Experimental Investigation of the Characteristics of a Chevron Type Gasketed-Plate Heat Exchanger

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Abstract—In this study, an experimental set-up is designed and constructed to determine the characteristics of an industrial gasketed-plate heat exchanger with chevron plates. Experiments are performed to measure the temperatures and volumetric flow rates at all ports and the pressure drops between inlet and outlet ports at different channel Reynolds numbers (450-5250). A gasketed-plate heat exchanger with different number of plates with city water as the working fluid for both hot and cold sides is utilized. Mass flow rates and inlet temperatures are changed for different cases during experiments. Pressure drop and temperature values are measured close to inlet and outlet ports to minimize the environmental effects. Volumetric flow rates are measured before the heat exchanger where a fully developed regime is obtained. Results are used to develop new correlations for the heat transfer coefficient and the friction factor for pressure drop calculations for the chevron plates tested. Obtained correlations are compared with correlations in literature.

Keywords—Gasketed-plate heat exchanger, experimental performance, chevron, heat transfer, correlation.

I. INTRODUCTION

Heat exchangers are used for space heating, air conditioning, waste heat recovery, power production, chemical processes and general heating and cooling processes [1]. They have a wide application field with many different types.

Recently, plate heat exchangers are commonly used when compared to other types of heat exchangers such as shell and tube type in heat transfer processes because of their compactness, ease of production, sensitivity, easy care after set-up and efficiency [2, 3, 4].

There are several correlations in the literature for different Reynolds numbers and chevron angles for gasketed plate heat exchangers [5]. However, these correlations do not give the right answers for all plate types because the plate geometry is

Figure 1: Schematic view of the experimental set-up
the major factor that affects heat transfer and pressure drop [6]. Thus, performance tests are necessary to specify the characteristics of a specific plate [7].

A literature survey has been performed to examine the experimental set-ups of similar studies and the method for developing the Nusselt number (Nu) and friction factor (f) correlations by using the experimental data. Afonso et al. [8] developed a convective heat transfer correlation for stirred yoghurt in the cooling stage of pasteurization in a plate heat exchanger by using a simplified and effective numerical method assuming the flow is two dimensional in flat plate geometry. To simplify the complex flow of yoghurt which is a non-Newtonian fluid; the flow of the fluid inside a complex corrugational pattern is assumed to be a bi-dimensional flow in flat plate geometry. The results of numerical simulations were compared with the experimental results and were verified. Bobbili et. al [9] investigated the flow and pressure drop across the port to channel in a plate heat exchanger for the Reynolds range of 1000-17000 using water as the working fluid for both hot and cold side of the exchanger. Using the pressure probes which are inserted into the inlet and exit ports of the channel, the channel pressure drops were measured along with the overall pressure drop of the plate heat exchanger for various flow rates. Analyzing the data of channel and mean channel pressure drops, a simplified non-dimensional channel velocity has been suggested.

Gut et al. [7] suggested a procedure for parameter estimation for plate heat exchangers by utilizing the experimental data from multiple heat exchanger configurations. The effects of the plate heat exchanger configuration on the parameters were investigated. Armfield FT-43 heat exchanger was used to verify the procedure. Using the developed correlations as a result of this study, a sizing problem was solved. Islamoglu and Parmaksizoglu [10] investigated the correlations for fully developed Nusselt number and friction coefficients for air flowing inside the corrugated channels of plate heat exchangers. Experimental data were obtained for the channel height of 5 and 10 mm for the Reynolds range of 1200 to 4000. As a result, they suggested that both the Nu and f coefficients increase with the rising height of the channel.

Muley and Manglik [11] used a single pass, U-type, counter flow plate heat exchangers with three different plates having different chevron angles in their study. The effect of chevron angle and the enlargement factor (ϕ) on pressure drop and heat transfer was investigated by using water for a flow rate range of 600<Re<1000 and the performance was compared with the performance of flat plates. As a result of these experiments, it was seen that as the chevron angle increases the amount of heat transfer is multiplied by 2-5 of that of flat plates. Moreover the pressure drop of the plate heat exchanger is 13 to 44 times higher than the pressure drop of the flat plate. As a result of this study, correlations for Nusselt number and friction coefficients were developed. Warnakulasuriya and Worek [12] studied the heat transfer and pressure drop characteristics of a commercial plate heat exchanger having an absorbent salt solution as the working fluid which serves as a sub-cooler in a refrigeration cycle. A computer program was developed and experimental data was curve fitted to establish power-law equations for Nu and f. The effect of viscosity variation due to temperature change was also taken into account.

II. OBJECTIVE OF THE STUDY

The aim of this study is to design and construct an experimental set-up to analyze the characteristics of a chevron type gasketed-plate heat exchanger at different Reynolds numbers and for various number of plates. The design and construction of the set-up, as well as the experiments performed using this set-up and obtained correlations are explained in detail in the following sections.
III. EXPERIMENTAL SET-UP

The experimental set-up has two separate cycles which are hot water and cold water cycles (Figure 1). An isolated hot water tank is designed and heated with electrical resistance which can heat up to 38 kW. To regulate and keep the flow rates as stable as possible, globe valves are used at the outlets of the 3kW pumps. Flow rates are read from magnetic flow meters as volumetric flow rate. Temperature and pressure drops are read from the vicinity of inlet and outlet ports with specialized adaptors. J type (Fe – Cu, %45Ni) thermocouples which have special covers are used to prevent fouling that can effect temperature measurements. Pressure transmitters are used to measure the pressure drop values between inlet and outlet ports. The instruments are calibrated in laboratory conditions and have accuracies which are shown in Table 1.

While test set-up is designed, it is considered to test not only one heat exchanger but also a wide range of heat exchangers. Thus, flexible connections are used at the heat exchanger ports and a solid and compact design with large tanks is used (Figure 2 and Figure 3). A more detailed description of the constructed test set-up is explained by Akturk et al [13].

Experiments are conducted for different temperatures and flow rates to determine the characteristics changing with the Reynolds number and Prandlt number.

The main issue for conducting experiments is maintaining steady state conditions for each measurement. Temperature, flow rate and pressure drop values have to be read in steady state conditions. Thus, firstly, the heated water and the cold water are sent to the heat exchanger with the help of pumps. After regulating the flows and reaching stable temperatures and flow rates, experiments are started. Temperature values are taken with a data taker in every second and pressure drop values are read every 30 seconds while the experiments run.

Table 1: Accuracies for measurement devices [14, 15, 16, 17]

<table>
<thead>
<tr>
<th>Device</th>
<th>Accuracy</th>
</tr>
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<tbody>
<tr>
<td>Pressure Transmitter</td>
<td>±%0.075</td>
</tr>
<tr>
<td>Flow Meter</td>
<td>±%0.4</td>
</tr>
<tr>
<td>Datalogger</td>
<td>±%0.45</td>
</tr>
<tr>
<td>Kreon 3D Aquilon Laser Scanner</td>
<td>5µm</td>
</tr>
</tbody>
</table>

IV. EXPERIMENTAL PROCEDURE

Plate measurement is performed with 5µm accuracy with a Kreon 3D Aquilon laser scanner and all required data is taken from the 3D model of the working plate such as port diameter, plate thickness, mean mass channel gap, etc. A chevron plate which has 0.1416m² expanded heat transfer surface area and 1.304 surface enlargement factor is used in the experiments (Figure 4). Plate heat exchangers with 10 and 21 plates are used with a U type arrangement and one pass for each fluid. The 10 plate package heat exchanger consists of 5 channel passes for hot fluid and 4 channel passes for cold fluid which have a counter flow corrugation pattern. The 21 plate package consists of 10 channel passes for each fluid with a counter flow corrugation pattern.
V. THEORETICAL CALCULATIONS

Basic equations of heat transfer are used and explained below to appreciate the data taken from experiments [5].

After calculation of mass flow rates from volumetric flow rates by $\dot{m} = \dot{V} \rho$; required heat duty is given by:

$$Q = \dot{m}_C p h \left( T_{h, in} - T_{h, out} \right)$$  \hspace{1cm} (1a)

$$Q = \dot{m}_C p c \left( T_{c, out} - T_{c, in} \right)$$  \hspace{1cm} (1b)

where fluid properties ($\rho$, $\mu$, $C_p$) are evaluated at bulk temperatures which are calculated as:

$$T_{c, b} = \left( T_{c, in} + T_{c, out} \right) / 2$$  \hspace{1cm} (2a)

$$T_{h, b} = \left( T_{h, in} + T_{h, out} \right) / 2$$  \hspace{1cm} (2b)

To find the overall heat transfer coefficient, total surface area and log-mean temperature must be known where log-mean temperature is calculated as:

$$\Delta T_L M = \frac{T_{h, in} - T_{c, out}}{\ln \left( \frac{T_{h, in} - T_{c, out}}{T_{h, out} - T_{c, in}} \right)}$$  \hspace{1cm} (3)

and overall heat transfer coefficient is obtained from:

$$Q = UA \Delta T_L M$$  \hspace{1cm} (4)

Channel Reynolds number can be considered to characterize the flow dominantly which is found by channel mass velocity, equivalent diameter and dynamic viscosity.

$$Re = \frac{G_{ch} D_e}{\mu}$$  \hspace{1cm} (5)

where,

$$D_e \approx 2b \& \ G_{ch} = \frac{\dot{m}_{ch}}{N_{cp} b L_w}.$$  

Nusselt number is obtained by applying a coefficient pilot method. While overall heat transfer coefficient is related with heat transfer coefficient as:

$$\frac{1}{U} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{k_w}$$  \hspace{1cm} (6)

without fouling resistance. Nusselt number is defined by,

$$Nu = \frac{hD_h}{k_f}$$  \hspace{1cm} (7)

where hydraulic diameter defined as:

$$D_h = \frac{2h}{\phi}.$$  

Most correlations define Nusselt number as a function of Reynolds number, Prandlt number and the ratio of dynamic viscosities at bulk to wall temperatures [18].

$$Nu = C Re^a Pr^b \left( \frac{\mu}{\mu_w} \right)^{0.14}$$  \hspace{1cm} (8)

By eliminating Nusselt with the help of the Eq.(7) and Eq.(8), Eq.(9) is obtained as:

$$h = \left( \frac{k_f}{D_h} \right) C Re^a Pr^b \left( \frac{\mu}{\mu_w} \right)^{0.14}$$  \hspace{1cm} (9)

The coefficient $b$ is taken as 1/3 in order to simplify the calculations as the most correlations have [18]. The coefficient $a$ will be between 0 and 1 so a trial and error procedure is applied in order to find $C, h_h, h_c$.

Considering Eq.(6) and by writing Eq.(9) twice for hot side and cold side, three equations and three unknowns $C, h_h, h_c$ are obtained. Solving these equations gives the desired solution of the heat transfer processes and Nusselt number can be found by the correlation developed.

Pressure drop due to the friction factor is expressed by Eq.(10); therefore, an f correlation can be found by using Eq.(10) and measured pressure drop during experiments.

$$\Delta P = 4 \left( \frac{L_{eff} N_p}{D_h} \right) \left( \frac{G_{ch}^2}{2 \rho} \right) \left( \frac{\mu_h}{\mu_w} \right)^{-0.17}$$  \hspace{1cm} (10)

VI. EXPERIMENTAL RESULTS

A set of data taken from the experiments are used to determine the characteristics of the plate. All the data are taken under steady state conditions. A wide range of Reynolds numbers is covered to observe all possible conditions. Reynolds numbers between 450 – 5250 are investigated.

By solving Eq.(9) for hot and cold sides and Eq.(6) together a Nusselt correlation can be found. Figure 5 illustrates that increase of coefficient a from 0.1 to 1 firstly gives a decreasing C average deviation up to $a = 0.6$ and then it increases until $a = 1$. This shows that the most reliable solution for $C$ is between 0.6 and 0.7. The location between 0.6 and 0.7 can be similarly investigated and the answer $C = 0.32592$ where $a = 0.6125$ can be found with a 4.07% $C$ average deviation. Therefore the
correlation for the plate tested is found as:

$$Nu = 0.32592Re^{0.6125}Pr^{0.13}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

heat transfer between two domains. Figure 6 shows that increasing Reynolds number affects heat transfer logarithmically.

Obtained correlation is compared with existent ones in literature (Table 2). Figure 7 illustrates that the plate tested has similar behavior with the plates used by Kumar [5] and Okada [19]. The plate tested has a characteristic between these two correlations where Focke’s [20] correlation deviates.

### Table 2: Some Nusselt correlations in literature and their application ranges

<table>
<thead>
<tr>
<th>Ref.</th>
<th>$\beta^\circ$</th>
<th>Re</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kumar [5]</td>
<td>30</td>
<td>&gt;10</td>
<td>$Nu = 0.348Re^{0.663}Pr^{0.13}\left(\frac{\mu}{\mu_w}\right)^{0.17}$</td>
</tr>
<tr>
<td>Focke [20]</td>
<td>30</td>
<td>600-16000</td>
<td>$Nu = 1.112Re^{0.6}Pr^{0.5}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>150-600</td>
<td>$Nu = 0.57Re^{0.7}Pr^{0.5}$</td>
</tr>
<tr>
<td>Okada [19]</td>
<td>30</td>
<td>400-15000</td>
<td>$Nu = 0.1528Re^{0.66}Pr^{0.4}$</td>
</tr>
</tbody>
</table>

Fluid flow characteristic can be controlled with design variables such as corrugation pattern, chevron angle, corrugation depth, flow distribution patterns etc. With the help of these parameters, flow regime can change and can become turbulent even in low flow rates. This is the desired objective for plate heat exchangers. Turbulent flow regime increases the

![Figure 5: Coefficient a versus C values and its percent derivation from average](image)

![Figure 6: Heat transfer characteristic of plate (Obtained correlation curve fit with experimental results)](image)

![Figure 7: Comparison of new founded correlation with existences (Nusselt number versus channel Reynolds number)](image)

A friction factor correlation is found by compiling pressure drop measurements due to Eq.(10) and fitting a curve. With 95% confidence bounds of $f$ can be found as:

$$f = 4291Re^{-1.278} + 0.3343$$

Figure 8 shows obtained correlation curve for friction factor and the experimental results with different number of plates.
Figure 8 illustrates that increasing Reynolds number results in lower friction factors. However, increasing Reynolds number can be achieved with high flow rates which will increase pressure drop dominantly more than f factor because of the $G^2_c$ component of the Eq.(10).

VII. CONCLUSIONS

Experiments are performed with a commercial plate heat exchanger with 30° chevron angle. New Nusselt and friction factor coefficient correlations are found. The obtained correlations can be used between a Reynolds number range of 450 - 5250.

A balance is needed between heat transfer and pressure drop variables. Increasing pressure drop will also increase cost, therefore an optimum choice must be selected. This is where selection programs take part.

A heat exchanger selection program which is given by plate heat exchanger producers can select the optimum choice according to the need of the consumers. However, to code a selection program it is also an obligation to know the characteristics of the plates. As a part of this study, a user friendly computer program will be developed to calculate the heat transfer and pressure drop of a plate heat exchanger. For the given heat transfer load, inlet and outlet temperature, maximum allowed pressure drop, the most applicable plate heat exchangers are presented to the selection of the user. Traditional heat transfer equations, logarithmic mean temperature method will be used together with the correlations for Nu number and f coefficients obtained as a result of this study, to calculate the minimum necessary effective heat transfer area and the number of plates to fulfill the desired heat transfer rate and the pressure drop of the plate heat exchanger.

VIII. FUTURE WORK

Different types of plates will also be tested and investigated using the set-up constructed. Based on the experimental results obtained from the set-up and the computational fluid dynamics analysis of the same cases, new correlations can be found for the different plate geometries to be tested and analyzed.

With the result of new experiments, the selection program can also be extended for new type of plate geometries.

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