Shell Side CFD Analysis of a Model Shell-and-Tube Heat Exchanger

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In the present paper, a model-size shell-and-tube heat exchanger with horizontal baffles is investigated numerically and compared to the measured values using the commercial software SC-Tetra V11. Determination of the heat transfer coefficients for the shell side depends on the type of the flow and the type of the baffles. Without baffles the shell side medium leaves in the shortest way, and dead zones are formed. With the usage of these baffles, the flow path is artificially formed and the flow velocity will be increased because of the decreased flow area. These two effects will cause a better heat transfer but the other hand this will increase the weight of the heat exchanger. There are very simple experimental correlations for the segment and disk-and-donut types baffles. However, must be known the real heat transfer coefficient in case of an optimal design. This study investigated the effect of the type of the baffles, the space between them and the baffle cut to the real heat transfer coefficient and the necessary material quantity and compared the simulated heat transfer coefficients with analytically calculated ones based on the literature.

1. Introduction

Heat exchangers devices used for transferring thermal energy between two fluids. They are widely used within the petrochemical, chemical and pharmaceutical industries, power stations and also in the households for different applications, such as heating, cooling, condensation and evaporation process (Master et al. 2006). The most common used type of them is the shell-and-tube heat exchangers. These have a lot of advantageous properties compared to the others: have a high ratio of volume and heat transfer area, relatively easy to manufacture, easy to clean and repair, and able to transfer high mass flow rates. Furthermore, easy to increase the performance of these equipment with the usage of baffles. With these components, the flow direction could be guided, the whole heat transfer area is involved in the heat transfer and due to the decreased flow section, the velocities and the turbulence will be higher, and this increased value will cause a higher heat transfer coefficient and heat performance.

The most important geometric parameters are the tube arrangement, the type, the orientation and the opening of the baffles, the space between the baffles, the position of the most extreme baffles and any other heat transfer improvement possibilities. Abd and Naji (2017) used the Kern's procedure to determine the external heat transfer coefficient, Ambekar et al. (2016) examined different segmental baffle types (single, double, triple and flower). Batalha et al. (2017) investigated the effect of the usage of different turbulence model, while Eryener (2006) used the number of transfer unit method with different tube layout. The other way to make artificial route to the shell side fluid to create scraped blades (Varga et al., 2017) or made helical baffles (Jiang et al., 2017) for the more favorable flow arrangement.

On the other hand, these baffles will increase the additional costs: the manufacturing and cleaning of these appliances more difficult than the simple ones, and the pressure drop will be also much higher, which requires a more powerful pump. Further, the analytical calculation is harder and shows a higher difference from the reality. The difficulty is caused by the different values of velocity in the different direction and the difficulty of the heat transfer processes.

The locations of the baffles inside the shell-and-tube HEs (Vukic et al., 2014) investigated the heat transfer of a two passes tube side and a one pass shell side heat exchanger with segmental baffles. In that study they showed that the performance is strongly depending on the number of baffles, baffle size, distance between

baffles, the shape of the baffle and the first and last baffle position to the inlet and outlet nozzle of the shell side. Mellal et al. (2017) continued this study, they investigated the effect of the segment type baffles. They tested two baffles spacing and six baffles orientation angles and showed that the decreased baffle spacing causes a higher performance, and the maximum angle of 180° between two adjacent baffles will cause the highest performance. Wen et al. (2015) investigated two different type of baffles: the sector-shaped plain baffles and the ladder-type fold baffles. Both baffles run the fluid much more tightly, than the segment type baffle, so the heat transfer at these cases will higher. The disadvantages of these type are the increased pressure drop and the manufacturing time and costs.

Applying CFD (Computational fluid dynamics) software becomes more and more important to investigate the heat transfer processes. Nowadays, the most investigated baffle type is the helical baffles. Especially at this appliance has the greatest importance of the numerical simulations, because of the difficulties at the manufacturing and maintenance. Due to the high performance computers and the high speed computing softwares the heat exchanger constructions can be investigated. Without the expensive manufacturing and long investigation time the equipment can be comparable to each other. The aim of this study to determine the reliability of the empirical correlations.

2. Modeling and simulation

2.1 Physical model

Heat exchangers were investigated in this paper with different shell side types. In all cases, the diameter of the shell (D_s) and the tubes (d_i and d_o) were constants. To investigate the effect of the baffles, the initial parameters of the tube side were also constant. That will occur a constant heat transfer coefficient on the tube side.
The chosen parameters for the model are showed in Table 1.

Table 1: Geometric parameter of the investigated equipment

Parameter	Sign	Value	Unit
inner diameter of shell	D_s	188	mm
inner diameter of tubes	d_i	16	mm
outer diameter of tubes	d_o	20	mm
length of the tubes	L	1,200	mm
number of tubes	n_t	37	pcs

Five different construction investigated with the initial parameters. Configuration of these equipment listed in the Table 2. The model of the ID4 variant shown in Figure1. In this figure, the relative position of the nozzles and the baffles are seen.
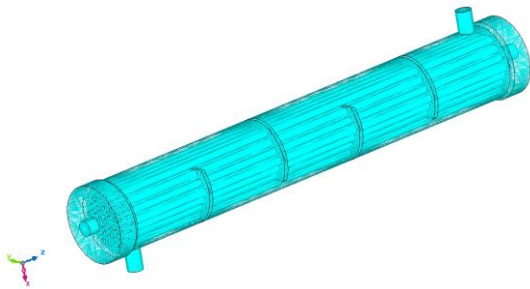

Figure 1: Geometric model of the heat exchanger

Table 2: Variables of the investigated equipment

ID	Number of baffles	Baffle space (mm)	Baffle cut
1	0	1,200	0 %
2	6	163	30 %
3	4	232	30 %
4	6	163	20 %
5	4	232	20 %

In Table 2, the baffle space means the distance of the nearest baffles, and the baffle cut means the ratio of the opened and whole cross section area. In Figure 2, the two type baffles can be seen.
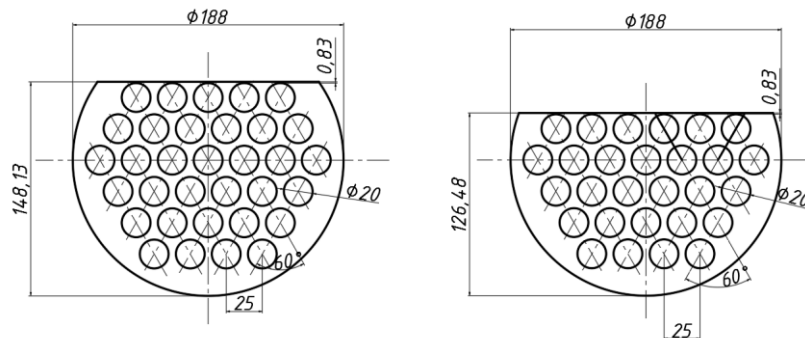

Figure 2: Geometric values of the baffles

2.2 Boundary conditions

At the shell side, the water was specified with an inlet temperature ($T_{s,in}$) of 30 °C and a mass flow rate (m_s) was 2 kg/s. The material of the tube side was also water, its inlet temperature ($T_{t,in}$) of 60 °C and a mass flow rate (m_t) was 3 kg/s. In both sides the outlet boundary condition was static pressure condition. The device made of carbon steel. There were walls between the tube side and tube and between tube and shell side. All of them are stationary walls. The material properties of the water determined at the temperature of 20 °C, and the effect of gravity was neglected. The model was simulated with SC-Tetra CFD software. The used turbulence model was the realizable k-EPS model, and in the solver settings, the pressure correction method was the modified SIMPLEC method.

2.3 Used mesh

The same meshing parameters used in all cases. The first simulation (ID1) showed a very good correlation with the measurement, and very high mesh elements were used, therefore the mesh independence test did not execute. The differences between the total element numbers caused by the size of the baffle. During the meshing, there was an important parameter, that in the tube wall there will be enough mesh layer. This mesh showed in Figure 3., while the used element number showed in the Table 3. In case of the investigation of the heat transfer the material of the shell and the baffles have not effect to the results, so these parts not modeled. At these surfaces adiabatic wall conditions used.
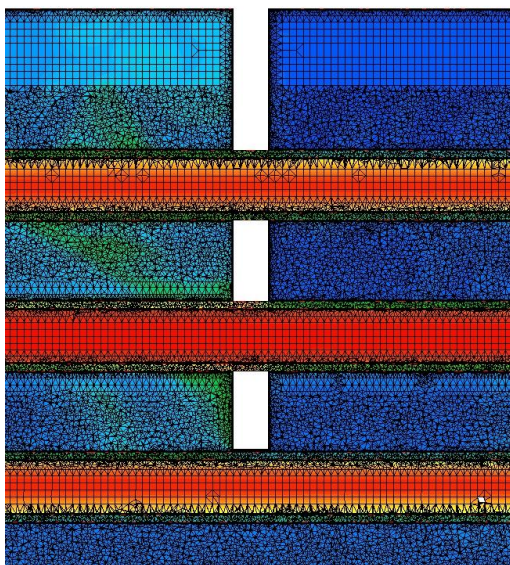

Figure 3: The used mesh near of a baffle

2.4 Results of the analysis

The results are converged in all cases. Figure 4 shows the temperature profile of the 4th variant. It is clearly seen, that the two fluids flow in counter current. The ends modeled as a cylinder, and the nozzles are concentric with these. It is seen from this temperature profile, that after the baffles a higher heat transfer coefficient can be achieved due to the higher velocities.
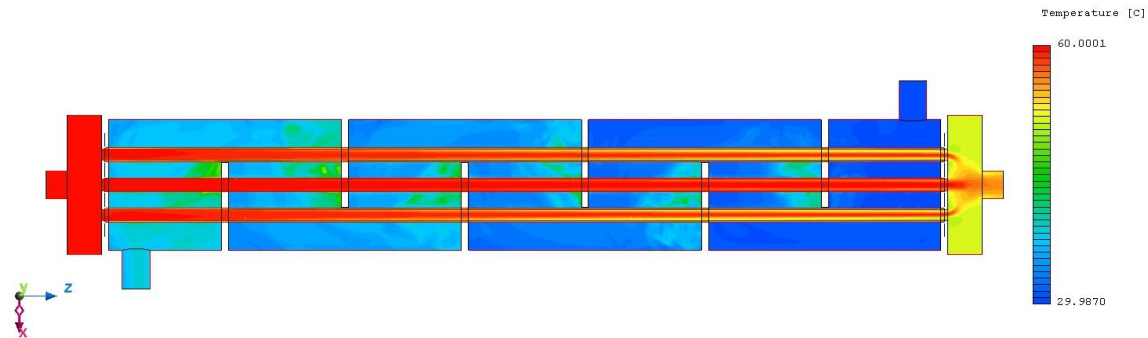

Figure 4: Temperature profile of ID4

Table 3: Total element numbers and outlet temperatures

ID	Total element number	Tube side outlet temperature (°C)	Shell side outlet temperature (°C)	Average performance (W)
1	113,442,281	56.34	35.3	45,123.73
2	105,404,341	54.9	37.68	64,110
3	110,941,146	55.98	36.19	51,103.99
4	107,876,269	55.68	36.58	54,616.87
5	114,430,746	56.04	36.13	50,476.69

3. Comparison of the CFD results with analytical results

3.1 Calculation of the tube side

In all cases, the heat transfer coefficient on the tube side was the same. The material properties are assumed at the average temperature. The total cross section of the tube side is:

$$A_t = n_t \cdot \frac{d_i^2 \cdot \pi}{4} = 7{,}439.291 mm^2 \tag{1}$$

From the mass flow rate and this cross section, the velocity is

$$v_t = \frac{m_t}{\rho \cdot A_t} = 0.404 \frac{m}{s} \tag{2}$$

Using the Dittus-Boelter correlation for determine the Nu-number:

$$Nu_t = 0.023 \cdot \mathrm{Re}_t^{0.8} \cdot \mathrm{Pr}_t^{1/3} \tag{3}$$

From these, the value of the Nu-number is 44.31 (which was constant at all five cases) and this determine a 1,667.03 W/(m²K) heat transfer coefficient. This study was not the goal of examining the tube side heat transfer, and due to its higher value than the outer heat transfer coefficient, it has a less impact to the overall heat transfer coefficient.

3.2 Calculation of the shell side

In all cases, due to the constancy of mass flow rates, the volume flow rates in the shell will be also constant. However, due to the different cross sections, the velocities on the shell side depend on the type of the baffles and the baffle spaces. The average velocity could calculate as a geometric middle of the longitudinal and the

transverse velocities. The longitudinal velocity depends on the baffle cut, while the transverse depends on the baffle space. Table 4 contains these velocities:

Table 4: Variables of the investigated equipment

ID	Longitudinal cross section (mm^2)	Transverse cross section (mm^2)	Longitudinal velocity (m/s)	Transverse velocity (m/s)	Mean velocity (m/s)
1	16,135.22	-	0.1241	-	0.1241
2	5,070.38	7,824	0.3948	0.2559	0.3178
3	5,070.38	11,136	0.3948	0.1798	0.2664
4	3,041.38	7,824	0.6583	0.2559	0.4104
5	3,041.38	11,136	0.6583	0.1798	0.3440

In every cases, the specific geometry will be the outer diameter of the tubes. The empirical correlation on the shell side of a heat exchanger is

$$Nu_s = 0.23 \cdot \mathrm{Re}_s^{0.6} \cdot \mathrm{Pr}_s^{0.33} \tag{4}$$

The Re_s in Eq(4) must be calculated in the function of the mean velocities, which are in Table 3. The used flow arrangement was the counter current to achieve a higher performance. The heat transfer area is:

$$A = n_t \cdot L \cdot \pi \cdot \frac{d_i + d_o}{2} = 2.51 m^2 . \tag{5}$$

Using the values of the outlet temperatures from Table 3 and the heat transfer coefficient from Eq(3), the calculated values are shown in the next table:

Table 5: Calculated thermal properties and its comparison of the CFD-results

ID	LMTD (°C)	Overall heat transfer coefficient (W/m^{2}K)	Transferred heat (analytical) (W)	Transferred heat (CFD) (W)	difference
1	25.51	639.06	40,933.97	45,123.73	9.29 %
2	23.58	882.67	52,271.57	64,110	18.47 %
3	24.88	841.35	52,556	51,103.99	-2.84 %
4	24.53	941.49	57,991.87	54,616.87	-6.18 %
5	24.93	901.01	56,418.42	50,476.69	-11.77 %

3.3 Comparison

After the calculations, the analytical results can be comparable with the CFD results. The differences showed in Table 5. It is clearly seen from these data, that this empirical correlation is usable in just a very small range. The negative differences mean an oversizing, which is a disadvantage in case of optimal sizing, while the positive differences are under sizing, the equipment does not able to transfer the necessary heat.

4. Optimal sizing

The total mass and the manufacturing cost are calculable from the geometric data for every case. All variants have the same shell and tube sheets, so the cost of the material and the manufacturing (which is proportional with the cutting lengths) are not presented in this study.
The investigated device is relatively small structure, but in the industry, these data have much higher effect. Due to the additional masses, the supporting structure will be increase, while the bigger cutting length will cause a higher manufacturing time and costs. From these considerations, the Figure 5 shows these additional costs. It is seen that even though ID4 has the highest heat performance, ID3 construction has the highest ratios in both cases.
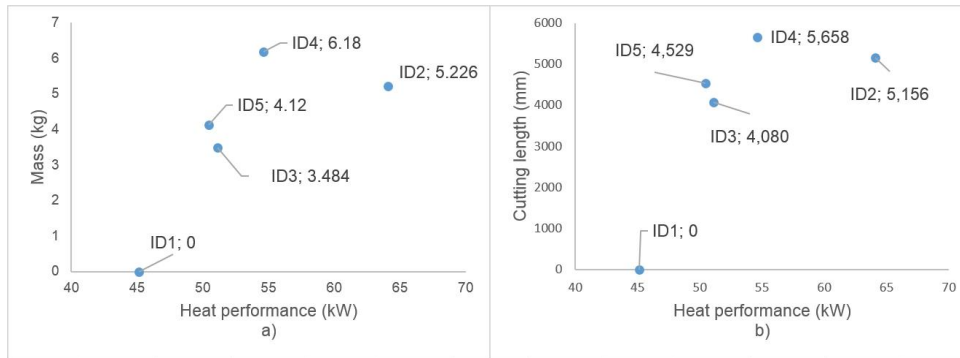
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Figure 5: Additional costs of the baffle usage in the function of a) total mass and b) cutting length

5. Conclusion

In this paper, heat transfer in segment type baffle shell-and-tube heat exchangers with different baffle geometries was studied numerically and analytically. The following conclusions are obtained.
(1) Usage of baffles in the shell side increases the overall heat transfer coefficient in every case.
(2) The empirical correlations in the literature can provide good preliminary results and can narrow the range of the geometric parameters to achieve a construction with a lower cost.
(3) The thermal optimum point and the total cost optimum point separated by each other, so different task demands different formation. From thermal engineering viewpoint, the ID2 has the highest performance, while the manufacturing costs also calculated the optimal equipment is the ID4. ID2 has 14.8 % higher performance than ID4, 18.3 % less material needs and 9.7 % less cutting costs than ID4, so the optimal choice for this task is ID4.

Acknowledgement

References

Abd A.A., Naji S.Z., 2017, Analysis study of shell and tube heat exchanger for clough company with reselect different parameters to improve the design, Case Studies in Thermal Engineering, 10, 455–467.

Ambekar A.S., Sivukamar R., Anantharaman N., Vivekenandan M., 2016, CFD simulation study of shell and tube heat exchangers with different baffle segment configurations, Applied Thermal Engineering, 108, 999–1007.

Batalha Leoni G., Suaiden Klein T., de Andrade Medronho R., 2017, Assessment with computational fluid dynamics of the effects of baffle clearances on the shell side flow in a shell and tube heat exchanger, Applied Thermal Engineering, 112, 497–506.

Eryener D., 2006, Thermoeconomic optimization of baffle spacing for shell and tube heat exchangers, Energy Conversion and Management, 47(11–12), 1478–1489.

Jiang B., Yan Sh., Zhang L., Xiao X., 2017, Numerical research of stream analysis on helical baffles heat exchangers, Journal of Engineering Thermophysics, 26(2), 272–290.

Master B.I., Chunangad K.S., Boxma A.J., Kral D., Stehlík P., 2006, Most Frequently Used Heat Exchangers from Pioneering Research to Worldwide Applications, Heat Transfer Engineering, 27(6), 4–11.

Mellal M., Benzeguir R., Sahel D., Ameur H., 2017, Hydro-thermal shell-side performance evaluation of a shell and tube heat exchanger under different baffle arrangement and orientation, International Journal of Thermal Sciences, 121, 138–149.

Varga T., Szepesi G. Siménfalvi Z., 2017, Horizontal scraped surface heat exchanger - Experimental measurements and numerical analysis, Pollack Periodica, 12(1), 107–122.

Vukic M.V., Tomic M.A., Zivkovic P.M., Ilic G.S., 2014, Effect of segmental baffles on the shell-and-tube heat exchanger effectiveness, Hemijska industrija, 68(2), 171–177.

Wen J., Yang H., Wang S., Xue Y., Tong X., 2015, Experimental investigation on performance comparison for shell-and-tube heat exchangers with different baffles, International Journal of Heat and Mass Transfer, 84, 990–997.