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EXPERIMENTAL INVESTIGATION AND PERFORMANCE ANALYSIS OF GASKETED-PLATE HEAT EXCHANGERS

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Abstract: In this study, the thermal and hydrodynamic performance analyses of sa elected gasketed-plate heat exchanger with different number of plates are performed experimentally. A gasketed-plate heat exchanger (GPHE) test set-up is designed and constructed to perform experimental measurements for thermal and hydrodynamic performance analyses of plate heat exchangers. The experiments are performed for an industrial chevron-type plate heat exchanger under different flow conditions for a wide range of Reynolds numbers between 450 and 5250. The temperatures at the inlet and outlet ports, the volumetric flow rates of the hot and cold fluids, and the pressure drops between the inlet and outlet ports are measured during the experiments. By using the experimental data, Nusselt number correlation for heat transfer analysis and a friction factor correlation for pressure drop calculations are developed for the chevron-type plate heat exchanger tested as a function of Reynolds number and Prandtl number. The results obtained from these new correlations are compared with several existing correlations in the literature, which were developed for different plates. Although the trends of the new correlations are similar with the ones in the literature, this study shows that every plate design needs its specific correlations for heat transfer and pressure drop calculations, since the specific correlations developed for the specific plates tested are different than the ones given in literature. Obtaining these kinds of correlations for local products is especially important for the related industry to improve with the help of local test set-ups.
Keywords: Plate heat exchanger, correlation, Chevron plate, experimental set-up

CONTALI-PLAKALI ISI DEĞİŞTİRGEÇLERİNİN DENEYSEL PERFORMANS ANALİZİ

Özet: Bu çalışmada, seçilen bir contalı-plakalı ısı değiştirgecinin farklı plaka sayılarında, ısıl ve hidrodinamik performans analizi deneysel olarak yapılmıştır. Bu doğrultuda, deneysel analizin yapılabilmesi için contalı-plakalı ısı değiştirgeci test düzeneği kurulmuştur. Deneyler, endüstriyel bir chevron tipi contalı plakalı ısı değiştirgecinin farklı akış koşullarında ve 450 ile 5250 arasında değişen Reynolds sayısı aralığında gerçekleştirilmiştir. Giriş ve çıkış manifoldlarındaki akışkan sıcaklıkları, sıcak ve soğuk akışkanlar için hacimsel debiler ve basınç düşümleri ölçülmüştür. Deneysel veriler kullanılarak, ısıl analiz için Nusselt sayısı ve Hidrolik analiz için sürtünme faktörü korelasyonları, Reynolds sayısı ve Prandtl sayısının bir fonksiyonu olarak geliştirilmiştir. Bulunan sonuçlar, literatürde farklı plakalar için elde edilmiş korelasyonlar ile karşılaştırılmıştır. Türetilen korelasyon, literatürdeki bazı korelasyonlar ile benzer eğilimler göstermiş olmasına karşın, bu çalışma göstermiştir ki; her plaka boyutsal özelliklerine bağlı olarak farklı karakteristikler gösterir ve her plakanın performans analizi ayrı bir şekilde yapılmalıdır, literatürde varolan benzerleri kullanılmamalıdır. Farklı plakalı ısı değiştirgeçleri için benzer korelasyonların türetilmesi ve bu doğrultuda test düzeneklerinin oluşturulması ve denysel çalışmaların yapılması, endüstriyel ve teknolojik verimlilik açısından büyük önem taşımaktadır.
Anahtar Kelimeler: Plakalı ısı değiştirgeci, korelasyon, chevron plakası, deney düzeneği

INTRODUCTION

Heat transfer is an energy transfer which is a result of temperature difference. Heat transfer applications are widely used in industrial, commercial and sanitary systems. In these systems, heat exchangers are utilized as transfer devices which transfer thermal energy from one fluid to another that have different temperatures and that may be in contact or separated. There are different types of heat exchangers [1] and one type is the gasketed-plate heat exchanger (GPHE) which has a group of plates compressed between a front and a rear support plates. The gasketed-plates have a corrugated surface design to increase the heat transfer area and the flow between the plates is directed by the gaskets. The corrugated surface increases the heat transfer area and it also affects the flow to become turbulent even in low Reynolds numbers [2, 3]. Plate heat exchangers are widely used due to their compact design and small sizes, working temperature range and efficiency, and easy manufacturing and maintenance [1, 3, 4]. In addition, new plates can be easily added, thus, allowing an increase in the performance of the heat exchanger. In general, the flow arrangements between the plates are U and Z type flow arrangements. Most commonly used plate corrugation type is the chevron type. In chevron type plate heat exchangers, the fluid enters the inlet port and spreads through the distribution patterns to the heat transfer zone. The heat transfer zone has a triangular pattern in chevron type plate heat exchangers and the angle of the triangular pattern is defined as the chevron angle of the plate. A plate heat exchanger can have symmetric or mixed combination of plates. In the symmetric plate heat exchanger, all plates are the same, having the same chevron angle, whereas in the mixed plate heat exchanger, each plate can have different chevron angles [3, 5, 6]. Because of the complex geometries of the plates, it is very difficult to analyze the thermal and hydrodynamic characteristics either analytically or numerically. Therefore, experimental studies are important and are usually used to obtain the correlations necessary for heat transfer and pressure drop calculations.

There are several studies on plate heat exchangers with different types of fluids and plate geometries [1]. There are also several correlations in literature that show the thermal and hydrodynamic characteristics of plate heat exchangers as a function of Reynolds number and chevron angle [1]. However, these correlations may not be used for all types of plates. The plate geometry is the most important factor which affects the heat transfer and pressure drop characteristics [7]. Therefore, performance tests are usually conducted to specifically define thermal and hydrodynamic characteristics of any plate design [8].

Rao and Das [9] performed an experimental study using an industrial heat exchanger that has 40 chevron type plates. The flow rates are controlled by using throttling valves in front of the pumps. A friction factor correlation was obtained as a function of Reynolds number within the range of 1000 to 7000. Rao et al. [10] investigated the flow maldistribution from port to channel in a plate heat exchanger both theoretically and experimentally. The experimental set-up contained a cooling tower to cool the heated water in the system. It was concluded that Z-type plate heat exchangers had more flow maldistribution compared to U-type plate heat exchangers. Dović and Švaić [7] investigated the effects of geometry on heat exchanger performance. The experimental set-up used by Cerezo et al. [11] has three cycles which are hot, cold and solution cycles and it is used to observe the absorption in the solutions. The mass flow rates and pressure drops for hot and cold sides and temperatures at each port were measured. Different Nusselt number correlations were developed for laminar and turbulent flow conditions. Afonso et al. [12] investigated the non-Newtonian thermal performance of yoghurt numerically in a cooling process with a plate heat exchanger. Bobbili et al. [4] studied with water for a Reynolds number range of 1000 to 17000 with two different numbers of plates: 21 and 81. A friction factor correlation was obtained for a Reynolds number range of 900 to 10000. They also stated that the systems which have the same inlet port diameter and the pipe inside diameter minimize the flow maldistribution. Muley and Manglik [13] experimentally investigated the heat transfer and pressure drop characteristics in a U type, one pass, cross flow, chevron type plate heat exchanger. They developed correlations for Nusselt number and friction factor for Reynolds numbers ranging from 600 to 10000 for different chevron angles. Khan et al. [14] conducted experiments with symmetric and mixed chevron angle plates and with water, for Reynolds numbers between 500 and 2500 and for Prandtl numbers between 3.5 and 6.5. As a result, a Nusselt number correlation which is a function of Reynolds number, Prandtl number and Chevron angle was developed. Islamoglu and Parmaksizoglu [15] performed experiments with corrugated surfaces and air as the working fluid. The heat transfer coefficients and friction factors were calculated and it was concluded that increasing the channel height increases the Nusselt number and friction factor. Warnakulasuriya and Worek [16] used an industrial type heat exchanger to investigate the characteristics in a cooling cycle in order to find the Nusselt number and friction factor correlations and to design better systems. The correlations were found by considering an empirical approach from the literature and by applying curve fitting to the experimental data. Durmus et al. [17] performed experiments with three different plates and water as the working fluid and the empirical correlations were obtained for parallel and counter flow arrangements for each plate. They obtained similar results as compared with the results of Gut et al. [8]. Tsai et al. [6] studied hydrodynamic characteristics of a selected heat exchanger both experimentally and numerically. The experiments were performed for various flow rates and the Computational Fluid Dynamics (CFD) analyses for similar cases helped to visualize the flow inside the heat exchanger. Muley [3] investigated heat exchangers with industrial type plates and developed the experimental Nusselt number and friction factor correlations for the heat transfer and pressure drop calculations, respectively. Muley [3] used the modified Wilson-Plot method to find the empirical correlation for heat transfer calculations. Claesson [18] investigated the thermal and hydrodynamic performance of compact brazed plate heat exchangers operating as evaporators in domestic heat pumps. This study differs from most of the similar studies such that salty water (% 24 ethanol-water mixtures) was used as working fluid on one side, which changed the flow rate and made the use of Wilson-Plot method invalid. R134a and R22 were used on the other side of the heat exchanger. Other studies on optimum design of plate heat exchangers with and without pressure drop specifications, and the experimental and theoretical analyses of the effect of flow maldistribution on the thermal performance of plate type heat exchangers can be found in [19, 20].

In this study, a gasketed-plate heat exchanger test set-up is designed and constructed to perform experimental measurements for thermal and hydrodynamic performance analyses of selected chevron type plate heat exchangers. The experiments are performed for the industrial type gasketed-plate heat exchangers with different number of chevron plates and under different flow conditions within the Reynolds numbers range of 450 to 5250. The new experimental correlations for Nusselt number and friction factor are found and compared with the correlations available in literature. As a result of this study, by using the plate specific correlations, a computer program is developed for a local company for heat exchanger selection for specified conditions in the practical applications.

DESIGN OF THE EXPERIMENTAL SET-UP AND THE MEASUREMENT SYSTEM

A new experimental set-up is designed and constructed as shown in Figures1 and 2. In the set-up, there are three water reservoirs: the hot, the cold and the recovery tanks. Electrical resistances are used in the hot water tank to heat up the water and the tank is insulated by using the foam boards. In the hot side of the test set-up, there is a junction valve after the heat exchanger outlet port which allows the flow direction to switch between the recovery tank and the hot tank. Thus, the hot fluid is allowed to circulate in the system to prevent any temperature stratification in the hot tank especially near the resistance heaters during the initial heating period at the beginning of the experiments. During experimental readings, this junction is used to allow the fluid to flow into the recovery tank in order to keep constant inlet temperature at the inlet port. On the other hand, on the cold side of the test set-up, the cold water is supplied from the city supply with almost constant temperature and flows into the recovery tank after the outlet port.

In the set-up, there are two pumps located under the water tanks to obtain a positive head to prevent cavitation. After the pumps, along the flow direction, the throttling valves are used to set the flow rates to the desired values. Magnetic flowmeters are placed on the straight long and vertical pipes to provide fully developed and fully loaded flow in the flowmeters in order to measure the flow rates correctly. Flexible connections are used to mount and unmount the heat exchanger in the test set-up in order to perform the tests of different heat exchangers easily with the same set-up.

The significant design criteria for the test set-up are decided by the help of the existing studies in literature as summarized above and the experiences gained from the previous experimental studies [21,22]. The most important criterion is the fluid temperature stability at the inlet ports to obtain a steady-state condition, i.e., having constant temperatures at the inlet ports, during the experimental measurements [21, 22].The second important point is the isolation of the measurement devices from each other and from the environmental effects [12, 14] for correct measurements.

Instrumentation

Temperature measurements are made with J type thermocouples which have special stainless casings to prevent fouling on thermocouple arcs which may affect the measurements. The temperature range is determined as 0 to 100°C as the water runs as the working fluid. Although, most of the studies in the literature were performed with K type thermocouples, J type thermocouples have more linear response than K types between the specified temperature ranges [23, 24]. A data logger is used to store the temperature values for every second and to observe if the system reaches to steady state or not.

(a) (b)

Figure 1. (a) 3-D CAD model of the test set-up, and (b) Views of the set-up (hot water tank, pumps, flow meters, heat exchanger and the connections).

In order to select the flowmeter for the test set-up, the hydrodynamic system characteristics for both hot and cold sides are calculated and the conditions without cavitation are considered for locating the water tanks at the specific heights from the pumps for the required Net Positive Suction Head (NPSH). Thus, the water tanks are located 3.15 m above from the pump inlets and the standard DN50 pipes are used for the connections. Globe valves are used to set the flow rates. These valves are used as throttling valves and they help the user to adjust the flow rates by looking at the magnetic flow meter which is located near the globe valves. It is known that throttling valves cause an extra loss coefficient which reduces the flow rates [25].

One of the most important factors for heat exchanger selection or design is the pressure drop characteristics. Pressure drop is a function of number of plates, number of passes, fluid viscosity, fluid velocity and surface parameters [4, 16]. The pressure transmitter is selected for the test set-up by considering the maximum allowable pressure drop as 100kPa. More detailed information on the experimental set-up and the instrumentation is given in [26].

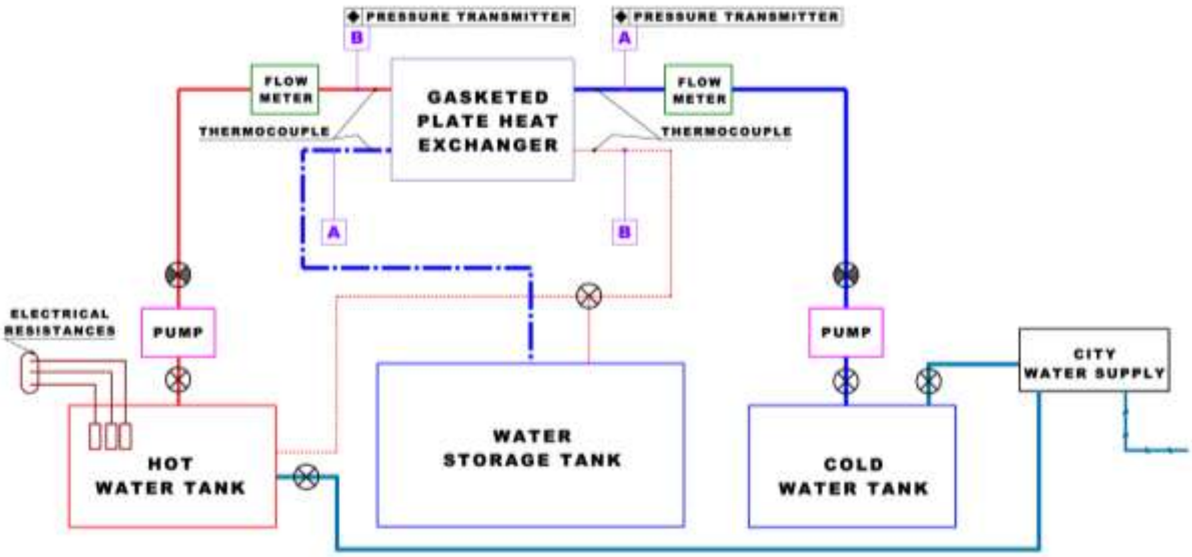


Figure 2. Schematic view of experimental test set-up.

Properties of the Gasketed-Plate Heat Exchangers

Plate heat exchangers with 10, 15 and 21 number of plates are used with a U type arrangement (Figure 3). The heat exchanger with 10 plates consists of 5 channel passes for hot fluid and 4 channel passes for cold fluid and has a counter flow corrugation pattern. The heat exchanger with 15 plates has 7 channel passes for each fluid and similarly, the heat exchanger with 21 plates consists of 10 channel passes, both with the counter flow corrugation pattern.

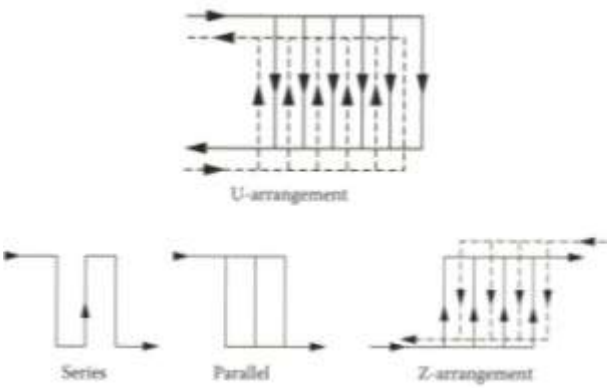


Figure 3. Flow pattern: (a) Schematic of a U-type arrangement – counter flow, single-pass flow (1x6/1x6) (b) Z-arrangement (1x4/1x4 configuration) [1]

The industrial plate geometry used in the experiments is first scanned with a 3D laser scanner which has 5µm accuracy [27]. Thus, a stereo-lithography (STL) data of the plate is obtained. According to the measurements, the plate has a 0.1416 m2 expanded heat transfer area and the surface enhancement factor is found as 1.304. Some of the plate properties are summarized on Table 1 and the characteristic dimensions of the chevron plate are given in Figure 4.

Table 1. Parameters for the heat exchanger from the STL data and the catalogues.

Parameter	Symbol	Unit	HEX
Maximum flow velocity		liter/hour	90000
Maximum working pressure and test pressure		Bar	16 & 21
Heat transfer area for unit plate (catalogue value)		m^2	0,125
Heat transfer coefficients (catalogue value)	U	W/m^2K	3489-5815
Plate material		-	AISI 316
Gasket		-	Rubber, NBR, EPDM, HNBR, VITON (FKM)
Plate width between gaskets	L_w	m	0,23
Plate height between ports	L_v	m	0,6058
Plate height between gaskets	L_p	m	0,537
Plate width between ports	L_h	m	0,1451
Port diameter	D_p	m	0,069
Chevron angle	β	°	30
Enhancement factor	ϕ	-	1,304
Surface area	A_{1p}	m^2	0,10856
Extended surface area	A_1	m^2	0,14159
Corrugation pitch	p_c	mm	9,84
Plate pitch	p	mm	3,3
Plate thickness	t	mm	0,45

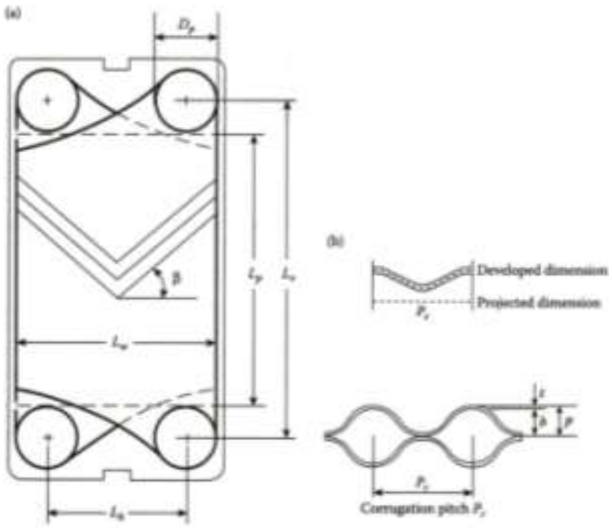


Figure 4. (a) Main dimensions of a chevron plate; (b) developed and projected dimensions of a chevron plate cross-section normal to the direction of troughs [1]

Experimental Conditions and Procedure

The experiments are performed for different inlet temperatures and flow rates to determine the heat transfer characteristics of a chevron plate group for different Reynolds numbers and Prandtl numbers. By changing the number of plates, a wide range of Reynolds numbers and also a wide range of pressure drops are obtained. The flow rates between 0.57 m3/h and 6.6 m3/h are used. The working temperature ranges of 53°C - 90°C for hot water and 9°C - 25°C for cold water sides are used for different experimental cases. As a result, Reynolds numbers from 450 to 5250 are obtained during the experiments.

According to the experimental procedure applied, after heating the water to a desired temperature, the system runs for just the hot side and then the electrical resistances are turned off. The flow temperature is read on the hot side and after reaching the steady state, the junction valve is adjusted for hot water not turn back into the hot side tank and the cold side pump starts pumping the cold water into the heat exchanger. When the outlet port readings reach to steady state values, the measurements are made. The temperature values are recorded for every second and the pressure drop is read every 20 seconds.

ANALYSIS

By using the experimental data set obtained for the specific plate, the thermal and hydrodynamic performance analyses are performed by using the procedure explained below [1, 15].

Nusselt Number Correlation

To calculate the total heat transfer, volumetric flow rates measured by the magnetic flowmeters are converted to the mass flow rates at the working temperature as .

From the first law of thermodynamics, i.e., the conservation of energy, the heat duty can be found as:

$$\dot{Q} = \dot{m}_{hot} C_{p,hot} \left(T_{hot,in} - T_{hot,out} \right) \quad (1a)$$

$$\dot{Q} = \dot{m}_{cold} C_{p,cold} \left(T_{cold,out} - T_{cold,in} \right) \quad (1b)$$

To select the reliable data sets, the hot and cold side heat transfer rates are compared in each test case, and a maximum 8% deviation is accepted between the heat duties of the hot and cold sides.
To calculate the overall heat transfer coefficient (U), the total heat transfer ($\dot{Q}$), the total heat transfer surface area (A) and the logarithmic mean temperature (ΔTLM) are used as [1]:

$$\dot{Q} = UA\Delta T_{LM} \quad (2)$$

By calculating inlet and outlet temperature differences between two fluids, ΔTLM can be calculated as [1]:

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{In\left(\frac{\Delta T_1}{\Delta T_2} \right)} \quad (2b)$$

The Reynolds number is defined by using the channel mass flow rate (G_{ch}), the equivalent diameter (D_e) and the dynamic viscosity (μ)as [1]:

$$\mathrm{Re} = \frac{G_{ch} D_e}{\mu} \quad (3)$$

where, the equivalent diameter is defined as $D_e \approx 2b$, and the channel mass flow rate is calculated from:

$$G_{ch} = \frac{\dot{m}_{ch}}{N_{cp} b L_w} \quad (4)$$

The overall heat transfer coefficient without fouling can be expressed as [1]:

$$\frac{1}{U} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{t}{k_w} \quad (5)$$

The Nusselt number can be calculated as below [1]:

$$\mathrm{Nu} = \frac{hD_h}{k_f} \quad (6)$$

where, the hydrodynamic diameter is taken as

$$D_h = \frac{2b}{\varphi} \; .$$

For heat transfer calculations, the Nusselt number can be found by a trial and error method. For this purpose, a modified Wilson-Plot method can be applied by using the experimental data. In literature, most of the correlations define the Nusselt number as a function of the Reynolds number, the Prandtl number and ratio of the dynamic viscosity at the average fluid temperature to the viscosity at the wall temperature of the working fluid [3]. These correlations can usually be written in a general form as shown below:

$$\mathrm{Nu} = C\,\mathrm{Re}^{a}\,\mathrm{Pr}^{b}\left(\frac{\mu_b}{\mu_w}\right)^{d} \tag{7}$$

In Equations 6 and 7, the Nusselt number term can be eliminated and the heat transfer coefficient can be obtained as:

$$h = \left(\frac{k_f}{D_h}\right) C\,\mathrm{Re}^{a}\,\mathrm{Pr}^{b}\left(\frac{\mu}{\mu_w}\right)^{d} \tag{8}$$

where μ and μw must be evaluated at fluid bulk and wall temperatures respectively. In these equations, the coefficients b and d can be found from the literature to simplify the calculations. Therefore, in this study, b and d are taken as constants as 1/3 and 0.14, respectively [3]. Another constant in the correlation equation is a, and it can also be found in literature to be between 0 and 1. Thus, a trial and error method can be used to find constant a and the corresponding C values.

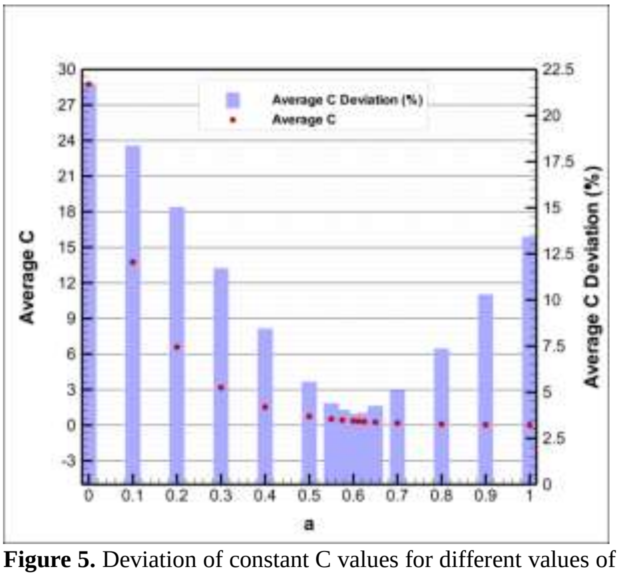


Figure 5. Deviation of constant C values for different values of constant a.

By using Equation 8, both for hot and cold sides and by using Equation 5, there will be three unknowns (C, h_h, h_c),and three equations to solve. By solving these equations for different values of constant a, different C, hh and hc values are found. By calculating the deviation of C values, an average C can be found for the Nusselt correlation. As it can be seen in Figure 5, when the value of constant a is increased from 0 to 1 with an increment of 0.1, the mean value of constant C decreases until a = 0.6, and increases after a = 0.7. Thus, most suitable a value is located between 0.6 and 0.7. Then, by repeating the same procedure for the interval between 0.6 and 0.7 for the values of a with a smaller increment, an average value of 0.32673 for the constant C can be found with a minimum deviation of 3.816 % of the average value for a = 0.6125. Thus, by using these values for the constants in Equation 7, the new Nusselt number correlation is found as below for the Reynolds number range of the experimental data:

$$\mathrm{Nu} = 0.32673\,\mathrm{Re}^{0.6125}\,\mathrm{Pr}^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad for\ 450 < Re < 5250 \tag{9}$$

Friction Factor Correlation

Pressure drop (ΔP), which is a function of friction factor (f), is described in literature [1, 5] as shown below:

$$\Delta P = 4f\left(\frac{L_{eff} N_p}{D_h}\right)\left(\frac{G_c^2}{2\rho}\right)\left(\frac{\mu_b}{\mu_w}\right)^{-0.17} \tag{10}$$

By using the measured pressure drop data and the geometrical information of the heat exchanger tested, a friction factor correlation can be found. A curve fitting method with 95% confidence level is applied to find a relation for the friction factor and Reynolds number as shown below:

$$f = 60550\,\mathrm{Re}^{-1.72} + 0.4299 \quad for\ 450 < Re < 5250 \tag{11}$$

Uncertainty Analysis

Uncertainties of a GPHE is defined by its performance characteristics and the variable that defines it such as the Reynolds number, Prandtl number, Nusselt number, and heat load. Therefore, the uncertainties of basic parameters that experimentally measured have to be calculated first. Uncertainty of a certain parameter is calculated by Equation 12 and the relative contribution of effecting parameters is calculated by Equation 13. [3, 28, 29, 30, 31]

$$U_r = \left[\left(\frac{\partial r}{\partial X_1}U_{X_1}\right)^2 + \left(\frac{\partial r}{\partial X_2}U_{X_2}\right)^2 + \ldots + \left(\frac{\partial r}{\partial X_j}U_{X_j}\right)^2\right] \tag{12}$$

$$r = \frac{\left(\frac{\partial r}{\partial X_j}U_{X_j}\right)^2}{U_r^2} \tag{13}$$

And the results of the uncertainty analysis are given in Table 2, which means that accuracy of the measurements is within the acceptable range.

Table 2. Uncertainty of the parameters.

Parameter	% Uncertainty
Reynolds number, *Re*	4,16
Prandtl number, *Pr*	2,19
Heat load, $\dot{Q}$ [W]	0,40
Overall heat transfer coefficient, *U* [W/m^{2}K]	0,41
LMTD, ΔT_{lm}[K]	0,01
Nusselt number, *Nu*	1,83
Fanning friction factor, *f*	4,18

RESULTS

The results of the thermal and hydrodynamic performance analyses and the comparisons of the new correlations with the ones in the literature are explained in detail in this section.

Nusselt Number Correlation

The new correlation curve for the Nusselt number which is given by Equation 9 and the experimental data are shown in Figure 6. Flow characteristics in a plate heat exchanger can be controlled and changed with corrugation patterns, chevron angle, and channel depth and flow distributor port geometry. Due to these parameters, flow can be turbulent even for low Reynolds numbers. This is a desired situation in plate heat exchangers since turbulent flow increases the heat transfer between two fluid domains. As it can be seen in Figure 6, as the Reynolds number increases, the heat transfer increases logarithmically. It can also be observed from Figure 6 that different number of plates used in the heat exchanger does not change the trend of Nusselt number with respect to Reynolds number.

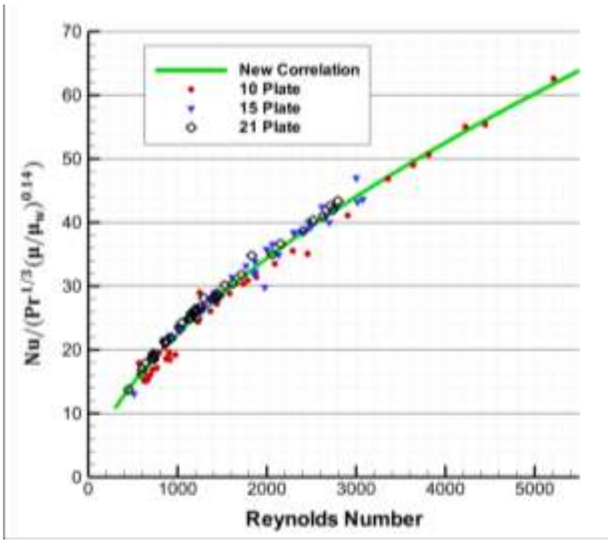


Figure 6. Nusselt number vs Reynolds number for different number of plates: new correlation and comparison with the experimental data.

Table 3. The new correlation for Nusselt number and the similar correlations from the literature (for 30° chevron angle).

	Correlation	Reynolds Number Range
New Correlation	$\mathrm{Nu} = 0.32673\,\mathrm{Re}^{0.6125}\,\mathrm{Pr}^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$	450-5250
Kumar et al. [1]	$\mathrm{Nu} = 0.348\,\mathrm{Re}^{0.663}\,\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.17}$	>10
Focke et al. [33]	$\mathrm{Nu} = 1.112\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{0.5}$	600-16000
	$\mathrm{Nu} = 0.57\,\mathrm{Re}^{0.7}\,\mathrm{Pr}^{0.5}$	150-600
Okada et al. [32]	$\mathrm{Nu} = 0.1528\,\mathrm{Re}^{0.66}\,\mathrm{Pr}^{0.4}$	400-15000

The new Nusselt number correlation is also compared to the similar correlations available in the literature in Table 3 and Figure 7. Table 3 shows that the new correlation has a similar trend with the correlations of Kumar et al. [1] and Okada et al. [32]. On the other hand, the correlation of Focke et al. [33] have much larger values of Nusselt number with respect to Reynolds number as compared with the new correlation and other two correlations from the literature.

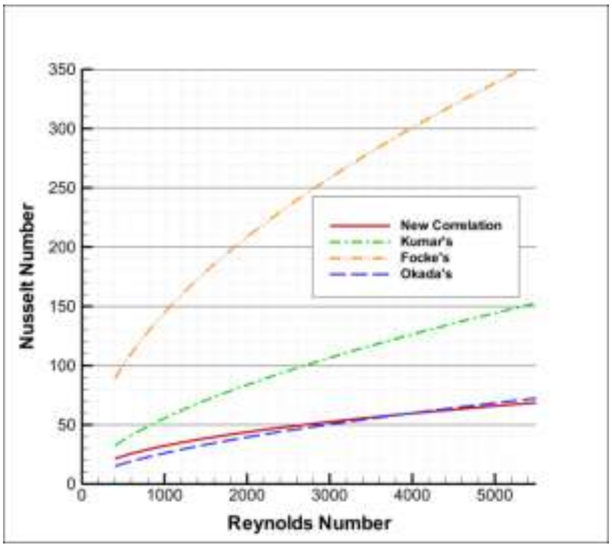


Figure 7. Nusselt number vs Reynolds number: Comparison of new correlation for Nusselt numbers with existing correlations in literature

Friction Factor Correlation

Figure 8 shows the new friction factor correlation curve and the comparison with the experimental data. It can be seen from this figure that the friction factor decreases with increasing Reynolds number. This decrease gets lower for higher Reynolds numbers and converges to a value as the Reynolds number increases. Hence, the friction losses become more effective in lower Reynolds numbers.

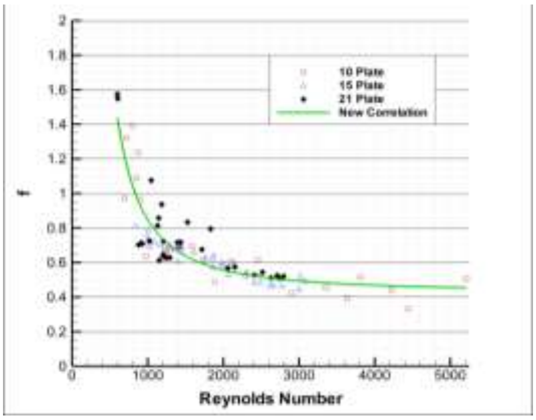


Figure 8. Friction factor coefficient vs Reynolds number: Experimental data for different number of plates and new correlation.

In Figure 8, the change in the friction factor coefficient of the experimental data with Reynolds number is plotted for different number of plates. This figure shows that the friction factor is not a function of the number of plates.

Table 4. The new correlation for friction factor coefficient and similar correlations from the literature.

	Correlation	Reynolds Number Range
New correlation	$f = \dfrac{60550}{\mathrm{Re}^{1.72}} + 0.4299$	450 - 5250
Kumar et al. [1]	$f = \dfrac{2.99}{\mathrm{Re}^{0.183}}$	>100
Focke et al. [33]	$f = \dfrac{6.7}{\mathrm{Re}^{0.209}}$	400 - 1600

Comparison of the new correlation with the existing correlations in literature is shown in Table 4 and in Figure 9. As it can be seen in this figure, the friction factors for the plates tested are less than the ones of Kumar et al. [1] and Focke et al. [33]. However, for lower Reynolds numbers, the friction factor has higher values with a steep change as can be observed from the Figure 9.

Figure 10 shows the change in the experimental pressure drop values for the plate heat exchanger with different number of plates with respect to Reynolds number and mass flow rate. As it can be seen in Figure 10 (a), increasing Reynolds number increases the pressure drop exponentially. However, pressure drop is proportional to the square of channel mass flow rate. Thus, pressure drop mostly depends on the mass flow rate rather than the friction factor. Therefore, while considering the pressure drop, it must be taken into consideration together with the mass flow rate rather than the Reynolds number. Figure 10 (b) shows how pressure drop changes with respect to mass flow rate. As it can be seen, for the same mass flow rate, the heat exchanger with fewer number of plates has larger pressure drop. As seen in Figure 10, this generalization cannot be made using Reynolds number because in plate heat exchangers, channel mass flow rate is a function of number of plates.

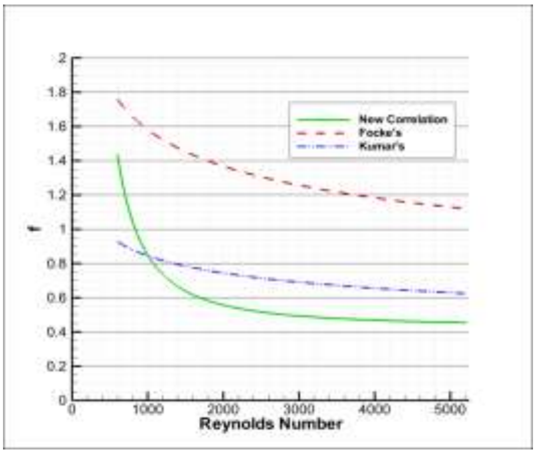
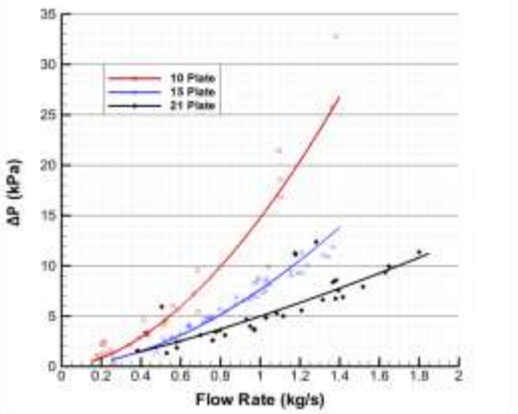


Figure 9. Comparison of new correlation curve for friction factor with existing correlations in literature.

a)



b)

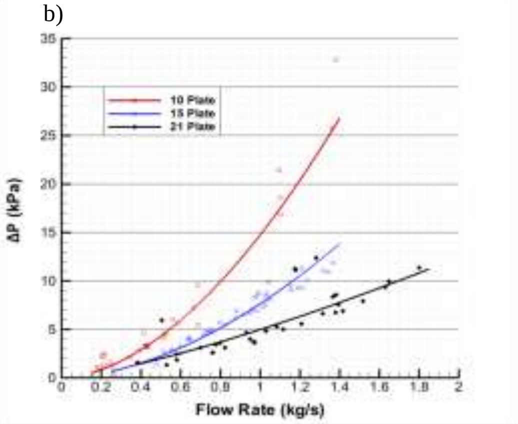


Figure 10. Pressure drops for plate heat exchangers with different number of plates with respect to a) Reynolds number and b) mass flow rate

DISCUSSION AND CONCLUSION

In this study, the experiments with gasketed-plate heat exchangers (GPHE) are performed to investigate thermal and hydrodynamic characteristics of full scale, industrial type and chevron-type gasketed-plate heat exchangers with different numbers of plates. A new experimental test set-up which is designed and constructed and the experimental results obtained for the selected plate type are described. For the selected plate type, heat exchangers with 10, 15, and 21 numbers of plates are investigated with water as the working fluid. During the tests, the flow rates between 0.57 m^3/h and 6.60 m^3/h and the inlet temperatures from 9°C to 90°C on the cold and hot sides are used.

As a result of the performance analysis made by using the experimental data, new empirical correlations for Nusselt number and friction factor coefficients are found as a function of Reynolds number in the range of 450 to 5250. Hence, the heat transfer characteristics of the tested plates were determined. The correlations were compared with the ones in the literature. Although they showed similar trends, they are all plate specific correlations.

On the other hand, as a result of this study, the wide range of experimental data obtained can be used to optimize the design of plates for thermal and hydrodynamic purposes. Flow maldistribution can be investigated for the tested plates. The experimental database can also be used as validation test cases for advanced CFD simulations. In addition, similar to this study, new and specific correlations for other type of heat exchangers and GPHE with different plate designs can be tested in the new test set-up. Different fluids can also be used to investigate viscosity effects on heat transfers and friction coefficients as a future work.

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NOMENCLATURE

Symbol	Explanation
A	Total heat transfer area [m^2]
b	Mean channel gap [m]
C_p	Specific heat capacity [$Jkg^{-1}K^{-1}$]
D_e	Equivalent channel diameter [m]
D_h	Hydrodynamic channel diameter [m]
f	Friction factor
G_{ch}	Channel mass flow rate [$kgm^{-2}s^{-1}$]
h	Convective heat transfer coefficient [$Wm^{-2}K^{-1}$]
k	Conductive heat transfer coefficient [$Wm^{-1}K^{-1}$]
L_w	Plate width inside gasket [m]
L_{eff}	Effective flow length between ports [m]
$\dot{m}$	Mass flow rate [kgs^{-1}]
N_{cp}	Number of channel per pass
N_p	Pass numbers
Nu	Nussel tnumber
P	Pressure [Pa]
ΔP	Pressure drop [Pa]
Pr	Prandtl number
$\dot{Q}$	Heat transfer rate [W]
Re	Reynolds number
t	Plate thickness [m]
T	Temperature [°C]
ΔT_{LM}	Mean logarithmic temperature [°C]
U	Overall heat transfer coefficient [$Wm^{-2}K^{-1}$]
$\dot{V}$	Volumetric flow rate [m^3s^{-1}]
β	Chevron angle [°]
ϕ	Surface enhancement factor
ρ	Density [kgm^{-3}]
μ	Dynamic viscosity [Pa.s]

Subs.	Explanation
b	Bulk
c or cold	Cold
ch	Channel
f	Fluid
h or *hot*	Hot
in	Inlet
out	Outlet
w	Wall

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