International Journal of ChemTech Research
CODEN (USA): IJCRGG, ISSN: 0974-4290, ISSN(Online):2455-9555
Vol.11 No.03, pp 342-350, 2018

Numerical-Experimental Comparison of the Overall Coefficient of Heat Transfer in a Shell and Tube Heat Exchanger

Luis F. Aguas[1], Fernando J. García[2], Guillermo E. Valencia[3]*

[1]Student, Faculty of Mechanical Engineering, Universidad del Atlántico, Km 7 Antigua Vía a Puerto Colombia, Barranquilla, Colombia.
[2]Student, Faculty of Mechanical Engineering, Universidad del Atlántico, Km 7 Antigua Vía a Puerto Colombia, Barranquilla, Colombia.
[3]Assistance Professor, Faculty of Mechanical Engineering, Universidad del Atlántico, Km 7 Antigua Vía a Puerto Colombia, Barranquilla, Colombia.

Abstract : This article presents a way for evaluating the overall coefficient of heat transfer, analyzed in a tube and shell heat exchanger, with a single pass through the tubes and shell, where hot fluid circulates through the tubes and cold fluid through the shell. This analysis is based on the different operating conditions of the fluids, by varying their volumetric flows in a range of 40°C to 60°C, recording the data obtained experimentally for each run. Simulation of this process was also carried out with the Aspen HYSYS plus® software, and its results were compared with the data obtained experimentally.
Keywords : heat exchanger, tube and shell, global coefficient, heat transfer.

1. Introduction

Heat exchangers are one of the most widely used equipment in the industry worldwide. Therefore, studies have focused on this type of devices and more specifically on tube and shell heat exchangers. These are used in the chemical industry, heat recovery, food processing, energy production[1,2], energy storage[3], among others. This has led engineers to direct their research towards performance optimization since factors such as fouling of pipes directly affect their performance[4]. Has been shown that a 45° tube distribution provide high velocity at the maximum number of tubes, as well as increasing flow uniformity distribution[5] and, in contrast, the 60° distribution offer low flow speed making it ideal for low viscosity fluids such as gases[6].

The search for optimization of exchangers has led engineers to study the baffle plates of these equipments. For these studies, CFD tools are widely used[7-11], thus achieving accurate results at low costs.

It has been demonstrated that using blinds in an exchanger a slanting flow pattern is generated on the shell side of the heat exchanger that is softer than the flow pattern in the segmented plate casing[12]. It

DOI : http://dx.doi.org/10.20902/IJCTR.2018.110346

was also demonstrated that these blinds have a more uniform temperature distribution, which optimizes thermohydraulic performance[12]. Meanwhile, for segmented plates, design analysis has been done using CFDs and empirical methods to test the influence of pipe modification for better heat exchanger performance and it was found that the rate of exergy destroyed is a function of the geometric parameters of the tubes and that the total rate of exergy destruction presented in the proposed design is about 8-22%[13]. Unlike the helicoidal deflectors where it was observed in the CFD that by varying the inclination of these and the length of the exchanger that these deflectors channel the flow to reduce pressure drops, but this effect decreases when the fluid has a high Prandtl number. Also, it was evidenced the increase in the transfer coefficient when increasing the number of deflectors for the same length[14].

Studies have also shown that by varying the deflector spacing results in an increase in the number of Nusselt and friction factor, a configuration of 10 baffle plates at 180° tilt offered the thermal performance of 3.55 in a Re of 3000 for the exchanger that was studied[15]. Another type of pipe configuration[16] has also been tested with a higher heat transfer rate. The economic part has been treated as a research topic, varying pipe diameters and plate separation, achieving an annual total cost reduction of approximately 26.99%[17]. On the economic side, algorithms for optimization have also been developed, such as CSA, with results of up to 77% energy savings and a reduction in investment costs of up to 13.1%[18].

The evidence shown in the studies performed has been achieved using CFD software, considering that and educational studies on laboratory experiments[19] and validations of methods for determining thermal resistance[20].

The main contribution of this work is directed towards the development of a comparative study of the global coefficient of transference of a heat exchanger of tube and shell in different operating conditions. This study will be accomplished by analyzing the data obtained experimentally in the laboratories and comparing them with those calculated by HYSYS plus software to compare methods and draw conclusions through the calculation of errors.

2. Materials and Methods

2.1 Equipment Description

The equipment used consists of two parts, a base unit which feeds the process fluids and a heat exchanger of tube and shell which performs the phenomenon under study. This equipment has an interface which allows us to translate the measurements taken and then analyze them in the software programmed and calibrated for this exchanger.

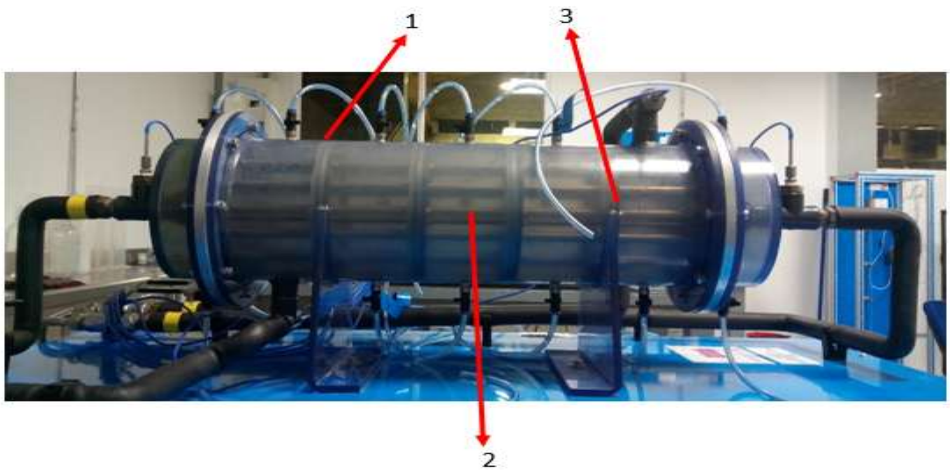

Figure 1.Characterized Shell and tube Heat Exchanger.

Table 1.Geometric configuration of the exchanger.

Location	Components	Parts	Internal diameter (m)	External diameter (m)	Thickness (m)	Length (m)
1	Coraza	1	0.148	0.160	6*10E-3	0.5
2	Tubos	21	8*10E-3	10*10-E-3	1,00E-02	0.5
3	Bafles	4	-	-	0.02	-

The base unit performs the functions of heating the water as the main function and can measure the flow rates of cold and hot water. It also allows pumping hot water to the entire installation by means of the variation of the position of the valves; it leaves to redirect the circulation directions of the cold water throughout the flexible tubes that join the parts of this equipment. On the other hand, the heat exchanger allows temperature measurements to be taken at different points of the heat exchanger, both in the cold and hot flow, allowing a more detailed analysis of the variations that are occurring. The heat exchanger is made up of a shell, inside it is formed by stainless steel tubes and four segmented deflectors placed with a transversal layout along the length of the tube.

2.2 Fundamental Equations

A set of equations were applied to estimate the overall heat transfer coefficient in the shell and tube heat exchanger, which have been extensively studied in the literature[21,22].

The overall heat transfer coefficient (U) was determined for the heat exchanger by means of an energy balance, assuming ideal conditions in the equipment used, resulting in

$$U = \frac{1}{R_{ter.eq}}, \quad (1)(1)$$

where$R_{ter.eq}$ is the equivalent thermal resistance of the machine.

Considering ideal device conditions, it is possible to calculate the equivalent thermal resistance for a shell and tube exchanger as the sum of the internal convection thermal resistances in the inner tubes R_{cvi}, the conductive resistance in the outer tube $R_{cd}$and the external convective resistance R_{cve}, as will be seen below.

$$R_{ter.eq} = R_{cvi} + R_{cd} + R_{cve}. \quad (2)$$

Since the thermal conductivity of stainless steel tubes is very high with respect to the resistance presented in the system, R_{cd}tends to zero, so the resulting equation (2) is

$$R_{ter.eq} = R_{cvi} + R_{cve}. \quad (3)$$

In order to determine the convective coefficient inside the stainless steel tubes, the Reynolds number was calculated as follows

$$Re = \frac{\rho V D_i}{\mu}, \quad (4)$$

whereρ is the density of the fluid, $D_i$is the inner diameter of the tube, μ is the dynamic viscosity and, V is the average velocity of the fluid, which was determined by

$$V = \frac{\dot{m}}{\rho A_f}, \quad (5)$$

where m is the mass flow of the fluid is $\dot{m}$and is the flow areaA_f, which was calculated through

$$A_f = \frac{\pi N_t {D_i}^2}{4N_p}, \quad (6)$$

whereN_t is the number of tubes and N_p is the number of steps present in the heat exchanger.

For the calculation of the Nusselt number, when Reynolds' range is between 40-4000, the relation

$$N_u = 0.683 Re^{0.466} Pr^{1/3}, \tag{7}$$

and if the range for Reynolds number is between 4000-40000, the ratio to be used is

$$N_u = 0.683 Re^{0.618} Pr^{1/3}, \tag{8}$$

wherePr is Pr's number.

Finally, the convective coefficient was determined according to

$$N_u = \frac{h_i D_i}{k_f}, \tag{9}$$

where the conductive coefficient (k_f) was determined with the properties of the fluid at the average working temperature.

To determine the external convective coefficient in the shell were made the above calculations, nevertheless to determine the number of Reynolds was used the follow equation

$$Re = \frac{\rho V D_{eq}}{\mu}, \tag{10}$$

whereD_{eq} is the equivalent diameter, which must be determined depending on the arrangement type of the heat exchanger tubes.

Finally, the number of Nusselt allowed the external convective coefficient to be determined by means of

$$N_u = \frac{h_i D_{eq}}{k_f}. \tag{11}$$

2.3 Description of the Experiment

In order to record the data generated by the experiment, software provided by the manufacturer of the EDIBON heat exchanger was used, in which the variations of the parameters can be observed and the desired conditions for the analysis controlled, as shown in Figure 2.

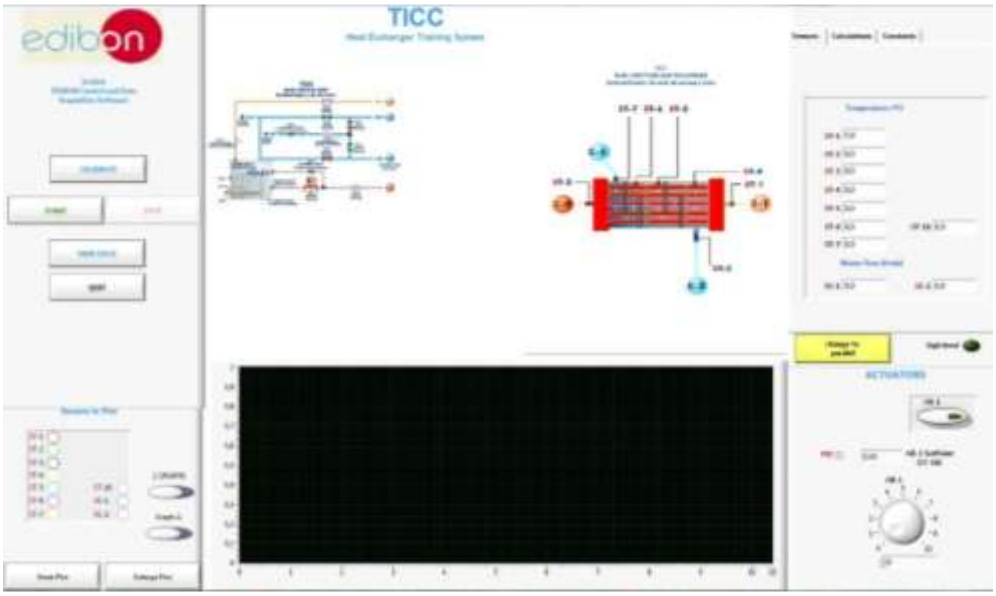

Figure 2. Data collection interface of the heat exchanger

Table 2.Experimental measurement scheme for obtaining the data

OPERATING CONDITION	TS-16 (°C)	VOLUMETRIC FLOW OF COLD WATER (L/min)	VOLUMETRIC FLOW OF HOT WATER (L/min)
Parallel/Counterflow	40-50-60	1,0 - 2,0 - 3,0	1,2 - 1,4 - 1,6 - 1,8 - 2,1 - 2,4

The procedure was based on varying the operating conditions of the system. The procedure was based on varying the operating conditions of the system. The first variable parameter was the temperature of the hot water flow, which was stabilized, and, the volumetric flow of the cold water was varied in a range from 1 to 3 L/min in order to generate a more detailed analysis of the varied intervals in three minutes; the revolutions per minute (rpm) of the pump establishing different states and allowing a review of the impact on the global coefficient of heat transfer.

After recording the data with the layout of the valves where both fluids have a parallel direction, the configuration of the valves was varied to generate counter flow in the system.

3. Results and Discussion

The following operating conditions were considered for the corresponding analyses of the heat exchanger specified above:

The heat exchanger operated in a stationary state.
The changes of kinetic energy and potential in the fluid were neglected.
Heat losses to the surrounding area were neglected.
Fluids are not mixed.

The conductive resistance in the tubes was neglected, since the stainless steel is highly conductive and the thickness of the tubes is very small.

3.1 Comparison of experimental results with Aspen HYSYS Plus®.

In order to carry out a comparative analysis of the experimental data, Aspen HYSYS Plus® software was used to simulate the operating conditions and characteristics of the pipe and shell heat exchanger provided by EDIBON. The required data were entered into the software of the working fluids, to obtain the calculations of the overall coefficient of heat transference by the area. In the following figures, we will observe the deviation of the data collected by the theoretical equations proposed in this article with respect to the Aspen HYSYS Plus® software.

In Figure 3, the result for the parallel flow in the heat exchanger was observed, in this study, the temperature in the storage tank was 40°C. In this case, the experimental behavior is below that obtained by the HYSYS plus® software. The average error rate described for these parameters was 1.63%, showing an essential closeness between the results.

Similarly, a similar behavior is shown in the Figure 4, taking into account that the temperature in the storage tank was modified to 50°C. Experimental results are still lower than those obtained in HYSYS plus®, showing an error of 2.01% which is a reasonably acceptable error.

Finally, for the parallel flow and 60°C in the storage tank is observed in Figure 5the behavior follows the previous trends with an error of 1.26% following the pattern presented in the two previous graphs.

This study also analyzed the behavior of the overall transfer coefficient for a counter flow arrangement in the heat exchanger, showing a variation from the parallel flow. In Figure 6 can be seen that for a temperature of 40°C in the storage tank the coefficient changes,

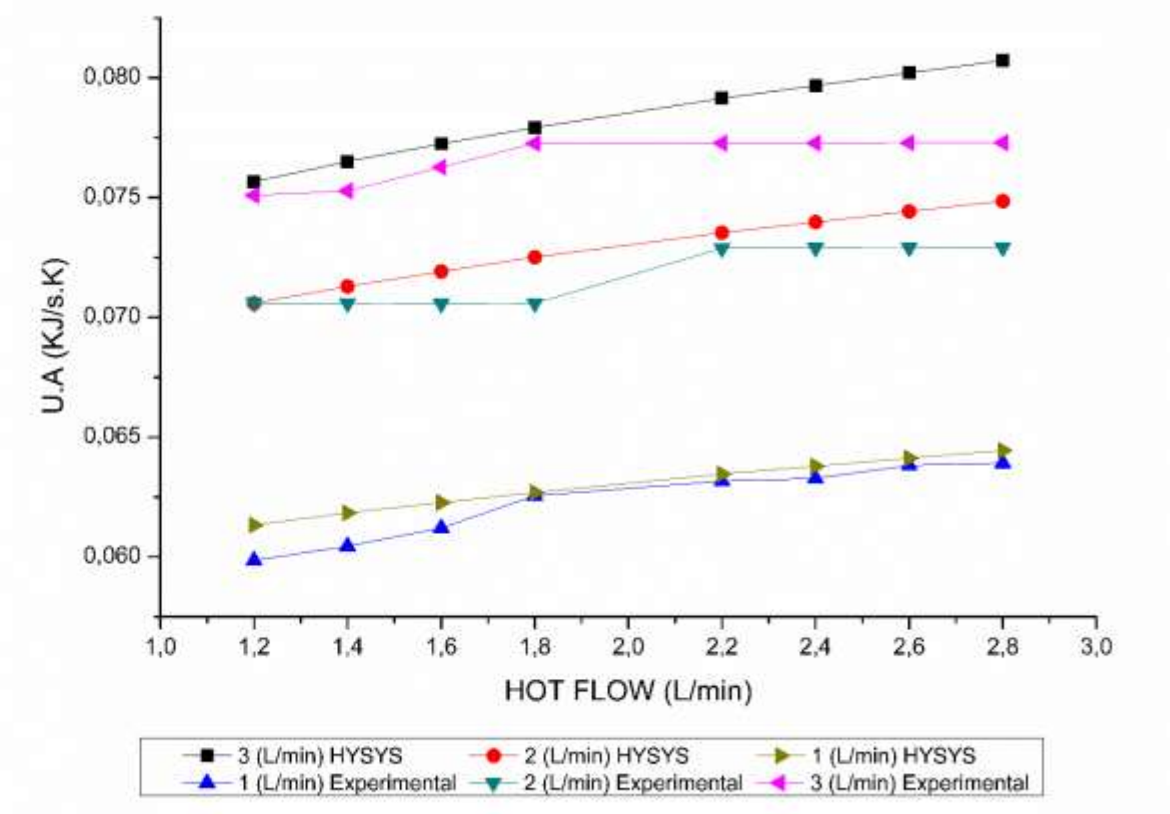

Figure 3.Overall transfer coefficient and hot volumetric flow rate at the heat exchanger inlet for 40°C.

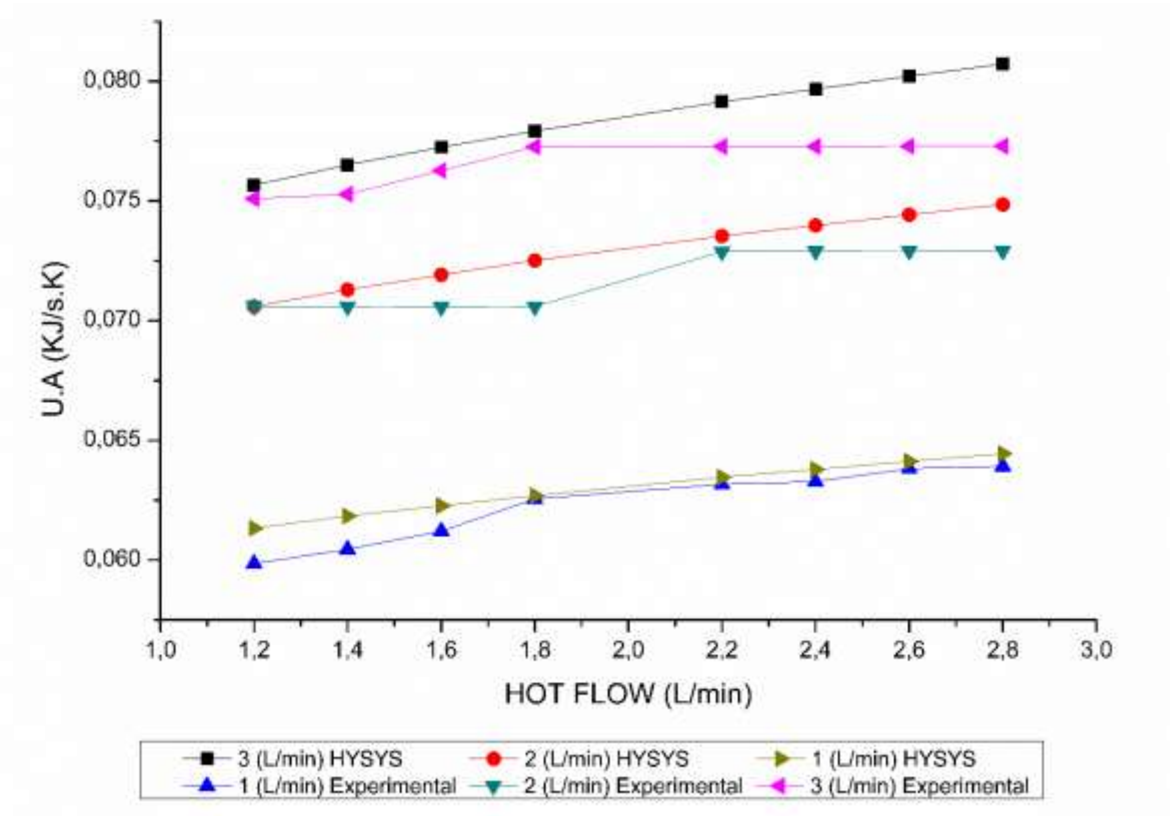

Figure 4.Overall transfer coefficient and hot volumetric flow rate at the heat exchanger inlet for 50°C.

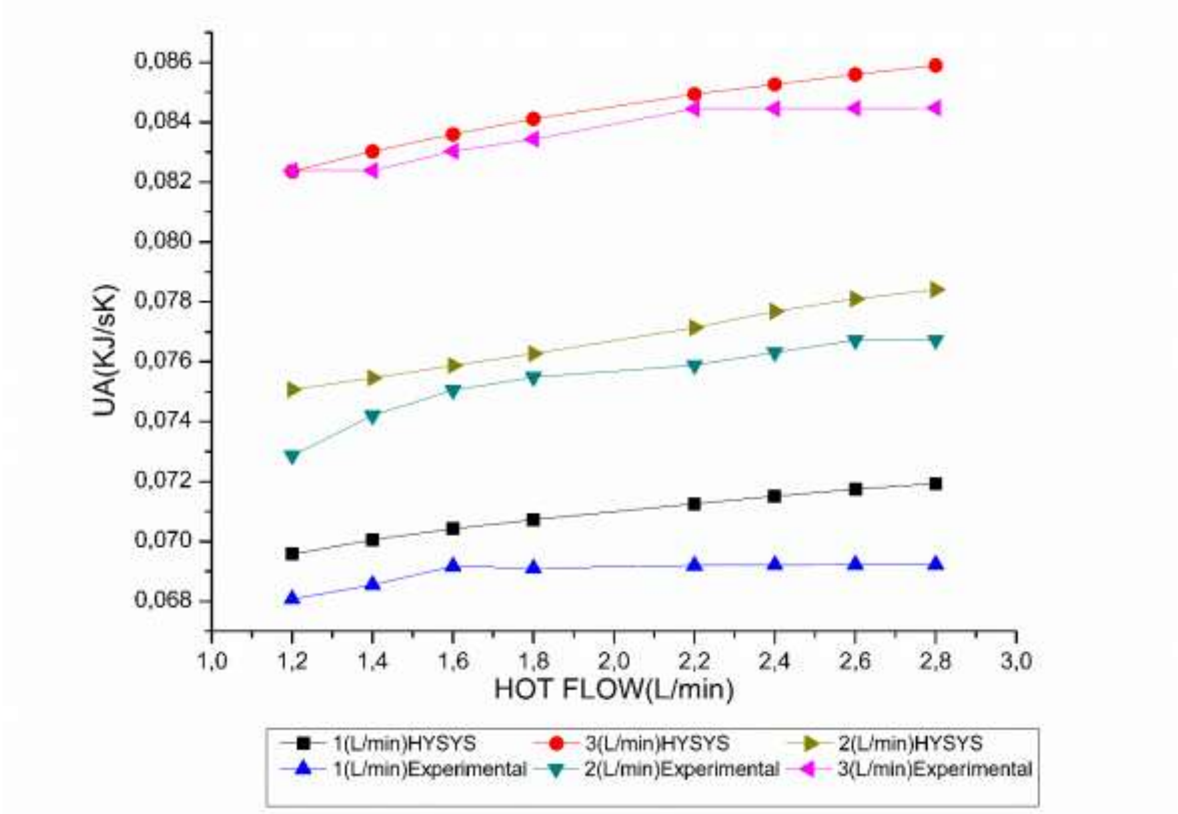

Figure 5.Overall transfer coefficient and hot volumetric flow rate at the heat exchanger inlet for 60°C.

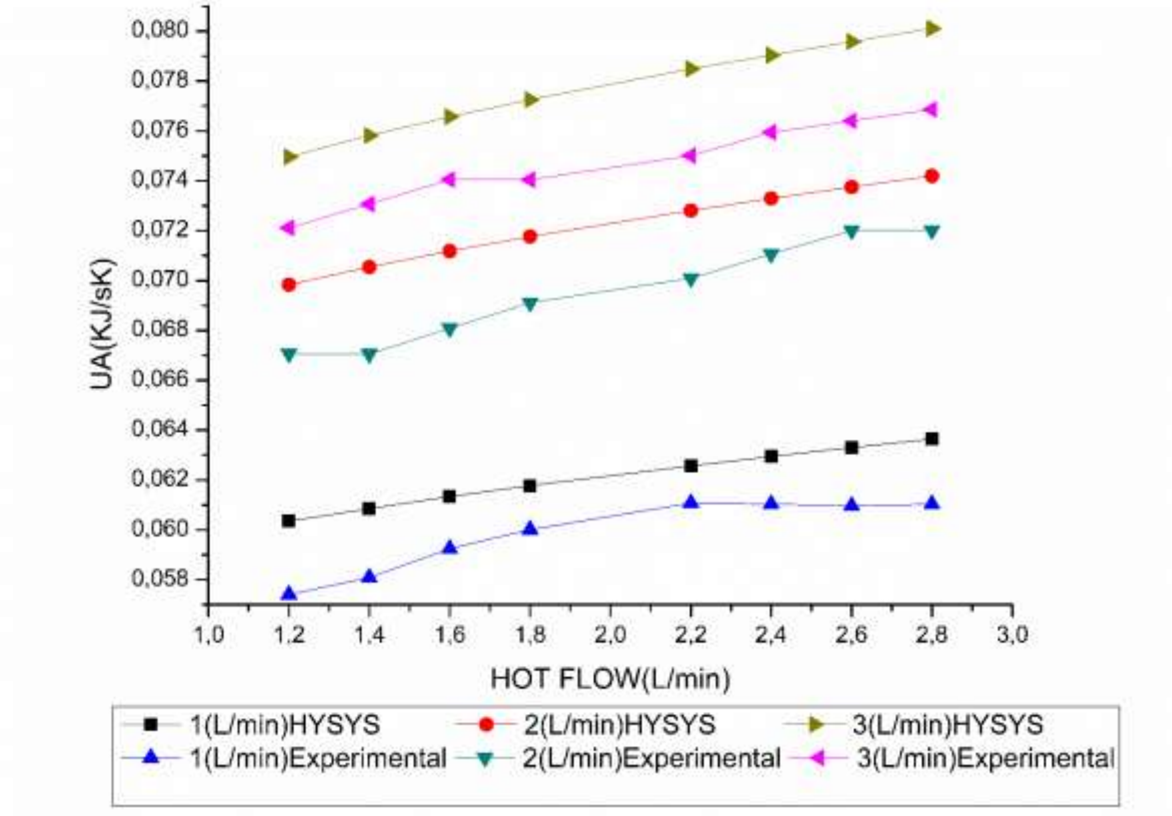

Figure 6.Global transfer coefficient and hot volumetric flow rate at the heat exchanger inlet for 40°C in countercurrent.

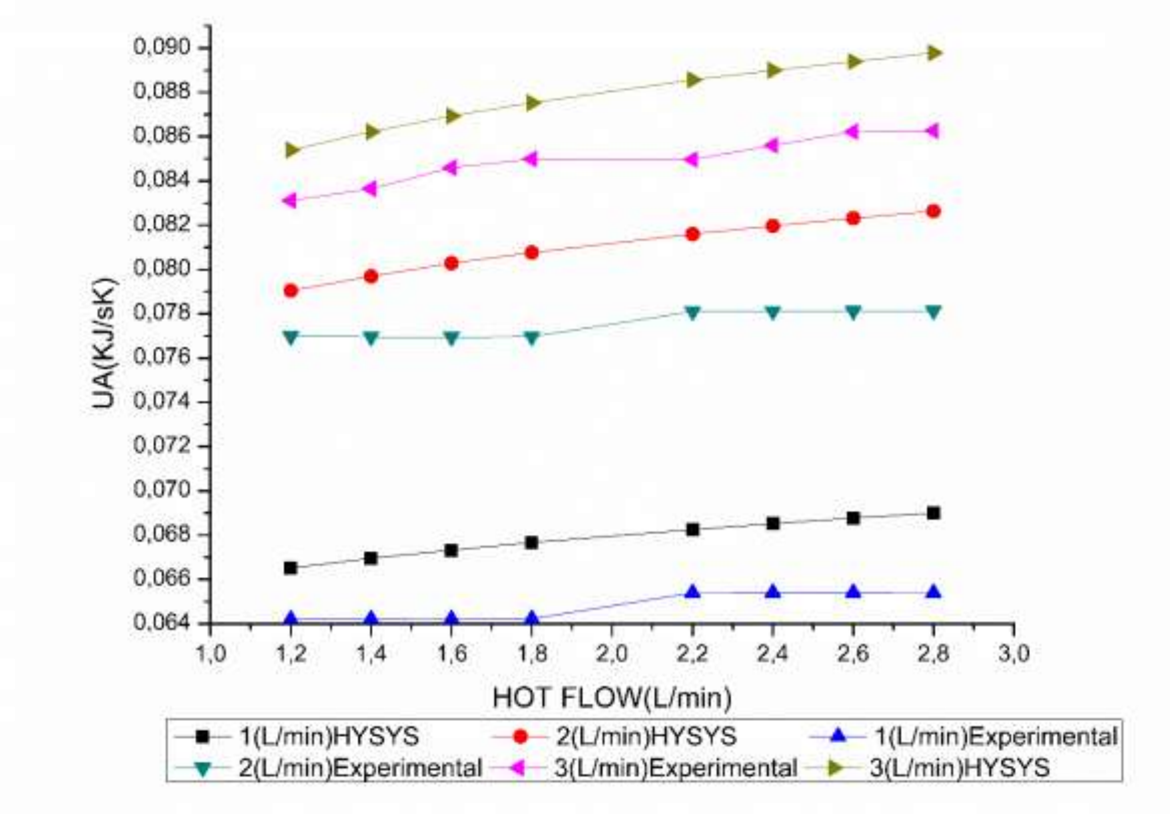

Figure 7.Global transfer coefficient and hot volumetric flow rate at the heat exchanger inlet for 50°C in countercurrent.

This change is shown as a decrease in the coefficient in the order of 1.08%. On the other hand, in Figure 7was seen an inverse behavior, a percentage increase of 1.44% in the overall transfer coefficient is presented.

4. Conclusions

Finally, it is possible to conclude that the results obtained experimentally showed a smaller error than expected, since the experimentation does not take into account variables as much of ambient conditions in the laboratory where it was tested as the deviations and errors present in the sensors nor the heat leakage through the exchanger casing.

Although the results performed better than expected, it can be said that the best way to obtain more accurate results is through HYSYS plus® software. Since it is in your mathematical model if you take into account external factors that cause the deviation found in this work.

References

1. I. Milcheva, F. Heberle, and D. Brüggemann, “Modeling and simulation of a shell-and-tube heat exchanger for Organic Rankine Cycle systems with double-segmental baffles by adapting the Bell-Delaware method,” Appl. Therm. Eng., vol. 126, pp. 507–517, Nov. 2017.
2. A. Erdogan, C. O. Colpan, and D. M. Cakici, “Thermal design and analysis of a shell and tube heat exchanger integrating a geothermal based organic Rankine cycle and parabolic trough solar collectors,” Renew. Energy, vol. 109, pp. 372–391, Aug. 2017.
3. J. Gasia, J. Diriken, M. Bourke, J. Van Bael, and L. F. Cabeza, “Comparative study of the thermal performance of four different shell-and-tube heat exchangers used as latent heat thermal energy storage systems,” Renew. Energy, vol. 114, pp. 934–944, Dec. 2017.
4. M. Markowski, M. Trafczynski, and K. Urbaniec, “Identification of the influence of fouling on the heat recovery in a network of shell and tube heat exchangers,” Appl. Energy, vol. 102, pp. 755–764, Feb. 2013.
5. K. Mohammadi and M. R. Malayeri, “Parametric study of gross flow maldistribution in a single-pass

shell and tube heat exchanger in turbulent regime," Int. J. Heat Fluid Flow, vol. 44, pp. 14–27, Dec. 2013.

6. O. Labbadlia, B. Laribi, B. Chetti, and P. Hendrick, "Numerical study of the influence of tube arrangement on the flow distribution in the header of shell and tube heat exchangers," Appl. Therm. Eng., vol. 126, pp. 315–321, Nov. 2017.
7. X. Zhang, P. Tseng, M. Saeed, and J. Yu, "A CFD-based simulation of fluid flow and heat transfer in the Intermediate Heat Exchanger of sodium-cooled fast reactor," Ann. Nucl. Energy, vol. 109, pp. 529–537, Nov. 2017.
8. G. Batalha Leoni, T. Suaiden Klein, and R. de Andrade Medronho, "Assessment with computational fluid dynamics of the effects of baffle clearances on the shell side flow in a shell and tube heat exchanger," Appl. Therm. Eng., vol. 112, pp. 497–506, Feb. 2017.
9. S. Wang, J. Xiao, J. Wang, G. Jian, J. Wen, and Z. Zhang, "Configuration optimization of shell-and-tube heat exchangers with helical baffles using multi-objective genetic algorithm based on fluid-structure interaction," Int. Commun. Heat Mass Transf., vol. 85, pp. 62–69, Jul. 2017.
10. A. El Maakoul et al., "Numerical comparison of shell-side performance for shell and tube heat exchangers with trefoil-hole, helical and segmental baffles," Appl. Therm. Eng., vol. 109, pp. 175–185, Oct. 2016.
11. E. Pal, I. Kumar, J. B. Joshi, and N. K. Maheshwari, "CFD simulations of shell-side flow in a shell-and-tube type heat exchanger with and without baffles," Chem. Eng. Sci., vol. 143, pp. 314–340, Apr. 2016.
12. Y. Lei, Y. Li, S. Jing, C. Song, Y. Lyu, and F. Wang, "Design and performance analysis of the novel shell-and-tube heat exchangers with louver baffles," Appl. Therm. Eng., vol. 125, pp. 870–879, Oct. 2017.
13. A. A. Tahery, S. Khalilarya, and S. Jafarmadar, "Effectively designed NTW shell-tube heat exchangers with segmental baffles using flow hydraulic network method," Appl. Therm. Eng., vol. 120, pp. 635–644, Jun. 2017.
14. X. Xiao, L. Zhang, X. Li, B. Jiang, X. Yang, and Y. Xia, "Numerical investigation of helical baffles heat exchanger with different Prandtl number fluids," Int. J. Heat Mass Transf., vol. 63, pp. 434–444, Aug. 2013.
15. M. Mellal, R. Benzeguir, D. Sahel, and H. Ameur, "Hydro-thermal shell-side performance evaluation of a shell and tube heat exchanger under different baffle arrangement and orientation," Int. J. Therm. Sci., vol. 121, pp. 138–149, Nov. 2017.
16. D. Panahi and K. Zamzamian, "Heat transfer enhancement of shell-and-coiled tube heat exchanger utilizing helical wire turbulator," Appl. Therm. Eng., vol. 115, pp. 607–615, Mar. 2017.
17. E. H. de Vasconcelos Segundo, A. L. Amoroso, V. C. Mariani, and L. dos S. Coelho, "Economic optimization design for shell-and-tube heat exchangers by a Tsallis differential evolution," Appl. Therm. Eng., vol. 111, pp. 143–151, Jan. 2017.
18. M. Asadi, Y. Song, B. Sunden, and G. Xie, "Economic optimization design of shell-and-tube heat exchangers by a cuckoo-search-algorithm," Appl. Therm. Eng., vol. 73, no. 1, pp. 1030–1038, Dec. 2014.
19. A. S. Kasumu, N. N. Nassar, and A. K. Mehrotra, "A heat-transfer laboratory experiment with shell-and-tube condenser," Educ. Chem. Eng., vol. 19, pp. 38–47, Apr. 2017.
20. M. Markowski, M. Trafczynski, and K. Urbaniec, "Validation of the method for determination of the thermal resistance of fouling in shell and tube heat exchangers," Energy Convers. Manag., vol. 76, pp. 307–313, Dec. 2013.
21. F. P. Incropera and D. P. DeWitt, Fundamentos de transferencia de calor. Pearson Educación, 1999.
22. Y. A. Çengel, Transferencia de calor, no. 660.28427 C4. 2004.

*****