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Effect of the Mass Flow Rate on the Heat Transfer Phenomena in a Shell and Tube Heat Exchanger

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Abstract

This article presents the development of some case studies on shell and tube heat exchangers using a graphical interface developed in MATLAB®. Through the case studies, the variation in the performance of the equipment was determined by means of the heat transfer coefficient in function of some parameters such as mass flow and tube arrangements, establishing initial conditions for each run that were established in the software. Likewise, the article shows in detail the methodology with which the study was developed, the fundamental equations used to calculate these results and the graphs obtained. Through this study, it was verified that the variables studied had such an influence on the parameters evaluated, with the objective of establishing selection and operation criteria when designing a shell and tube heat exchanger.

Keywords: Heat Exchanger,

INTRODUCCIÓN

Heat exchangers are devices designed for the transfer of heat between two fluids, one high temperature fluid to the other low temperature fluid [1]. These are widely used in the chemical, pharmaceutical, petrochemical, power plant and power plant industries and in various other applications, such as power plants, where different heat exchanger designs have been applied to meet thermal requirements with large heat transfer areas, and rigorous specifications of the materials used in the components of this equipment, such as pipe joints, mirrors, nozzles and welding flanges [2].

One of the most widely used heat exchangers in the chemical industry and energy production is the Shell and Tube Heat Exchanger (STHE), which consists of a cylindrical casing with several cylindrical tubes inside, where one fluid flows through the internal tubes while the other surrounds it. In addition, the baffles send the flow from the casing to increase its travel through the heat exchanger and improve the transfer process. Therefore, this equipment plays an important role in the world's energy demand, since the chemical and petrochemical industry represents 24% of the global industrial energy consumption, while in energy production, it is important not only to produce it, but also to reuse the waste heat generated in the process.

In addition, several research projects have been developed for this type of equipment. A new model for the characterization and performance of this type of exchanger was developed, where the heat exchange with a GHE tube was able to dissipate up to 2034 W[3]. In addition, one study showed in detail the performance of an integrated shell and finned tube heat exchanger with a CI engine configuration to extract heat from the exhaust gases and a thermal energy storage tank used to store the available excess energy[4], resulting in 14% of the fuel energy being stored as heat in the combined storage system, which is available at a reasonably higher temperature for a suitable application. On the other hand, the performance of the heat transfer process using phase change materials (PCM)[5] was investigated experimentally during the melting and solidification of five heat exchangers that function as latent heat storage systems, showing the values of the average thermal power under different operating conditions, comparing them with the heat exchangers studied.

The objective of this article is to study the variation of the overall heat transfer coefficient in the event of changes in operational parameters such as flow through the casing and tubes, as well as geometrical parameters of the equipment such as the reference of the tube arrangement.

METHODOLOGY

The following are the fundamental relationships used to perform the thermal calculations of the equipment, which are reported in literature [6].

For the calculation of heat transfer using the colburn factor j_H, a correlation was used for fluids circulating inside the tubes, but this does not apply to fluids passing through a bank of tubes with segmented deflectors, so equation (1) can be used, using random values of equivalent diameter and mass.

$$j_H = \frac{h_o D_e}{k} = 0.36\left(\frac{D_e G_s}{\mu}\right)^{0.55}\left(\frac{c\mu}{k}\right)^{1/2}\left(\frac{\mu}{\mu_\omega}\right)^{0.14}, \quad (1)$$

which is valid for values of 2000<Re<100000000, where the linear and mass velocity changes continuously across the tube bundle, as the shell width and number of tubes varies from zero at the top and bottom to a maximum at the center of the shell.

The cross-sectional area of flow as a function of the shell area a_s is given by equation (2) and the mass velocity G_s is given by equation (3).

$$a_s = \frac{DI \times C' B}{P_T \times 144}\ feet, \quad (2)$$

$$G_s = \frac{W}{a_s}\ lb/(h)(feet^2). \quad (3)$$

For the calculation of the equivalent diameter for the shell, the hydraulic radius obtained by the arrangement arranged in the pipe head is taken as four times, so for the square arrangement equation (4) is used.

$$D_e = \frac{4\times\left(P_T^2 - \frac{\pi d_O^2}{4}\right)}{\pi d_o}\ feet, \quad (4)$$

where P_T is the pipe spacing, and d_o is the outside diameter of the pipe. For the delta arrangement, the wet perimeter of the element corresponds to half a tube, as shown in equation (5).

$$d_e = \frac{4\times\left(\frac{1}{2}P_T \times 0.86 P_T - \frac{\frac{1}{2}\pi d_O^2}{4}\right)}{\frac{1}{2}\pi d_o}. \quad (5)$$

For the calculation of the true effective temperature difference Δ5 to replace the MLDT in countercurrent, taking into account that the temperature of the fluid in the shell is at an average isothermal temperature in any cross-section, the heating area in each step is equal, the total heat transfer coefficient is constant, the flow rate of each of the fluids is constant, the specific heat of each fluid is constant, there are no changes in the evaporation or condensation phase in one part of the exchanger, and the heat losses are negligible, resulting in the energy balance equation (6).

$$\Delta t = \left(\frac{T_1 - T_2}{UA/WC}\right)_{real} = \left(\frac{t_2 - t_1}{UA/wc}\right)_{real}. \quad (6)$$

By simplifying the parameters in the same way as for the double-tube exchanger, equations (7) and (8) result.

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{wc}{WC}, \quad (7)$$

$$S = \frac{t_2 - t_1}{T_1 - t_1}. \quad (8)$$

Performing the respective differential procedures, we have equation (9).

$$\left(\frac{UA}{wc}\right)_{real} = \frac{1}{\sqrt{R^2+1}} ln \frac{2 - S\left(R + 1 - \sqrt{R^2+1}\right)}{2 - S\left(R + 1 + \sqrt{R^2+1}\right)}. \quad (9)$$

Therefore, the fractional relationship 55, as a function of the true temperature difference and the MLDT is given by equation (10).

$$F_T = \frac{\sqrt{R^2+1}\, ln\frac{1-S}{1-RS}}{(R-1) ln\frac{2 - S\left(R+1-\sqrt{R^2+1}\right)}{2 - S\left(R+1+\sqrt{R^2+1}\right)}}. \quad (10)$$

Finally, it is important to clarify that the calculations of $\left(\frac{UA}{wc}\right)_{real}$ y F_T, can be performed by using the graphs of the correction factors F_T vs MLDT as S functions with R as the parameter.

In the case of pressure drop for fluids that are heated or cooled, including input and output losses are calculated using equation (11).

$$\Delta P_s = \frac{f G_s^2 D_s (N+1)}{2 g \rho D_e \varphi_s} = \frac{f G_s^2 D_s (N+1)}{5.22\times 10^{10} D_e s \varphi_s} \frac{lb}{feet^2}, \quad (11)$$

where N is the number of crossings, and s is the specific gravity of the fluid. The pressure drop in the pipes can be calculated with equation (12).

$$\Delta P_s = \frac{f G_t^2 Ln}{5.22\times 10^{10} D_e s \varphi_s}\ lb/feet^2, \quad (12)$$

where n is the number of steps, L is the length of the tube, and Ln is the total length of the path in feet. Finally, to calculate the heat produced and the area of the exchanger, equations (13) and (14) are used, respectively:

$$Q = UA\Delta T_{LM}, \quad (13)$$

$$A = \frac{Q}{U\Delta T_{LM}}. \quad (14)$$

RESULTS AND DISCUSSIONS

Below are the results of two case studies developed to understand the effect of mass flow on the overall heat transfer coefficients and the overall heat transferred in the equipment, as well as the influence this parameter has on the pressure drop and the obstruction factor of the equipment.

Influence of mass flow on the behaviour of the Ud, Uc and Q.

The behavior of the heat exchange process was analyzed taking into account the kerosene as a working fluid on both sides of the exchanger, varying its flow from an initial value of 1 lb/h to 5 lb/h, for inlet and outlet temperature conditions of T_{in}=100 °F and T_{out}=70 °F , and inlet and outlet temperatures of T_{in}=25 °F and T_{out}=50 °F respectively, an equal shield length of 1.5 ft, a selected number of tubes of 6, and length between deflectors of 1 inch. On the other hand, the arrangement chosen for the tubes was ¾ inch, with a 15/16" triangular section and an 8" shell. Figure 1 shows a comparison between the results obtained for a shell array (3/4 BWG 10) and an array (3/4 BWG 11), when changes in flow were made to both the shell (Figure 1a) and the tubes (Figure 1b).

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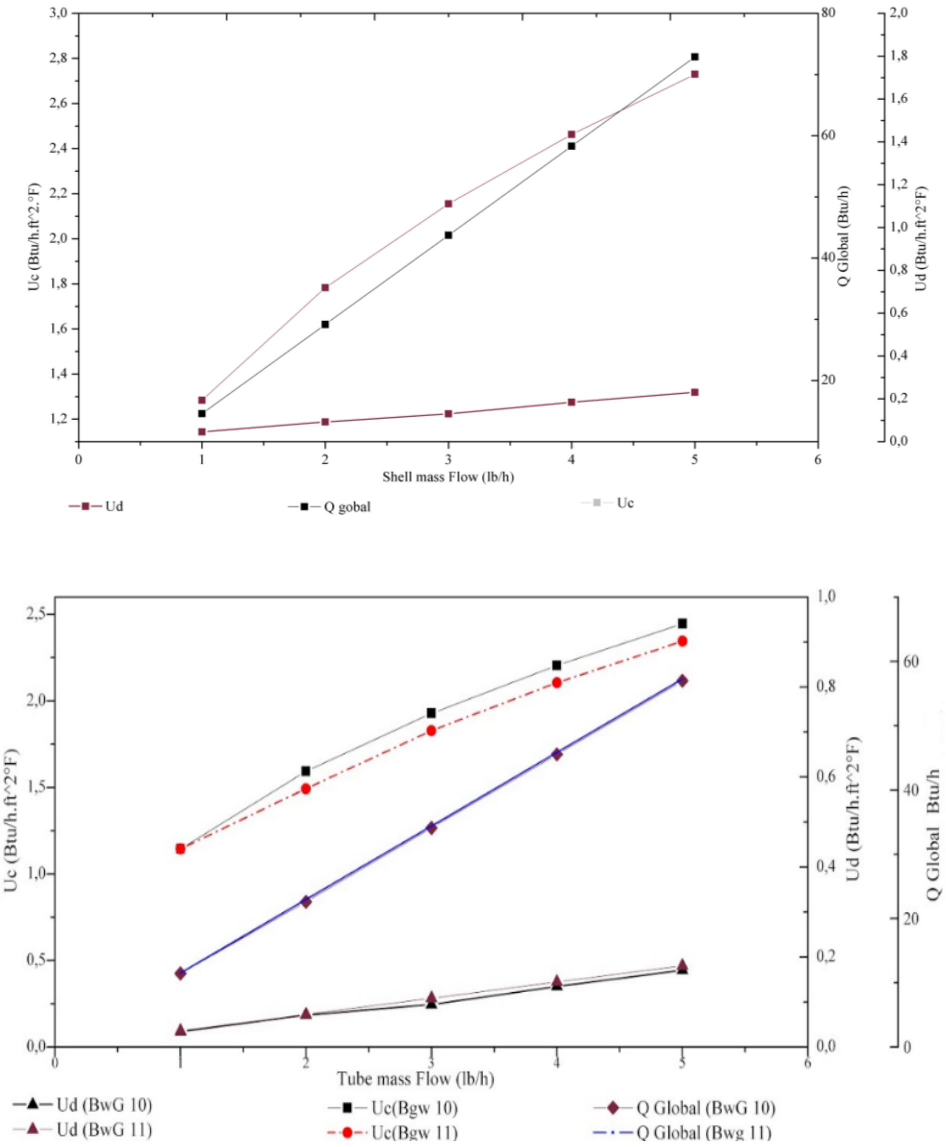

Figure 1. Behaviour of the heat transfer -Q, overall heat transfer coefficient- Uc and Ud as function on the mass flow through the a) shell, b) tubes.

The results for both the total heat Q, and the overall heat transfer coefficients Uc and Ud, show that the heat exchanger when the flow through the shell was varied remained constant regardless of the variation in the arrangement of the tubes during the study, which indicates that the behavior of these variables does not depend significantly on the type of arrangement selected, or show no significant changes for these two types of arrangements studied. However, the increase in mass flow within the limits defined in the case study while maintaining the temperatures makes this parameter increase proportionally for both global heat transfer coefficients, for a value of 130% for you, while Uc increases by more than 100%, resulting in a 250% increase in global heat transferred. The change in the mass flow on the tube side in the face of the same temperature difference on both the shell side and the tubes, also presented an increase in the heat transfer capacity of the equipment, but the type of arrangement causes average differences of no more than 5% for the arrangements studied, which means that these two types of configurations, despite being very similar, are very influential in the thermal performance of the equipment, and constitute a parameter to be taken into account in the selection and design processes of the tube and shell heat exchangers.

Influence of mass flow on pressure drop and obstruction factor.

When studying the influence of the mass flow of both the shell and the tubes on the pressure drop and the obstruction factor of the tubes that make up the heat exchanger, the results shown in Figure 2 were obtained, maintaining the same operating parameters defined in the previous case study.

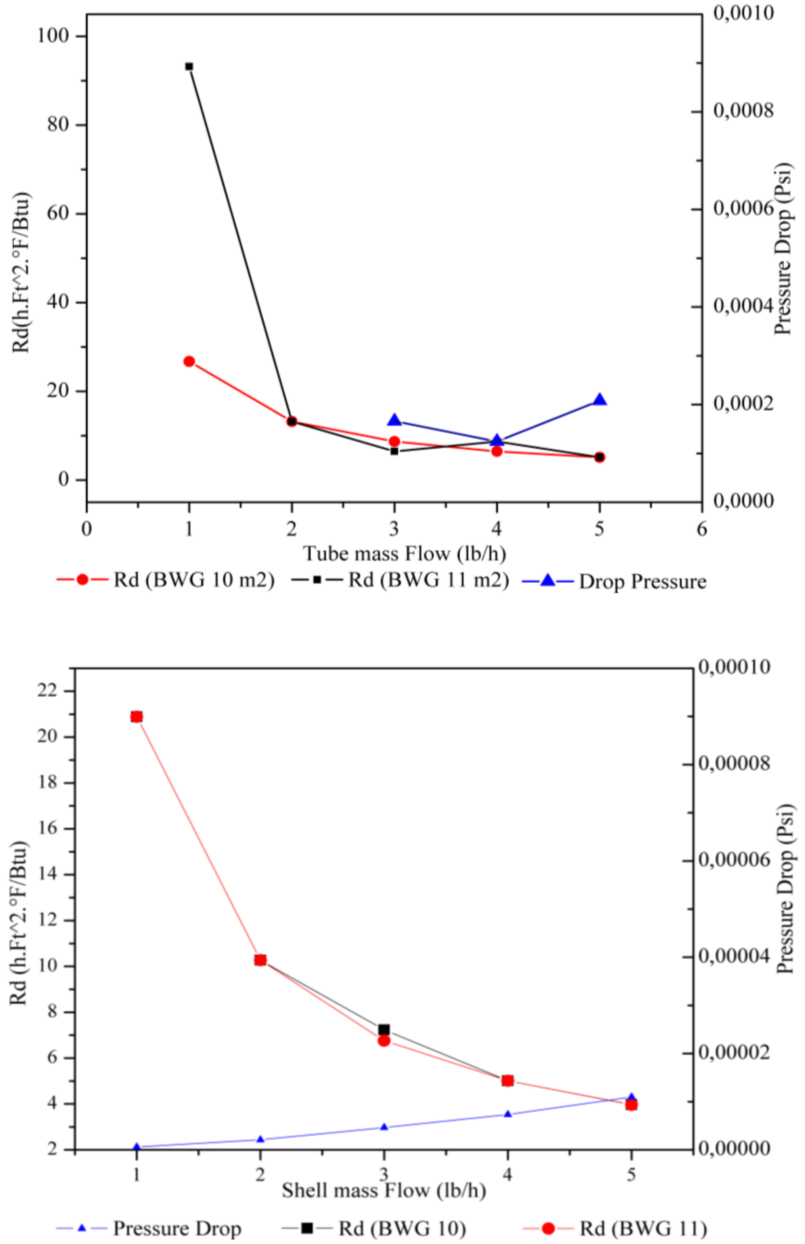

Figure 1. Behaviour of the Rd and pressure drop in the event of changes in mass flow through the a) shell, b) tubes

It was observed that for both tube arrays the pressure drop on the shell and tube side remained the same, which indicates that the changes in kerosene flow only increase the pressure drop due to the hydraulic losses in the equipment, but these do not differ for the two types of arrays tested, which is due to the fact that these two arrays have the same geometries with very similar dimensions. A detailed analysis of the pressure drop behavior shows a linear behavior, but the behavior is actually quadratic, a difference that is due to the small changes evaluated in the flow. On the other hand, when evaluating the changes in the flow of the tubes as shown in figure 2b, a decrease in the pressure drop is observed when the flow is 4 lb/h, a result that is not in accordance with the phenomenological behavior of the exchanger, which is explained by the low mass flow values evaluated in the case study, but an increasing linear trend is observed at high flow values. As for the obstruction factors shown in Figure 2, no significant changes are shown, as expected given that the temperatures in both cases were the same, in addition to the varied mass flow through the shell, however in the run a point of discrepancy was observed for a flow of 3lb/h, which is due to the approximations in significant figures considered in the study. Figure 2b shows that the selection of the BWG 10 array has a better performance in terms of obstruction factor than the BWG11 array, for the same conditions evaluated, which is desirable from the point of view of heat transfer.

CONCLUSION

The analysis of parameters and their influence on the design of tube and shell heat exchangers was presented, determining the effect of the variation of the circulating mass flow in the shell and tubes, the choice of a tube arrangement, and the comparison of two of these arrangements, which were made taking into account the overall heat variation of the system and the pressure drop in the tubes. We compared the performance of two case studies from the establishment of initial operating conditions that remained fixed and two arrangements described at the beginning of the study, also analyzed the changes in the total heat transfer coefficients (You and Uc), and the global heat transferred, in addition to the obstruction factor that the equipment would have under the same operating conditions.

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