

PLATE HEAT EXCHANGER WITH POROUS STRUCTURE FOR POTENTIAL USE IN ORC SYSTEM

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Abstract: *The experimental analysis of passive heat transfer intensification in the case of plate heat exchanger has been carried out. The passive intensification was obtained by a modification of the heat transfer surface, which was covered by a metallic porous microlayer. The experiment was accomplished in two stages. In the first stage the commercial stainless steel gasketed plate heat exchanger was investigated, while in the second one – the identical heat exchanger but with the modified heat transfer surface. The direct comparison of thermal and flow characteristics between both devices was possible due to the assurance of equivalent conditions during the experiment. Equivalent conditions mean the same volumetric flow rates and the same media temperatures at the inlet of heat exchangers in the corresponding measurement series. Experimental data were collected for the single-phase convective heat transfer in the water-water and water-ethanol configuration. The heat transfer coefficients were determined using the Wilson method.*

Keywords: POROUS STRUCTURE, HEAT TRANSFER INTENSIFICATION, PHE, ORC TECHNOLOGY

1. Introduction

Efficient heat production and distribution is very important from the economical and natural resources depletion points of view. Therefore an extensive research and development efforts have been undertaken in the area of heat transfer intensification over the past couple of decades. They refer to the single-phase convection and also to the boiling/condensation conditions. Nowadays we can observe a tendency to miniaturization in every field of life, but especially in technical applications. At the same time, in the area of energy technology very important are the high heat fluxes transfer problems. This is the reason why these new challenges require high efficiency of system components, especially highly efficient and small capacity heat exchangers. Plate heat exchangers have been widely used in power engineering, chemical processes and many other industrial applications due to their good effectiveness and compactness. Nevertheless there are still investigations going toward even more efficient and smaller size ones. They are going to be obtained by the heat transfer intensification and this new kind of plate heat exchangers could be prospectively applied for example in the heat recovery systems. Another example of perspective application of plate heat exchangers is the tendency to increase the importance of the so called dispersed generation, based on the local energy sources and the working systems utilizing both the fossil fuels and the renewable energy resources. Generation of electricity on industrial scale together with production of heat can be obtained for example through employment the ORC systems. It is mentioned in the EU directive 2012/27/EU for cogenerative production of heat and electricity. The authors are involved in a large scale national project with the objective of development of a commercially available ORC CHP unit for industrial applications.

General overview of heat transfer (in the flow passages) augmentation by passive methods can be found in [1], while Stone [2] concentrated on the heat transfer intensification in compact heat exchangers. Research connected with corrugated plate heat exchangers are going in many directions. It may be concentrated on the heat transfer coefficient and formulation of heat transfer correlation [3], on the pressure drop and friction factor correlation [4] or both of them [5].

Recently a large number of investigations on plate heat exchangers were reported in the professional literature. Unfortunately, rather limited data for units with high performance microsized, enhancement structures were available. Among them could be found works by Matsushima and Uchida [6]. A novel nano- and microporous structures were shown by Furberg et al. [7]. Müller-Steinhagen [8] described and analyzed a vacuum plasma sprayed 250 [μm] thick layer of spherically shaped Inconel 625 particles onto a plate and frame heat exchanger surface. The

knowledge and experiences connected with the passive heat transfer enhancement in the case of plate heat exchangers were also presented by Wajs and Mikielwicz [9,10].

In this paper the experimental analysis of passive heat transfer intensification in the case of model plate heat exchanger has been presented. The passive intensification was obtained by a modification of heat transfer surface, which was this time covered by a metallic porous microlayer. As it was mentioned in the abstract, the experiment was done in two stages, for two heat exchangers, that is the commercial stainless steel gasketed one and the identical heat exchanger but with the modified heat transfer surface. Experimental data were collected for the single-phase convective heat transfer in the water-water and water-ethanol system. The heat transfer coefficients were determined using the Wilson method.

2. Plate Heat Exchanger

The model of twisted plate heat exchanger offered at the domestic/world market by Sondex was the subject of presented investigations. In this kind of heat exchanger the heat is transferred in one pass. The model was made of 316 stainless steel according to AISI standard and consisted of three plates, whose thickness was 0.5 [mm]. The surface roughness of working plate was equal to 0.46 [μm] (parameter R_a) and 3.36 [μm] (parameter R_z), respectively. The total length of the heat exchanger was 450 [mm], while the overall heat transfer area was equal to 0.039 [m²]. The distance between the plates was kept constant and the EPDM seal was fixed in the "hang on" system. Permissible working pressure was equal to 1.6 [MPa]. The schematic view of heat exchanger plate is presented in Figure 1. To meet the needs of experiment second stage the porous layer was created on the heat transfer surface. The special metal finishing was applied to increase the surface roughness. As an abrasive agent the broken alundum of 500 [μm] average grain size was used. The alundum grains were carried by the stream of compressed air under the pressure of 0.6 [MPa]. This metal finishing increased the surface roughness about three times in comparison with the original plate.

3. Experiment

Water – Water System

The single-phase convective heat transfer investigations in the water-water system were carried out on a dedicated facility for testing of heat exchangers, Figure 2. The first test stand enabled the heat transfer by convection between the hot and cold water. The hot water was circulating in the system with an electric flow heater, while the cold water was a tap water. In both circuits fine filters were installed. The heat was transferred due to the counter-current

flow of working media. The fluid flow rates were measured by the Cobold rotameters with the accuracy of ± 3 [dm³/h]. The heater was controlled by the power supply in the range from 0% to 100% of heating power. As a variable parameter the input temperature of heat exchanger was taken. During experiments the volumetric flow rate of hot/cold water was varied in the range from 50 to 400 [dm³/h]. The water supply pressure was about 0.4 [MPa]. Both heat exchangers were supplied with the hot water of temperature equal to 80 and 60 [°C], respectively in the first and second investigated cases. The cold water temperature was in each measurements' series equal to 15.5 ± 0.5 [°C].

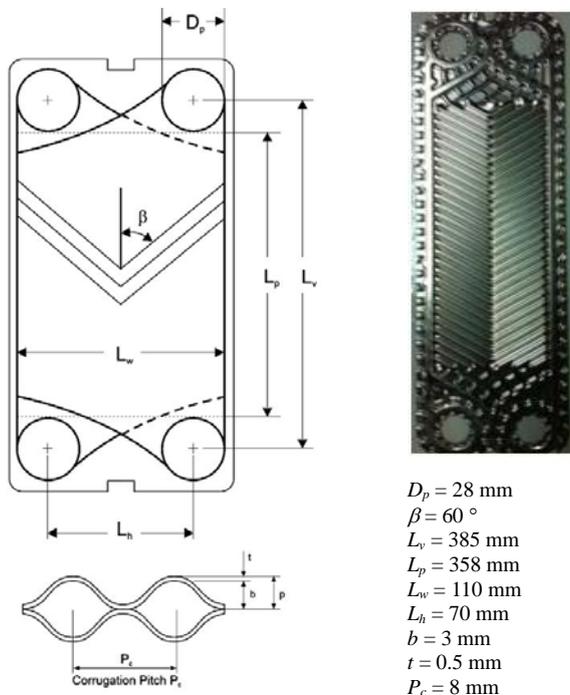


Fig. 1 Schematic and dimensions of heat exchanger plate.

Water – Ethanol System

The second test stand enabled the heat transfer tests by convection between the hot water and ethanol, Figure 3. Water was the heating medium, while ethanol - the coolant. The stream of water was first directed to the rotameter and then to the electrical heater to obtain a proper parameters at the inlet to heat exchanger. The heater was controlled by an autotransformer, which allowed smooth change of heater power and then the precise water temperature settings. The ethanol was circulating in a closed system equipped with thermostatic bath, which heated it to a certain level before entering the heat exchanger. For the needs of experiment an additional heat exchanger, supplied with the tap water (cold) was provided to the thermostatic bath. Because of that the thermal energy gained by the ethanol could be withdrawn from it, what assured the stationary state of the analysis.

During experiments the mass flow rate of hot water was varied in the range from 50 to 125 [dm³/h], while the ethanol mass flow rate was varied in the range from 35 to 160 [dm³/h]. Temperature of the hot water supplying the heat exchanger was 80 and 60 [°C], whereas the ethanol temperature was in each measurements' series equal to 30 ± 0.5 [°C]. In both type of experiments the pressure drop was measured by differential pressure transducer (Huba Control sensor) with accuracy of 1% of the full scale. Thermocouples of J-type were used to measure temperature in four locations i.e. at the inlet and outlet of heat exchanger cold side and at the inlet and outlet of heat exchanger hot side. Prior to experiments all thermocouples were calibrated to yield the accuracy of measurements of ± 0.5 [°C]. The reference temperature for thermocouples measurements was equal to 0 [°C]. On the basis of measurement results the heat flux (q), the Logarithmic Mean Temperature Difference (LMTD) in the heat exchanger and the

overall heat transfer coefficient (k) were calculated. The overall heat transfer coefficient was determined with the aid of the Peclet's law based on the heat transfer area equal to 0.039 [m²] and average value of the heat rate transferred through the wall in a given measurement series.

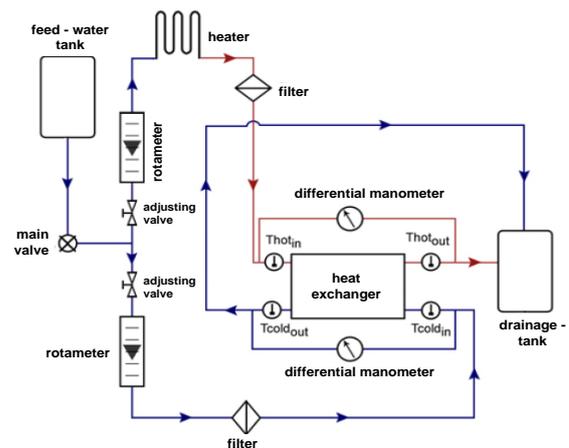


Fig. 2 Scheme of experimental facility – water-water system.

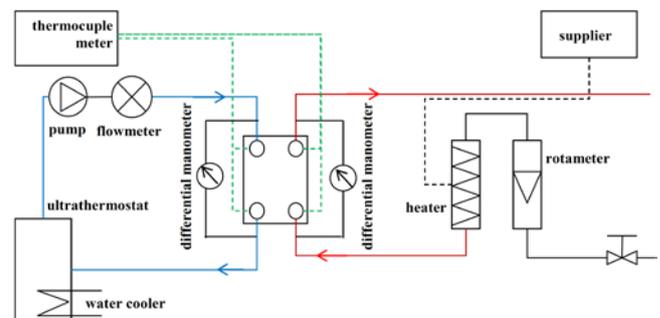


Fig. 3 Scheme of experimental facility – water-ethanol system.

4. Determination of Heat Transfer Coefficient

The experimental investigations of heat exchangers require determination of mean heat transfer coefficients on both sides of the wall separating exchanging heat fluids. Usually that requires installation of thermocouples for measurements of wall temperature separating two fluids. If the heat exchanger has a complex surface geometry then accurate measurement of the mean surface temperature faces significant difficulties for example in the course of disassembling installation a large number of thermocouples must be attached and subsequently everything must be reassembled again. Such difficulties can be alleviated if the Wilson's method [11] is applied. The method is very simple and can be applied to the analysis of different types of heat exchangers [12]. The heat transfer coefficient values obtained in hot and cold passes are shown below.

In the case of water-water system their values versus Reynolds number for one chevron channel (as usually presented in the papers) are presented in Figure 4 and Figure 5. As it was mentioned before, during tests the inlet temperature of hot and cold water was kept constant.

In case of water-ethanol system the convective heat transfer coefficient versus Reynolds number for one chevron channel is presented in Figures 6 and 7. During tests the inlet temperature of hot water and ethanol (cooling fluid) was kept constant. The Reynolds number was calculated with application of the formula:

$$(1) \quad Re_{1ch} = \frac{GD_H}{\mu}$$

where hydraulic diameter D_H , is usually taken as double corrugated depth ($D_H = 2b$), G - mass flux, μ - viscosity. The viscosity of both fluids was taken from Refprop [13] software for average temperature of hot passage $(T_{h-in} + T_{h-out})/2$ and cold passage $(T_{c-in} + T_{c-out})/2$ in the heat exchanger, respectively.

The heat transfer coefficients in the water-water system took higher values for the commercial heat exchanger in all studied cases (Figures 4 and 5). However it should be emphasized that for the smaller flow rates this predominance is not obvious. In the water-ethanol system (for small flow rates) the heat transfer coefficient on the ethanol (cold) side took higher values for the modified heat exchanger in all studied cases, but still on the water (hot) side it was higher for the commercial one, see Figures 6 and 7.

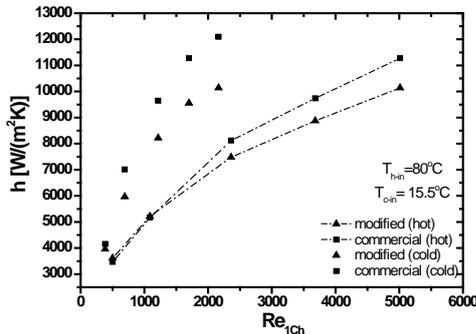


Fig. 4 Heat transfer coefficient versus Reynolds number; water-water system, $T_{h-in} = 80^\circ\text{C}$.

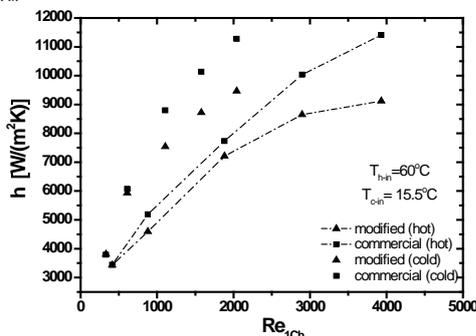


Fig. 5 Heat transfer coefficient versus Reynolds number; water-water system, $T_{h-in} = 60^\circ\text{C}$.

5. Determination of Flow Resistance

Generally, the total pressure drop (ΔP_{exp}) consists of four factors, namely the frictional term (ΔP_f), elevation term (ΔP_g), the pressure losses at the test section inlet and outlet ports (ΔP_p), and the acceleration term (ΔP_a). The latter term is included in the analysis only if the phase change of particular fluid would be observed. Therefore in the case of reported study, the acceleration term was omitted because there was no phase change at this stage of experiment.

The gravitational component was not taken into account due to the horizontal position of heat exchangers. To evaluate the friction factor associated with the water flows, the frictional pressure drop (ΔP_f) was calculated by subtracting the pressure losses at the ports of heat exchanger from the measured total pressure drop:

$$(2) \quad \Delta P_f = \Delta P_{exp} - \Delta P_p$$

The pressure drop at the inlet and outlet ports of heat exchanger was empirically suggested by [14]. This is approximately 1.5 times the head due to the flow expansion at the inlet:

$$(3) \quad \Delta P_p \approx 1.5 \left(\frac{G_p^2}{2\rho} \right)$$

where ρ is the density of fluid, while the mass flux inside the port, G_p , is defined as:

$$(4) \quad G_p = \frac{4m}{\pi D_p^2}$$

In Eq. (4) m is the mass flow rate, whereas D_p is the port diameter. The friction factor is described by formula:

$$(5) \quad f = \frac{\Delta P_f D_H \rho}{2 G_{1ch}^2 L_p}$$

where G_{1ch} is the mass flux in one chevron channel, L_p - the active length of heat exchanger.

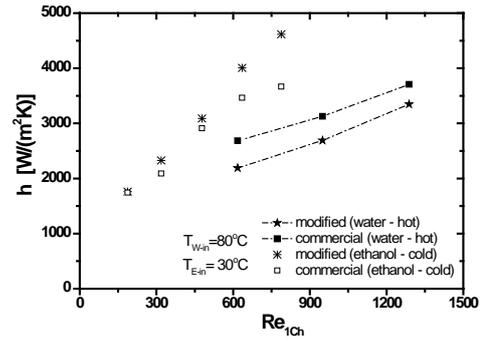


Fig. 6 Heat transfer coefficient versus Reynolds number; water-ethanol system, $T_{h-in} = 80^\circ\text{C}$.

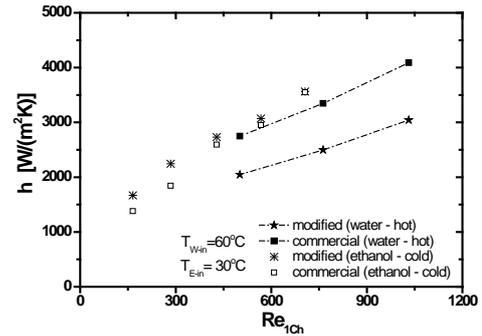


Fig. 7 Heat transfer coefficient versus Reynolds number; water-ethanol system, $T_{h-in} = 60^\circ\text{C}$.

In the case of water-water system the experimental investigations of hydraulic resistance were conducted with the same thermal conditions in both (hot and cold) passages of the heat exchanger, in which the water temperature was equal to 15.5 °C. The pressure drop as a function of volumetric flow rate applied in the experiment is presented in Figure 8. The friction factor profile calculated with utilization of Eq. (5) is shown in Figure 9. The flow characteristics in water-water system demonstrates smaller values of pressure drop for the modified heat exchanger. It is connected with lower values of friction factor for that case.

The flow characteristics in the water-ethanol system is presented in Figure 10. It shows that for very low flow rates the overall pressure drop is higher for modified heat exchanger than for commercial one. However this tendency is opposite for higher values of flow rates. It corresponds to the friction factor presented as a function of Reynolds number in Figure 11. With increasing Reynolds number the friction factor of modified surface decreased and finally became smaller than for the commercial plate.

6. Conclusions

The experimental analysis of heat transfer enhancement for plate heat exchanger was described. The results of heat transfer for the exchanger with modified surface were always compared with the results of the commercial one. Regarding the heat transfer coefficients (obtained with the use of the Wilson's method) of water-water system, the results were always higher for the commercial (original) heat exchanger. It means that the porous layer did not intensify the heat transfer in this case. Analysis of water-ethanol system gave very interesting data – the heat transfer coefficient on the ethanol side for small flow rates took higher values for the modified heat exchanger. It is in agreement with the results presented in [9,10], where the heat transfer intensification was also observed for smaller flow rates. The first attempt to the understanding of this phenomena was undertaken. Authors considered the values of water and ethanol surface tension. The surface tension of ethanol is about four times smaller than the surface tension of water. Therefore the wettability of ethanol is

larger than water and it can explain the better results of heat transfer in the case of porous layer.

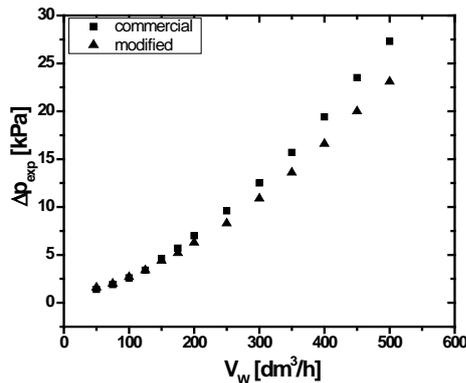


Fig. 8 Flow characteristics in water-water system.

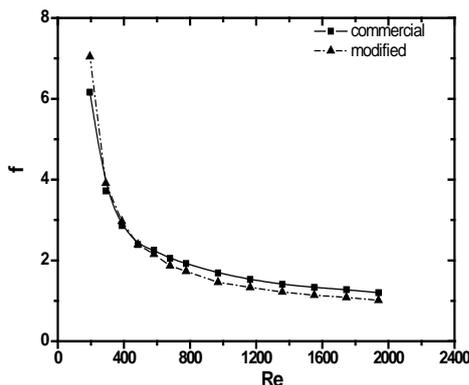


Fig. 9 Characteristics of friction factor in water-water system.

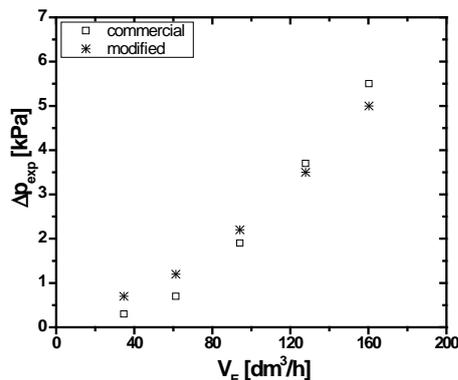


Fig. 10 Flow characteristics for water-ethanol system.

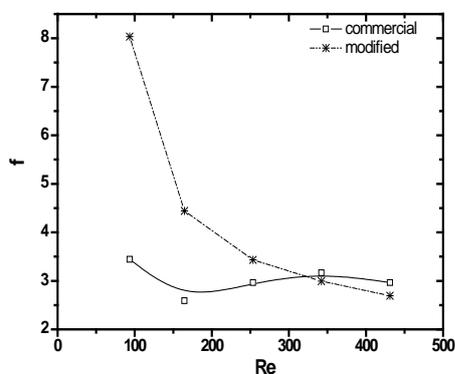


Fig. 11 Characteristics of friction factor in water-ethanol system.

Analytical analysis of this phenomena is in progress. The smaller overall pressure drop values for the heat exchanger with porous layer in the water-water system corresponds to the lower values of friction factor. It looks like the porous layer caused decrease of the friction, probably due to the breaking of boundary

layer and turbulization. In the case of water-ethanol system the Authors obtained opposite results. It means that the overall pressure drop was slightly higher for the modified heat exchanger surface, what corresponds with the higher value of friction factor. The explanation could be also connected with the surface tension and wetting ability of ethanol. It looks like the porous layer caused higher ethanol friction, because due to the smaller surface tension it goes "deeper" into the pores.

Presented data shows that described surface finishing is not suitable for working fluids with high values of surface tension (for example water), but can be utilized in the system, in which the working fluid has low value of surface tension (for example ethanol, refrigerants). Therefore there is open area of such passive enhancement in the ORC systems.

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