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THERMAL-HYDRAULIC PERFORMANCE ANALYSIS COLD SIDE OF THE PLATE HEAT EXCHANGER USING WATER-WATER

ТЕПЛОГИДРАВЛИЧЕСКИЙ АНАЛИЗ ПРОИЗВОДИТЕЛЬНОСТИ ХОЛОДНОЙ СТОРОНЫ ПЛАСТИНЧАТОГО ТЕПЛООБМЕННИКА, ИСПОЛЬЗУЯ ТИПЫ СРЕД ТЕПЛОНОСИТЕЛЕЙ ВОДА-ВОДА

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Abstract. In this research, the thermal-hydraulic performance of plate heat exchanger (Ridan HHN no. 04) used in domestic water system is investigated experimentally. The hot water inlet keeps temperature and flow rate constant at 70 ℃ and 0.1 L/s, the cold–water inlet remained 11℃ with different velocity. The heat transfer coefficient can reach maximum when velocity of cold water is 0.2 m/s. Then the convective heat transfer coefficient increases with enhancement of Reynolds number and it is stable after more than 1200. Moreover, it is observed that Fanning friction factor decreases with an increase of the Reynolds number and it is showed by the empirical correlation. Therefore, it is possible to find that the increase of the Peclet number results in an increase of the Nusselt number as well when Peclet number small than 200. Finally, we get reference idea how to be cooling in the particular case using plate heat exchanger.

Аннотация. В этом исследовании экспериментально изучены теплогидравлические характеристики пластинчатого теплообменника (Ridan HHN №04), используемого в бытовой системе водоснабжения. На входе горячая вода поддерживает постоянную температуру и расход 70 °C и 0,1 л/с, а холодная вода на входе сохраняет 11 °C с разной скоростью. Коэффициент теплопередачи может достигать максимума, когда скорость холодной воды составляет 0,2 м/с. Тогда коэффициент конвективной теплопередачи возрастает с увеличением числа Рейнольдса и стабилен свыше 1200. Кроме того, наблюдается, что коэффициент трения Фэннинга уменьшается с увеличением числа Рейнольдса, и это показано эмпирической зависимостью. Таким образом, можно обнаружить, что, когда число Пекле меньше 200, его возрастание приводит к увеличению числа Нуссельта. В конечном итоге, мы получаем идею, как пользоваться охлаждением в конкретном случае, используя пластинчатый теплообменник.

Keywords: plate heat exchanger, thermal-hydraulic, convective heat transfer coefficient, Peclet number.

Ключевые слова: пластинчатый теплообменник, термогидравлические, коэффициент конвективной теплопередачи, число Пекле.

Introduction

Plate heat exchanger (PHE) were originally designed for hygienic applications in the food and pharmaceutical industries. However, in those years, PHE have extensively used in the power and process industries as well. In addition to the ease of sanitation and maintenance, the main advantages of PHE are the compactness, flexibility, and high thermal effectiveness. Basically, the PHE consists of a compressed pack of thin metal plates, where each plate has a sealing gasket around the perimeter and can have up to four corner orifices (ports). The gaskets are designed so that hot and cold fluids flow through alternate flow channels without mixing. Heat is exchanged between neighbor channels through the metal plate, which have a corrugated surface for improving turbulence and mixing. The alignment of the plate orifices forms continuous ducts (manifolds) inside the plate pack that distribute the fluids into the narrow channels between plates [1].

The fluid entering the plate pack at its top or bottom is split into N parallel flow channels and is collected the flow in the duct on the opposite side. This group of channels in which the flow is in the same direction is denominated “pass”. The flow arrangement of a PHE is often presented as：

$$P_{hot} \cdot N_{hot} / P_{cold} \cdot N_{cold}\,, \tag{1}$$

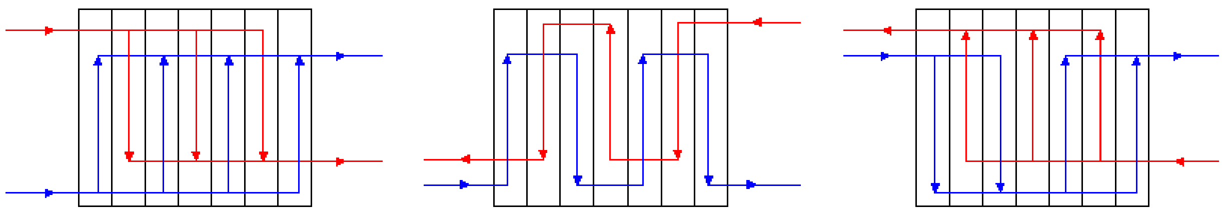
where P is the number of passes;

N is the number of channels.

The number of channels in the hot or cold sides of the PHE is given by P*N, the total number of the channels in the PHE is thus defined as:

$$N_c = P_{hot} \cdot N_{hot} + P_{cold} \cdot N_{cold}\,, \tag{2}$$

And the number of thermal plates is given by N_c-1. Figure 1 shows a few examples of possible configuration for a PHE with seven channels. The configuration consists of the flow arrangement and the location of the inlet and outlet connections of hot and cold fluids.



(a) (b) (c)

Figure 1. Some possible configurations for a PHE with seven flow channels (six thermal plates):
a — 1*4/1*3 (looped Z); b — 4*1/3*1 (series); c — 2*2/1*3 (complex).

Overview of information sources on the problem scientific research

Bajura [2] analytical investigated the performance of the flow distribution systems for both intake and exhaust manifolds. A mathematical model describing the flow behavior at a discrete branch point was formulated in terms of a momentum balance along the manifold. Fang et al. [3] investigated the flow distribution in manifolds of the heat exchanger by proposing a discrete model matching the real physical phenomenon to predict the pressure distribution in headers. Acrivos et al. [4] demonstrated how the wall friction and varying fluid momentum lead to pressure variations in a manifold divided fluid stream. Martin [5] considered the effects of the longest flow path to derive a relatively simple but physically reasonable equation for the friction factor as a function of chevron angle and the Reynolds number and showed the comparison between crossing and longitudinal flow. Bassiouny and Martin [6] analytically studied the plate heat exchanger to calculate the axial velocity, total pressure drop, pressure distributions in both the inlet and outlet conduits as well as the flow distribution in the channels between the plates. Akturk et al. [7] conducted the experiment of thermal and hydraulic analysis with a different number of plates in gasketed plate heat exchanger (GPHE). They developed the correlations for the convective Nusselt number and friction factor of plate heat exchanger. Yildirim and Söylemez [8] presented the simple algebraic formula for estimating the optimum area of heat exchangers of plate type for the thermoeconomic optimization. Gulenoglu et al. [9] experimentally studied the thermal and hydraulic performance for three different plate geometries and developed new correlations for the Nusselt number and friction factor. Khan et al. [10] experimentally studied the heat transfer for multiple plate configurations and developed a correlation for the Nusselt number. Focke et al. [11] experimentally investigated the effect of corrugation inclination angle on the thermal hydraulic performance of the plate heat exchanger for the symmetric chevron angle plate configurations. The pressure drop and heat transfer were reported to increase with an increase in chevron angle. Faizal and Ahmed [12] studied the pressure drop and heat transfer in PHE with different spacing between the corrugated plates. Han et al. [13] investigated both, numerically and experimentally, the temperature, pressure, and velocity fields in chevron corrugated-plate heat exchanger and found that the highest temperature occurred around the upper port while the lowest temperature in the cold fluid inflow around the lower port. The fluid pressure gradually reduced along the flow direction in pressure field. Rush et al. [14] performed an experimental investigation of the local heat transfer and flow behavior for laminar and transitional flows in sinusoidal wavy passages. Aliabadi et al. [15] experimentally studied the thermal-hydraulic performance of copper-water nanofluid flow in different plate fin channels. They developed an empirical correlation for the Nusselt number and Fanning friction factor of the base fluid and nanofluids for plate-fin channels. Aliabadi et al. [16] studied the thermal-hydraulic performances of the plate fin heat exchanger for different geometrical parameters (corrugation amplitude/length and winglet height/pitch), using the ethylene glycol mixtures as the working fluid. Tereda et al. [17] investigated the port-to-channel flow maldistribution for a varying port diameter at a fixed number of plates and corrugation angle. Rao et al. [18, 19] experimentally investigated the port flow maldistribution in plate heat exchangers for small and large plate packages at high and low corrugation angles and found that the flow maldistribution increases with overall pressure drop. Mueller and Chiou [20] presented an adequate review of the work devoted problem on the flow maldistribution.

In above-cited literature, the majority of researchers presented characteristics in hot side by uniform flow in the channels, thus indicating an ideal case of a few cold side in their studies. Few experimental studies have been presented to study thermal-hydraulic performance in cold side of plate heat exchanger. Thus, the objective of the present work is to study pressure drop in a wide range of low Reynolds number and compare their performances in the plate heat exchanger. The experiments have been carried out under the steady conditions for PHE.

Experimental setup procedure

Details of the experimental set-up of PHE are shown in the schematic layout (Figure 2). Red and blue arrow marks, trace the hot and cold water respectively. Scheme works as follow: circulating water is stored in electric boiler (1), condenser (17), and cooling tower (19). Reciprocating pump (2) propels the heated water which comes from the electric boiler and heats to the desired temperature with 0.1 L/s to the plate heat exchanger where it is converted with cold water. Maximum operating pressure and temperature of the electric boiler are 4 Ba and 80 ℃. Cold water is used in PHE coming from the condenser by the pump (16). Investigation of the flow distribution in PHE is carried out for 13 plates (12 channels). The turbine type rotary mechanical flow meters (4 & 14) and four PT-100 thermometers (6, 7, 10 & 12) are provided in hot and cold water flow path to measure the volume flow rates and temperatures, respectively. Thermometers are placed near to the ports of PHE in the stainless steel pipe section at the inlet and outlet of water. Four pressure transmitters (5, 8, 9 & 13) are placed near the thermometers to measure the fluid pressure. Flow rates of the working fluid being pumped by pumps (2 & 16), is controlled by the speed of the pump. Pressure transmitters and thermometers are connected to a Programming Logic Controller (PLC), which is further connected to a human interface unit (HMI), measure and control the flow rates of hot and cold fluid streams. They also measure the pressure and temperature at the inlet and outlet of both the fluid streams.

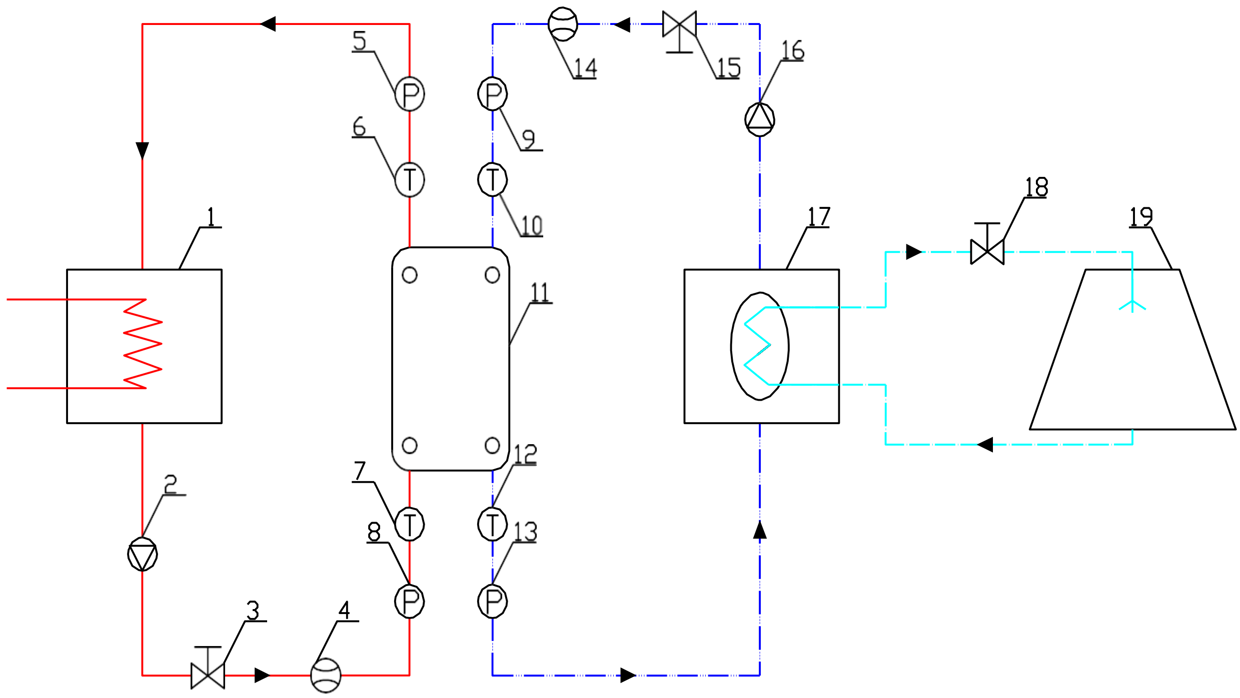


Figure 2. Schematic layout of experimental set-up: 1 — electric boiler; 2 — first pump; 3 — first vale; 4 — first flow meter; 5 — first pressure transmitter; 6 — first temperature transmitter; 7 — second temperature transmitter; 8 — second pressure transmitter; 9 — third pressure transmitter; 10 — third temperature transmitter; 11 — PHEs ; 12 — fourth temperature transmitter; 13 — fourth pressure transmitter; 14 — second flow meter; 15 — second valve; 16 — second pump; 17 — condenser; 18 — third valve; 19 — cooling tower.

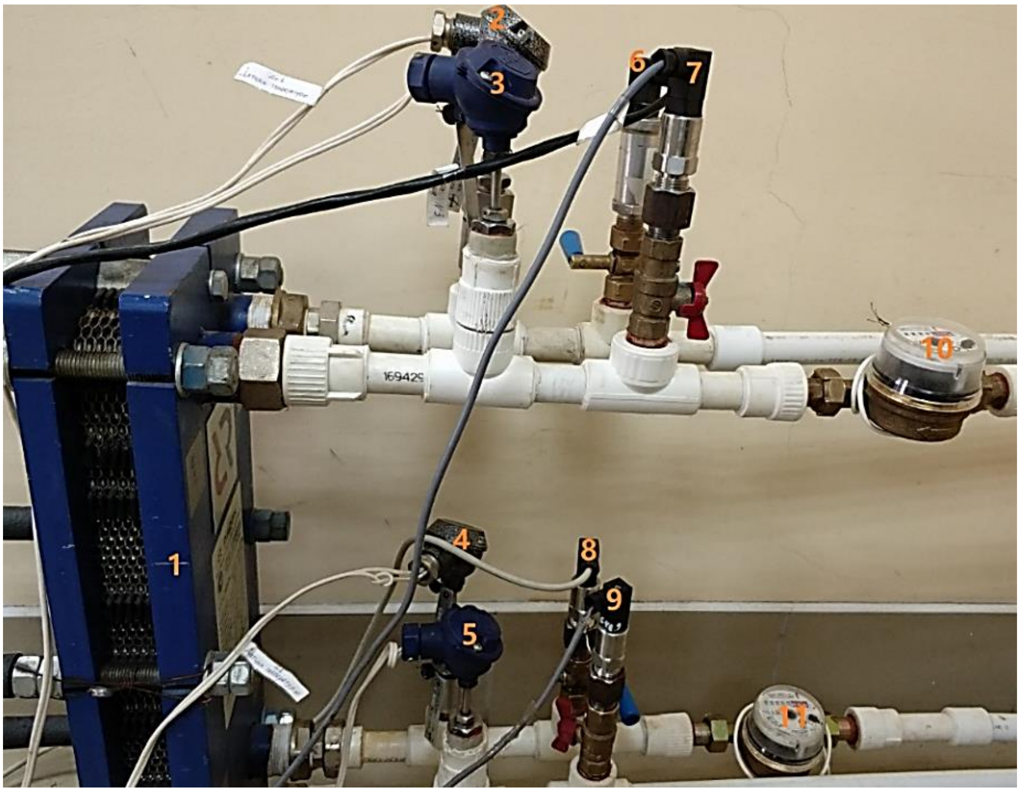
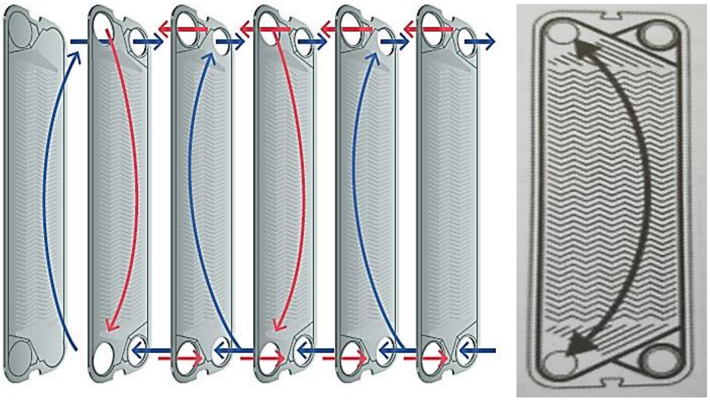


Figure 3. Experimental test section: 1 — plate heat exchanger; 2, 3, 4, 5 — M5100 pressure transmitters; 6, 7, 8, 9 — thermometers; 10, 11 — mechanical flow meters.

A pictorial view of the experimental test section as shown in Figure 3. It consists of PHE, Turbine type mechanical flow meters (10 & 11) and M5100 pressure transmitters (2, 3, 4 & 5) are provided in hot and cold fluid stream as already discussed. Thermometers (6, 7, 8 & 9) are provided near the inlet and outlet port of cold and hot fluid streams. The working fluid is pumped by pump.



(a) (b)

Figure 4. Flow arrangement for plate heat exchanger and details for plate heat exchanger: a — side view of plates with the flows arrangement of cold and hot water of PHE; b — front view of the plate.

Figure 4 (a) presents the flow arrangement of PHE. Its geometrical and flow path details are given with the figure. And figure 4 (b) shows the details of one plate. Geometrical parameters of the same are listed in Table 1.

In isothermal condition, cold water is supplied to all the channels of PHE, thus there is no heat transfer involved in the PHE. This analysis is carried out to study hydraulic performance at a constant temperature. The experiments have been carried out at 12 °C for the isothermal condition. In non-isothermal condition, cold and hot water are supplied in alternative channels of the PHE. Cold water is supplied at 12 °C and hot water is maintained in the range of 70-71 °C.

Table 1.

GEOMETRICAL CHARACTERISTICS OF CHEVRON PLATE

Geometrical characteristics of chevron plate		
Number	*Particulars*	*Dimensions*
1	Port diameter, d_p	0.032m
2	Port to port length, L_{ch}	0.381m
3	Port to port width, Lw	0.07m
4	Corrugation picth p	0.011m
5	Amplitude of corrugation, b	0.0028m
6	Thichness of plate, t	0.0005m
7	Chevron angle, β	60°
8	Heat transfer area (plate)	0.04m²
9	Plate material	AISI 316 (Stainless steel)
10	Gasket material	Nitrile rubber

The details of measuring instruments used in the experiment are shown in Table 2 below.

Table 2.

THE MODEL, RANGE, AND ACCURACY OF THE MEASURING INSTRUMENTS

Number	*Instrument*	*Measure*	*Model*	*Range*
1	Mechanical flow meters	Flow rate	Pt 500	1000-30000 LPH
2	Thermocouple	Temperature	PT-100, R.T.D	0 to 180 ℃
3	Pressure transmitters	Inlet and outlet pressure of PHEs	M5100	0-2.5 MPa

Data reduction

The experimental data have been obtained under steady state conditions and the operating flow rates were taken to have a range of Reynolds number from 105 to 570 for PHE. The mean pressure drop data obtained using pressure transmitters was used in the following equation to evaluate Fanning friction factor (f):

$$f = \frac{\Delta P D_{eq}}{2 L_{ch} \rho V^2}, \tag{3}$$

where ΔP is pressure drop, Pa;

D_{eq} is hydraulic diameter, m;

L_{ch} is port to port length, m;

ρ is the density of the water, kg/m^3 ;

V is the velocity of the water, m/s.

The behavior of hydraulic resistance with Reynolds number is shown in Figure 8.

The correlation for the Fanning friction factor in terms of Reynolds number is developed in fitting formula (12). The Reynolds number for plate heat exchanger is defined on the basis of hydraulic diameter D_{eq} , as:

$$\mathrm{Re} = \frac{V D_{eq}}{\upsilon} \tag{4}$$

where υ is Kinematic viscosity, m^2/s.

The characteristic length of the channel was the equivalent diameter [5, 21], which is defined as follows:

$$D_{eq} = 2b \qquad (\because b << Lw), \tag{5}$$

The inlet port velocity is evaluated as:

$$V = \frac{V_{h.i}}{A_p}, \tag{6}$$

where $V_{h.i}$ is inlet volume flow rate, m^3/s;

A_p is the inlet sectional area, m^2.

With respect to the average heat transfer coefficient (h), it was calculated by:

$$h = \frac{Q}{S\Delta T_m}, \tag{7}$$

where Q is the heat transfer rate, W;

S is the heating wall surface area, m^2.

ΔT is the logarithmic mean temperature difference which is given by:

$$\Delta Tm = \frac{(T_{h.in} - T_{c.out}) - (T_{h.out} - T_{c.in})}{Ln\left[\dfrac{T_{h.in} - T_{c.out}}{T_{h.out} - T_{c.in}}\right]} \tag{8}$$

where $T_{c.in}$ is cold inlet flow temperature, K;

$T_{c.out}$ is cold outlet flow temperature, K;

$T_{h.in}$ is hot inlet flow temperature, K;

$T_{h.out}$ is hot outlet flow temperature, K.

In the heat transfer at a surface within a fluid, the Nusselt number (Nu) is the ratio of the convective to the conductive heat transfer across normal to the boundary and it is given by:

$$Nu = \frac{hD_{eq}}{\lambda}, \tag{9}$$

where h is the convective heat transfer coefficient, w/(m^2*k);

λ is the thermal conductivity, w/(m*k).

The Prandtl number ($\Pr$) is the ratio of the momentum and the thermal diffusivities, and it describes the static properties of the fluid substance.

The generalized Prandtl number is defined by:

$$\Pr = \frac{\mu C_p}{\lambda}, \tag{10}$$

where C_p is the specific heat, J/(kg*k);

μ is Newtonian viscosity, Pa*s.

Also, the Peclet number is defined as:

$$Pe = \mathrm{Re}\,\Pr, \tag{11}$$

Results and discussion

Figure 5 shows the dependence of the heat transfer rate with respect to the velocity of the fluid. It is noted that heat transfer rate is added rapidly with increase of velocity which no more than 0.05 m/s; and it almost keep constant with velocity increasing after 0.05 m/s. It indicated that plate heat exchanger achieved full heat exchange and added pump power had a little influence on heat transfer rate when velocity greater than 0.05 m/s.

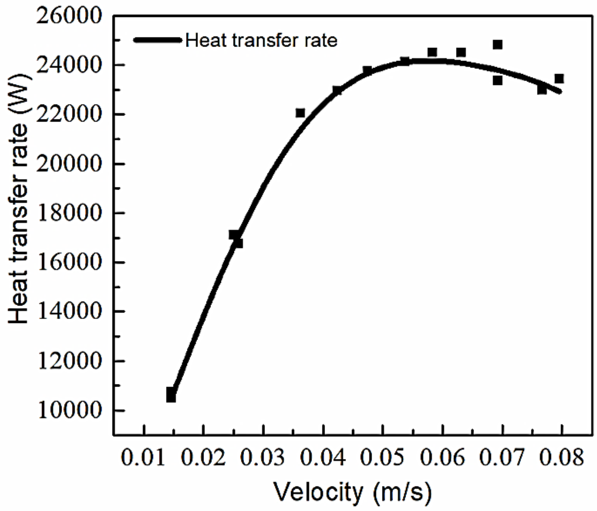


Figure 5. Behavior of the heat transfer rate as function on the velocity in PHE

The comparison of heat transfer rate between cold side and hot side with velocity number is shown in Figure 6. The cold side's results are found to be in good agreement with hot side's results which the relative error is no more than 15%. And we can use mean heat transfer rate or cold side's calculate that can ensure the correctness calculation.

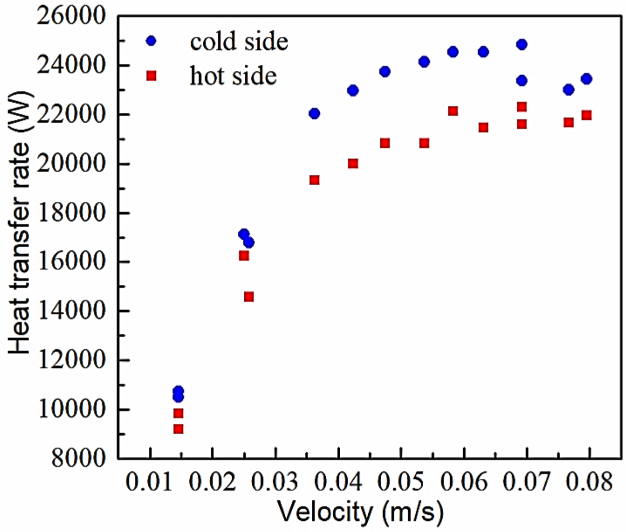


Figure 6. The correlation of heat transfer rate between hot side and cold side

Figure 7 shows the convective heat transfer coefficient versus Reynolds number at different velocity. Based on these results the convective heat transfer coefficient increases with enhancement of Reynolds number and after a certain Reynolds number the convective heat transfer coefficient is stable. As shown in Figure 6, data tendency changes quickly at the range of Reynolds number from 150 to 200 that it maybe depends on increasing of turbulence intensity at higher Reynolds number and decrement in fluid thermal boundary layer thickness due to the reduction of fluid viscosity near the wall. Also heat exchange comes into full development stage and thermal boundary layer remains unchanged after Reynolds number more than 300.

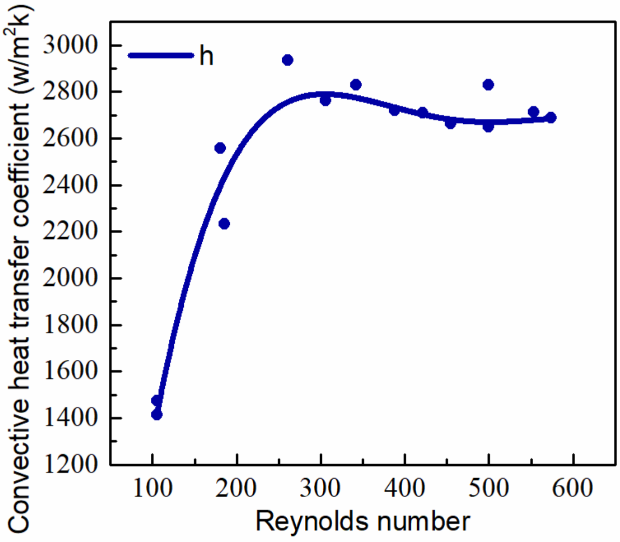


Figure 7. Convective heat transfer coefficient changed with Reynolds number

Figure 8 shows the behavior of the Fanning friction factor with Reynolds number. For the Reynolds number range investigated in this work, the following empirical correlation of the Fanning friction factor as a function of the generalized Reynolds number is proposed.

$$f = 5.60*10^{6}\,\mathrm{Re}^{-1.853}, \tag{12}$$

It is observed that Fanning friction factor decreases with an increase in the Reynolds number. This is due to a tremendous increase in turbulence at higher Reynolds number within plates. At higher Reynolds number, fluid molecules get lesser time to interact with plate surface, and hence lower friction between the plates and fluid particles. The fluctuations in the Fanning friction factor at higher Reynolds number are due to the experimental error like the vibration in the system when the motor speed goes above 2500 RPM.

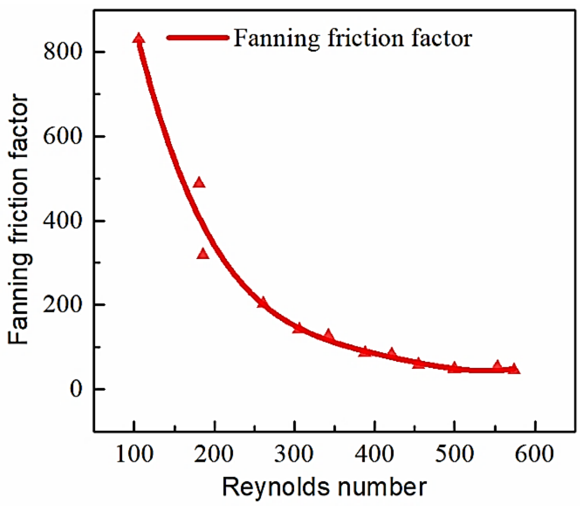


Figure 8. Variations of Fanning friction factor with Reynolds number

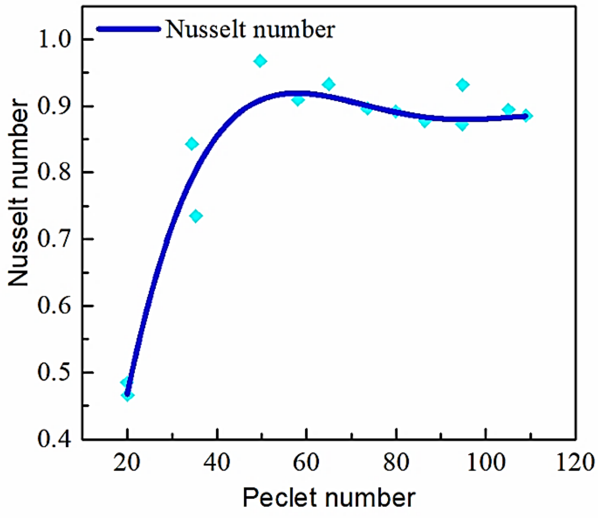


Figure 9. Behavior of the Nusselt number as a function on the Peclet number in PHE

The variation of the Nusselt number as a function of the Peclet number (Pe = RePr) for all the case studies with plate heat exchanger is presented in Figure 9, in which it is possible to observe that the increase of the Peclet number results in an increase of the Nusselt number as well when Peclet number small than 200. The increase of the Nusselt number indicates an enhancement in the heat transfer coefficient due to the convection increases. The Nusselt number almost keep constant after Peclet number more than 200, that is mean thermal resistance over convective resistance is hardly to change with Peclet number.

Conclusion

Thermal hydraulic performance analysis cold side of PHE are experimentally investigated under steady state conditions. The hot water inlet keep temperature and flow rate constant at 70℃ and 0.1 L/s, the cold water inlet remained 11℃ flows different velocity. The heat transfer coefficient mainly varied from 1480 to 2900 W m^{-2} K^{-1}, and it can reach maximum when velocity of cold water is 0.2m/s. Then the convective heat transfer coefficient increases with enhancement of Reynolds number and after 1200 the convective heat transfer coefficient is stable. However, pressure drop in the PHE decreases with increase in Reynolds number. Moreover, it is observed that Fanning friction factor decreases with an increase of the Reynolds number and it is showed by the empirical correlation. Therefore, it is possible to find that the increase of the Peclet number results in an increase of the Nusselt number as well when Peclet number small than 200. The increase of the Nusselt number indicates an enhancement in the heat transfer coefficient due to the convection increases. The Nusselt number almost keep constant after Peclet number more than 200, that is mean thermal resistance over convective resistance is hardly to change with Peclet number. Through this research work, new correlation in term of Fanning friction factor and Reynolds number has been developed, and it provides reference idea how to cooling in the particular case using plate heat exchanger.

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