Optimization research of sextant fan baffle curvature radius in shell and tube heat exchanger

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Abstract. For a high shell side pressure drop of the conventional segmental baffles in shell and tube heat exchanger, a novel sextant fan baffle was put forward. To research the influence of baffle curvature radius of the sextant fan baffled shell and tube heat exchanger (SFTHX) on the shell side pressure drop, the heat transfer coefficient and the comprehensive heat transfer performance, six different curvature radius baffles were numerically simulated and experimental studied in this paper. Based on the numerically simulation results, under the same inlet flow conditions, a better comprehensive heat transfer performance can be found in SFTHX with the baffle curvature radius of 1 D, which is higher by 0.84-6.85% more than that of the others. Moreover, the experimental investigation data of SFTHX with baffle curvature radius of 1 D indicates that the numerically simulation can well predict the flow and heat transfer characteristics with the experiment.

1 Introduction
Shell and tube heat exchangers (STHXs) have been the most widely used equipment in chemical industrial fields [1, 2]. In its structure, baffles are one of the most important parts. The function of baffles is that they force the fluid of shell side to flow across the tubes to ensure high heat transfer rates and provide support for tube bundle. The most commonly used baffles, called segmental baffles, cause the shell side fluid to flow in a zigzag manner across the tube bundle. The action improves heat transfer by enhancing turbulence or local mixing on the shell side; however, it also increases the shell side pressure drop and requires a great pumping power. High range of dead zones, back flows and high risk of vibration failure on the tube bundle are other disadvantages of the conventional segmental baffles. Then, some other types of baffle arrangements, such as circle orifice baffle, overlap helical baffles, helical baffle [3-6], were developed and applied in STHXs to minimizes the principle shortcomings in designing of the conventional segmental baffles, which cause reduction in shell side pressure drop and improves heat transfer performance.

In this paper, based on the investigation of the sextant fan baffled shell and tube heat exchanger [7], six baffles with different curvature radius were simulated to focus on the curvature radius impact on the heat transfer performance of SFTHX. The chosen baffles with different curvature radius are: for SFTHX-A: 0 D, for SFTHX-B: 0.5 D, for SFTHX-C: 0.75 D, for SFTHX-D: 1 D, for SFTHX-E: 1.25 D, and for SFTHX-F: 1.5 D. Moreover, the heat transfer performance of SFTHX with the optimized curvature radius baffle was investigated in the experimental study.
2 Numerical simulation

2.1 Geometric configuration
Table 1 shows the structural parameters of SFTHX, Figure 1 gives the geometry model of the sextant fan baffle, and Figure 2 displays the installation way of the sextant fan baffle with different curvature radius in the simulation model.

![Geometry model of the sextant fan baffle](image1)

**Figure 1.** Geometry model of the sextant fan baffle.

| Table 1. Structural parameters of SFTHX. |
|------------------------------------------|
| Diameter  | mm  | 209 |
| Length    | mm  | 1886|
| Tube number  | --  | 22  |
| Tube Arrangement Mode  | --  | Regular triangle |
| Inlet Diameter of Shell-side  | mm  | 50  |

![Installation way of sextant fan baffle](image2)

**Figure 2.** Installation way of sextant fan baffle.

2.2 Numerical simulation method
During the numerical simulation with Fluent software package, the major assumptions are: (1) The flow is incompressible and steady; (2) Thermal radiation and natural convection are neglected; (3) SFTHX has ideal surfaces and uniform surface dimensions throughout the array; (4) The structure of impingement plate, spacer and tie rod should be simplified; (5) The interval between the baffle plate and the tube, as well as the interval between the baffle plate and the shell should be neglected; (6) The flow regime and the heat transfer follow the continuity equation, energy equation and momentum equation.

During the numerical simulation, considering the quality and adaptive, Hyper Mesh meshing using unstructured tetrahedral mesh was adopted, and the appropriate grid unit size was selected to guarantee the minimum grid orthogonal quality of 0.25. RNG $k-\varepsilon$ turbulence model was taken based on the preliminary work [8]. In addition, segregated implicit was employed to guarantee the stability of the calculation convergence, SIMPLE algorithm for Pressure-Linked and Velocity-Linked Equations was used to couple the continuity and Navier-Stokes equations, the wall function was introduced for con-
sidering the influence of the boundary layer on the turbulent flow of the fluid, second order upwind method was adopted to deal with the momentum, turbulent kinetic energy and turbulent dissipation rate [9]. Furthermore, in SFTHXs, the degree of turbulence were applied for the import fluid, and the free boundary condition was adopted for the export fluid.

2.3 Heat transfer coefficient
Heat transfer coefficient versus the shell side volumetric flow rate within 4 to 10 m$^3$/h for SFTHX-A to SFTHX-F is represented in Figure 3. It can be observed that the heat transfer coefficient increases by the increase of volumetric flow rate. In detail, it is found that there is a slight difference among the six SFTHXs at lower volumetric flow rate, and the difference becomes obvious along with the increasing volumetric flow rate. Among the six SFTHXs, SFTHX-D shows a better heat transfer coefficient, which is higher by 0.74%-6.85% than the others. Hence, it can be deduced that the curvature radius play a certain effect on the heat transfer coefficient in SFTHXs.

![Figure 3. Heat transfer coefficient of SFTHXs.](image)

2.4 Pressure drop
Pressure drop is of great importance in design of SFTHXs because pumping costs are highly depended on pressure drop. Therefore, a lower pressure drop leads to a lower operating costs. Figure 4 indicates the variation of pressure drop versus the shell side volumetric flow rate within 4 to 10 m$^3$/h. It worth mentioning that, the parameter represents the most proper comparison regardless of tubes lengths. Note that, Figure 4 shows the rising pressure drop at increasing volumetric flow rate. Meanwhile, a lower pressure drop of shell side is caused by SFTHX-D. However, similar with the numerically simulation results of the heat transfer coefficient, the pressure drop has a sight difference among the six SFTHXs.

![Figure 4. Shell side pressure drop of SFTHXs.](image)

2.5 Comprehensive heat transfer
J-F factor is selected to represent the comprehensive heat transfer performance of SFTHXs. The relationships between J-F factor and the volumetric flow rate in Figure 5 shows that J-F factor decreases slowly with the increasing volumetric flow rate of shell side. In every SFTHX with the different curvature radius baffle, the variation tendency is related to the characteristics of fluid flow and heat transfer. Compared with the other SFTHXs, SFTHX-D leads to a better comprehensive heat transfer per-
formance. In the range of the volumetric flow rate within 4-10 m³/h, the comprehensive heat transfer performance for SFTHX-D is 0.84-6.85% greater than that of the others.

3 Experimental investigation

3.1 Experimental process

The installation way of SFTHX-D in the experimental study is shown in Figure 6, and the schematic diagram of the cold model experimental process of SFTHX-D is illustrated in Figure 7. In the experiment, the water inlet temperature of shell side was the ambient temperature and the water inlet temperature of tube side was 335 K adjusted by the temperature control cabinet. Meanwhile, the shell side volumetric flow rate was in the range of 4 to 10 m³/h.

3.2 Comparison of the experimental investigation and numerically simulation

One of the important purposes of the comparison of the experimental investigation and numerically simulation was to develop a model that can be verified with the experimental data in order to use the simulation results in the situation not supported by experiments. Figure 8 shows the comparison of J-F factor between the numerical simulations and the experimental data for SFTHX-D.
In Figure 8, J-F factors decreases with the increasing volumetric flow rate under the experimental condition. And numerical simulation results of J-F factor follow very similar trends to the experimental data. From Figure 8, it also can be seen that J-F factor from the experimental investigation is higher than that from the numerically simulation. In addition, the numerical calculation for J-F factor are at most 6.66% different from the experimental results. The reasons for the difference in J-F factor can be considered as follows: (1) Experiment uncertainty factors. There are some uncertainty factors in the experiment, such as the measurements of temperatures, pressure drops and volumetric flow rates; (2) Rough surface. The baffles and some other inner structure in SFTHX have a rough surface, which are not considered in the numerical models; (3) Baffle shape uncertainty. Due to manufacturing tolerances, distinct geometrical discrepancies exist between actual and simulated baffles. In addition, the geometrical uncertainty factors mainly focuses on baffle height, curvature radius, baffle space and baffle length. However, uniform surface dimensions throughout the array are assumed in the numerical simulations; (4) Simplification and neglection. There are some simplified and neglected factors of the inner structure in the numerical models of SFTHX, which are existed in the experimental study.

![Figure 8. Comparison of J-F factor between experimental data and numerical simulation.](image)

Although there is a difference between the experimental data and the numerically simulation results, however, it can be found that the above presented numerical method can be successfully predicted the heat transfer performance of SFTHX with different curvature radius baffle. That is to say, the numerically simulation can well predict the flow and heat transfer characteristics with the experiment.

4 Conclusions

In this paper, the numerically simulations of SFTHXs with different curvature radius baffle were conducted to investigate the characteristics of the pressure drop of shell side, the heat transfer coefficient and the comprehensive heat transfer performance. From the numerically simulation results, curvature radius has a certain effect on the characteristics of fluid flow and heat transfer performance. Among SFTHXs, the heat transfer coefficient in SFTHX-D is about 0.74%-6.85% higher than that of the others under the velocities of shell side within 4-10 m$^3$/h. In addition, there is a sight difference of the pressure drop among six SFTHXs. Taken together, the comprehensive heat transfer performance of SFTHX-D is about 0.84-6.85% higher than that of the others, which displays a better comprehensive heat transfer performance. Furthermore, the comprehensive heat performance of experimental investigation for SFTHX-D was performed in the cold model experimental process. From the experimental investigation results, it can be deduced that there is a good agreement between the experimental data and numerical predictions.

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References
[1] Fan, A.W., J. J. Deng, J. Guo & W. Liu (2011). A numerical study on thermo-hydraulic characteristics of turbulent flow in a circular tube fitted with conical strip inserts, Appl. Therm. Eng., 31, 2819-2828.
[2] Gu, L. H., X. Ling & H. Peng (2012). An experimental and numerical investigation of air side heat transfer and flow characteristics on finned plate configuration. Heat Mass Transfer. 48, 1707-1721.
[3] Neshat, E., S. Hossainpour & F. Bahirae (2014). Experimental and numerical study on unsteady natural convection heat transfer in helically coiled tube heat exchangers. Heat Mass Transfer., 50,877-885.
[4] Nasiruddin, S. & K. Siddiqui (2007). Heat transfer augmentation in a heat exchanger tube using a baffle. International Journal of Heat and Fluid Flow, 28(2), 318-328.
[5] Wang, Y. S., Z. C. Liu & X. Y. Zhang (2012). Fluid flow and heat transfer study for flower baffle heat exchanger. Journal of Chemical Industry and Engineering(China), 63(S1), 99-105.
[6] Qian, C. F., H. Y. Gao & H. Y. Sun (2011). Shell-side fluid flow and heat transfer in curved baffle heat exchanger. Journal of Chemical Industry and Engineering(China), 62(5), 1233-1238.
[7] Jin, M., J. Fang & L. Zhan (2015). Performances of heat transfer and fluid flow in the shell and tube heat exchanger with novel sextant fan baffles. International Conference on Advances in Energy, Environment and Chemical Engineering (AEECE-2015), 122-125
[8] Jin, M., L. Zhan & G. X. Yu (2014). Effect of the Flow Models on the Numerical Simulation of Shell and Tube Heat Exchanger. Adv. Mater. Res., 1008-1009, 910-913.
[9] Taher, F. N., S. Z. Movassag, K. Razmi & R. T. Azar (2012). Baffle space impact on the performance of helical baffle shell and tube heat exchangers. Appl. Therm. Eng., 44, 143-149.