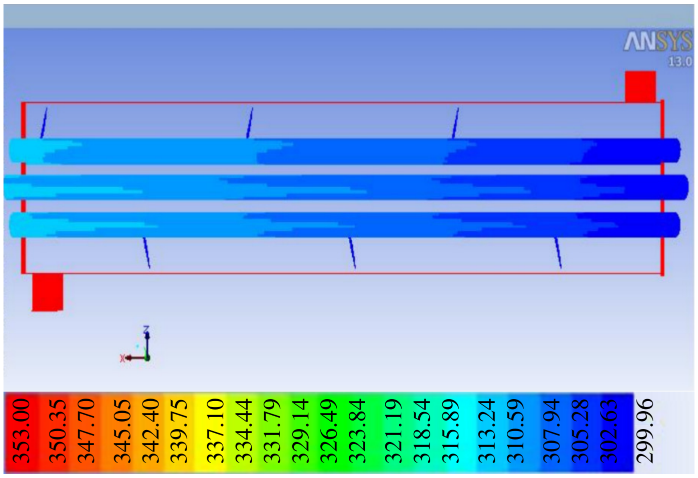

Figure-6
Temperature Distribution for 10^0 baffle inclination

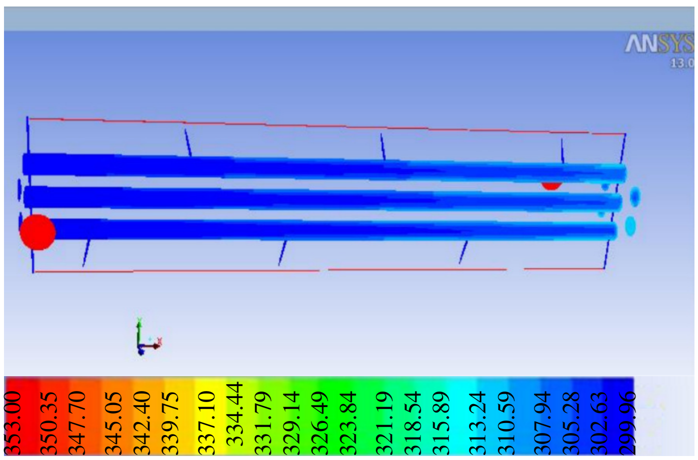

Figure-7
Temperature Distribution of 20^0 baffle inclination

Temperature of the hot water in shell and tube heat exchanger at inlet was 353K and in outlet it became 347K. In case of cold water, inlet temperature was 300K and the outlet became 313K. Similar results were reported by Usman and Goteberg, 2011[8]. Tube outlet Temperature Distribution was given below in exchanger (figure-8). Similar enhanced heat transfers in heat exchangers were reported by Murugesan and Balasubramanian[9,10].

Variation of Velocity: The flow distribution across the cross section at different positions in heat exchanger is understood by the examination of Velocity profile. Velocity profile of Shell and Tube Heat exchanger at different inclinations of Baffle are shown in Figures 9, 10 and 11. The heat exchanger is modeled considering the plane symmetry. The velocity profile at inlet is same for all three inclination of baffle angle i.e 1.44086 m/s. Outlet velocity vary tube to helical baffle and turbulence occur in the shell region as reported earlier[11].

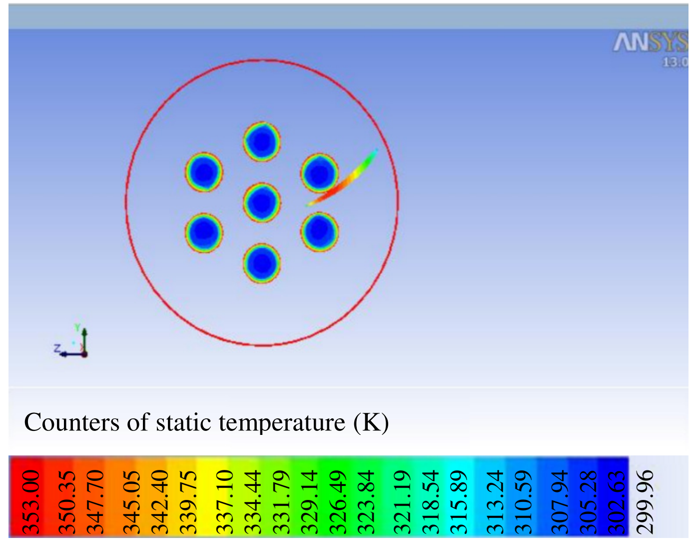

Figure-8
Temperature Distribution across Tube outlet in 0^0 baffle inclination

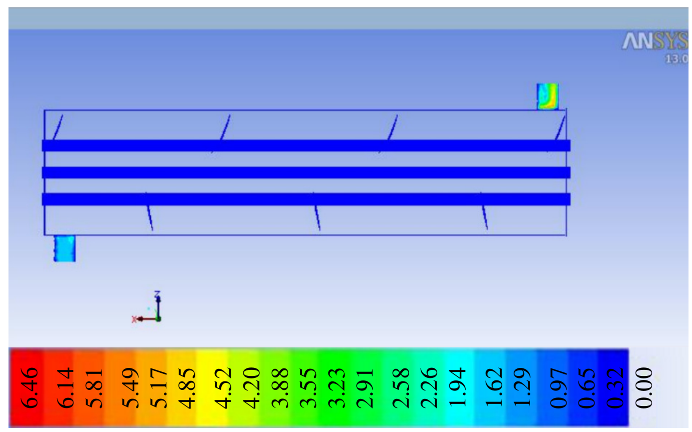

Figure-9
Velocity profile across the shell at 0^0 baffle inclination

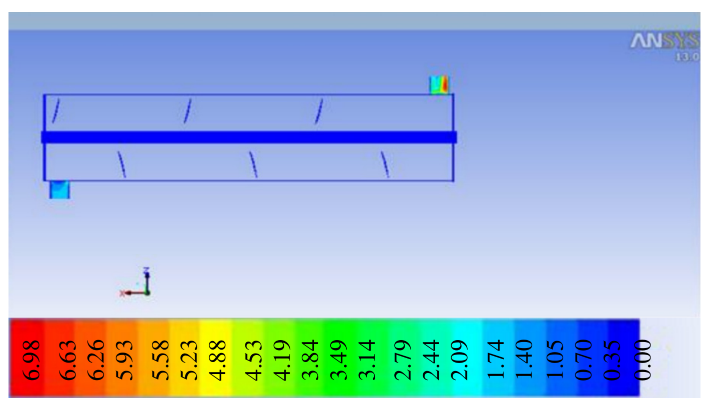

Figure-10
Velocity profile across the shell at 10^0 baffle inclination

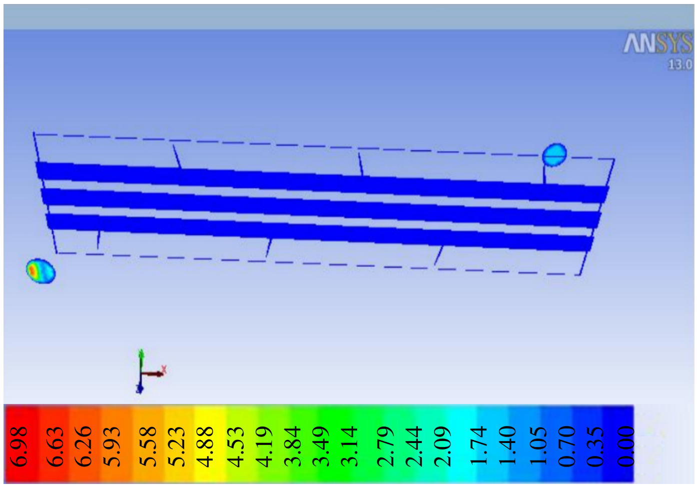

Figure-11
Velocity profile across the shell at 20^0 baffle inclination

Variation of Pressure: Figure-12, 13 and 14 shows the pressure distribution across shell and tube heat exchanger at different baffle inclinations. With every increase in baffle inclination angle, the pressure decline inside the shell is decreased and the pressure was found varying widely from inlet to outlet.

The contours of static pressure are shown in all the figures to give a detailed idea.

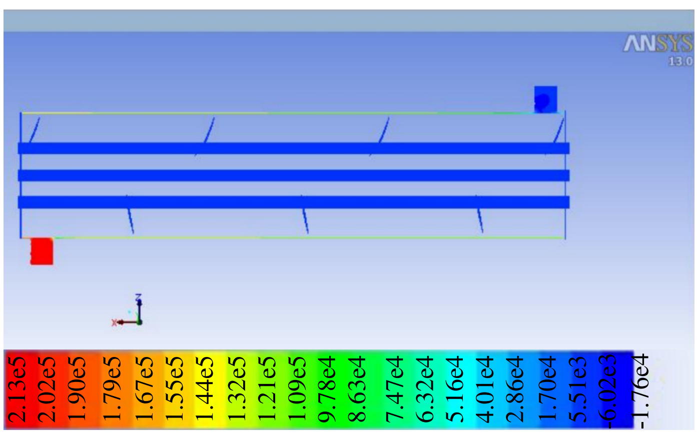

Figure-12
Pressure Distribution across the shell at 0^0 baffles inclination.

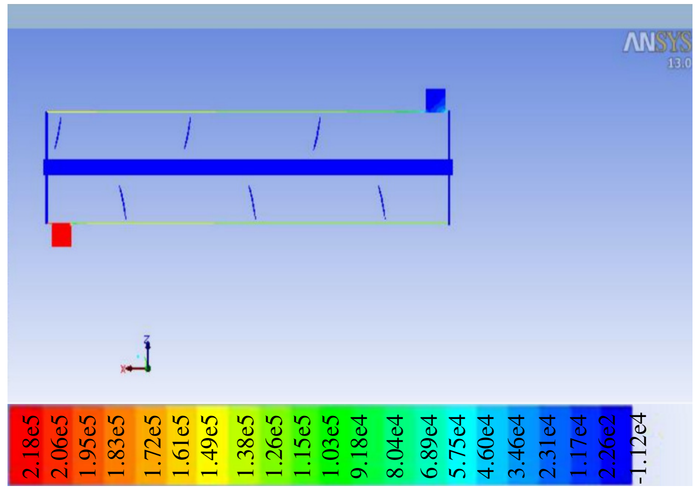

Figure-13
Pressure Distribution across the shell at 10^0 baffle inclination

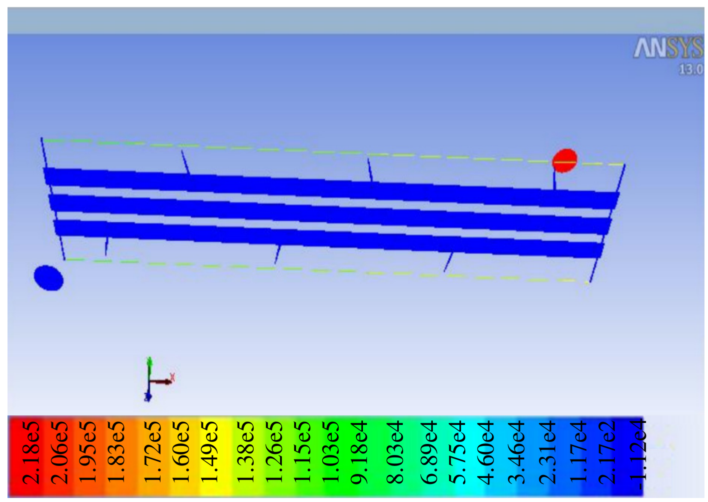

Figure-14
Pressure Distribution across the shell at 20^0 baffle inclination

Changes in outlet temperatures are shown in Table 2 and in Figure- 15.

Similar heat transfers are reported by Emerson[12] and Li and Kottek[13]

Table-2
Change in Outlet Temperature with respect to baffle inclination angle

Baffle Inclination Angle (Degree)	Outlet Temperature of Shell side (Kelvin)	Outlet Temperature of Tube side (Kelvin)
0	346	317
10	347.5	319
20	349	320

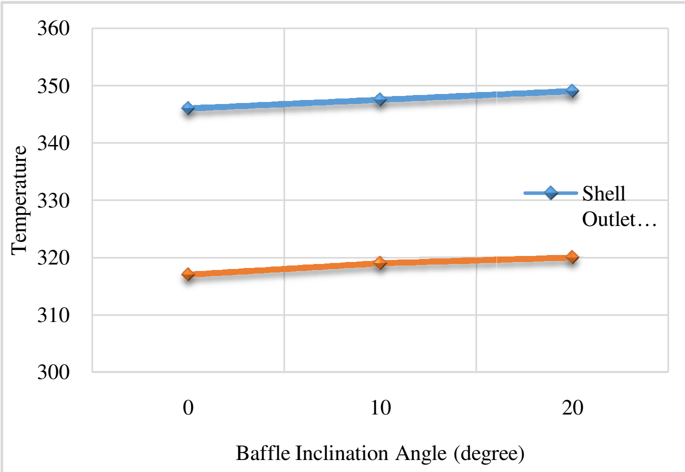

Figure-15
Plot of Baffle inclination angle vs Outlet Temperature of shell and tube side

It has been found that the increasing baffle inclination angle from 0^0 to 20^0 affect the outlet temperature of shell side significantly. Pressure Decline inside Shell with respect to baffle inclination angle is provided in table 3.

Similar results were reported by Diaper and Hesler, 1990[14], Sunil and Pancha[15] and Zhang et al., 2008[16] using Computational Fluid Dynamics software. Pressure Decline inside Shell with respect to baffle inclination angle is provided in figure-16. Velocity inside the shell with respect to baffle inclination angle from 0^0 to 20^0 is detailed in table 4 and figure-17.

The shell-side pressure declines when the baffle inclination angle is increased from 0^0 to 20^0. For shell and tube heat exchanger a 10^0 baffle inclination angle results in pressure drop of 4 per cent and 20^0 baffle inclination angle results in pressure drop of 16 per cent when compared to the pressure drop at 0^0 baffle inclination angle as shown in figure-16. So, it is generalized that with increase in baffle inclination, pressure decline decreases, so that it affect in heat transfer rate, which is increased.

Table-3
Pressure Decline inside Shell with respect to baffle inclination angle

Baffle Inclination Angle (Degree)	Pressure Decline Inside Shell (kPA)
0	230.992
10	229.015
20	228.943

Table-4
Velocity inside Shell with respect to baffle inclination angle

Baffle Inclination Angle (Degree)	Velocity inside shell (m/sec)
0	4.2
10	5.8
20	6.2

Similar performance resulting from baffle inclinations in heat exchangers are reported by Thirumarimurugan et al., 2008[17].

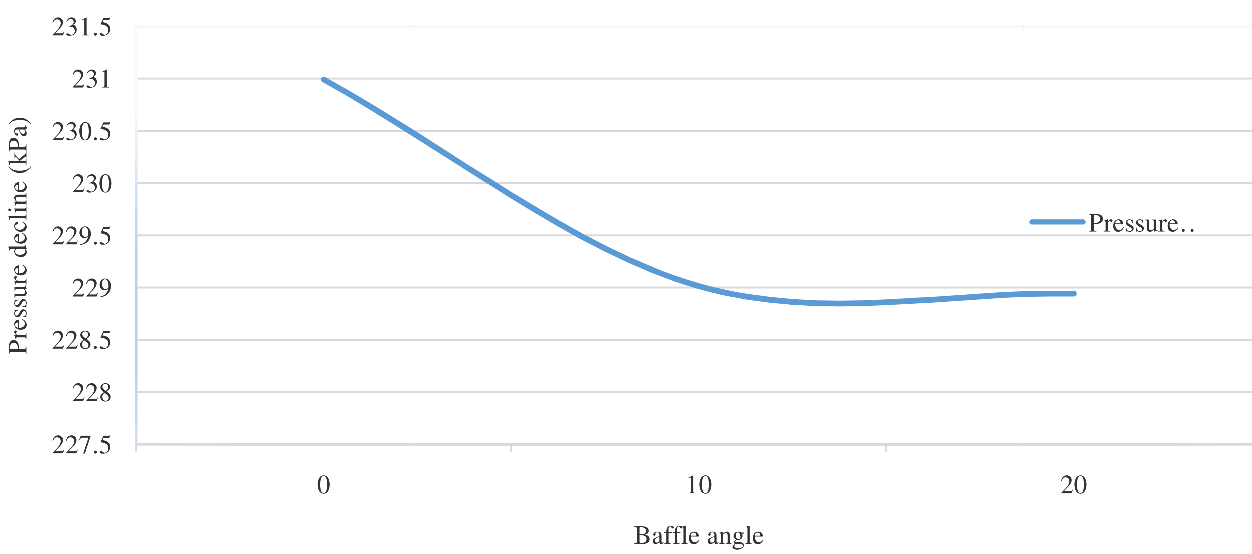

Figure-16
Plot of Baffle angle vs Pressure Decline

The outlet velocity is increasing with increase in baffle inclination so that more will be heat transfer rate with increasing velocity.

Heat Transfer Rate: The Plot showing that with increasing baffle inclination, heat transfer rate increase. For better heat transfer rate, helical baffle is used and results is shown in figure-18 and table 5. The overall values obtained in simulation are given in table 6.

Table-5
Heat Transfer Rate across Tube side with respect to baffle inclination angle

Baffle Inclination Angle (Degree)	Heat Transfer Rate Across Tube side (W/m^2)
0	3557.7
10	3972.9
20	4182

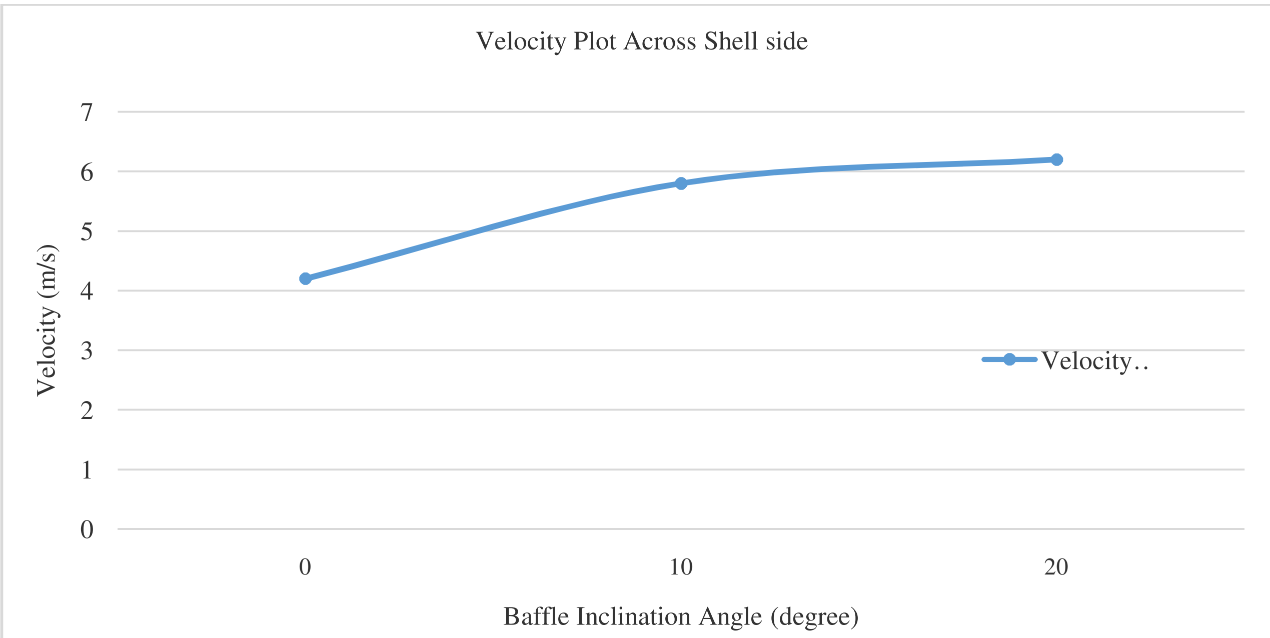

Figure-17
Plot of Velocity profile inside shell

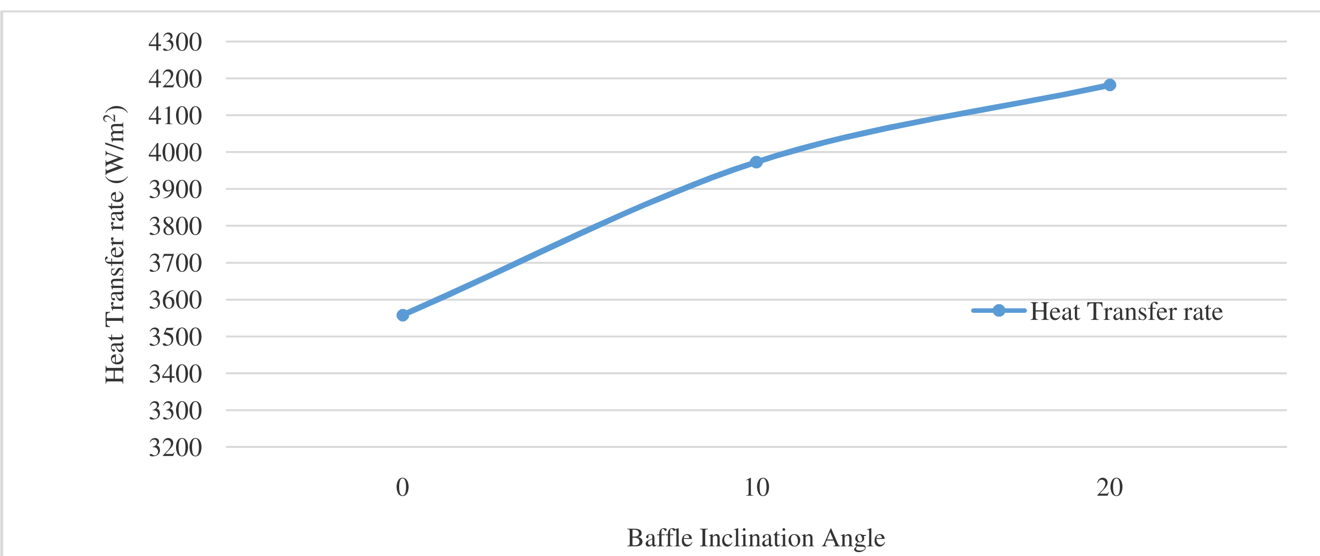

Figure-18
Heat Transfer Rate along Tube side

Table-6
Overall Calculated value in Shell and Tube heat exchanger in simulation

Baffle inclination (in Degree)	Shell Outlet Temperature (Kelvin)	Tube Outlet Temperature (Kelvin)	Pressure Drop	Heat Transfer Rate(Q) (in W/m^2)	Outlet Velocity(m/s)
0^0	346	317	230.992	3554.7	4.2
10^0	347.5	319	229.015	3972.9	5.8
20^0	349	320	228.943	4182	6.2

Conclusion

The following are the salient points emerged from this study.

The simulation was converged at 160th iteration for zero degree baffle inclination. Simulation of ten degree baffle inclination was converged at 133rd iteration. Simulation of twenty degree baffle inclination was converged at 138th iteration. Temperature of the cold water in shell and tube heat exchanger at the inlet was 300K and at the outlet was 313K. In case of hot water, inlet temperature was 353K and at the outlet became 347K. The velocity profile at outlet vary tube to helical baffle and turbulence occur in the shell region. The velocity profile at inlet was same for all the three inclination of baffle angle i.e 1.44086 m/s. The pressure decline inside the shell is decreased with the increase in baffle inclination angle. The pressure vary widely from inlet to outlet.

Outlet temperature of shell side was much affected while the baffle inclination angle was increased from 0^0 to 20^0. This was because of decrease in shell side pressure decline. The pressure decline was found to decrease by 4 per cent with 10^0 and 16 per cent with 20^0 baffle inclination. The outlet velocity also increases with increase in baffle inclination and cause a further increase in heat transfer.

The prediction of pressure decline and heat transfer of the model was found to be with an average error of 20 per cent. Rapid mixing and change in flow direction was observed in inlet and outlet region and found to be the only exception for the assumption in geometry and meshing. Reliable results was observed with the model by considering the standard k-e and wall function. It could also be seen that the mass flow rate when increased beyond 2kg/s; the pressure decline suddenly increases with practically nil variation in outlet temperature for the given geometry. The unsupported behavior of center row of tubes makes the baffle use ineffective when the baffle angle is above 20^0. Hence, the helix baffle inclination angle of 20^0 makes the best performance of shell and tube heat exchanger.

References

1. Lutcha J. and Nemcansky J., Performance improvement of tubular heat exchangers by helical baffles, *Chemical Engineering Research and Design,* **68**, 263-270 **(1990)**
2. Stehlík P., Nemcansky J., Kral D. and Swanson L.W., Comparison of correction factors for shell-and-tube heat exchangers with segmental or helical baffles, *Heat Transfer Engineering*, **15(1),** 55-65 **(1994)**
3. Kral D., Stehlik P., Ploeg V.D.H.J. and Master B.I., Helical baffles shell-and tube heat exchangers, 1: *Experimental verification, Heat Transfer Engg.,* **17(1),** 93–101 **(1996)**
4. Murugesan M.P. and Balasubramanian R., To Study the Fouling of Corrugated Plate Type Heat Exchanger in the Dairy Industry, *Res. J. Engineering Sci.,* **2(1),** 5-10 **(2013)**
5. Temam R. (Ed.), Claude-Louis Navier and Gabriel Stokes Navier-Stokes Equations: Theory and Numerical Analysis, Institut for Scientific Computing and Applied Mathematics, Indiana University, USA **(1977)**
6. Gaddis D. Standards of the Tubular Exchanger Manufacturers Association, TEMA Inc, 9th ed., Tarrytown, USA **(2007)**
7. Khairun H.O., CFD simulation of heat transfer in shell and tube heat exchanger, Bachelor in chemical Engineering (Gas Technology) Thesis **(2009)**
8. Usman U.E. and Goteborg, Heat Transfer Optimization of Shell-and-Tube & Heat Exchanger through CFD, Master's Thesis, Chalmers University of Technology, Sweden **(2011)**
9. Murugesan M.P. and Balasubramanian R., The Effect of Mass Flow Rate on the Enhanced Heat Transfer Characteristics in A Corrugated Plate Type Heat Exchanger, *Res. J. Engineering Sci.,* **1(6),** 22-26 **(2012)**
10. Murugesan M.P. and Balasubramanian R., The Experimental Study on Enhanced heat Transfer Performance in Plate Type Heat Exchanger, *Res. J. Engineering Sci.,* **2(2),** 16-22 **(2013)**
11. Haseler L.E., Wadeker V.V. and Clarke R.H., Flow Distribution Effect in a Plate and Frame Heat Exchanger, *I Chem E Symposium Series,* **129**, 361-367 **(1992)**
12. Emerson W.H. Shell-side pressure drop and heat transfer with turbulent flow in segmentally baffled shell-tube heat exchangers, *Int. J. Heat Mass Transfer,* **6**, 649–66 **(1963)**
13. Li H. and Kottek V., Effect of baffle spacing on pressure drop and local heat transfer in shell and tube heat exchangers for staggered tube arrangement, *Int. J. Heat Mass Transfer,* **41(10),** 1303–1311 **(1998)**

14. Diaper A.D. and Hesler L.E., Cross flow Pressure Drop and Flow Distributions within a Tube Bundle Using Computational Fluid Dynamic, Proc. Heat Transfer Conf., Israel, 235-240 **(1990)**

15. Sunil K.S. and Pancha M.H., Comparative thermal performance of shell and tube heat Exchanger with continuous helical baffle using different angles, *International Journal of Engineering Research and Applications,* **2(4),** 2264-2271 **(2012)**

16. Zhang J.F., He Y.L. and Tao W.Q., 3 D numerical simulation of shell and tube heat exchanger with middle-overlapped helical baffle, *Journal of School of energy and power engineering,* **132**, 1-8 **(2008)**

17. Thirumarimurugan M., Kannadasan T. and Ramasamy E., Performance Analysis of Shell and Tube Heat Exchanger Using Miscible System, *American Journal of Applied Sciences,* **5**, 548-552 **(2008)**