Swanand Gaikwad* and Ashish Parmar

Numerical simulation of the effect of baffle cut and baffle spacing on shell side heat exchanger performance using CFD

https://doi.org/10.1515/cppm-2020-0033
Received April 21, 2020; accepted September 27, 2020; published online October 19, 2020

Abstract: Heat exchangers possess a significant role in energy transmission and energy generation in most industries. In this work, a three-dimensional simulation has been carried out of a shell and tube heat exchanger (STHX) consisting of segmental baffles. The investigation involves using the commercial code of ANSYS CFX, which incorporates the modeling, meshing, and usage of the Finite Element Method to yield numerical results. Much work is available in the literature regarding the effect of baffle cut and baffle spacing as two different entities, but some uncertainty pertains when we discuss the combination of these two parameters. This study aims to find an appropriate mix of baffle cut and baffle spacing for the efficient functioning of a shell and tube heat exchanger. Two parameters are tested: the baffle cuts at 30, 35, 40% of the shell-inside diameter, and the baffle spacing's to fit 6,8,10 baffles within the heat exchanger. The numerical results showed the role of the studied parameters on the shell side heat transfer coefficient and the pressure drop in the shell and tube heat exchanger. The investigation shows an increase in the shell side heat transfer coefficient of 13.13% when going from 6 to 8 baffle configuration and a 23.10% acclivity for the change of six baffles to 10, for a specific baffle cut. Evidence also shows a rise in the pressure drop with an increase in the baffle spacing from the ranges of 44–46.79%, which can be controlled by managing the baffle cut provided.

Keywords: baffle cut; baffle spacing; CFD; heat transfer coefficient; pressure drop; shell and tube heat exchanger.

1 Introduction

Heat Exchangers, in general, have been an integral part of the chemical and mechanical industry since the 18th century. They are used in many industries, such as space heating, refrigeration, power stations, etc. Energy conservation is an exigent issue; there is a dire need for advancements in the current methodologies to consume minimum and produce maximum. In the design of any heat exchanger, the area available for heat transfer and the heat transfer coefficient (U) obtained are of prime importance. Active and passive methods are in study to improve the heat transfer coefficient. Active methods mainly include external power input to enhance heat transfer in the form of surface vibrations, electrostatic fields, jet impingements, etc. On the contrary passive methods consist of surface or geometrical modifications to the flow channel by incorporating inserts or additional devices increasing the transfer surface area or by introducing forms of turbulence, etc. There are many types of heat exchangers in service in the industry, such as plate heat exchangers, micro-heat exchanger, shell and tube heat exchangers, double-pipe, etc. Amongst these, the shell and tube heat exchangers are the most profitable and hold up to 40% of market share in petrochemical and process industries [1]. The present paper focuses on the improvement of the shell side characteristics of the heat exchanger by the introduction of baffles which direct the flow and enhance the heat transfer coefficient by creating turbulence [2].

***Corresponding author: Swanand Gaikwad**, Department of Chemical Engineering, Sardar Vallabhbhai National Institute of Technology, Gujarat, 395007, Surat, India, E-mail: swanandgaikwad2021@gmail.com
Ashish Parmar, Department of Chemical Engineering, Sarvajanik College of Engineering Technology, Gujarat, 395001, Surat, India

A lot of study on the effect of baffles on the heat transfer enhancement can be found in the literature; round-cut baffles called as segmental baffles are employed, which also provide support to the tubes and intensify the fluid flow, but this heat transfer enhancement comes at the expense of increased pressure drop. It can, at times, form dead zones near the edges of the baffles, but an apt balance between the increase in the heat transfer coefficient and pressure drop, if maintained, can be beneficial in processes. Wang et al. propose an innovative approach to block the triangle zones where leakage occurs in shell side of STHX. With the usage of these types of fold baffles, it was reported that the overall heat transfer coefficient was improved by 7.9–9.7%; moreover, it concluded that the aspect of optimization of baffle configuration could be explored to increase heat transfer rate and lower pressure drop [3]. A study on the effect of baffle spacing only to the performance in an oil cooler was done by Zebua et al. The results showed that temperature effectiveness decreases with increased baffle spacing at the same time, the heat transfer coefficient was higher when the baffle spacing was lower as compared to the other cases for a variety of inlet temperatures [4]. Bhalkikar et al., in his work on the study of baffle cut on pressure drop and heat transfer coefficient, concluded that the shell side heat transfer coefficient is indeed affected by the baffle geometry. Still, not much difference was observed between 25 and 30% baffle cut in terms of heat transfer coefficient; the pressure drop was though lower for the latter case and preferable [5]. A study by Mustapha Mellal et al. on shell side performance in an STHX under different baffle arrangement and orientation revealed that a change in the baffle spacing leads to a change in the heat exchanger performance. A decrease in the baffle spacing increased the Nusselt number and friction factor. He also shed light on the importance of baffle orientation and stated that the best performance in terms of heat transfer coefficient was obtained with 180° inclination, creating a zig-zag flow mode with a short bypass [6].

A study of baffle space impact on the performance of helical baffle in an STHX has also been done, in which Taher et al. concluded that for a helical baffle, pressure gradient was observed to increase with a decrease in the baffle spacing provided. If the mass flow rate and working conditions were kept constant, it was observed that longer baffle spacing resulted in lower heat transfer coefficients upon simulation [7]. Zhang et al. studied the helix angles of a helical baffle varying from 10° to 30°, comparing them to conventional segmental baffles showing that the helical baffles yielded a higher heat transfer coefficient for the same pumping power, 10° having the highest pressure drop and heat transfer coefficient but the worse performance and 30° having the lowest values in the parameters mentioned above [8]. A combination of the parameters, namely baffle cut and baffle inclination, was performed by Govindaraj Kumaresan et al., whose results showed that a blend of 30% baffle cut and 35° baffle inclination provides a high heat transfer rate coupled with minimum pressure drop compared to other baffles at 25°,30° and 40° inclinations and the conventional segmental baffle [9]. Also, as a general observation that the inclined baffles performance is enhanced than the conventional segmental baffle. The types of baffles such as single segmental, double segmental, and helical baffles were also juxtaposed and compared with the help of numerical simulations by Pranita Bichkar et al., who observed that the pressure drop was lower in the case of helical baffles due to the elimination of dead zones and the vibrational damage was reduced when the double segmental baffles were employed, the less dead zones formed in helical baffles accounted for a better heat transfer rate. [10], Usman Salahuddin et al. do work. Reviewed significant work done on the helical baffles and compared that to the segmental baffles and discussed how in most cases, discontinuous, sextant, folded at 40° inclination and low baffle spacing in some order would give the best results; he also suggested that sealing strips would most likely help to improve the performance of shell and tube heat exchangers with continuous helical baffles [11].

Passive methods of heat transfer augmentation are also popular, following the work of Sarafraz et al., nanofluids are a contemporary method used to intensify forced convection, they influence the pressure drop and friction factor, the heat transfer enhancement was achieved to a numeric of 67% [12]. Tiwari et al. do further research on the effect of different types of nanofluids. in his work using a plate heat exchanger and passing different nanofluids through the system and concluding that the CeO_2/water yields the best performance amongst the studied four nanofluids [13]. A study by L. Cheng et al. describes the use of vibrations, which are considered detrimental as a heat transfer enhancement technique and how it decreases the fouling resistance and increases the heat transfer coefficient [14]. From the citied literature review, it is evident that continuous efforts are being made to optimize the performance of heat exchangers and design is one such parameter on

Table 1: Dimensions of shell and tube heat exchanger.

Description	Values
Heat exchanger length	600 mm
Tube diameter	20 mm
Shell side diameter	90 mm
Tube pitch	Triangular, 30 mm
Baffle cut's	30, 35, 40%
No. of baffles included	6, 8, 10
No. of tubes	7

which work can be done; also, there is work available on baffle spacing and the effect of baffle cut on shell side performance separately, but not much work is done in the domain of their combination, wherein both the parameters are studied simultaneously. In the present work, an effort has been made to observe the effect of baffle spacing wherein 6, 8, 10 baffles are accommodated in the same space along with a baffle cut of 30, 35, and 40%, to find the optimum mixture of these parameters for the efficient performance of an STHX using the Computational Fluid Dynamics tool, CFX.

2 Geometric modeling and methods

2.1 Geometric modeling and meshing

The three-dimensional geometric modeling is done by the design-modeler module present in ANSYS as per the parameters listed in Table 1. The wireframe model is shown in Figure 1, and the geometric model with boundary

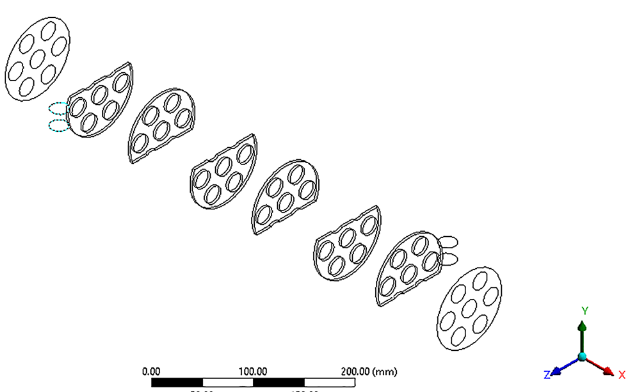


Figure 1: Wireframe model of heat exchanger.

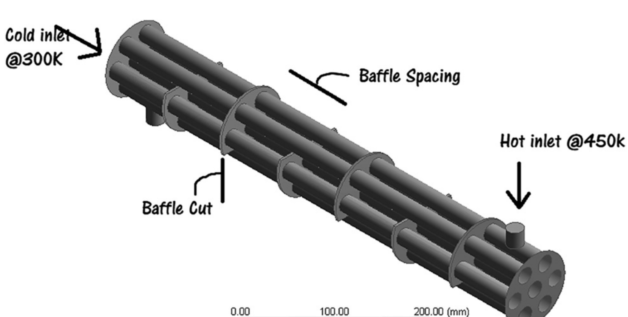


Figure 2: Model of shell and tube heat exchanger.

conditions is shown in Figure 2. The no. of baffles varies as the spacing between two baffles changes. The tubes are arranged in a triangular pitch manner, and different values for the baffle cut, namely 30, 35, 40%, are used in geometric modeling.

For the model generated meshing is done with the help of the Mesh module present in ANSYS, discretization of the entire domain has been done using CFD, and the geometry is divided into solid and fluid domains. The fluid domain meshing has been kept finer than prior. The entire model is discretized using the hexahedral meshing, as it requires less computational effort. The fluid domain has more refined elements compared to the rest of the model.

The complete geometry is divided into six parts, namely inlet, outlet, wall, shell, fluid, baffle. Complete effort of using the most refined mesh and the smallest elements has been made keeping in mind the hardware capabilities of the machine used. The orthogonal quality tells us about the fraction of formed elements and the types of elements formed; a test of the orthogonal quality showed that most of the elements were a complete Hexahedral with a total number of 358965 nodes and 1473646 elements. The average skewness ratio was obtained to be 0.22. The meshed model of the STHX is shown in Figure 3, and a glimpse of the meshing done on the pipe is shown in Figure 4.

2.2 Governing equations

The fluid flow and heat transfer analysis use the fundamental laws of conservation of mass, momentum, and energy. The mathematical expression governing the physical phenomenon of incompressible flow can be written as [9, 15]:

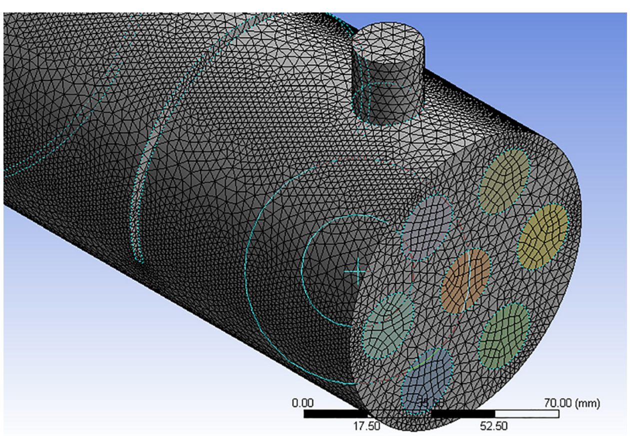


Figure 3: Surface meshing of STHX.

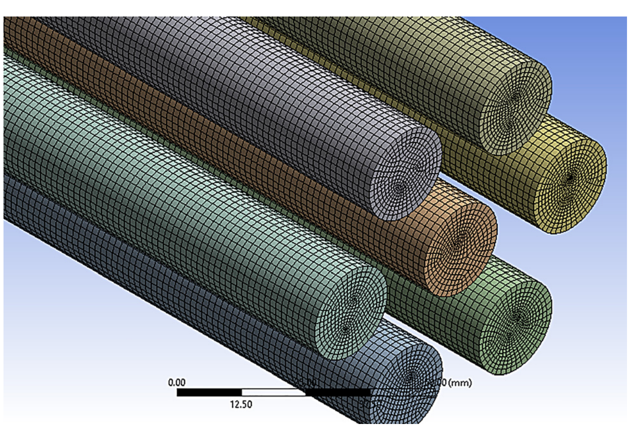


Figure 4: Meshing type of pipe.

$$\textit{Continuity equation}: \nabla.\vec{V} = 0 \tag{1}$$

$$\textit{Momentum equation}: \frac{\partial V}{\partial t} + \left(\vec{V}.\nabla\right)\vec{V} = -\frac{1}{\rho}\nabla p + \upsilon\nabla^2\vec{V} \tag{2}$$

$$\textit{Energy equation}: \rho C_p\left[\frac{\partial T}{\partial t} + \left(\vec{V}.\nabla\right)T\right] = k\nabla^2 T + \varnothing \tag{3}$$

The standard k-ε turbulence model is being used, and the shell side heat transfer coefficient cannot be calculated using the correlations used for flow through a tube. Hence we estimate the parameters G_b, the mass velocity of the shell side fluid as if it were all moving parallel to the tubes, and G_c, the mass velocity of the shell side fluid as if it were all moving across the tubes.

The required mass velocities become [16–18],

$$G_b = \frac{\dot{m}}{\frac{\pi}{4}\left(f_B D_s^2 - N_{bt} d_0^2\right)} \tag{4}$$

$$G_c = \frac{\dot{m}}{P_B D_s\left(1 - \frac{d_0}{p_t}\right)} \tag{5}$$

which can be then used to find the shell side heat transfer coefficient

$$Nu_o = \frac{h_o d_o}{k} = 0.2\left(\frac{d_o\sqrt{G_b G_c}}{\mu}\right)^{\hat{}0.6} Pr^{0.3} \tag{6}$$

where [16],
- D_s, shell inside diameter
- f_B, the fraction of the shell cross-section that makes up the baffle window
- h_o, shell side heat transfer coefficient
- μ, viscosity
- N_{bt}, the number of tubes in the baffle window
- P_B, the baffle pitch
- P_t, the baffle pitch
- d_o, the tube outside diameter

2.3 Operating conditions and assumptions

The operating conditions of the STHX used in the analysis are tabulated in Table 2. A counter-current flow operation has been set up with the same fluid at constant physical properties at the inlet and outlet. No-slip wall condition has been assumed for the baffle and the shell walls, and the shell is assumed to be perfectly insulated and to make the flow pattern in the shell side simpler, we have made the following assumptions:-
(1) Steady-state heat transfer is assumed, and the flow in the shell side is turbulent.
(2) Natural convection induced by the fluid is neglected.
(3) The heat exchanger is insulated, and hence the heat loss and the interaction to the surrounding are neglected.
(4) An interface between the tubes and the shell side fluid is maintained.

The governing equations are discretized using the first wind up scheme, all of the algebraic equations are solved iteratively using the SIMPLE algorithm. The CFX software used the finite volume method for solving the mass, momentum, and energy conservation equations stated above.

Table 2: Operating conditions used for analysis.

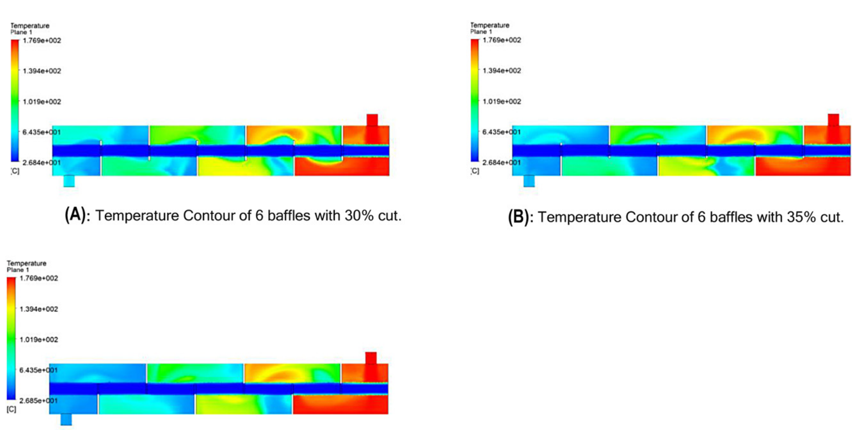
Quantity	Values
Fluid on the shell side	Water
Fluid on the tube side	Water
Tube inlet temperature	300 K
Shell inlet temperature	450 K
Velocity on both sides	1 m/s
Flow pattern	Counter-current
Wall thickness	2 mm
Wall material	Copper
Mass and momentum	No-slip wall

3 Results and discussions

The results of the numerical simulation with baffle cuts of 30, 35, 40%, and baffle spacing to accommodate 6, 8, 10 baffles in an STHX are discussed below, mainly focusing on the shell side heat transfer coefficient when compared to one another. Also, a discussion on the variation in pressure has been done for the studied cases. A linear relationship between the increase in the number of baffles and the shell side heat transfer coefficient was expected, but the results outline slight modifications to that theory; also, the observed results showed that it was not necessary to keep reducing the baffle spacing only, for target heat transfer coefficient, the role of baffle cut to modify this heat transfer coefficient and pressure drop could also be considered.

3.1 Shell side heat transfer coefficient

The shell side heat transfer coefficient is calculated using the formulas provided for the mass velocities. It is observed that as the number of baffles along with the heat exchanger increases, the effect of the baffle cut as a parameter starts playing a significant role in the shell side heat transfer coefficient and the heat transfer coefficient rises with the inclusion of more baffles along the length of the heat exchanger. The lowest value of 144.225 kW/m^{2}K was observed for a six baffle with 40% cut, and the highest value of 195.722 kW/m^{2}K was encountered for 10 baffles with a 30% cut system, Figures 5A–7C, gives an insight into the temperature contour of the studied cases.

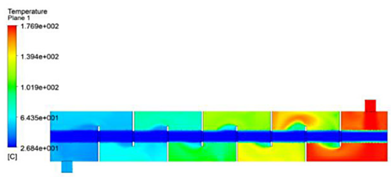


(A): Temperature Contour of 6 baffles with 30% cut.

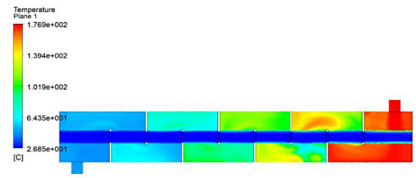
(B): Temperature Contour of 6 baffles with 35% cut.

(c): Temperature Contour of 6 baffles with 40% cut.

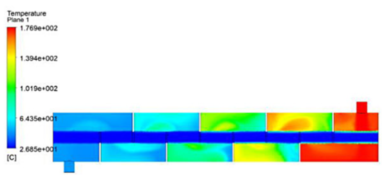
Figure 5: (A) Temperature Contour of six baffles with 30% cut. (B) Temperature Contour of six baffles with 35% cut. (C) Temperature Contour of six baffles with 40% cut.



(A): Temperature Contour of 8 baffles with 30% cut.

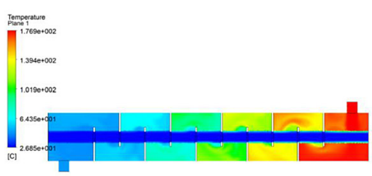


(B): Temperature Contour of 8 baffles with 35% cut.

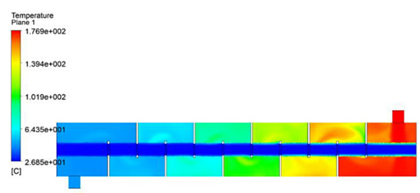


(C): Temperature Contour of 8 baffles with 40% cut.

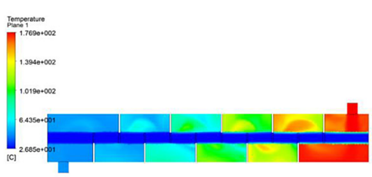
Figure 6: (A) Temperature Contour of eight baffles with 30% cut. (B) Temperature Contour of eight baffles with 35% cut. (C) Temperature Contour of eight baffles with 40% cut.



(A): Temperature Contour of 10 baffles with 30% cut.



(B): Temperature Contour of 10 baffles with 35% cut.



(C): Temperature Contour of 10 baffles with 40% cut.

Figure 7: (A) Temperature Contour of 10 baffles with 30% cut. (B) Temperature Contour of 10 baffles with 35% cut. (C) Temperature Contour of 10 baffles with 40% cut.

The temperature contours show how the fluid at 450 K at the shell side inlet is being cooled due to the presence of passing cold fluid in the tube side; there are different flow patterns generated in correspondence to the baffle spacing and the baffle cut used in the heat exchanger. A change in the flow pattern is observed mainly due to the baffle cut, which indirectly controls the velocity of the fluid, and velocity being a function of the heat transfer coefficient affects the STHX's performance. The heat transfer coefficient in the shell side is improved due to the effective circulation of shell-side fluid around the bank of tubes. The fluid would comparatively take a more extended amount of time to pass across a baffle with a lower percentage of baffle cut. The heat transfer coefficients calculated gives us information that it is not necessary to increase the number of baffles when a specific value of baffle cut can provide us with approximately the same shell side heat transfer coefficient; for instance, in this study, the heat transfer coefficient obtained in an STHX for an eight baffle, 30% cut is close to that obtained for a 10 baffle, 40% cut, which indicates that the prior can be installed for the better economy of the HX the relation between the baffle cut and spacing of baffles was also observed and how the heat transfer coefficient depends on it and the significance of baffle cut upon an increase in the baffle spacing can be best culminated by Figure 8. It is also observed that as the baffle spacing is reduced in the heat exchanger, the effect of change in the baffle cut assumes more importance, i.e., the difference in readings of various baffle cuts increases as the baffle spacing reduces.

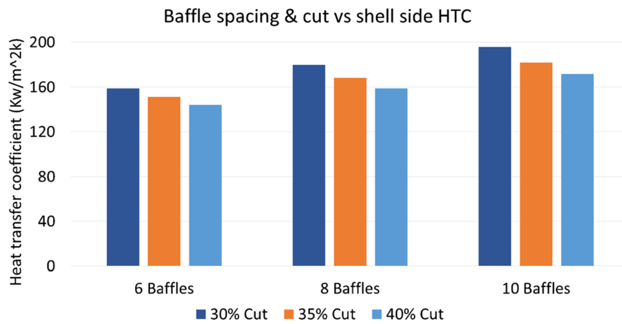


Figure 8: Graphical representation of heat transfer coefficient.

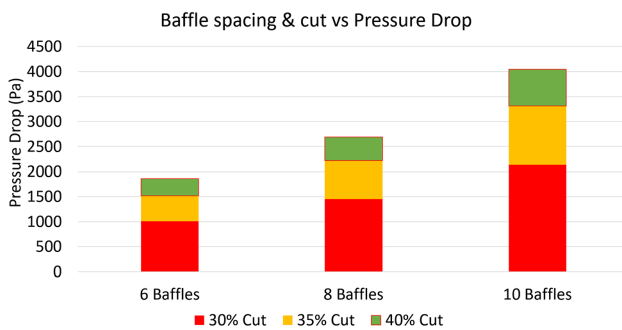


Figure 9: Graphical representation of Pressure Drop.

3.2 Variation in pressure distribution

The CFD analysis is performed for all the configurations. As the pressure drop is of prime importance while designing a heat exchanger (HX), a higher net pressure drop means that more pumping power is required to ensure the fluid passes through the HX, which indirectly increases operating costs; a high-pressure stream is favorable because it provides minimal fouling which itself is a significant problem in heat exchangers. The high-pressure drop is induced due to the combination of vertical and horizontal flow directions of the fluid. The pressure drop in the system with 10 baffles and 30% cut showed the highest value of 2133.154 Pa; this can be accounted for by the fact that the more the number of turns the fluid has to take, the more pumping power required. The lowest pressure drop of 343.501 Pa was observed in the six baffles system with a 40% cut, which can be explained by the ease with which the fluid could cover the length of the heat exchanger and the relative abundance of the area available for the fluid to pass through. It is worth noting that the pressure drop in the case of 10 baffles, 40% cut system was lower than the one observed in eight baffle system with 35% cut, which implies that the pressure drop doesn't need to increase concomitantly as the number of baffles increase, adjusting the baffle cut in an exchanger system can also help alleviate pressure drop losses. Also, a decrease in the baffle cut showed higher values of pressure drop for an equal number of baffles as the shell side fluid would have more difficulty passing through the given space or area owing to a high-pressure drop. The pressure drop calculation is done by taking the difference in the pressure values at the two ends of the shell side in an STHX. Figure 9 gives us a graphical representation of the trends of pressure drops observed in this numerical investigation. Upon closer inspection, it can be stated that the gap between values of pressure drop with different baffle cuts for the same number of baffles increases rather quickly as the number of baffles in the system increases, which should be verified by further research and applicatory when designing heavy-duty exchangers.

It can be interpreted from the Figures 8, 9 that a high heat transfer coefficient on the shell side comes at the expense of a higher pressure drop, and an adequate balance between these two parameters shall be maintained for the best possible configurations out of the studied 9 cases. A higher number of baffles can lead to an increase in the operating costs but can be recovered by the efficiency offered in terms of the heat transfer coefficient. Also, in some instances, from the results provided, it can be assessed that a system with more number of baffles and a higher baffle cut can be chosen to save the costs due to pressure drop. As a higher number of baffles are available, it can be assumed that the shell side heat transfer coefficient shall be higher in magnitude.

4 Conclusions

In the present work, the pressing need of the heat exchanger industry, the tradeoff optimization between the pressure drop and the heat transfer coefficient has been studied to provide an idea of the effects of the change in two parameters, namely, baffle spacing and baffle cut simultaneously and observe how these two will affect the performance of an STHX. An attempt to find the optimum geometric configuration has been carried out. The results can be concluded further as the shell side heat transfer coefficient increases with a decrease in the baffle spacing, and a higher baffle cut reduces the heat transfer coefficient on the shell side; also the significance of baffle cut on the shell side heat transfer coefficient increases as the baffles spacing reduces the investigation shows an increase in the shell side heat transfer coefficient of 13.13% when going from 6 to 8 baffle configuration and a 23.10% acclivity for the change of six baffles to 10 The pressure drop is also discussed which indicates that a reduction in the baffle spacing leads to an increase in the pressure drop and an increase in the baffle cut reduces the pressure drop. An optimization of 49.13% is observed in pressure drop when we slightly tweak the baffle cut by 5% for a given six baffle system. The study is mainly beneficial in elucidating that a higher heat transfer coefficient can be obtained for fewer baffles when coupled with the appropriate baffle cut, and similar behavior can be observed for the pressure drop parameter. It can be interpreted and suggested that the right balance between both the parameters should be maintained for attaining the most feasible configuration.

Nomenclature

CFD	Computational Fluid Dynamics
STHX	Shell and Tube Heat Exchanger
T	Temperature (K)
C_p	Specific heat at constant pressure (J/Kg K)
$\vec{V}$	Velocity Vector
ρ	Density (kg/m^3)
$\varnothing$	Viscous dissipation
k	Turbulence kinetic energy (m^2/s^2)
p	Pressure (N/m^2)
Nu_o	Shell side Nusselt number
Pr	Shell side Prandtl number

Acknowledgments: We would like to express our gratitude to the institute SVNIT for sharing their resources with us during the course of this research and thank our families,we hope to dwell deeper into the fields of engineering and science in the future.
Author contributions: All authors have accepted responsibility for the entire content of this manuscript and approved its submission.
Research funding: None declared.
Competing interests: Authors state no conflict of interest.

References

1. Shah RK, Sekulic DP. Fundamentals of heat exchanger design New Jersey: John Wiley & Sons, Inc.; 2003.
2. Incropera FP. Fundamentals of heat and mass transfer. 7th ed. New Jersey: John Wiley; 2011.

3. Wang S, Wen J, Yang H, Xue Y, Tuo H. Experimental investigation on heat transfer enhancement of a heat exchanger with helical baffles through blockage of triangle leakage zones. Appl Therm Eng 2014;67:122–30.
4. Ambarita H, Zebua MA. Numerical Simulation of the effect of baffle spacing to the effectiveness of a shell and tube heat exchanger. In IOP conference series: materials science and engineering; 2018; Indonesia.
5. Chetan Namdeo P, Bhalkikar NS. CFD analysis of shell and tube heat exchanger to study the effect of baffle cut on the pressure drop and heat transfer coefficient. Int J Sci Res Dev 2014;2:649–54.
6. Mellal M, Benzeguir R, Sahel D, Ameur H. Hydro-thermal shell side performance evaluation of a shell and tube heat exchanger under different baffle arrangement and orientation. Int J Therm Sci 2017;121:138–49.
7. Taher FN, Movassag SZ, Razmi K, Azar RT. Baffle space impact on the performance of helical baffle shell and tube heat exchangers. Appl Therm Eng 2012;44:143–9.
8. Zhang M, Meng F, Geng Z. CFD simulation on shell and tube heat exchangers with small-angle helical baffles. Front Chem Sci Eng 2015;9:183–93.
9. Kumaresan G, Santosh R, Duraisamy P, Venkatesan R, Kumar NS. Numerical analysis of baffle cut on shell side heat exchanger performance with inclined baffles. Heat Tran Eng 2017 September;39:1156–65.
10. Bichkar P, Dandgaval O, Dalvi P, Godase R, Dey T. Study of shell and tube heat exchanger with the effect of types of baffles. In 2nd International conference on materials manufacturing and design engineering; 2018; Aurangabad:195–200 pp.
11. Salahuddin U, Bilal M, Ejaz H. A review of the advancements made in helical baffles used in shell and tube heat exchangers. Int Commun Heat Mass Tran 2015;67:104–8.
12. Sarafraz MM, Hormozi F. Intensification of forced convection heat transfer using biological nanofluid in a double-pipe heat exchanger. Exp Therm Fluid Sci 2015;66:279–89.
13. Tiwari AK, Ghosh P, Sarkar J. Performance comparison of the plate heat exchanger using different nanofluids. Exp Therm Fluid Sci 2013;49:141–51.
14. Cheng L, Luan T, Du W, Xu M. Heat transfer enhancement by flow-induced vibration in heat exchangers. Int J Heat Mass Tran 2009;52:1053–7.
15. Ajithkumar MS, Ganesha T, MC M. CFD analysis to study the effects of inclined baffles on fluid flow in a shell and tube heat exchanger. Int J Res Advent Technol 2014;2:164–75.
16. Price RM. Shell and tube heat exchangers: calculations. [Online]; 2003 [cited 2020 March 10/03/2020]. Available from: http://facstaff.cbu.edu/rprice/lectures/stcalcs.html.
17. Brodkey RS, Hershey HC. Transport phenomena: a unified approach. New York: McGraw-Hill; 1988.
18. Kern DQ. Process heat transfer. India: McGraw-Hill; 1950.

Supplementary material: The online version of this article offers supplementary material (https://doi.org/10.1515/cppm-2020-0033).