# Numerical Comparison of Shell-Side Performance for Shell and Tube Heat Exchangers with Novel, Helical and Segmental Baffles

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*Abstract*— Shell-and-tube heat exchanger, is one of the most widely used heat exchange apparatus in various industrial process and research fronts. Baffle selection is critical to control and improve the thermo-hydraulic performance of this type of heat exchanger. In this paper, three-dimensional computational fluid dynamics (CFD) simulations, using the ANSYS 17 FLUENT, have been performed to study and compare the shell-side flow distribution, heat transfer coefficient and the pressure drop between the recently developed novel, helical baffles and the conventional segmental baffles. In this numerical comparison, the whole heat exchangers consisting of the shell, tubes, baffles and nozzles are modelled. The model is then used to compute and compare the thermo-hydraulic performance for all the three cases. The results show that the use of helical baffles results in higher thermo-hydraulic performance compare to novel and segmental baffles.

***Key words:*** Novel baffle STHX, Helical baffle SHTX, Segmental baffle STHX, CFD, Heat Exchanger, Numerical comparison, RNG, k-ε model

## I. INTRODUCTION

Heat exchangers have an important role in various engineering processes. According to [1] more than 35- 40% of heat exchangers are of the shell-and-tube heat exchangers (STHXs) type, this is mainly due to their wide range of allowable design pressures and temperatures, their rugged mechanical construction, and ease of maintenance. STHXs contain a number of tubes packed in a shell with their axes parallel to that of the shell (Fig1). The process of heat transfer takes place as one fluid flows inside the tubes, while the other fluid flows on the shell side across the tube bundles. Baffles are used to control the shell-side flow distribution as well as enhancing heat transfer. Hence, the form and structure of the baffles are of crucial importance for the performance of this type of heat exchangers.

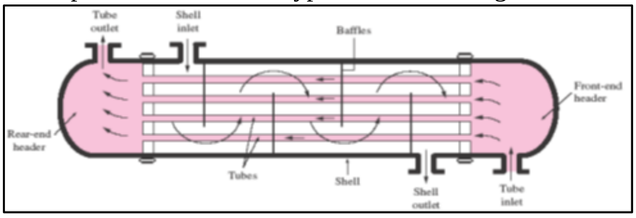



Fig.1: Tube and Shell type Heat Exchanger (Counter Flow)

Evaluation of the shell side thermo- hydraulic performances and the design and rating of STHXs can be difficult. It is further complicated due to the presence of various leakages and secondary flows. Nowadays, the evaluation of these performances is almost exclusively done using commercial software such as Heat Transfer Research, Inc. (HTRI). HTRI developed the Stream Analysis method described [2], which is considered one of the most rigorous methods available for computing the shell side coefficients. However, most of the data needed for its implementation remain proprietary. Several empirical methods calculating shell-side heat transfer coefficient and pressure drop have been well developed. The most accurate is the Bell-Delaware method [3], the method can be used in its complete or simplified version [4]. Although not highly accurate, the simplified version is straight forward and can be easily used. The complete version is more accurate but relatively lengthy and involved.

Segmental baffles (Fig 2.b) are most commonly used in conventional STHXs to support tubes and change flow direction. The fluid flow in a tortuous, zigzag manner across the tube bundle in the shell side, the use of segmental baffles improves heat transfer by enhancing turbulence or local mixing on the shell side of the exchanger, however, the conventional STHXs with segmental baffles present some major inconveniences : (1) the pressure drop across the shell is very high due to flow separation at the edge of the baffles with subsequent flow contraction and expansion; (2) low heat transfer efficiency due to the flow stagnation in the “dead zones”; (3) the strong induced vibrations reduce the operation time of the STHXs; (4) large shell side fouling resistance. Under the same heat load higher pumping power is usually needed to offset the higher pressure drop while using the conventional segmental baffles STHXs. Several researches has been carried out to understand the shell side thermo-hydraulic characteristics of segmental baffles STHXs and overcome its inconveniences. Edward S.Gaddis and Volker Gnielinski introduced few correction factors using Delaware method for calculating shell side pressure drop [7]. W.Roetzel and D.Lee have experimentally investigated the leakage flow in STHE with the segmental baffles and found leakage (SL) has great influence on overall heat transfer coefficient [8]. Simin Wang, Jian Wen, and Yanzhong Li improved the configuration of shell and tube heat exchanger through the installation of sealers in the shell side. Uday C Kapale, and Sathish Chand developed a theoretical model for shell side pressure drop. The model incorporates the effect of the pressure drop in inlet and outlet nozzles along with the losses in the segments created by baffles. However, the major problems remain same. As a result, it is necessary to use new type of baffles to improve the shell side heat transfer rates/pressure drop ratio.

Various types of baffles have been designed to overcome these problems and improve the thermo-hydraulic performance of the conventional STHXs. [10] discuss a variety of different strategies available to process and equipment designers to improve industrial heat transfer. Such as helical baffle rod type plate baffle spiral plate baffle novel type plate baffle.

Helical baffles were first proposed by Lutcha J and Nemcansky J in the year 1990. They found that helical baffles caused near plug flow conditions within the shell space and induced rotational flow. The plug flow conditions increased the heat exchanger effectiveness. Stehlik et al. [10] compared heat transfer and pressure drop correction factors of an optimized segmental baffle heat exchanger to those of a helical baffle heat exchanger. M.R.Jafari Nasr, and A.Shafeghat derived equations for both turbulent and laminar regimes relating pressure drop to heat transfer coefficient and heat transfer area for helical baffled heat exchanger. They developed a straight forward design procedure for helical coil heat exchanger [9]. Yong – Gang Lei et al. carried out numerical simulation to understand the effect of different baffle inclination angles on fluid flow and heat transfer of heat exchanger with helical baffles [11]. They found that the enhanced performance increases with the increase of baffle inclination angle when $\alpha < 45°$, and decreases when $\alpha > 45°$. D U Wenjing, Wang Hongfu and Cheng Lin have investigated the role of shape and quantity of the helical baffles in the shell side heat transfer rate and fluid flow performance [12]. Luhong Zhang et al. [13] conducted comparative experiments of non-continuous helical baffle heat exchangers and segmental baffle.

In [9], the optimal angle for helical baffles was found to be 40°. Also, the results suggest that STHX with helical baffles are a proper replacement for STHXs with segmental baffles. Helical baffles (Fig 2.c) in STHXs represent an alternative to segmental baffles by circumventing the aforementioned drawbacks of conventional segmental baffles. Helical baffles are used as tubes support and can produce a perfectly helical flow pattern across the tubes bundles. They consist of baffles with an inclined helix angle as measured from the perpendicular to the axis of the exchanger. Typical angle values vary from 10 to 40°. Helical baffles offer the following advantages: (1) ratio of heat transfer rates/pressure drop improves at shell side; (2) reduced bypass effects; (3) reduced fouling of shell-side; (4) prevent vibration induced by flow; (5) requires less maintenance.

Experimental and numerical results for Rod baffles and their enhancement are presented in [14]- [15]. These baffles can eliminate flow induced vibrations while having small pressure drops and good overall thermo-hydraulic performances.

Novel plate baffles (Fig 2.c) are new type heat transfer devices and no publications comparing their performance to traditional baffles could be found. Novel plate baffles are reformed from segmental and rod baffle. The shell side fluid flows longitudinally through the gaps between the orifice edges and tube walls. They have good thermo-hydraulic performances while being less liable to foul, eliminates stagnant recirculation zones and avoid flow induced vibration compared to the conventional STHXs with segmental baffles. Yang at le [16] performed Numerical and experimental investigation on a novel shell-and-tube heat exchanger with plate baffles. He found that the novel plate baffles heat exchanger shows better thermos-hydraulic performance compare to rod type baffle and provides a good alternative solution for industrial designers compared to rod baffles heat exchanger.

According to the experimental results in [17], the use of continuous helical baffles resulted in nearly 10% increase in heat transfer coefficient, compared to that of conventional segmental baffles based on the same shell side pressure drops. Until now, the majority of helical baffles used in STHXs are non-continuous approximate helicoids, due to difficulty in manufacturing the continuous helical baffles. thermo-hydraulic performance of non-continuous helical baffles can be decreased because of their high fluid leakage compared to continuous helical baffles [18]. Therefore, this study will adopt a STHX with continuous helical baffles. It should be noted that nowadays and with the rapid development of CFD, Numerical simulation of STHXs has been a necessary supplement to experiments and theory. Moreover, it offers an economic alternative. The detailed mathematical and numerical modelling of STHXs with helical baffles is discussed in [19]- [20]. The results show that the different numerical models compare reasonably well with experimental data. In [21], it was found that exit length to shell side velocity ratio of 2.5 is required for proper convergence using CFD code OpenFOAM-2.2.0. The effect of the flow field on shell side heat transfer coefficient and a comparison with analytical methods are presented for various flow rates.

In this work, the commercial software ANSYS FLUENT is adopted to conduct the numerical study for the same STHX with three different baffle types, to evaluate their thermo-hydraulic performance. Complete STHX with tubes, tube sheet, shell, baffles and inlet outlet nozzles are modelled for three cases with segmental, helical and novel plate baffles.

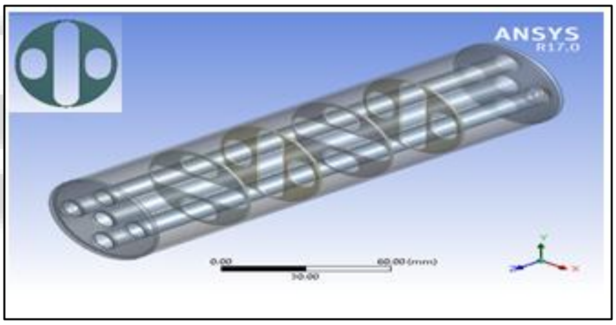



Fig. 2: a) Model of tubes bundles with different baffles type: novel baffles

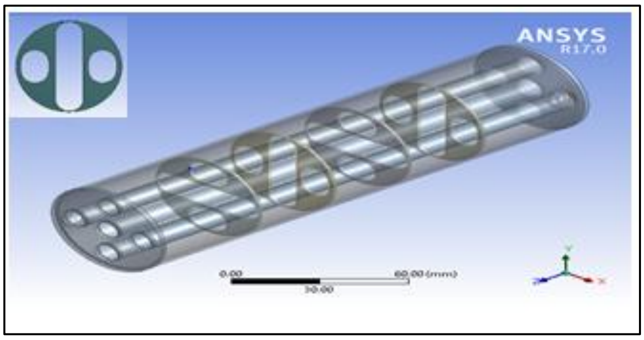



Fig. 2: b) Model of tubes bundles with different baffles type: Segmental baffles

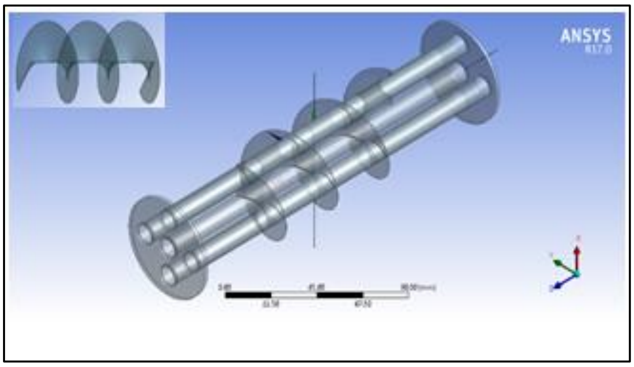



Fig. 2: c) Model of tubes bundles with different baffles type: Helical baffles

## II. METHODOLOGY

### A. *Model Geometry*

The configurations of the STHX with segmental, helical and novel plate baffles are shown respectively in Fig.3. The size of the current STHX is small, thus the computation load of modelling the whole device for the three type of baffles can be tolerated. Another one point should be highlighted that baffle spacing of the three modelled heat exchangers is kept the same with each other, which is to ensure all of the values of geometry parameters are consistent except the baffle type. It could be believed that under the same conditions, comparisons between the three different baffles type are more convincing [26]. The material of the shell is stainless steel and for tubes and baffle material selected is copper. The working fluid in both the shell and tube side of the heat exchanger is water.

Water is considered as a Newtonian and incompressible fluid with constant thermo-physical properties. Furthermore, the fluid flow and heat transfer processes are turbulent and in steady-state. The viscous heating and compression work are both trivial and thus are neglected in the energy equation. The heat exchanger is assumed to be newly built and thus has a negligible fouling resistance. The leakage between tube and baffle and between baffle and shell is negligible and thus ignored.

In this study, a hydrodynamic model based on the unstructured-grid finite volume method was developed using ANSYS Fluent software. This model was based on the numerical solution of continuity, momentum and energy equations.

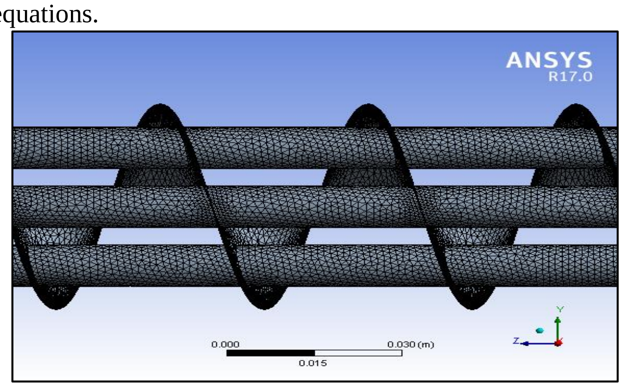



Fig. 3: a) Mesh: helical baffles with tube bundle

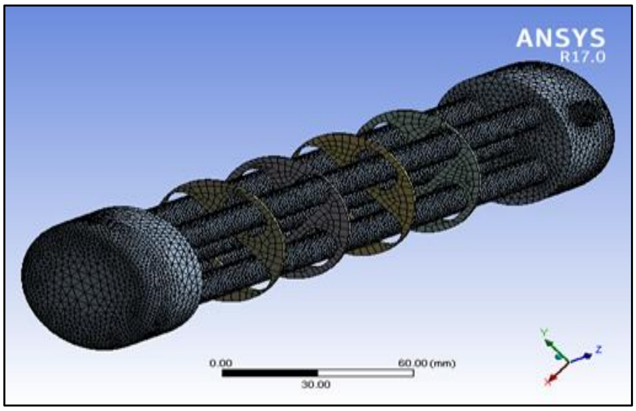



Fig. 3: b) Mesh: novel baffle with tube bundle

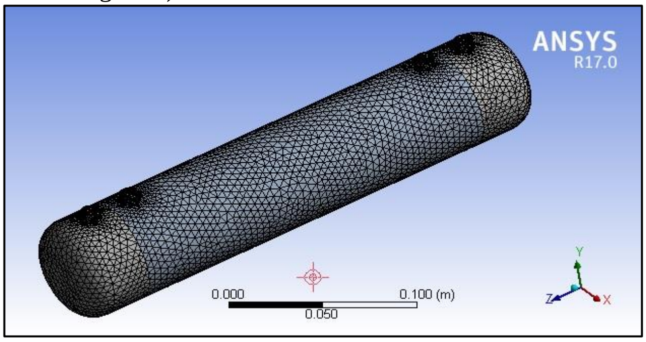



Fig. 3: c) Meshing of All Parts of Heat Exchanger

RNG k–ε turbulence model is adopted in the current study. The renormalization group (RNG) k–ε model of Yakhot and Orszag is adopted in the simulation because the model provides improved predictions of near-wall flows. The RNG k– ε model was derived by a statistical technique called renormalization method, which is widely used in industrial flow and heat transfer because of its economy and accuracy. The governing equations for continuity, momentum, energy, k and ε in the computational domain can be expressed as follows:

Continuity equation: $\frac{\partial \rho}{\partial x} + div(\rho u) = 0$

### B. *Momentum equations:*

X-Momentum equation:

$$\frac{\partial(\rho u)}{\partial x} + div(\rho uu) = -\frac{\partial p}{\partial x} + div(\mu gradu) + S_{Mx}$$

Y-Momentum equation:

$$\frac{\partial(\rho v)}{\partial y} + div(\rho vu) = -\frac{\partial p}{\partial y} + div(\mu gradv) + S_{My}$$

Z-Momentum equation:

$$\frac{\partial(\rho w)}{\partial z} + div(\rho wu) = -\frac{\partial p}{\partial z} + div(\mu gradw) + S_{Mz}$$

Energy equation:

$$\frac{\partial(\rho i)}{\partial t} + div(\rho iu) = -pdivu + div(kgradT) + \Phi + S_i$$

Turbulent kinetic energy: $\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_j)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \eta_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \rho\varepsilon$

Turbulent dissipation energy: $\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \varepsilon u_j)}{\partial x_j} =$

$$\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1^* \frac{\varepsilon}{k}\eta_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - C_2 \rho \frac{\varepsilon^2}{k}$$

Where $\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$, $C_1^* = C_1 - \frac{\eta(1-\eta/\eta_0)}{1+\beta\eta^3}$,

$$\eta = (2E_{ij} * E_{ij})^{\frac{1}{2}} \frac{k}{\varepsilon}, E_{ij} = \frac{1}{2}\left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right],$$

$$C_\mu = .085, \quad C_1 = 1.42, \quad C_2 = 1.68,$$
$$\beta = 0.012, \quad \eta_0 = 4.38$$

### *C. Domains definitions, meshes and boundary conditions:*

As mentioned before, three STHXs with different baffle types are modelled. For each of the three studied STHXs, three domains are defined, two fluid domain (water in the tube and shell side) and the solid domain (tubes bundle, baffles). Fig.4 shows the meshing formed in Fluent after the setup of Fine element sizing with High smoothing characteristics. Total number of elements formed are 993682, 661094 and 1329669 and nodes formed are 234831, 182514 and 308371

For segmental, novel and helical baffle STHX.

| | |
|---|---|
| Diameter of shell | 50mm |
| Length of shell | 200mm |
| Tube internal diameter | 9.5mm |
| Tube external diameter | 11mm |
| No. of tubes | 5 |
| Baffle thickness | 1mm |
| Tube arrangement | triangular |

Table 1: Structural Parameters Of The Sthx

The momentum boundary condition of no slip and no penetration is set for all the solid walls. The thermal boundary condition of zero heat flux is set for the shell wall and inlet and outlet nozzle walls, while the walls of tubes, baffles, and tube bundle, which also represent the solid-fluid interfaces between the two fluid domain and the solid domain, have the thermal boundary condition of coupling heat transfer (two interfaces with coupled wall). The inlets for the shell and tube sides are set as boundary conditions of velocity-inlet, the outlets are set as pressure-outlet. The outlets are assumed to have a pressure of zero so the inlet pressure is equal to the pressure drop on both shell and tube sides.

The commercial ANSYS FLUENT is used to calculate the fluid flow and heat transfer in the computational domains. The governing equations are iteratively solved by the finite-volume formulation with the SIMPLE algorithm. The second-order upwind scheme is adopted for the momentum, energy, turbulence and its dissipation rate. The pressure term is treated with the standard scheme. Default under relaxation factors of the solver are used, which are 0.3, 0.7, 0.8, and 0.8 for the pressure, momentum, turbulent kinetic energy, and turbulent energy dissipation, respectively. The convergence criterion is that the normalized residuals are less than $1e^{-4}$ for the flow equations and $1e^{-8}$ for the energy equation.

## III. Results and Discussions

As per the methodology mentioned above, total 3 experiments have been carried out for each case in a system consist of 4GB DDR3-1600Mhz Ram, AMD A8-7410 APU – 2.2Ghz Processor with 4 Cores. This whole experiments took approximately 105hrs of time for completion.

As specified in the boundary conditions in FLUENT, the shell and tube side outlets are assumed to have a pressure of zero, thus, the pressure drop is equal to the inlets pressure for both shell and tube side. The overall thermo-hydraulic performance on the shell side is expressed by the heat transfer coefficient per unit pressure drop, i.e., h/ΔPa

The pressure drop is of great importance in the design of shell-and-tube heat exchangers, pumping cost are highly related to pressure drop, and therefore lower pressure drop results in lower operating costs.

Fig.4 depicts the variation of the pressure drop versus the shell side mass flow rate for the three heat exchangers. The pressure drop increases proportional to the mass flow rate for the three heat exchangers. The pressure drop for the STHX with helical baffles is about 23% and 18% on average lower than the pressure drop for the STHX with segmental baffles and novel baffle respectively. The pressure drop for the STHX with novel baffle is lower than that for segmental baffle STHX as water flows through the opening around tubes thus lower resistance to flow. The reason is that the flow distribution with segmental baffles on the shell side is zigzag, flow separation at the edge of baffles causes abrupt momentum change and severe pressure drop. Whereas the primary flow direction of helical baffles does not change dramatically.

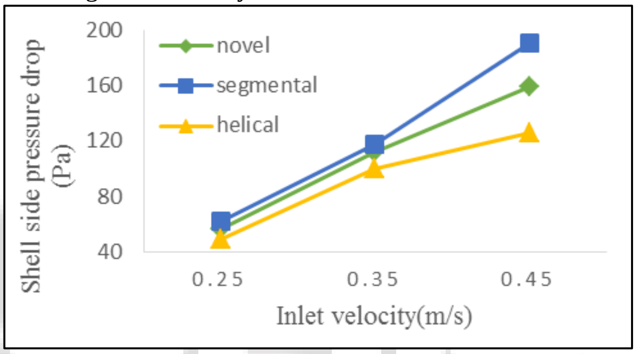



Fig. 4 Shell side pressure drop versus inlet velocity

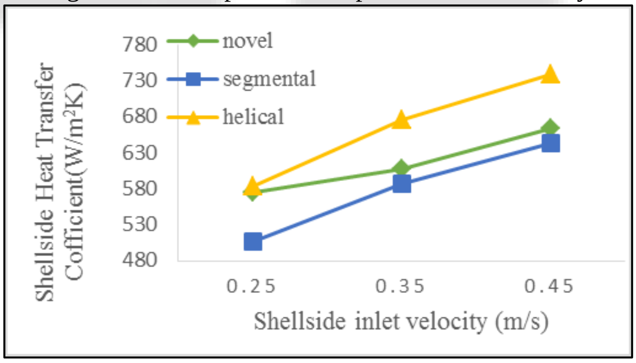



Fig. 5 Shell side heat transfer coefficient versus inlet vel.

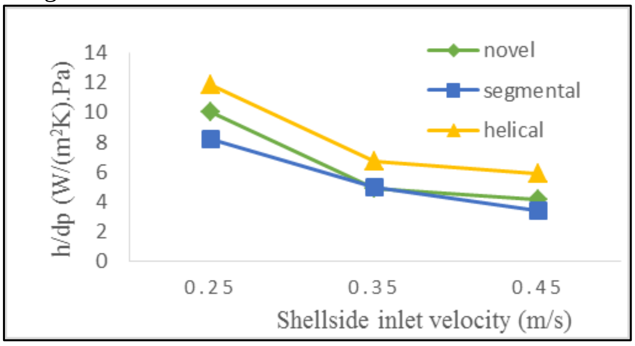



Fig. 6 Heat transfer coefficient per pressure drop versus inlet velocity

Fig.5 represents the comparisons of the shell side heat transfer coefficient for the three heat exchangers. It can be observed that the shell side heat transfer coefficient increases with the increases in the shell side mass flow rate. Heat transfer coefficient for helical baffle is maximum followed by novel and minimum for segmental baffle.

Fig.6 shows the comparisons of the ratio of the heat transfer coefficient to pressure drop for the three heat exchangers with different baffles. It is clearly seen that the STHX with helical baffles has the best performance ratio among the three STHXs.

After successful completion of all the experiments, temperature contours are obtained for all the cases, shown below.

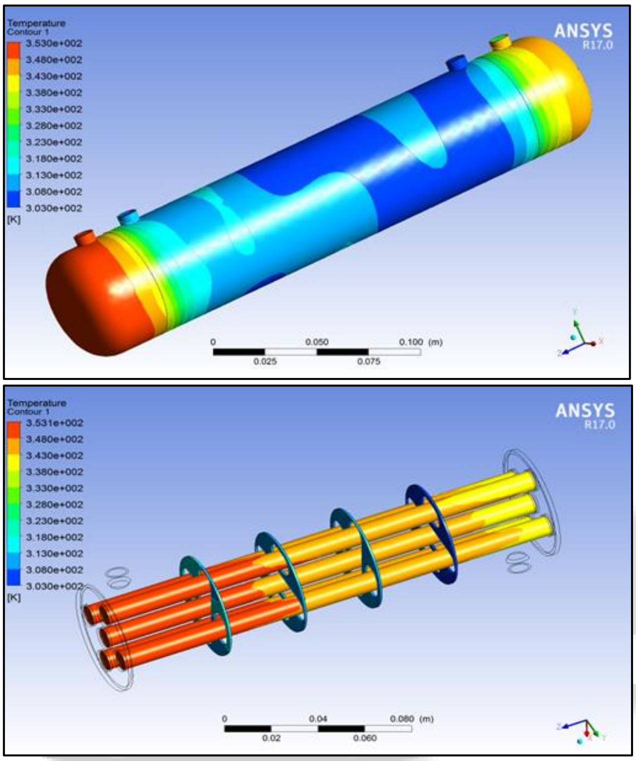
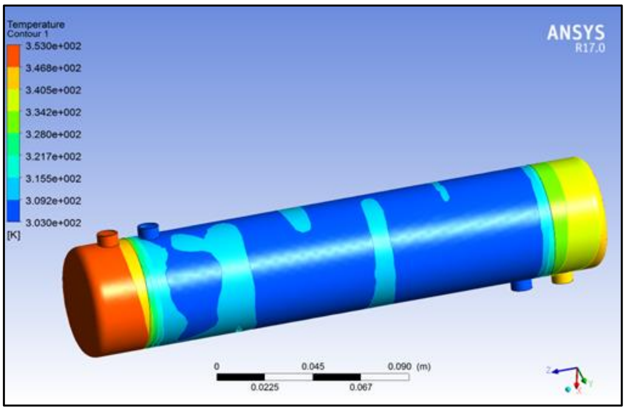



Fig. 7: a) Temperature Contour of Outer Surface and inner tube surface novel baffle



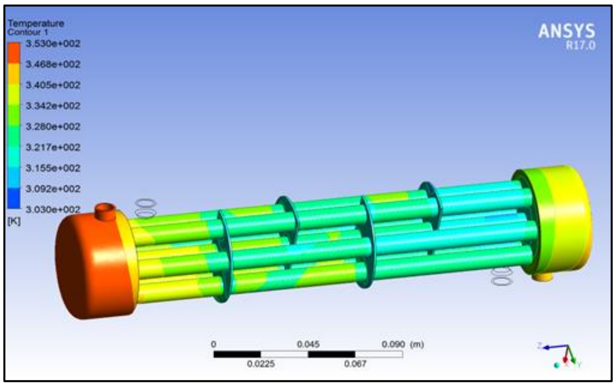




Fig. 7: b) Temperature Contour of Outer Surface and inner tube surface segmental baffle

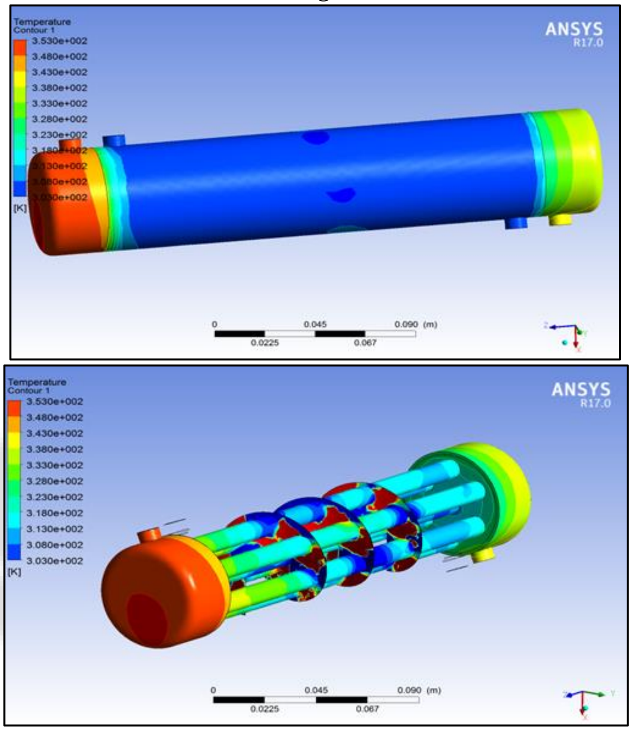



Fig. 7: c) Temperature Contour of Outer Surface and inner tube surface helical baffle

## IV. CONCLUSION

In the present study, a numerical model is used to compute and compare the thermo-hydraulic performances of shell-and-tube heat exchangers with different baffle types: segmental, helical and novel baffles. Flow analysis in shell side, showed that velocity distribution in helical baffles is more uniform and homogenous compared to segmental and novel baffles. This leads to less dead zones and less fluid recirculation areas inside the shell. The results indicate that, compared to the conventional segmental baffles, the use of helical baffles provide a good balance between heat transfer and pressure drop characteristics. Novel baffle shows better performance characteristics compares to segmental baffle but lower than that for helical baffles. Thus, the highest thermo-hydraulic performance is achieved by using helical baffles. Although manufacturing of novel baffle is easier compare to helical baffle and could be considered as an effective alternative to conventional segmental baffles.

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