ТЕХНИЧЕСКИЕ НАУКИ / TECHNICAL SCIENCE

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EXPERIMENTAL INVESTIGATION ON PERFORMANCES OF PLATE HEAT EXCHANGER'S COLD SIDE FOR LUBRICATION/WATER - WATER HEAT TRANSFER

*©**Wang W.**, ORCID: 0000-0001-8493-4828, Ogarev Mordovia State University; Jiangsu University of Science and Technology, Saransk, Russia, Willie_CN520@163.com*
*©**Makeev A.**, Ph.D., ORCID: 0000-0001-5356-2144, Ogarev Mordovia State University, Saransk, Russia, tggi@rambler.ru*
*©**Povorov S.**, Ogarev Mordovia State University; Jiangsu University of Science and Technology, Saransk, Russia, acrosrm@gmail.com*

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ХАРАКТЕРИСТИК ТЕПЛОПЕРЕДАЧИ ХОЛОДНОЙ СТОРОНЫ ПЛАСТИНЧАТОГО ТЕПЛООБМЕННИКА ДЛЯ ТИПОВ СРЕД ТЕПЛОНОСИТЕЛЕЙ МАСЛО/ВОДА - ВОДА

*©**Ван В.**, ORCID: 0000-0001-8493-4828, Национальный исследовательский Мордовский государственный университет им. Н. П. Огарева; Цзянсуский университет науки и технологии, г. Саранск, Россия, Willie_CN520@163.com*
*©**Макеев А. Н.**, канд. техн. наук, ORCID: 0000-0001-5356-2144, Национальный исследовательский Мордовский государственный университет им. Н. П. Огарева, г. Саранск, Россия, tggi@rambler.ru*
*©**Поворов С. В.**, Национальный исследовательский Мордовский государственный университет им. Н. П. Огарева; Цзянсуский университет науки и технологии, г. Саранск, Россия, acrosrm@gmail.com*

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 lubrication/water inlet keep temperature and flow rate constant at 72 °C and 0.8–0.1 L/s, the cold–water inlet remained 12 °C with different velocity. Then the convective heat transfer coefficient increases with enhancement of Reynolds number. Moreover, it is observed that Fanning friction factor decreases with an increase of the Reynolds number and it is showed by the figure. 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 in water — water smalls than 2500. Finally, we get heat exchanging performances of cold side in lubrication/water — water using plate heat exchanger.

Аннотация. Экспериментально изучены теплогидравлические характеристики пластинчатого теплообменника (Ridan HHN №04), используемого в бытовой системе водоснабжения.

На входе горячее масло/вода поддерживает постоянную температуру и расход 72 °C и 0,8–0,1 л/с, а холодная вода на входе сохраняет 12 °C с разной скоростью. Тогда коэффициент конвективной теплопередачи увеличивается с увеличением числа Рейнольдса. Кроме того, наблюдается, что коэффициент трения Фэннинга уменьшается с увеличением числа

Рейнольдса, и это показано на рисунке. Таким образом, можно обнаружить, что, когда число Пекле меньше 2500, его возрастание приводит к увеличению числа Нуссельта.

В конечном итоге, получаем характеристики теплообмена холодной стороны пластинчатого теплообменника для типов сред теплоносителей масло/вода — вода.

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

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

Introduction

The plate heat exchangers are widely used in warming, heating, cooling applications, food, and cosmetic and chemistry industry. The plate type heat exchangers are initially developed for the pasteurized liquid food domain which mostly requires hygienic application. But, these heat exchangers have a large application area in chemistry and food sector because of being compact and having the quality to be easily cleaned [1–5].

Some examples to active method include the use of mechanical auxiliary elements, turning of surface, mixing of fluid with mechanical parts, constituting of electron-static areas in flow area, vibration of system, etc., Some examples to passive method include covering of surface, changing of surface, forming of the same projection parts of the rough surface, locating of the tabulators in flow area, etc. [6–7]. Mishra et al. [8] incorporated a genetic–algorithm–based optimization technique for a cross–flow plate and fin heat exchanger to minimize the total entropy generated within a prescribed heat duty. Cheng [9] proposed the concept of entropy resistance based on the entropy generation analysis as an alternative method for heat exchanger analysis. Fakheri [10] used the second law of thermodynamics to determine the exchanger thermal efficiency through the ratio of the actual heat transfer rate to the optimum heat transfer rate. By the way, Zheng et al. [11] further studied flooded boiling of pure ammonia and ammonia/lubricant mixture on a plain horizontal tube subjected to a vapor quality at the inlet of the evaporator. Khan et al. [12] used Mixed chevron angle plate configuration to do experiment with miscible oil–ammonia. Zhang et al. [13] investigated the performance of oil cooler with helical baffles (OCHB) and compared with that of the segmental one (OCSB) with practical size, and also found that the shell side heat transfer coefficients and the shell–side pressure drop of the OCHB are respectively lower than and far lower than those of the OCSB.

In this study, the thermal–hydraulic characters of the plate heat exchangers on heat transfer, Nu, f and Pe numbers base, is experimentally examined.

Experimental setup procedure

Details of the experimental set-up of PHE are shown in the schematic layout (Figure 1). Red and blue arrow marks with trace the hot lubrication/water and cold water respectively. Circulating lubrication/water is stored in electric boiler (1), condenser (17), and cooling tower (19). Reciprocating pump (2) propels the heated lubrication/water which comes from the electric boiler and heats to the desired temperature with 0.08-0.1 L/s to the plate heat exchanger where it is converted with cold water, Reciprocating pump (16) propels the cooled water which comes from the condenser and cools to the desired temperature with different volume velocity to the plate heat exchanger where it is converted with hot lubrication/water. Maximum operating pressure and temperature of the electric boiler are 4 Ba and 80 °C.

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 lubrication/water 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 pump's speed. Mechanical flow meters (turbine type flow meter), 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.

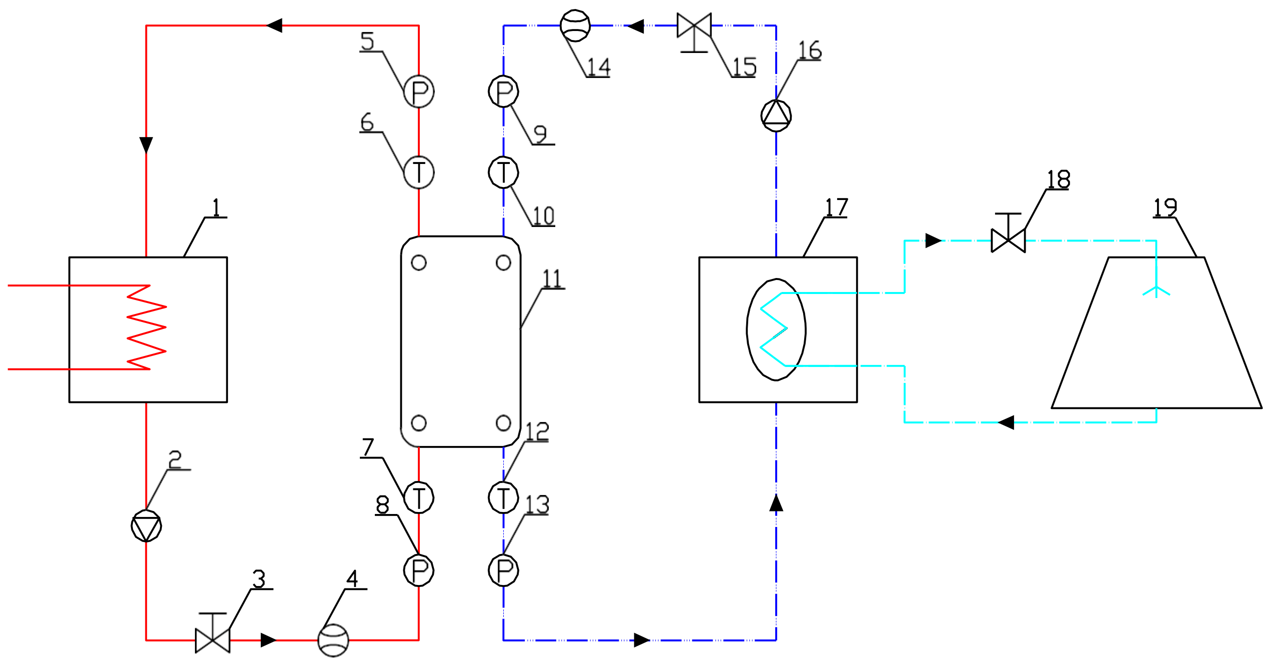

Figure 1. Schematic layout of experimental set-up: 1 — electric boiler; 2 — first pump; 3 — first vale; 4 — first flow meter; 5 — first pressure transmitter; 6 — first temperature transmitter; 7 — second temperature transmitter; 8 — second pressure transmitter; 9 — third pressure transmitter; 10 — third 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.

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 220 to 2400 for PHE. The mean pressure drop data obtained using pressure transmitters was used in the following equation to evaluate Fanning friction factor (f):

$$f = \frac{\Delta P_{ch} D_{eq}}{2 L_{ch} \rho V^2} \quad (1)$$

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 measured pressure drop of PHE includes the frictional pressure drop in the channels and the pressure drop in the ports.

$$\Delta P = \Delta P_{ch} + \Delta P_p \tag{2}$$

where ΔP_p is pressure drop in port.

The channel pressure drop is dined by using the Darcy friction factor model as below:

$$\Delta P_{ch} = 4f\left(\frac{L_p}{D_{eq}}\rho\frac{V^2}{2}\right) \tag{3}$$

where f is the channel friction factor that is determined by correlation obtained from Focke et al. [14]; L_p is the port to port length, m.

To calculate the pressure, drop in the ports Shah and Focke [15] correlation is used as follows:

$$\Delta P_p = 1.5\frac{\rho u^2}{2} \tag{4}$$

where the u is velocity in port, m/s.

The behavior of hydraulic resistance with Reynolds number is shown in

Figure 3. The Reynolds number for plate heat exchanger is defined on the basis of hydraulic diameter D_{eq}, as:

$$\mathrm{Re} = \frac{VD_{eq}}{\upsilon} \tag{5}$$

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

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

$$h_i = \frac{Q_i}{S_i \Delta T_m} \tag{6}$$

where Q_i is the heat transfer rate (use $Q = \dfrac{Q_h + Q_c}{2}$ to calculate), W;

S_i is the heating wall surface area, m^2;

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

$$\Delta T_m = \frac{(T_{h.in} - T_{c.out}) - (T_{h.out} - T_{c.in})}{Ln\left[\frac{T_{h.in} - T_{c.out}}{T_{h.out} - T_{c.in}}\right]} \quad (7)$$

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:

$$Nu = \frac{hD_{eq}}{\lambda} \quad (8)$$

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

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

Also, the Peclet number is defined as:

$$Pe = \mathrm{Re}\,\mathrm{Pr} \quad (9)$$

Results and discussion

Figure 2 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 in water–water is stable, but the it continued grows up in lubrication–water. The data in water–water bigger than lubrication–water because of different heat capacity. As shown in Figure 2, data tendency changes quickly at the range of Reynolds number from 300 to 500 that it maybe depends on increasing of turbulence intensity at higher Reynolds number and decrements in fluid thermal boundary layer thickness due to the reduction of fluid viscosity near the wall. Also, for water–water, heat exchange comes into full development stage and thermal boundary layer remains unchanged after Reynolds number more than 500.

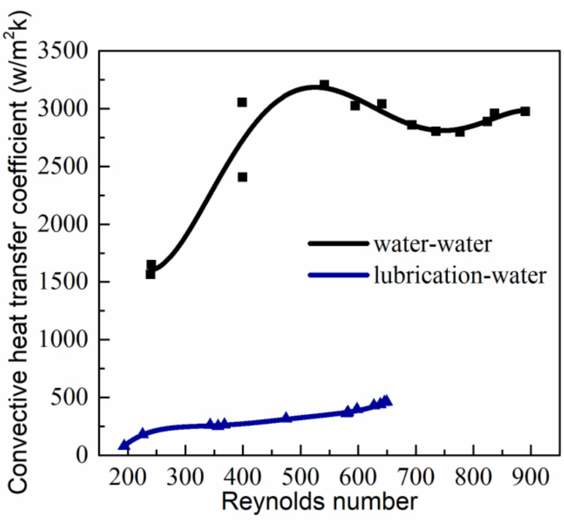

Figure 2. Convective heat transfer coefficient changed with Reynolds number.

Figure 3 shows the behavior of the Fanning friction factor with Reynolds number. 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. We can find that lubrication–water has low heat and velocity loss than water–water.

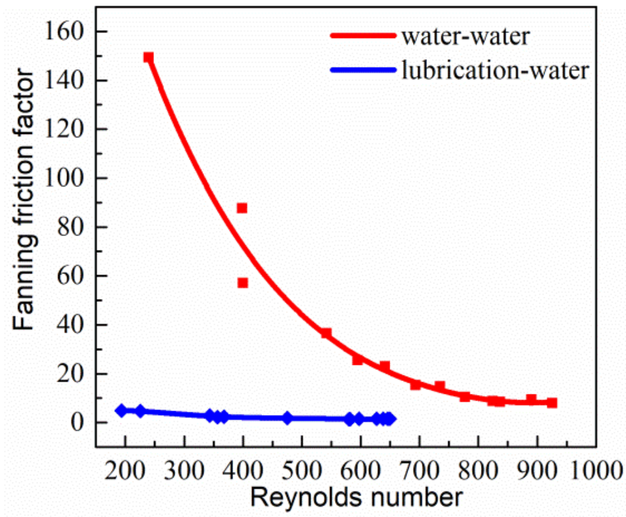

Figure 3. Variations of Fanning friction factor with Reynolds number.

The variation of the Nusselt number as a function of the Peclet number for all the case studies with plate heat exchanger is presented in Figure 4, 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 2500 in water–water, however, Nusselt number directly adds with Peclet number in lubrication–water. 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 2500 in certain situation, that is mean thermal resistance over convective resistance is hardly to change with Peclet number.

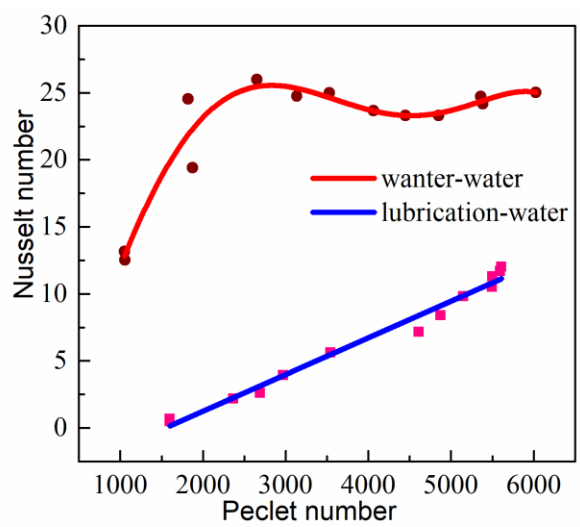

Figure 4. Behavior of the Nusselt number as a function on the Peclet number in PHE.

Conclusion

Thermal–hydraulic performance analysis cold side of PHE are experimentally investigated under steady–state conditions. The hot lubrication/water inlet keep temperature and flow rate constant at 72 °C and 0.1 L/s, the cold–water inlet remained 12 °C flows different velocity. Then the convective heat transfer coefficient increases with enhancement of Reynolds number and the result in water–water higher than lubrication–water. Moreover, it is observed that Fanning friction factor decreases with an increase of the Reynolds number and it is showed by the figure. Therefore, it is possible to find that the increase of the Peclet number results in an increase of Nusselt number as well when Peclet number small than 2500 in water–water. The increase of the Nusselt number indicates an enhancement in the heat transfer coefficient due to the convection increases. The Nusselt number will keep constant after the certain Peclet number, that is mean thermal resistance over convective resistance is hardly to change with Peclet number. Through this research work, it provides thermal exchanging feature in water–water and lubrication–water using plate heat exchanger.

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