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A THEORICAL ANALYSIS FOR PRESSURE DROP AND HEAT TRANSFER IN CORRUGATED PLATE HEAT EXCHANGERS

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Abstract. *Plate heat exchangers are devices commonly used in industry due to their high efficiency and ease cleaning. Although its first use was in food sector, nowadays this equipment can be found in most industrial segments, like chemistry and oil industry. Due the facility of fabrication, the corrugated gasket plate heat exchanger is amply utilized in those segments. However, most of the correlations available in the literature were developed to predict the thermal-hydraulic performance of plate heat exchangers applied to the food segment. Thus, the main objective of this work is to study the existent correlations and adapt them for prediction of the thermo-hydraulic behavior of such devices when applied to other industry segments, in special the oil sector.*

Keywords: corrugated plates, heat exchanger, heat transfer, pressure drop

1. INTRODUCTION

Plate heat exchangers are devices widely used in industry, since 1930. The first application was for the trade of heat in liquid-liquid flows, in the food sector due to their efficiency and ease of cleaning (Subbiah, 2012). Nowadays, these exchangers are employed in most of the major industries, including oil and chemistry, owing to its excellent heat transfer performance and high structural compactness (Zhi-jian *et al*, 2008). Among the several configurations, the gasket plates is the most common model of heat exchangers, which, however, present tight limitations concerning high pressures and temperatures. One of the most popular gasket plate heat exchanger is the corrugated type (Fig. 1), due the facility of the device mounting.
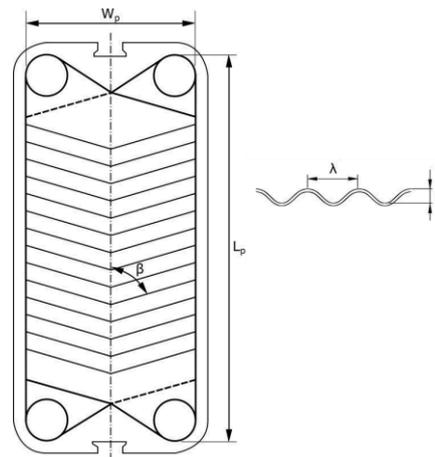

Figure 1: Schematic drawing of corrugated plate heat exchanger's plate.

Figure 1 shows a schematics of a corrugated plate, where W_p is the plate horizontal effective width and L_p is the vertical length (both dimensions include the nozzles), λ is the corrugation pitch, β is the chevron angle and a is the corrugation depth.

Many researchers discuss about what are the best correlations for predicting the pressure drop and the heat transfer in this heat exchanger. Most of them are based on experimental results and the input data include the plate geometrical parameters, such as the chevron angle (β), and operating parameters, such as mass velocity (G) and mass flow ($\dot{m}$). It

should be noted that these last two parameters influence directly the Reynolds number. Moreover, some authors analyzes the pressure drop by evaluating the Fanning friction factor (f) and others the Moody friction factor (ξ).

Plate heat exchange correlations from the literature can be applied from laminar flows, with low values for Reynolds numbers, up to turbulent flows, usually for Reynolds number equal or higher than 2000. However, some authors propose correlations which apply only to specific plate geometries, such as the one of Arsenyeva *et al.* (2016), which is limited to a specific range of the ratio between the corrugation depth and the corrugation pitch.

2. GEOMETRIC AND THERMAL-HYDRAULIC PARAMETERS

According Talal *et al.* (2017) the major parameters used for thermal and hydraulic analysis of corrugated plate heat exchangers are the hydraulic diameter, Reynolds number, Nusselt number and friction factor. These authors established two different definitions of hydraulic diameters:

$$d_e = 2a \tag{1}$$

where d_e is valid for circular cross section tubes and d_h, used for noncircular tubes (and which represents a dimension equivalent to a circular tube), given by:

$$d_h = \frac{2a}{\varphi} \tag{2}$$

where φ is valid for the corrugated plate heat exchanger and is a function of geometric parameters, being given by:

$$\varphi(X) = \frac{1}{6}\left(1 + \sqrt{1 + X^2} + \sqrt{1 + \frac{X^2}{2}}\right) \tag{3}$$

$$X = \frac{\pi a}{\lambda}$$

The Reynolds number is defined as:

$$Re = \frac{\rho V D_h}{\mu} \tag{4}$$

where ρ, and μ are the density and viscosity of the fluid respectively, V is the mean velocity of the fluid, D_h is the flow diameter.

The Reynolds also are represented in terms of mass velocity (G), defined as follows:

$$Re = \frac{G d_h}{\mu} \tag{5}$$

where d_h is the hydraulic diameter and G is defined as follow:

$$G = \frac{\dot{m}}{A} \tag{6}$$

where A is the area of fluid flow and is represented by:

$$A = N_p \cdot a \cdot W_p \tag{7}$$

where N_p is the number of flow passages in the plate, a is the corrugations depth and W_p is the horizontal width of the plate.

The Nusselt number is:

$$Nu = \frac{h D_h}{\kappa} \tag{8}$$

where h and κ are the film heat transfer coefficient and the thermal conductivity respectively.

The Moody friction faction (ξ), to be employed later in this paper, is defined as:

$$\xi = \frac{2\Delta p d_h}{\rho V^2 L_p} \tag{9}$$

The Fanning friction factor (*f*) can be defined as a function of Moody friction factor, as follows:

$$\xi = 4f \tag{10}$$

3. THERMAL AND PRESSURE DROP ANALYSIS

From the geometrical based parameters, one can obtain the heat transfer coefficient and the pressure drop of the fluid flowing through the channels formed by parallel corrugated plates.

The correlation suggested by Shah *et al.* (2003) for pressure drops, shows the dependency of the Fanning friction factor of the geometry is and is given by:

$$\frac{1}{\sqrt{f}} = \frac{\cos \beta}{\sqrt{\left(0.045 \tan \beta + 0.09 \sin \beta + \frac{f_0}{\cos \beta}\right)}} + \frac{1 - \cos \beta}{\sqrt{3.8 f_1}} \tag{11}$$

This correlation can be used for laminar and turbulent flows, making adjustments of the f_0 and f_1 parameters, which are functions of the Reynolds number, using the following expressions:

$$f_0 = \begin{cases} \dfrac{16}{Re} & for\ Re < 2000 \\ \dfrac{1}{(1.56\ \ln(Re) - 3.0)^2} & for\ Re \geq 2000 \end{cases} \tag{12}$$

and,

$$f_1 = \begin{cases} \dfrac{149.25}{Re} + 0.9625 & for\ Re < 2000 \\ \dfrac{9.75}{Re^{0.289}} & for\ Re \geq 2000 \end{cases} \tag{13}$$

Therefore, the pressure drop for corrugated plate exchangers is given by:

$$\Delta p = \frac{1.5 G_p^2 n_p}{2\rho_i} + \frac{4 f L_p G^2}{2 d_e}\left(\frac{1}{\rho}\right)_m + \left(\frac{1}{\rho_o} - \frac{1}{\rho_i}\right) G^2 \pm \rho_m g L_p \tag{14}$$

where G_p is the fluid mass velocity in the entrance regions, ρ_i and ρ_o are the in and out fluid densities, respectively. This correlation is valid for *10°<β<80°*. The accuracies for industrial plates is ±40% for Fanning friction factor and ±30% for Nusselt number.

The Nusselt number correlation proposed these authors, used to predict the thermal behavior of plate heat exchangers is:

$$Nu = 0.205 Pr^{1/3}\left(\frac{\mu_m}{\mu_w}\right)^{1/6} (f Re^2 \sin(2\beta))^{0.374} \tag{15}$$

where Pr is the Prandtl number, μ_m and μ_w are the mean and local (over the wall) fluid viscosities, respectively.

Arsenyeva *et al.* (2016) proposed another correlation for friction coefficient and Nusselt number. In these correlations, the Moody friction factor is used, obtained from the following expression:

$$\xi = 8\left[\left(\frac{12 + p_2}{Re}\right)^{12} + \frac{1}{(A + B)^{\frac{3}{2}}}\right]^{\frac{1}{12}} \tag{16}$$

where A and B constants that depend on the Reynolds number and on the geometrical parameters, which are, in turn, defined as:

$$A = \left[p_4 \ln\left(\frac{p_5}{\left(\frac{7 \cdot p_3}{Re}\right)^{0,9} + 0.27 \cdot 10^{-5}}\right)\right]^{16} \tag{17}$$

and,

$$B = \left(\frac{37530 \cdot p_1}{Re}\right)^{16} \tag{18}$$

where p_1-p_5 are constants, which dependent on the geometrical parameters of the plate and are defined as:

$$p_1 = e^{(-0.15705 \cdot \beta)} \tag{19}$$

$$p_2 = \frac{\pi \cdot \beta \cdot \gamma^2}{3} \tag{20}$$

$$p_3 = e^{\left(-\pi \cdot \frac{\beta}{180} \cdot \frac{1}{\gamma^2}\right)} \tag{21}$$

$$p_4 = (0.061 + (0.69 + \tan(\beta))^{-2.63}) \cdot (1 + (1 - \gamma) \cdot 0.9 \cdot \beta^{0.01}) \tag{22}$$

$$p_5 = 1 + \frac{\beta}{10} \tag{23}$$

where, γ is the ratio between the double corrugation depth and the corrugation pitch:

$$\gamma = \frac{2 \cdot a}{\lambda} \tag{24}$$

The friction factor correlation can be used for laminar and turbulent flow patterns, so as for Shah's correlation. In this case, the pressure drop proposed for Arsenyeva *et al.* (2016) is given by:

$$\Delta p = \left(\xi \frac{L_p}{d_e} \frac{\rho V^2}{2} + \xi_{DZ} \frac{\rho V^2}{2}\right) \chi + 1.3 \frac{\rho V_{porti}^2}{2} \tag{26}$$

where V is the fluid velocity in the exchanger, V_{porti} is the fluid velocity in the nozzles, χ is the number of passes for the fluid and ξ_{DZ} is the coefficient of local hydraulic resistance in distribution zones in the plate, defined according to Arsenyeva *et al.* (2013) as:

$$\xi_{DZ} = 38 \frac{\xi_\beta(Re)}{\xi_\beta(2700)} \tag{27}$$

The Nusselt number is given by:

$$Nu = 0.065 Re^{\frac{6}{7}} \left(\frac{\psi\xi}{F_x}\right)^{\frac{3}{7}} Pr^c \left(\frac{\mu}{\mu_w}\right)^{0,14} \tag{28}$$

where F_x is the coefficient of surface area enlargement due to corrugation, ψ is the share of pressure loss due to friction on the wall in total loss pressure.

According Arsenyeva *et al.* (2013) the heat transfer results compared with data with an error of less than 15% when compared with another literature correlations.

4. RESULTS

These two correlations were implemented in EES software and the results obtained are compared in Figs. 2 to 10. The test parameters were: constant mass flow of both fluids for varying inlet temperatures. The same chevron angle was considered for both heat exchange between fluids simulated cases.

The geometric parameters selected for the present analysis were: chevron angle (β) of 30°, corrugation depth (*a*) of 0.004 m, corrugation pitch (λ) of 0.01 and the distances W_p and L_p equal to 0.5m and 1.0 m respectively.

The heat transfer between two liquids using a corrugated plate heat exchanger was the selected case for the present study. The results presented concern the hot fluid side in the heat exchanger. Both fluids were water and the hot water was considered flowing vertically upwards (see Fig.1). Different mass flows were considered (1.6 to 2.5 kg/s) and the hot fluid inlet temperature ranged from 90 to 120°C.

Figures 2 and 3 show the curves that related the Reynolds number to the temperature, using the correlations described in this text. The difference between these results can be attributed to the difference of the definitions of the hydraulic diameter utilized in the determination of the Reynolds number, adopted by the authors: Shah's correlation used the hydraulic diameter (d_h) and Arsenyeva's correlation used the equivalent diameter (d_e). Actually, the definitions of mass velocity and of the fluid flow velocity are related to this diameter. Therefore, different values for the Reynolds number for the same fluid flow can be obtained.
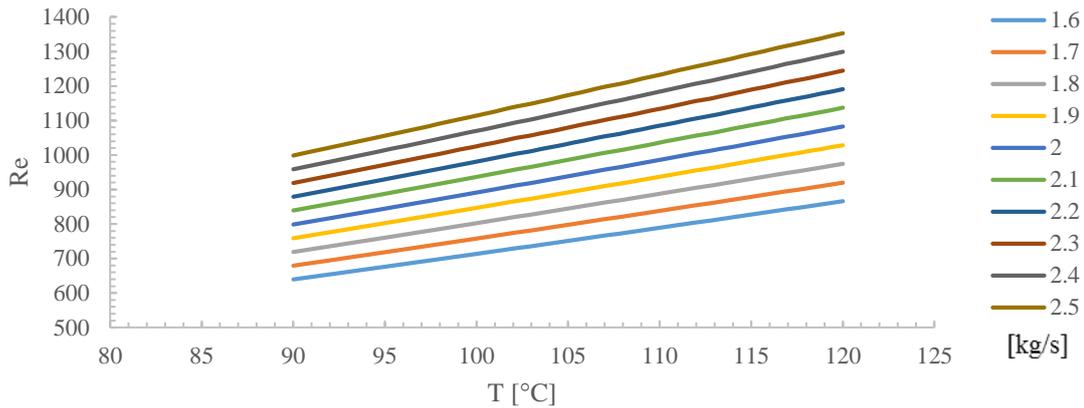

Figure 2: Reynolds number variation with temperature variation. Shah's correlation

The indexes ranging from 1.6 to 2.5, used in both graphics, represent the different values of mass flow used in the tests. As expect, larger is the mass flow, larger is the Reynolds number. It is possible to see that the Reynolds number increase with the mass flow increased allied to the temperature increase.
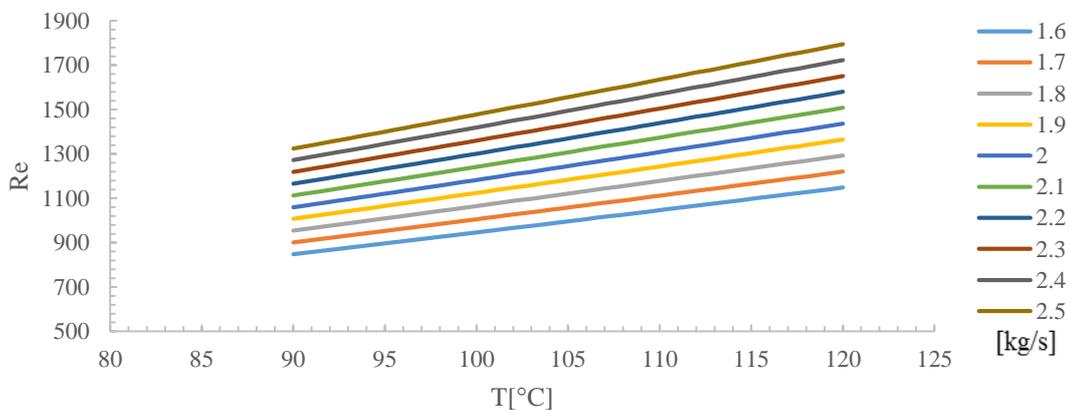

Figure 3: Reynolds number variation with temperature variation. Arsenyeva's correlation.

Figures 4 and 5 show the variation of pressure drop as a function of the temperature. The curve behavior is as expected, once the friction factor decrease with the decrease of the fluid viscosity and, in turn, viscosity decreases with the increase of temperature.

Once the tests were made with liquid-water, the third parcel in Shah's correlation (momentum effects) (See Eq. 14) are neglected, as the inlet and outlet densities present no significant variation with the water temperature and so this parcel value is so much smaller if compared to the others in the pressure drop correlation.
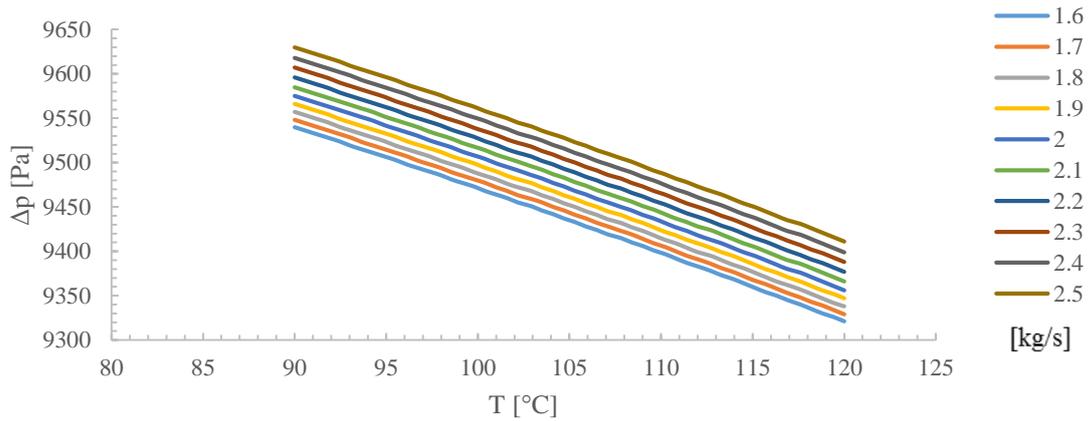
Figure 4: Pressure drop variation with temperature variation. Shah's correlation

The difference in the level and behavior of the pressure drop curves can be explained by the fact that the Shah's correlation takes into consideration the gravity effect in the pressure drop and the value of this parcel is much larger then the other terms on the equation. Actually, Arsenyeva's correlation does not consider the gravity effects, but only the pressure loss (due the viscous friction) and port effects (or entrance effects).
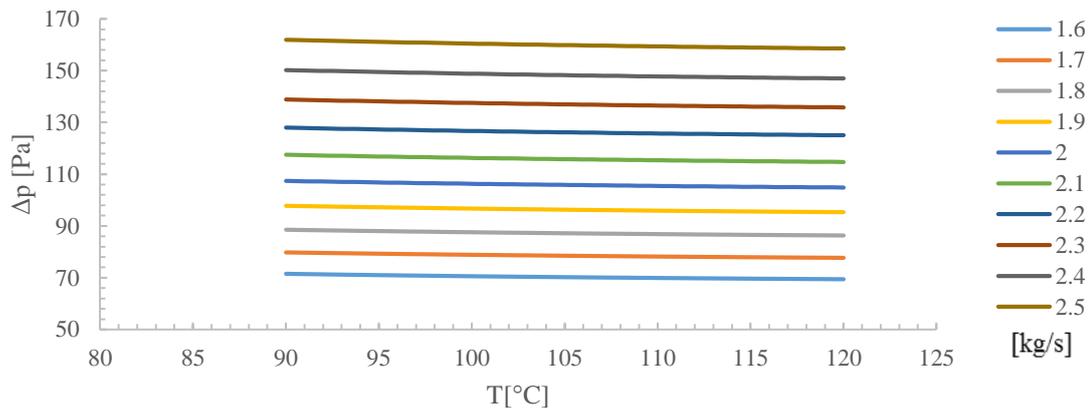
Figure 5: Pressure drop variation with temperature variation. Arsenyeva's correlation.

The graphics in figure 6 and 7 show the behavior of the Nusselt number with the temperature increase. As expected, the increase of temperature and, consequently, the increase in the Reynolds number, results in an increase of Nusselt number, once one is proportional to the other.
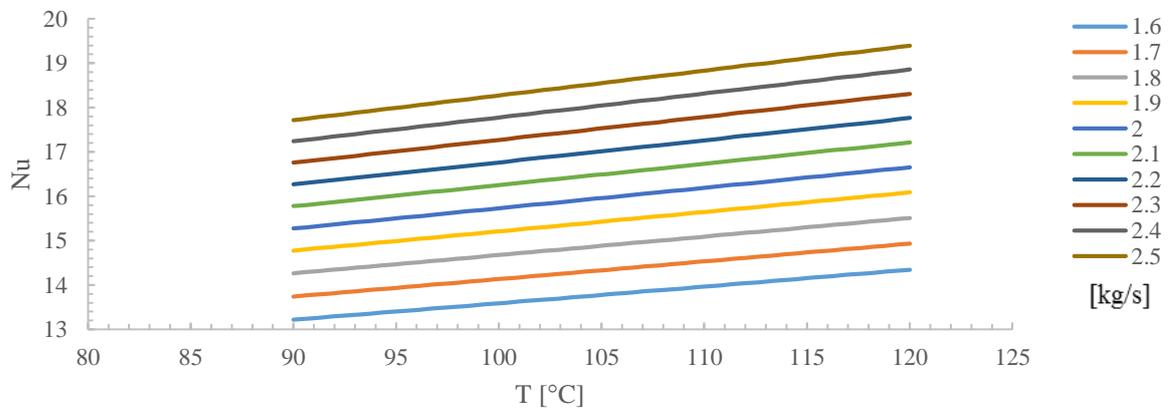

Figure 6: Nusselt number variation with the temperature increased. Shah's correlation.

The difference between the graphics in the figures 5 and 6 are explained by the different values of the Reynolds number obtained from the correlations, as previously stated in the Figures 2 and 3.
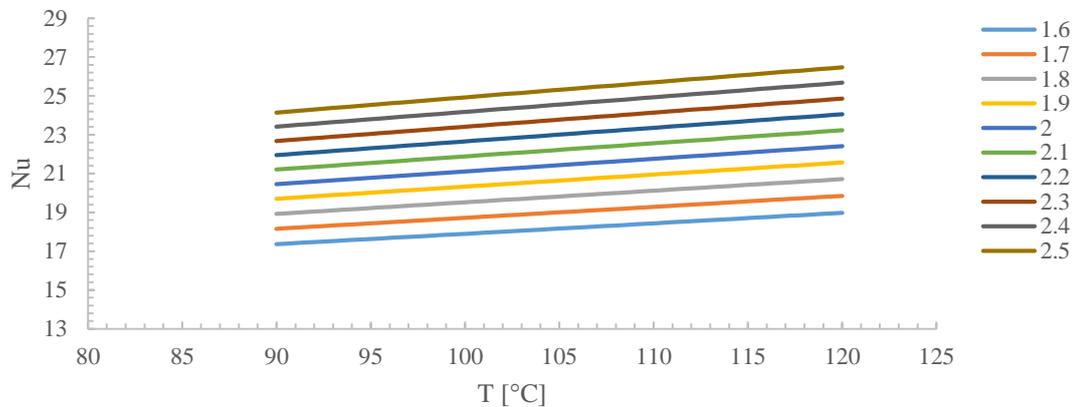

Figure 7: Nusselt number variation with the temperature increased. Arsenyeva's correlation.

In figures 8 to 10 the results for correlations in the more difficult test conditions are compared, which are the maximum mass flow ($\dot{m} = 2.5\ kg/s$) and the maximum temperature (T=120°). In Figure 8 one can see that the Nusselt number obtained from the Arsenyeva's correlation area larger than those obtained from Shah's correlation.
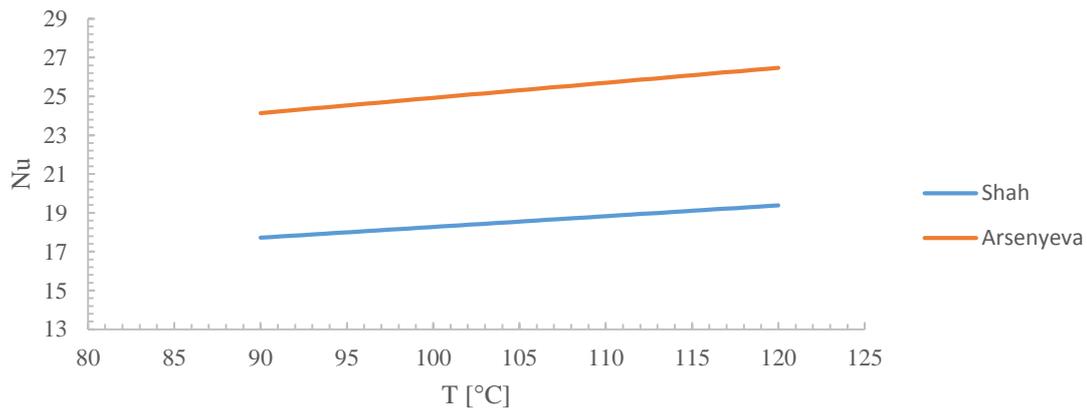

Figure 8: Comparison of Nusselt number.

Figure 9 shows the large difference obtained for the pressure drop predictions between the two correlations. This large difference of more than 9000 Pa is explained by the fact that in Shah's correlations, all the pressure drop parcels are computed, including: entrance, exit, momentum, friction and elevation (for a vertical exchanger). On the other hand, Arsenyeva's correlation does not consider the elevation effects, which, for the water flow, represents more than 50% of the total pressure drop.
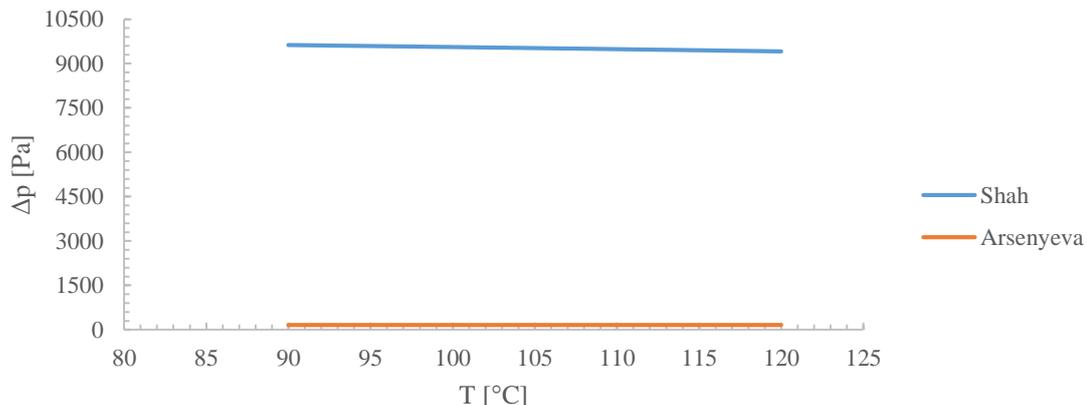

Figure 9: Comparison of pressure drop.

Figure 10 represents the Arsenyeva's correlation pressure drop behavior with the temperature increase, when elevation effects are taken into consideration, As seen in this graphic, the values for pressure drop when the elevation effects are included in Arsenyeva's correlation are approximately the same obtained using the Shah´s correlation. One can also see that the different consideration of hydraulic diameter of both correlations affected very little the pressure drop predictions.
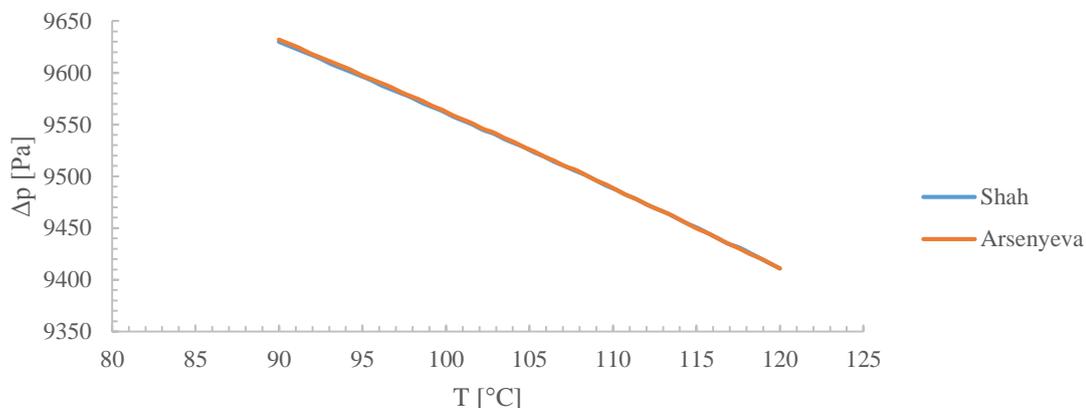

Figure 10: Pressure Drop comparison with elevation effects included.

5. CONCLUSIONS

The results show that the pressure drop and Nusselt number values can be obtained from correlations available for corrugated plate heat exchangers. The analysis was made considering the inlet temperature variable, when the mass flow is kept constant. The temperature variation affect directly the fluid properties and, consequently, the fluid mass velocity and the core velocity.

As expected, the Nusselt number increases with the Reynolds number. Furthermore, the pressure drop for both correlations decrease with the temperature decrease, as the fluid viscosities decrease with the increase of temperature.

Moreover, the pressure drop values for the both correlations are quite different and this difference was explained by the fact that Arsenyeva *et al.* (2016) correlation does not consider the pressure drop effects due to gravity forces and, consequently, elevation effects are neglected. When the elevation effects are included on the Arsenyeva *et al.* (2016) correlation, the difference becomes negligible, although the correlations are based on different hydraulic diameter definitions.

6. REFERENCES

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