Open Access

SCHOLARENA

Journal of Innovations in Energy science

RESEARCH ARTICLE

Numerical Investigation of Shell and Tube Heat Exchanger with Spiral and Segmental Baffle

AR Urade[1*] and SS Magdum[2]

[1]Assistant Professor, MED, GH Raisoni Institute of engineering and Technology, Wagholi, India
[2]UG Student, MED, GH Raisoni Institute of engineering and Technology, Wagholi, India

*Corresponding author: AR Urade, Assistant Professor, MED, GH Raisoni Institute of engineering and Technology, Wagholi, India, Tel: 9604548214; E-mail: a.r.urade@gmail.com

Citation: AR Urade, SS Magdum (2020) Numerical Investigation of Shell and Tube Heat Exchanger with Spiral and Segmental Baffle. J Innovations Energy Sci 1: 102

Abstract

The shell and tube heat exchanger plays a vital role in a various industries like chemical processing plant, nuclear power plant, thermal power plant etc. The Shell and tube heat exchanger is the cross flow heat exchanger. In this paper a numerical simulation of shell and tube type heat exchanger with spiral baffle and segmental baffles is carried out. In this numerical simulation of STHX of 1 shell and 7 tubes containing spiral baffle at 50 mm pitch and 5 number of segmental baffle. In this paper various operational parameters like flow rate of cold and hot water, temperature and geometrical parameters like spiral pitch, segmental baffle size are studied and heat transfer characteristics, and thermal efficiency is evaluated by heat transfer equation. In this paper the shell and tube heat exchanger modelling, meshing and fluent as solver is done using Ansys 14.5. In this paper thermal efficiency of the heat exchanger is increased by varying the parameters like the pitch of spiral baffle and the sizes, number of the segmental baffle. The numerical result shows that the increment in shell side pressure drop is about twice The thermal efficiency is enhances 10% which demonstrates that both spiral as well as segmental baffle can effectively improve the heat transfer performance of heat exchanger. The numerical result are validated with available literature and found to be valid.

Keywords: Spiral Baffle; Segmental Baffle; Spiral Pitch; Flow Breaker Segmental Baffle; STHX; Thermal Efficiency; Computational Resources

Introduction

STHX is a class of indirect contact heat exchangers used in industrial processes, such as gas processing, petrochemical industries, offshore heat recovery, food preservation and so on, in order to exchange heat at higher efficiencies between fluids at different temperatures. About 35-40% of the heat exchangers used in industrial processes are of shell and tube type [1,2]. They are immensely used because of the advantages they possess such as lower cost, easy maintenance, repair, and possible upgrades. Baffles play a crucial role to argument the thermal performance in the shell side of a STHX. Hold the tubes in position, both production and operation and this increase the fluid velocity and the effective heat transfer coefficient as well as the heat transfer efficiencies. The most commonly used baffle in the segmental baffle. It causes the shell-side fluid to flow in a zigzag manner over the tube bundles and enhances mixing on the shell-side fluid and the spiral baffle provide cyclic motion of the fluid over the bundles of tube as well as the shear viscous to the shell boundary, thereby increasing the heat transfer, but also results in high pressure drop which is undesirable. Along with this, dead zones leading to fouling, back flow, and excessive tube vibration also come into play [1,3,4].

Many research woks has been found broad application with various advantage of higher the thermal efficiency, larger the effective heat transfer area, anti-fouling factor performance, Numerous research works have been done on a number of improved structures to eliminate or minimize the shortcomings faced due to the use of segmental baffles. Instead of the only use of the segmental this baffle are used with spiral baffle in this paper work.

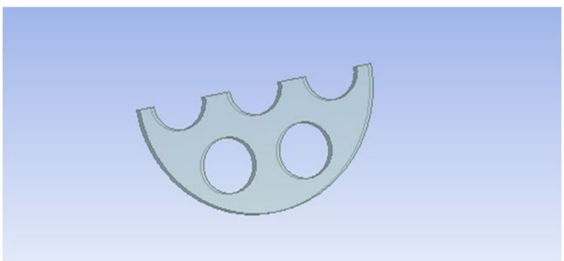


Figure 1: Half Circular segmental baffle

The above Figure 1 shows a half circular baffle with some thickness. This Segmental baffle to give the direction to the fluid upstream and downstream in the shell.

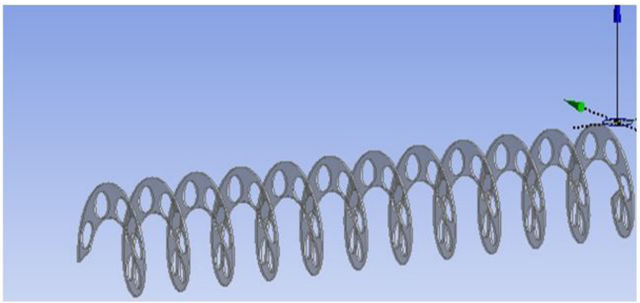


Figure 2: Rectangular Spiral Baffle

The above Figure 2 shows a spiral baffle with some spiral pitch. This baffle is to give the spiral motion to the fluid and it is well distributed entire shell length.

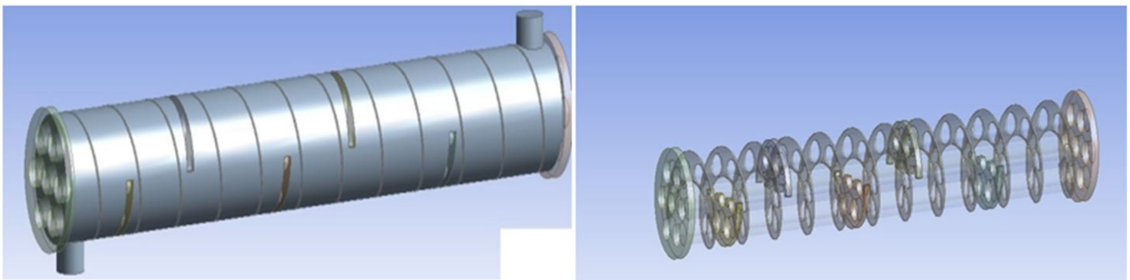
Numerical Model

Geometric Modelling

The geometrical configuration of the spiral as well as segmental baffle STHX for two different is presented in Figure 3. The spiral baffle is drawn from the circumferential part of shell by using sweep creation technique. And the segmental baffle is placed at certain distance in the shell by using manual connection so as together they fix the tube position. Figure 3 the segmental baffle is perpendicular to each other. For the whole model a detailed heat exchanger with heat transfer tubes. All parameters of numerical simulations are identical with the experimental setup. Regular k-ε equations are adopted for turbulent flow zone (Table 1).

Component	*Qty*	Parameter	Length
Shell	1	Diameter	90 mm
		Length	600 mm
Tube	7	Diameter	20 mm
		Length	600 mm
½ Segmental Baffle	5	Diameter	90 mm
		Thickness	8 mm
Spiral Baffle	1	Type	Rectangle
		Length	600
		Pitch	50

Table 1: Structural Geometry

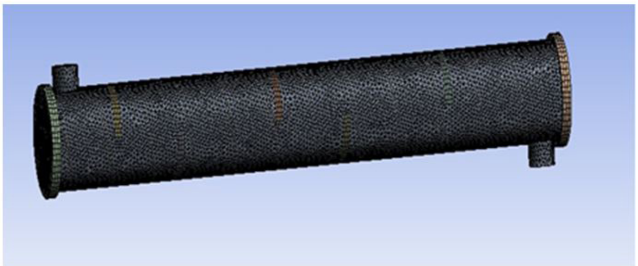


(a) (b)

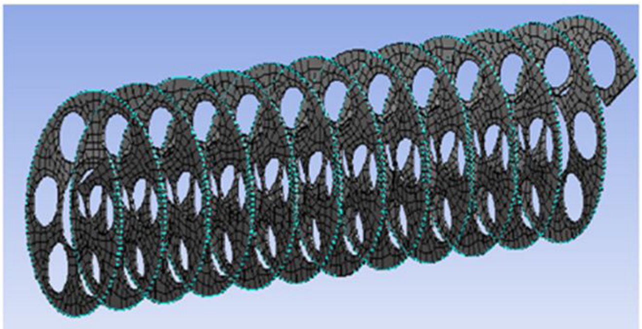
Figure 3: Geometrical Diagrams of Spiral and Segmental baffle with STHX a) Computational Model b) Internal Shell geometry Model

Mesh Generations

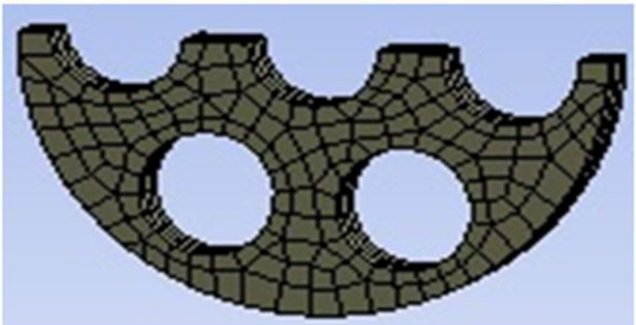
The geometric modelling and mesh generation are carried out with commercial Ansys 14.5. The mesh generations test is completed for each model. The mesh diagram for unit model, periodic model and whole model presented in Figure 4. The whole model is discretised with hexahedral and square meshes for the most flow region is with tetrahedral meshes for the segmental and spiral baffle region [5].



(a)



(b)



(c)

Figure 4: Meshing of Whole Model and Model Element a) Meshing Model b) Spiral Baffle Mesh c) Segmental Baffle

Boundary Condition

The main focus of the present study is the flow and heat transfer performance in the shell-side of STHXs.

Following assumptions are acceptable:

1. The flow and heat transfer is steady and turbulent;
2. The working fluid is incompressible;
3. The thermal-physical properties of the working fluid are considered as temperature dependent;
4. The thickness of baffles is neglect.

The boundary conditions are described as follows:

1. The shell side inlet: velocity inlet, u = w = 0, v = constant, Tin = 312, K (39 °C) (uniform inlet temperature), inlet turbulence intensity, I=10 %.
2. The shell side outlet: outflow outlet, $\frac{\partial u}{\partial n}=\frac{\partial v}{\partial n}=\frac{\partial w}{\partial n}=0$ and $\frac{\partial T}{\partial n}=0, \frac{\partial K}{\partial n}=0, \frac{\partial \in}{\partial n}=0$.
3. The heat exchange tube wall surfaces: non-slip boundary, u = v = w = 0, Tw = 373K (100 °C) (hot tube walls).
4. Other wall surfaces: non-slip boundary, u = v = w = 0, $\frac{\partial T}{\partial n}=0$.

The finite difference method and the second order upwind difference scheme are applied, and simple algorithm is adopted for coupling between pressure and velocity field. The second order schemes are momentum and energy computation and the standard difference scheme are available for pressure [6].

The convection coefficient of water with tube steel set as constant (380 W/m2 K) and upstream bulk temperature is set as a 312 K. The shell wall of the model is assumed to be non-heat exchange wall. Heat conduction of baffles in heat exchanger using the shell conduction is non-considerable thick wall in fluent [7]. The order setting parameters adopted defaults settings according to user guide in fluent. Water is set as a working fluid, and the corresponding parameters are presented in Table 2. The working fluid is incompressible, isotropic, Newtonian and continuous; the effect of gravity is negligible and viscous.

Parameter	Value with unit
Specific Heat Capacity (Cp)	4182 J/Kg
Dynamic Viscosity (μ)	0.001003 Kg/ms
Density (p)	998.2 Kg/m^3
Thermal Conductivity (k)	0.6 W/m k
Heat Transfer coefficient w.r.t. steel	380W/ m^2 K

Table 2: Thermo-Physical properties of water

Analytical Treatment

Table 3

Mass flow rate of hot water	0.4 kg/s
Mass flow rate of cold water	1.4 kg/s
Inlet temperature of hot water	40 °C-70 °C
Outlet temperature of hot water	70 °C-80 °C
Inlet temperature of cold water	27 °C
Outlet temperature of cold water	40 °C-60 °C

Table 3: Experimental Parameter

The log mean temperature difference (LMTD) between hot and cold water is calculated as follows:

$$\Delta t = \frac{(Thi - Tci) - (Tho - Tco)}{\text{In}\left(\frac{Thi - Tci}{Tho - Tco}\right)} \tag{1}$$

$$\Delta t = 44.85$$

The convective heat transfer rate between two fluids is

$$Q = hA\Delta t \tag{2}$$

$$Q = 7208.64$$

The effectiveness of the shell and tube heat exchanger

$$\in = \frac{tco - tci}{thi - tci} \tag{3}$$

$$\in = 0.705$$

Where, Q is heat transfer amount, A is heat transfer area, and h is a convective heat transfer coefficient. In the above calculations four different temperatures may vary in computational cases. As illustrated in R.K. Rajput [8], the difference between arithmetic means temperature differences (AMTD) between logarithmic temperature differences (LMTD). Due to parallel flow heat transfer as the temperature drop at one side of shell is written as the (thi - tci) and other end is (tho – tco). The result in this work using temperature differences is almost identical.

Result and Discussion

Effect of Segmental and Spiral Baffle

The path lines in shell side of heat exchanger are shown in Figure 5. It can be clearly observed that the fluid passes through the tube bundles.

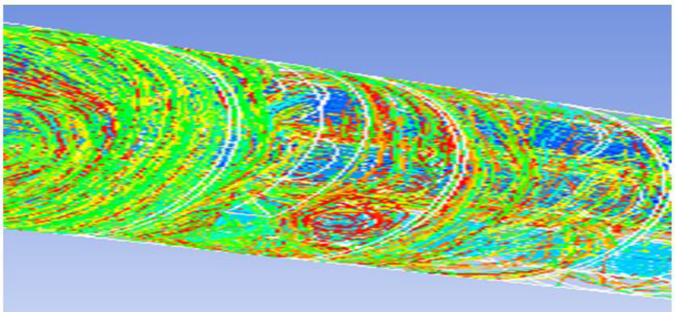


Figure 5: Pathline of Fluid Flow

The variation trends of pressure drop with mass flow rate are shown in Figure 6. It can be seen that the pressure drop increases with increasing mass flow rate due to spiral baffle. Even though the flow path of one cycle with spiral baffle is longer. The pressure drop increases through the decrease in baffle pitch below the 50 mm [9].

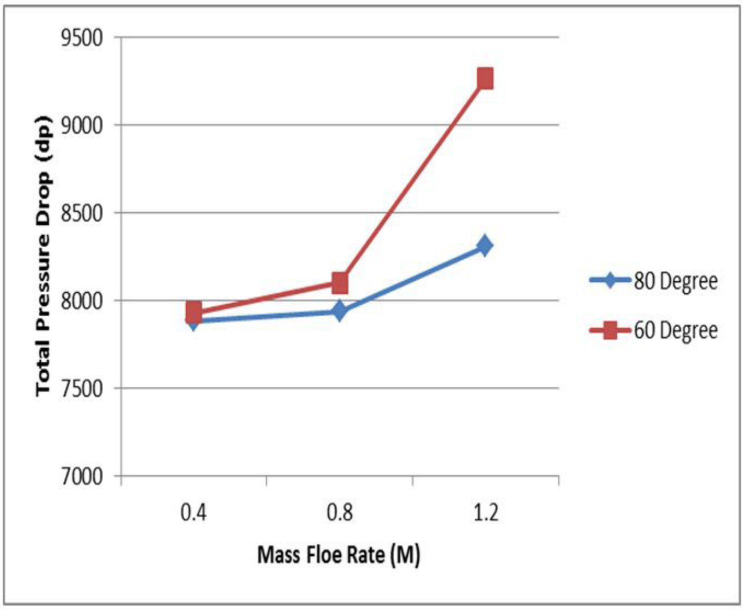


Figure 6: Pressure Drop (dp) VS Mass Flow Rate (M)

Figure 7 shows the comparison of shell side heat transfer among the periodic model within the range of mass flow rate tested. The result shows that the heat transfer rate increases with increase in mass flow rate and pitch of the spiral baffle. The variation trend can be understood from two aspects. First, at fixed shell inner diameter and helix pitch and also cross flow area decrease with diameter with decrease in helix pitch. Second decrease in helical angle of spiral baffle.

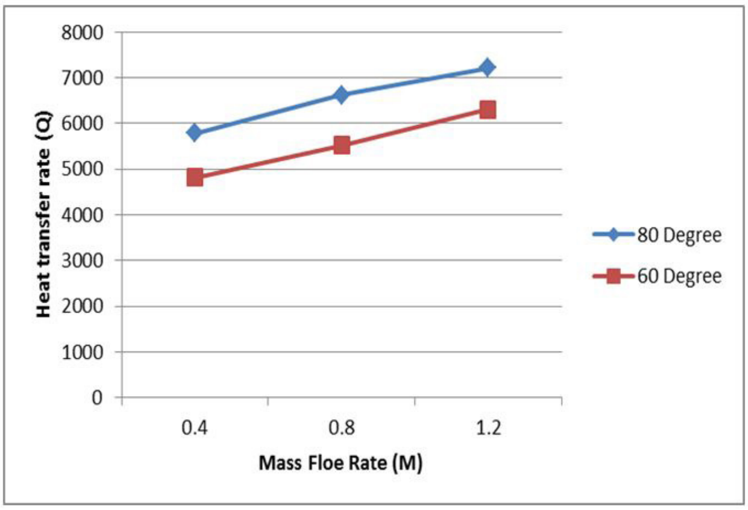


Figure 7: Heat Transfer Rate (Q) VS Mass Flow Rate (M) (at temperature 80 °C and 60 °C)

Figure 8 shows the comparison of shell side heat exchanger effectiveness among the periodic model within the range of mass flow rate tested. The result shows that the heat exchanger effectiveness increases with increase in mass flow rate. Due to increase in cold side temperature the heat exchanger effectiveness is increases. The variation trend can be understood from two aspects. First, at fixed shell inner diameter and helix pitch and also cross flow area decrease with diameter with decrease in helix pitch. Second decrease in helical angle of spiral baffle and no of segmental baffle increases in the shell side length [10].

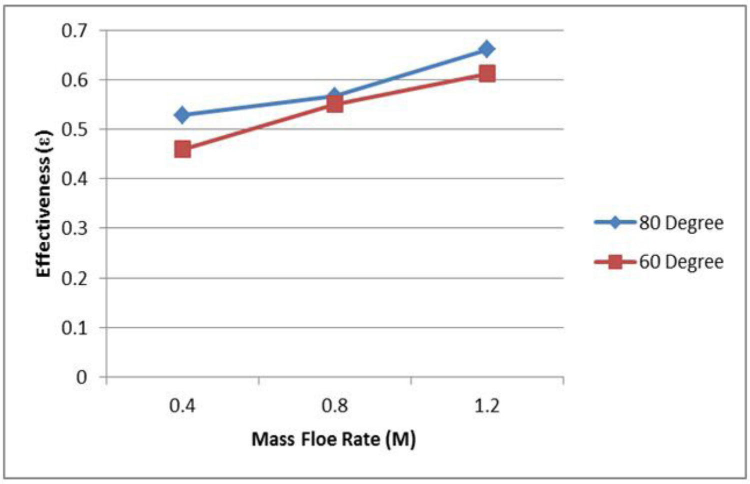


Figure 8: Heat Exchanger effectiveness (ε) VS Mass Flow Rate (M)

Conclusion

From numerical simulation of shell a tube heat exchanger with helical and segmental baffle it is observed that about pressure drop decreases along the length of tube, while the heat transfer rate and effectiveness is increased.

References

1. Peng B, Wang QW, Zhang C, GN Xie, LQ Luo, et al. (2007) An Experimental Study of Shell and Tube Heat Exchangers with Continuous Helical Baffles. ASME J Heat Transfer 129: 1425-31.

2. Wenjing DU, Hong Fu W, Lin C (2014) Effects of Shape and Quantity of Helical Baffles on the Shell Side Heat Transfer and Flow Performance of Heat Exchanger. Chin J Chem Eng 22: 243-51.

3. Wang QW, Chen GD, Xu J, Ji YP (2010) Second-Law Thermodynamic Comparison and Maximal Velocity Ratio Design of Shell-and-Tube Heat Exchangers with Continuous Helical Baffles. ASME J Heat Transfer 132: 101801.

4. Kral D Stehlik, P Van Der Ploeg HJ, Masters BI (1996) Helical Baffles in Shell-and-Tube Heat Exchangers, Part I: Experimental Verification. Heat Transfer Eng 17: 93-101.

5. Jie Yang, Lie Ma, Jessica Bock, Anthony M Jacobi, Wei Liu (2014) A comparison of four numerical modelling for enhanced shell-and-tube heat exchanger with experimental validation. Appl Therm Eng 65: 369-89.

6. Urade AR, Magdum SS (2017) A Review on Continuous helical baffle Shell and tube heat exchanger. National Conference in Research and Development in Mechanical Engineering pp. 24.

7. Jian Fei Zhang, Ya Ling He, Wen Quan Tao (2009) 3 D Numerical Simulation on shell and tube type heat exchanger with middle- overlapped helical baffle continuous baffle – Part- 2: Simulation result of periodic model and comparison between continuous and non-continuous helical baffle. Int J Heat Mass Transfer 52: 5381-9.

8. RK Rajput (2019) Heat and Mass Transfer. 3rd Edn, S Chand Publication.

9. Guidong Chen, Qui-Wang Wang (2009) Experimental and numerical studies of shell and tube heat exchanger with helical baffles. ASME Heat Transfer Equipment pp. 601-9.

10. Partha PS, Majumade A (2016) Continuous Helical Baffle Shell and Tube Heat Exchanger. J Therm Sci Eng Appl 8: 031002-1-031002.