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Experimental comparison of performances of three different plates for gasketed plate heat exchangers

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ABSTRACT

In this study, an experimental set-up for testing chevron type gasketed plate heat exchangers is utilized to investigate the thermal and hydraulic characteristics of three different plate geometries. The experiments are performed using various number of plates, several flow rate and inlet and outlet temperature values so that the Reynolds numbers (300–5000) and Prandtl numbers vary for all the plates that have 30° of chevron angle. Plate-specific correlations are derived for Nusselt number and friction factor by using the experimental results.

1. Introduction

Heat exchangers are widely used devices for mainly heating, cooling and heat recovery processes in industrial applications from micro channel cooling to nuclear power plants. In recent years, there is an increase in the usage of gasketed plate heat exchangers especially in chemical industries such as brewery and food processing due to their flexibility, compactness, ease of cleaning and assembly/disassembly [1–3]. In order to enhance heat transfer and maintain the thermal and hydraulic characteristics, the channel geometries, plate pack arrangements and the flow distributions have to be investigated and determined in detail.

Pinto and Gut [4] defined the configuration of a plate heat exchanger by six parameters; the number of channels, the number of passes on each side, the fluid distribution locations, feed positions and type of flow in channels to develop an optimization method for the best configuration(s) of gasketed plate heat exchangers. They show that approximately 5% of the pressure drop and channel velocity calculations and 1% of the thermal simulations are required for the solution for determining a successful and optimal configuration with a minimum number of exchanger evaluations. Islamoglu and Parmaksizoglu [5] studied the effect of channel height on the enhanced heat transfer characteristics in a corrugated heat exchanger channel and performed experiments with air as the working fluid to determine forced convection heat transfer coefficients and friction factor. Measurements were performed for channel height values of 5 and 10 mm with a Reynolds number range of $400 \leq Re \leq 1200$. They observed that with increasing channel height, there is a substantial increase in the fully developed Nusselt number and friction factor. Gut et al. [6] presented a parameter estimation method for plate heat exchangers that handled experimental data from multiple configurations. The test was carried out with a heat exchanger with flat plates and the parameter estimation results were compared to those obtained from the usual method of single pass arrangements. They observed that the obtained heat transfer correlations are associated with the configurations experimentally tested and the corresponding flow distribution patterns.

Rao et al. [7,8] studied the effect of flow maldistribution from port to channel on the thermal performance and pressure drop of single and multipass plate heat exchangers, experimentally. The results showed that maldistribution was more severe in Z-type compared to U-type plate heat exchangers and multipass arrangement was found to reduce the maldistribution effect significantly and predictions were made to limit the addition of plates beyond a certain value. Bobilli et al. [9] experimentally investigated the flow maldistribution in small and large plate package heat exchangers with a Reynolds number range of 1000–17,000, with a corrugation angle of 20–80°, water as the working

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fluid. They showed that the flow maldistribution increased by increasing the overall pressure drop in a plate heat exchanger. Muley and Manglik [10] studied the heat transfer and isothermal pressure drop data for single phase water flow in three different plate arrangements with chevron angles of 30°, 60° and one mixed 30°/60° of plate heat exchanger with single pass U-type counter-flow chevron plates, a Reynolds number range of 600 to 10^4 experimentally. Data for Nusselt number and friction factor was presented. They found that at constant pumping power, depending on the Reynolds number, chevron angle and the ratio of effective to projected surface area of plate corrugations, the heat transfer was enhanced by up to 2.8 times that of an equivalent flat plate channel.

For developing a new correlation method, Fernández-Seara et al. [11] practiced the Wilson plot method and reported characteristic experimental results for a smooth tube and a spirally corrugated tube. In another study of Fernández-Seara et al. [12] modifications on Wilson plot method to determine the convection coefficients in heat exchanger devices were investigated and they showed how Wilson plot method could deal with the determination of convection coefficients based on the measured experimental data and the subsequent construction of appropriate correlations.

Some correlations developed from previously reported studies for obtaining thermal and hydraulic characteristics of various plate heat exchangers with their application ranges are tabulated in Table 1.

2. Experimental methodology

2.1. Experimental setup

The experimental set-up which uses tap water as working fluid is designed [13–16] to permit various sizes of plate heat exchangers to be tested with different number of plates so that the experiments can be performed with wide range of capacities with flexible joints and high capacity tanks. The set-up is composed of two discrete cycles with hot and cold fluids as shown in Fig. 1. The detailed description of the experimental setup is explained by Akturk et al. [16].

The water goes to 3 kW pumps from the hot water tank, where the water is heated up by 38 kW resistance heatings, and from the cold water tank. Then the pumped water passes through the ±0.4% accurate electromagnetic flow meter [17] where the volumetric flow rate of the fluid is measured and goes to the plate heat exchanger. The temperature and pressure drops are read by J type (Fe–Cu, %45Ni) thermocouples, ±0.45% accurate Datalogger [18] and ±0.075% accurate pressure transmitter [19]. Discharge water from the heat exchanger is collected at a waste water tank during experiments. There is a closed cycle option for hot water which is used to stabilize the temperature distribution in the tank before starting experiments.

To observe different characteristics for plate packs, the experiments are carried out for various temperatures and flow rates for U-type flow arrangement shown in Fig. 2. After waiting the system to be stabilized, temperature values are recorded by digital data taker and pressure differences and volumetric flow rates are recorded every 20 s.

2.2. Geometrical properties of plates

The three different chevron plates that are experimentally tested have different geometrical properties as shown in Table 2. Fig. 3 shows the dimensional parameters that define the geometrical properties. Plate 1 and Plate 2 both have the same geometrical properties except for the plate length between ports. Plate 3 has larger dimensions.

3. Correlation development methodology

Calculations are performed with the experimental data by using the basic heat transfer equations from the literature [2].

$$T_{\mathrm{c,b}} = (T_{\mathrm{c,in}} + T_{\mathrm{c,out}})/2 \tag{1}$$

$$T_{\mathrm{h,b}} = (T_{\mathrm{h,in}} + T_{\mathrm{h,out}})/2 \tag{2}$$

$$T_{\mathrm{w}} = (T_{\mathrm{c,b}} + T_{\mathrm{h,b}})/2 \tag{3}$$

The thermophysical properties (ρ, μ, C_p, Pr, k) of the fluid can be found by bulk temperatures at the cold side, hot side and the wall [21]:

For density, ρ [kg/m^3]: $271 \leq T \leq 373$ K

$$\rho = \Big[999.83592 + 16.94517T - 7.98704\cdot 10^{-3}T^2 4.6170461 \cdot 10^{-5}T^{-3} + 1.0556302\cdot 10^{-7}T^4 2.8054253\cdot 10^{-10}T^5\Big] [1 + 0.01687985]^{-1} \tag{4}$$

For thermal conductivity, k [W/m K]: $273 \leq T \leq 400$ K

Table 1
Correlations for Nusselt number and friction factor from literature and their ranges of application.

Reference	Ø	β (°)	Correlation	
			Nusselt number	Friction factor
Focke et al. [22]	1.464	30	$Nu = 0.77Re^{0.54}Pr^{0.5}$ ($120 \leq Re \leq 1000$)	$f = 57.5Re^{-1} + 0.093$ ($260 \leq Re \leq 3000$)
			$Nu = 0.44Re^{0.64}Pr^{0.5}$ ($1000 \leq Re \leq 42{,}000$)	$f = 0.8975Re^{-0.263}$ ($3000 \leq Re \leq 50{,}000$)
Chisholm and Wanniarachchi [23]	1.17	$30 \leq \beta \leq 80$	$Nu = 0.768Re^{0.59}Pr^{0.4}$ ($1000 \leq Re \leq 40{,}000$)	$f = 0.973Re^{-0.25}$ ($1000 \leq Re \leq 40{,}000$)
	1.288		$Nu = 0.799Re^{0.59}Pr^{0.4}$ ($1000 \leq Re \leq 40{,}000$)	$f = 1.098Re^{-0.25}$ ($1000 \leq Re \leq 40{,}000$)
Bond [24]	1.17	30	$Nu = 0.329Re^{0.529}Pr^{0.33}(\mu/\mu_w)^{0.17}$ ($23 \leq Re \leq 468$)	$f = 3.01Re^{-0.457}$ ($47 \leq Re \leq 468$)
			$Nu = 0.113Re^{0.719}Pr^{0.33}(\mu/\mu_w)^{0.17}$ ($Re > 468$)	$f = 0.735Re^{-0.213}$ ($Re > 468$)
	1.288		$Nu = 0.345Re^{0.529}Pr^{0.33}(\mu/\mu_w)^{0.17}$ ($52 \leq Re \leq 515$)	$f = 2.886Re^{-0.457}$ ($52 \leq Re \leq 515$)
			$Nu = 0.116Re^{0.713}Pr^{0.33}(\mu/\mu_w)^{0.17}$ ($Re > 515$)	$f = 0.72Re^{-0.213}$ ($Re > 515$)
Maslov and Kovalenko [25]		60	$Nu = 0.78Re^{0.5}Pr^{1/3}$ ($50 \leq Re \leq 20{,}000$)	$f = 95.6Re^{-0.25}$ ($50 \leq Re \leq 20{,}000$)
Tovazhnyanski et al. [26]	1.16	30	$Nu = 0.074Re^{0.73}Pr^{0.33}(\mu/\mu_w)^{0.25}$ ($2000 \leq Re \leq 25{,}000$)	$f = 0.204Re^{-0.215}$ ($2000 \leq Re \leq 25{,}000$)
Talik et al. [27]	1.22	60	$Nu = 0.248Re^{0.7}Pr^{0.4}$ ($1450 \leq Re \leq 11{,}460$)	$f = 0.3323Re^{-0.042}$ ($1450 \leq Re \leq 11{,}460$)
Present study $300 \leq Re \leq 5000$	1.17	30	$Nu = 0.32867Re^{0.68}Pr^{0.1/3}(\mu/\mu_w)^{0.14}$	$f = 259.9Re^{-0.9227} + 1.246$
			$Nu = 0.3277Re^{0.675}Pr^{0.1/3}(\mu/\mu_w)^{0.14}$	$f = 1371Re^{-1.146} + 1.139$
	1.288		$Nu = 0.17422Re^{0.7}Pr^{0.1/3}(\mu/\mu_w)^{0.14}$	$f = 0.003743Re^{0.5981} + 0.9132$