# Design and Comparative Study of Shell and Tube Heat Exchanger with Different Baffle Configurations

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***Abstract*****:** ***In this study, the pressure drop, heat transfer coefficient and performance evaluation factor (PEF) of a shell and tube heat exchanger with eight different geometrical baffle configurations are numerically investigated. The work was carried out using ANSYS. To investigate the influence of baffle on performance, four baffle spacing (86 mm, 100 mm, 120 mm and 150 mm), four baffle cuts (30%, 40%, 50% and 70%) and two baffle types (single segmental type and double segmental type) are considered. Low baffle spacing provides the most heat transfer rate and efficient overall performance of heat exchanger with the expense of pressure drop. With increase in baffle cut percentages, both pressure drop and heat transfer coefficient decreases but overall performance of heat exchanger increases. The results show that heat exchanger with double segmental type baffle (DHX) has the highest impact on heat transfer and overall performance of heat exchanger in comparison to single segmental type (SHX).***

***Keywords*****:** **Pressure drop, Heat transfer coefficient, Baffle cut, Performance evaluation factor, Single segmental.**

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## I. INTRODUCTION

With the advancement of technology, the industries have constantly demanding the requirement of better efficient and cost saving mechanical equipment. Heat exchangers are one of those widely used equipment which can directly affect the outcome of an industrial plant. The main purpose of heat exchanger is to cool down an over-heated part or fluid and heating up of an element or fluid in industries. The fluid which transfers the heat inside a heat exchanger may be separated by a solid wall to prevent mixing or they may be in direct contact. Thus, heat exchangers keep machinery within the safe operating temperature. In industry, the requirement of heat transfer capacity is quite large and therefore, shell and tube heat exchangers are widely used. It is not only capable enough to efficiently transfer the large amount of heat through cooling or heating but also it can be cleaned and maintained easily.

Naqvi et al. [1] numerically studied a heat exchanger and performance analysis was conducted using anti-vibration clamping baffle with twisted square tubes and helical baffles with cylindrical tubes. The result shows the decrease of pressure drop in anti-vibration clamping baffles and increase in heat transfer rate as compared to segmental baffle with cylindrical tubes. Ozkol et al. [2] performed an analysis on determination of optimum geometry using a genetic algorithm solver by giving performance limits. They used number of heat transfer units (NTU) for the analysis. The experiment of performance test on an air-cooled finned tube supercritical $CO_2$ sink heat exchanger was carried out by Vojacek et al. [3]. Gnielinski correlation was used for calculating the overall heat transfer coefficient in their work. Xu et al. [4] performed a study on the heat transfer characteristics of compact heat exchanger based on experimental data. This study was based on $\eta$-NTU relationships and the parametric optimization of compact air-air heat exchanger. The computational fluid dynamics (CFD) simulation on shell and tube heat exchanger shows that, single shell with helical type baffle configuration provides appreciable overall heat transfer coefficient but with high pressure drop [5]. Arani et al. [6] also applied CFD to analyse the thermo-hydraulic behaviour of a shell and tube heat exchanger with the new baffles and ribbed tube and found certain improved results in heat transfer.

The study of Cao et al. [7] on aerodynamic noise and heat transfer in shell and tube heat exchangers used the helical and segmental baffles. Their results reveal that at the same

Re, the aerodynamic noise is lower in helical type than segmental type baffles. Kunwer et al. [8] performed the numerical analysis to compare the effectiveness of STHX without baffle, with segmental and align baffles. Mohammadi et al. [9] numerically investigated the total heat transfer rate and pressure drop for a shell and tube heat exchanger with six different porous baffles. It was observed that porosity had the least impact in comparison to baffle cut. The permeability of $\emptyset = 10^{-9}$ m$^2$ provides the highest heat transfer rate and also the lowest pressure drop in this setup. The numerical simulation was performed by Lei et al. [10] to see the influence of baffle inclination angle on the heat transfer and pressure drop in heat exchanger. It was found that that Nu of tube bundles starts increasing with increase in the baffle inclination angle when $\alpha<30°$. The pressure drop variation is found to be larger in smaller inclination angle. Li et al. [11] conducted both experimental and numerical simulation on SHX with longitudinal flow to analyse the thermal hydraulic performance and efficiency of energy. Their results show that the longitudinal flow pattern affected lesser on loss of pressure and heat transfer. However, the distribution of pressure drop provides the possibility of better energy conservation. Three-dimensional numerical simulation performed by Taher et al. [12] on a helical shell and tube heat exchanger using different baffle spacing shows the increase in pressure gradient with decrease of baffle spacing. In longer baffle spacing heat transfer coefficient is lower at the same mass flow rate and working conditions. For the same pressure gradient condition, larger baffle spacing has highest heat transfer coefficients. Yang et al. [13] studied the flow and heat transfer performance on unilateral ladder type helical baffle heat exchanger (ULHBHX) using different types of baffle configurations. The result shows that the heat transfer coefficients of shell side of the ULHBHX with folded baffles gives better results and helical schemes shows better results than segmental schemes.

The study on the different literatures reveals the performance analysis of shell and tube heat exchanger with different type of baffles. However, there is scope for studying the effects baffle cut and baffle spacing on the heat exchanger performance. The present study aims to carry out the investigation on the influence of single and double segmented baffles with various baffle cuts and spacing on heat transfer and pressure drop of a shell and tube heat exchanger.

## II. METHODOLOGY

The geometrical specifications, user defined thermal and flow parameters of the respective heat exchanger are shown in Table 1 and Table 2. The geometrical dimensions of studied shell and tube heat exchanger is same for all the models. The commercial ANSYS 16.0 software was used to perform the simulation and analysis.

Table 1: Geometrical specifications of heat exchanger

| Parameters | Specifications |
|---|---|
| Shell diameter | 90 mm |
| Tube diameter | 23 mm |
| STHX length | 600 mm |
| No. of tubes | 5 |

**Table 2**: User defined thermal and flow parameters

| Parameters | Defined values |
|---|---|
| Shell side inlet temperature | 300K |
| Tube side inlet temperature | 380K |
| Backflow temperature at outlet | 300K |
| Velocity of fluid at inlet | 0.5m/s |

### A. *Boundary conditions and assumptions*

The boundary conditions used in the design for the heat transfer simulation are summarized as follows:

1. The walls of heat exchanger are set to be in no slip condition.
2. The fluid inlet temperature at the shell side and tube side are maintained at constant temperature.
3. The inlet of the shell side is set as mass flow inlet condition.
4. The inlet of the tube side is set as velocity inlet condition.
5. Fouling is considered as negligible.

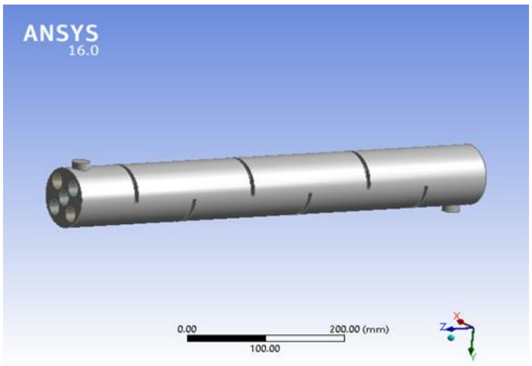


**Fig. 1.** Single segmental baffle heat exchanger.

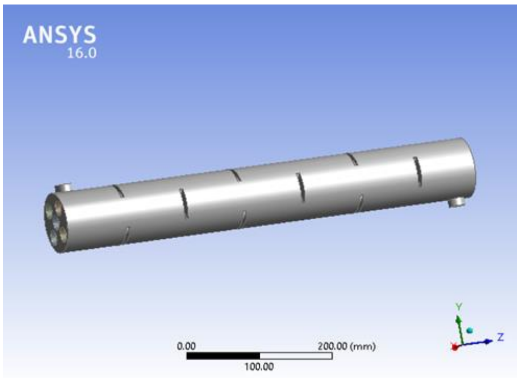


**Fig. 2.** Double segmental baffle heat exchanger.

### B. *Governing Equations*

The shell and tube heat exchanger under consideration has least fouling resistance which is ignored in the boundary conditions. The flow modelling is considered for turbulent flow. The fluid is assumed to be in the steady state and has no time dependent terms. The fluid properties are considered to be constant throughout heat transfer process. The $k-\varepsilon$ turbulence model is used for the calculation of eddy viscosity. CFD is based on the Navier Stokes equation which is given in Eq. 1.

$$\frac{\partial(\rho\varphi)}{\partial t}+\frac{\partial(\rho u\varphi)}{\partial x}+\frac{\partial(\rho v\varphi)}{\partial y}+\frac{\partial(\rho w\varphi)}{\partial z}= \Gamma\left(\frac{\partial^2\varphi}{\partial x^2}+\frac{\partial^2\varphi}{\partial y^2}+\frac{\partial^2\varphi}{\partial z^2}\right)+S(\varphi) \quad (1)$$

where φ = 1, $u, v$ and $w$ are velocity components along $x, y$ and $z$ coordinates respectively.

The governing equations for continuity, momentum and energy are shown in Eqs. 2-4.

Continuity equation

$$\frac{\partial\rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \quad (2)$$

Momentum equation

$$\rho\left(\frac{\partial v_x}{\partial t}+v_x\frac{\partial v_x}{\partial x}+v_y\frac{\partial v_x}{\partial y}+v_z\frac{\partial v_x}{\partial z}\right)=-\frac{\partial P}{\partial x}+ \mu\left(\frac{\partial^2 v_x}{\partial x^2}+\frac{\partial^2 v_x}{\partial y^2}+\frac{\partial^2 v_x}{\partial z^2}\right) \quad (3)$$

Energy equation

$$\rho c_p\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)= k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)+\mu\varphi \quad (4)$$

where $v, x$ is the velocity vector, $P$ is the pressure and $\rho$ is the density. $k$ and $\varphi$ represent thermal conductivity and the dissipation function respectively.

The governing equations and relevant terms for $k-\varepsilon$ turbulence model are presented in Eqs. 5-8.

Turbulence kinetic energy

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho u_i k)=\frac{\partial}{\partial x_i}\left(\left(\mu+\frac{\mu_i}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right) +S_k \quad (5)$$

Turbulence dissipation energy

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho u_i\varepsilon)=\frac{\partial}{\partial x_i}\left(\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_i}\right) +S_\varepsilon \quad (6)$$

where $S_k$ and $S_\varepsilon$ are source terms.

$$S_k=\tau_{ij}^R\frac{\partial u_i}{\partial x_j}-\rho\varepsilon+\mu_t P_B \quad (7)$$

$$S_\varepsilon=C_{\varepsilon 1}\frac{\varepsilon}{k}\left(f_1\tau_{ij}^R\frac{\partial u_i}{\partial x_j}+\mu_t C_B P_B\right)-C_{\varepsilon 2}f_2\frac{\rho\varepsilon^2}{k} \quad (8)$$

where $k$ is the first transported variable i.e., turbulent kinetic energy and $\varepsilon$ is the second transported variable i.e., the rate of dissipation of turbulent kinetic energy. $P_B$ represents the turbulent generation due to buoyancy force.

### C. *Grid generation*

The grid independency test is a study of mesh obtained during simulation which ensures the results are independent of the mesh and improves the reliability of the results. In this test, convergence criterion is evaluated firstly and then it is examined whether different mesh discretization is affecting the output parameter of the simulation such as temperature, pressure, mass flow rate and velocity etc. The grid independency test is done for our designed model and data are presented in Figure 3.

The plot seen in Figure 3 illustrates that the output parameter for our designed model which is temperature is independent of the grid size or mesh type. The three different mesh types (coarse, medium and fine) are used for obtaining temperature. The data shows the difference of these three mesh types as very small, which is less than about 4%. This ensures the designed model can be used for the proposed work. The independency of the output parameter over grid size provides the reliability of the model. By comparing the results, mesh 3 model is taken for the study.

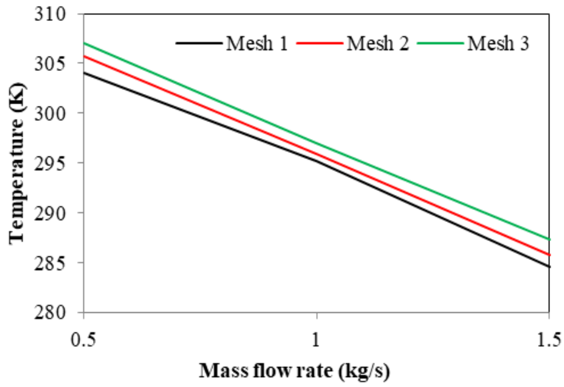


**Fig. 3.** Mesh Independency for temperature

## III. RESULTS AND DISCUSSION

### A. *Variation of pressure along the length in single segmental and double segmental baffle heat exchanger*

The variation of pressure along the length of the shell side is presented in Figure 4. The results obtained illustrate that the average pressure at each distance is decreasing along the length. Here, the pressure variation is considered for the shell side through which cold fluid (water) passes. The distance for the length of the shell side is equally considered and pressure at each distance is obtained by average pressure calculator in the simulation. The two design configurations are compared in the Figure 4. From the Figure 4, it is observed that the pressure drop in single segmental baffle type (SHX) is more than the double segmental baffle type (DHX) which indicates more pumping power requirement. The decrease in pressure drop in double segmental baffle is due to the elimination of dead zones near baffle which results in more heat transfer. Therefore, the configuration of DHX seems to be more preferable.

### B. *Variation of pressure drop with baffle spacing*

The variation of pressure drop and heat transfer coefficient with baffle spacing are presented in Fig.5 and Fig.6 respectively. To improve overall performance of a heat exchanger pressure drop and heat transfer coefficient plays most important role. In this comparative study, all the comparative factors such as pressure drop, heat transfer coefficient and performance evaluation factor (PEF) are considered for shell side through which cold fluid passes. The trend seen from Fig.5 illustrates that the pressure drop is higher when baffle spacing is less and with the increase in baffle spacing pressure drop reduces gradually. In DHX pressure drop is found to be lower than in SHX because of lesser dead zone formation. The results reveal that average pressure drop of DHX is 6.89% lower than the SHX at the maximum baffle spacing (150 mm BS).

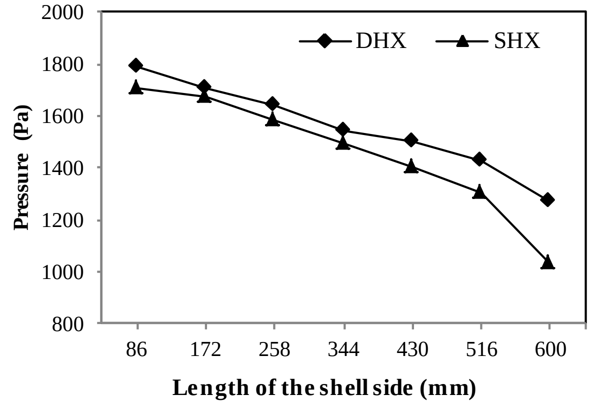


**Fig. 4**. Pressure along the length of shell side of heat exchanger.

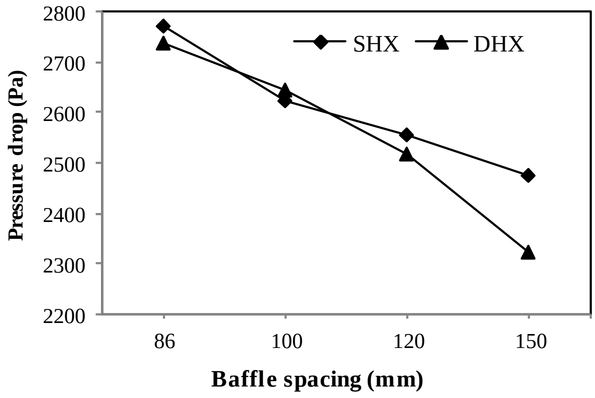


**Fig. 5.** Variation of pressure drop with baffle spacing.

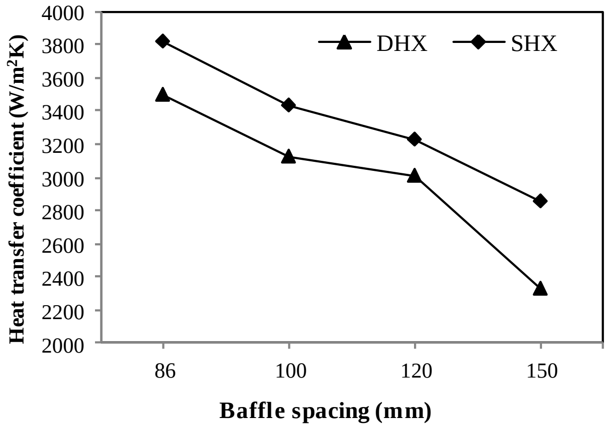


**Fig. 6**. Heat transfer coefficient with baffle spacing.

## *C. Variation of heat transfer coefficient with baffle spacing*

In Figure 6, heat transfer coefficient factor is calculated and graphically presented with the help of the results obtained from simulation. The sequential decreasing behaviour of heat transfer coefficient is observed with increase in baffle spacing. In this case, at 86 mm baffle spacing the shell side heat transfer coefficient reached the highest value for both SHX and DHX. It is observed that, average heat transfer coefficient of SHX is 7.89% higher than DHX at the smallest baffle spacing (86 mm BS). It is found that when baffle spacing is smaller there is increase in formation of eddies which provides more heat transfer as compared to larger baffle spacing where lesser eddies are formed. From the Figure 5 and Figure 6, it is also observed that both the figures give two different characteristic performances, therefore, for optimal selection of shell and tube heat exchanger another parameter is introduced as PEF which is analysed as same operating and boundary conditions.

## *D. Variation of performance evaluation factor with baffle spacing*

For the selection of optimal heat exchanger the ratio of heat transfer to pressure drop is a crucial factor. At the same thermal and flow conditions, it is always challenging to obtain maximum heat transfer rate with minimum pressure drop. This ratio of heat transfer rate to pressure drop is termed as performance evaluation factor (PEF) in this study. The significance of PEF is to obtain the value of heat exchange at same pumping power which can be compared for different geometric conditions. The variation of PEF with baffle spacing is presented in Figure 7. The obtained results are compared between SHX and DHX and can be concluded that at 86 mm baffle spacing, the best value of PEF is found for both baffle type of baffles. Figure 7 also reveals that DHX shows 3.65% higher PEF over SHX.

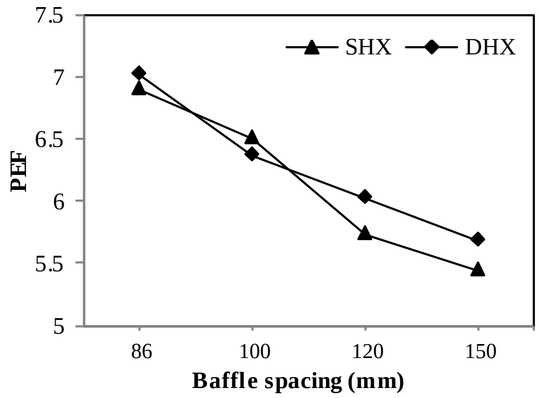


**Fig. 7**. Variation of PEF with baffle spacing

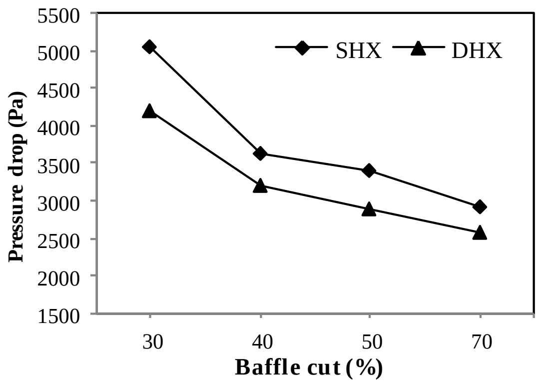


**Fig. 8**. Variation of pressure drop with baffle cut.

*E. Variation of pressure drop with baffle cuts*

Four number of baffle cuts viz., 30%, 40%, 50% and 70% were considered in order to study the effects of baffle cuts on heat exchanger performance. Although, baffles improve the heat transfer, at the same time it is also responsible for pressured drop. Figure 8 indicates that at 30% baffle cut the pressure drop is higher and with increase in baffle cut percentage pressure drop reduces. In SHG type baffle heat exchanger, flow pattern is zigzag in shell side and thus, at the edge of baffles flow separation occurs resulting the change in momentum and severe pressure losses. The average of pressure drop in DHX is 7.47%-20.54% lower than SHX. This result suggests the use of DHX is more preferable in this type of configuration to reduce the cost of pumping power.

*F. Variation of heat transfer coefficient with baffle cuts*

Figure 9 shows the variation of heat transfer coefficient with baffle cut for SHX and DHX. The trend indicates that the heat transfer coefficient is highest at 30% baffle cut and gradually decreases with increasing baffle cut percentages. This is due to the fact that in smaller baffle cut the obstruction for fluid to pass is higher which implies the fluid gets more mixing and thus more heat transfer than larger baffle cut percentages. The heat transfer coefficient is lower at the starting in DHX than SHX, however, the heat transfer coefficient improves with further increase in baffle cuts. The results reveal that average of heat transfer coefficient in DHX is 6.4% - 8.1% higher than SHX.

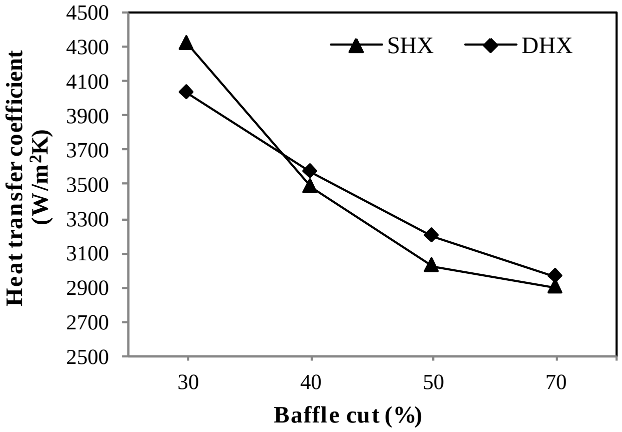


**Fig. 9.** Variation of heat transfer coefficient with baffle cut.

*G. Variation of performance evaluation factor with baffle cuts*

The pressure drop and heat transfer coefficient are the crucial parameters for effective evaluation of heat exchanger performance. In Figure10, variation of PEF with baffle indicates that increasing baffle cut percentage gives higher PEF. This PEF parameter is a deciding factor in the selection of geometrical configuration for higher heat transfer rate and lower pressure drop. In this study, DHX shows better performance when compared with SHX. The average of PEF for DHX is about 9.03% higher than the SHX.

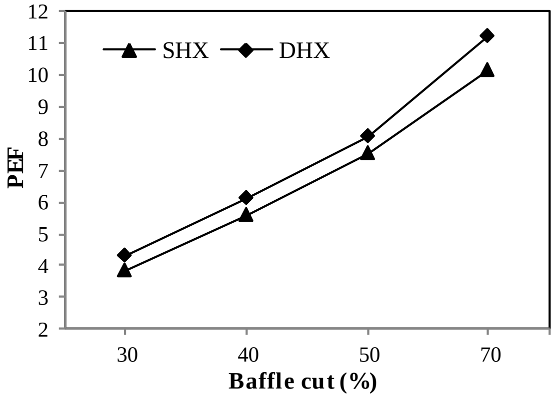


**Fig. 10**: Variation of performance evaluation factor with baffle cut.

## IV. CONCLUSION

This study covered the design and comparative study of a shell and tube heat exchanger with different geometric configurations including different parameters such as pressure drop, heat transfer coefficient, PEF, baffle cuts, and baffle spacing. The study shows that DHX type baffle significantly reduces pressure drop in shell side and enhance the overall thermal performance of the designed heat exchanger in comparison to SHX type baffle. The variation of baffle spacing and baffle cuts has been found to have direct effect on the performance of the heat exchanger. With decrease in baffle spacing the overall performance of the heat exchanger is improved and with increase in baffle cut percentage the overall performance of the heat exchanger is enhanced. Through this work, an attempt to find the optimum geometric configuration is made to enhance rate of heat transfer with minimum pressure drop.

### Nomenclature

| | |
|---|---|
| $Cp$ | Specific heat capacity, J/K |
| $g$ | Gravitational acceleration, m/s$^2$ |
| $k$ | Turbulent kinetic energy |
| $m$ | Mass flow rate, kg/s |
| $P$ | Pressure, Pa |
| $Q$ | Heat transfer rate, W/m$^2$K |
| $Re$ | Reynolds number |
| $u$, $v$ and $w$ | Velocity components along $x$, $y$ and $z$ coordinates respectively, m/s |
| $\Delta P$ | Pressure drop |
| $\rho$ | Density, kg/m$^3$ |
| $\mu$ | Dynamic viscosity |
| $\varepsilon$ | Rate of dissipation of turbulent kinetic energy |
| $\varphi$ | Dissipation function |

## REFERENCES

[1] S.M.A. Naqvi, K.E. Elfeky, Y. Cao, Q. Wang, "Numerical analysis on performances of shell side in segmental baffles, helical baffles and novel clamping anti-vibration baffles with square twisted tubes shell and tube heat exchangers," Energy Procedia, vol. 158, pp. 5770-5775, 2019.

[2] I. Ozkol, G. Komurgoz, "Determination of optimum geometry of the heat exchanger body via a genetic algorithm," Numerical Heat Transfer, Part A: Applications, vol. 48 (3), pp. 283-296, 2005.

[3] A. Vojacek, V. Dostal, F. Goette, M. Rohde, T. Melichar, "Performance Test of the Air-Cooled Finned-Tube Supercritical $CO_2$ Sink Heat exchanger," Journal of Thermal Science and Engineering Applications, vol. 11, 2019.

[4] X. Xu, X. Zhang, P. Ke, C. Wang, H. Yang, C. Yang, "Study on the heat transfer characteristic of compact heat exchanger based on experimental data," Procedia Engineering, vol. 121, 2015.

[5] A.S. Ambedkar, R. Sivakumar, N. Anantharaman, M. Vivekenandan, "CFD simulation study of shell and tube heat exchangers with different baffle segment configurations," Applied Thermal Engineering, vol. 108, pp. 999-1007, 2016.

[6] A.A.A. Arani, R. Moradi, "Shell and tube heat exchanger optimization using new baffle and tube configuration," Applied Thermal Engineering, vol. 157, 113736, 2019.

[7] Y. Cao, H. Ke, J.J. Klemes, M. Zeng, Q. Wang, "Comparison of aerodynamic noise and heat transfer for shell and tube heat exchangers with continuous helical and segmental baffles," Applied Thermal Engineering, vol. 88, 116341, 2019.

[8] R. Kunwer, S. Pandey, S.S. Bhurat, "Comparison of selected shell and tube heat exchangers with segmental and helical baffles," Thermal science and engineering progress, vol.20, 100712, 2020.

[9] M.H. Mohammadi, H.R. Abbasi, A. Yavarinasab, H. Pourrahmani, "Thermal optimization of shell and tube heat exchanger using porous baffles," Applied Thermal Engineering, vol. 170, 115005, 2020.

[10] Y.G. Lei, Y.L. He, "Effects of baffle inclination angle on flow and heat transfer of a heat exchanger with helical baffles," Chemical Engineering and Processing, vol. 47, pp. 2336-2345, 2008.

[11] N. Li, J. Chen, T. Cheng, J.J. Klemes, P.S. Varbanov, Q. Wang, W. Yang, X. Liu, M. Zeng, "Analysing thermal hydraulic performance and energy efficiency of shell and tube heat exchangers with longitudinal flow based on experiment and numerical simulation," Energy, vol.202, 117757, 2020.

[12] F.N. Taher, S.Z. Movassag, R.T. Azar, K. Razmi, "Baffle space impact on the performance of helical baffle shell and tube heat changers," Applied Thermal Engineering, vol. 44, pp. 143-149, 2012.

[13] S. Yang, Y. Chen, J. Wu, H. Gu, "Influence of baffle configurations on flow and heat transfer characteristics of unilateral type helical baffle heat exchangers," Applied Thermal Engineering, vol.133, pp. 739-748, 2018.

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