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EFFECT OF SURFACE FINISH ON HEAT TRANSFER PERFORMANCE OF PLATE HEAT EXCHANGER

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ABSTRACT

In the paper an experimental analysis of passive heat transfer intensification technique employed in the case of plate heat exchanger is presented. The passive intensification was obtained by a modification of the heat transfer surface. The roughness of surface was increased by a usage of glass micro-beads. Single-phase convective heat transfer in the water-water system was studied.

The experiment was accomplished in two stages. In the first stage the commercial 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's temperatures at the inlet of heat exchangers in the corresponding measurements' series. Due to this the systematic experimental data show that larger roughness of heat transfer surface leads to an increase of heat transfer coefficient on the side of cooling water (increase by about 30 ÷ 35%) and simultaneously to an increase of flow resistance (up to 30% when the volumetric flow rate is equal to 500 l/h). On the side of heating water it was found that the heat transfer coefficient increased by about 25%, while the flow resistance by about 22% (the volumetric flow rate of 500 l/h).

INTRODUCTION

The techniques of heat transfer improvement (intensification) in conventional applications have been under scrutiny in literature for more than century and a large number of information was gathered up to now [1]. Generally speaking, the intensification methods can be classified as passive (no additional energy have to be supplied) and active (an additional energy is required). Efficiency of such methods strongly depends on heat transfer conditions and mechanisms which can change from single phase convective heat transfer to the flow boiling. With the prospects of energy efficiency, miniaturization, product reliability, and the potentially large economic advantages, an extensive research and development effort has been undertaken in the area of enhanced heat transfer over the past couple of decades [1–4].

NOMENCLATURE

A	[m^2]	surface
C_c, C_h, C_3	[-]	constants of linear regression
h	[(J/kg·K)]	enthalpy
k	[W/(m^2·K)]	overall heat transfer coefficient
$\dot{m}$	[kg/s]	mass flux
P	[Pa]	pressure
$\dot{Q}$	[W]	rate of heat
R	[μm]	roughness parameter
T	[K]	temperature
$\dot{V}$	[m^3/s]	volumetric flow rate
w	[m/s]	velocity
x		Cartesian axis direction
y		Cartesian axis direction

Greek letters

α	[W/(m^2·K)]	convective heat transfer coefficient
δ	[m]	wall thickness
λ	[W/(m·K)]	thermal conductivity

Subscripts

c	cold
h	hot
in	inlet
out	outlet

Over the past 20 years, the compact plate heat exchangers have replaced the traditional shell-and-tube heat exchangers, since the former are more energy and space efficient and are cheaper to produce. Despite this trend, only a few attempts to enhance compact plate heat exchangers with high performing microsized, or smaller, enhancement structures have been reported. Nowadays most of the activities in that area relate to enhancement of heat transfer in boiling range.

Presented below is a survey of some methods which lead to enhancement of pool boiling in plate heat exchangers.

Hillis and Thomas [5], as part of an evaluation of heat exchangers for a large 40 MW ocean thermal energy conversion pilot plant in Hawaii, tested the performance a small-frame plate heat exchanger with ammonia as refrigerant. The heat exchanger plates featured a 60 deg chevron angle corrugation pattern and were coated in Linde's high-flux surface: a porous aluminum particle layer. Boiling heat transfer coefficients of about 30 kW/(m^2·K) were recorded at a heat flux of 26 kW/m^2, equivalent to a fivefold improvement compared to uncoated surface.

Müller-Steinhagen [6] vacuum plasma sprayed a 250 μm thick layer of spherically shaped Inconel 625 particles on to a plate and frame heat exchanger surface. The particles had a diameter of 105–170 μm and enhanced the boiling heat transfer coefficient of R134a with up to 100%.

Matsushima and Uchida [7] tested a brazed plate heat exchanger with a novel pyramid-like structure in R22. The structural features were 1.5 mm in height, hence, not in the microsized region, but the evaporation heat transfer coefficients were estimated to be 1.5–2 times higher than those of regular herringbone-type plates.

Longo et al. [8] applied 50–200 μm sized, pyramid-like surface features to a plate heat exchanger with herringbone macroscale corrugation, resulting in a 40% increase in the boiling heat transfer coefficient of R22. This enhancement was larger than the increase in heat transfer area, suggesting a real improvement in the boiling heat transfer mechanism.

Recent developments within nano- and microtechnologies have made possible the creation of well defined three-dimensional connected porous network structures, Davis [9], which, so far, have mainly been used in applications such as catalysis molecular sieves, fuel cells, sorption, and separation [10]. These developments have opened up new possibilities to structure high performing boiling surfaces with well defined micro- and submicron topology, using methods more precise than mechanical deformation.

A novel nano- and microporous structure was recently shown to enhance pool boiling heat transfer in 134a with over one order of magnitude compared to a plain machined copper surface by Furberg et al. [11]. He presented an experimental study of the performance of a plate heat exchanger evaporator with and without this novel enhancement structure applied to the refrigerant channel. The flow boiling tests were conducted for 134a with heat fluxes ranging from 4.5 kW/m^2 to 17 kW/m^2. Various distance frames were also used to widen the refrigerant channel in order to isolate the influence of refrigerant mass flux and thereby gain a better understanding of boiling heat transfer in plate heat exchangers in general and the enhanced surface in particular.

In this paper, for the purpose of investigating Organic Rankine Cycle evaporators and future technical applications, an analysis of passive heat transfer intensification in the case of plate heat exchanger was done. A new technique of increasing the surface roughness is proposed, namely through abrasive blasting with the utilization of glass micro-beads. Such technique is relatively not expensive and produces the enhancement effect. Experimental data were collected for water-water case, where the heat transfer coefficient was calculated using the Wilson method. That method seems to be, in the authors' opinion, the only one for finding the heat transfer coefficient for such a complex heat exchanger structure. In the near future it is planned to perform experiments on boiling with low boiling point fluids to show the capacity of proposed method of surface modification.

PLATE HEAT EXCHANGER

The twisted plate heat exchanger offered at the home/world market by Sondex was the subject of presented investigations. In this kind of heat exchanger the heat is transferred in one pass. It was made of stainless steel (no. 316 according to AISI standard) and consisted of twelve plates, whose thickness was 0.5 mm. The total length of heat exchanger was 270 mm, its capacity was 2 l and its weight was 19.5 kg. The overall heat transfer area was equal to 0.468 m^2. The distance between the plates was kept constant and the EPDM seal was fixed in the system "hang on". Permissible working pressure was equal to 16 bar. The schematic view of heat exchanger is presented in Figure 1, while its main features are listed in Table 1. The individual plate is shown in Figure 2.

Table 1 Main features of commercial heat exchanger

plate dimensions length/width [mm]	capacity of one channel [l]	number of plates	roughness of plate surface [μm]
451 / 141	0.170	12	R_a=0.46, R_z=3.34

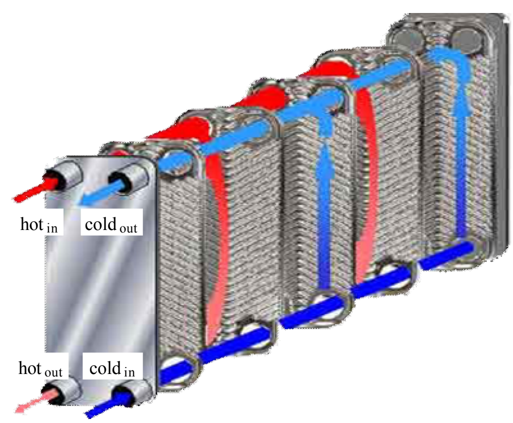

Figure 1 Scheme of plate heat exchanger


Figure 2 View of a single plate

For the purpose of investigations the heat exchanger plates were subjected to a abrasive blasting with the utilization of glass micro-beads. Granulation of the beads was of 300-400 µm, while the density of glass was 2.5 g/cm^3 and its hardness was 6 according to the Mohsa's scale.

The roughness changes of heat exchanger plate surface, as a result of abrasive blasting, were examined by the Ship Design and Research Center in Gdansk. The measurements were done with the Surftest 211 (Mitutoyo). At first the measurement device was calibrated with application of the roughness' standard 178-601 delivered by Mitutoyo company. The flat parts of heat exchanger plates were examined, while the sampling length was 0.25 mm. The results are presented in Table 2 and applied designations are as follows: A – primary surface state, B – surface after the abrasive blasting.

Table 2 The measurement results of surface roughness

sample	measured parameter	average value [µm]
A	R_a	0.46
	R_z	3.34
B	R_a	2.8
	R_z	15.9

Parameter R_a is an average arithmetical roughness in the range of sampling length l (Figure 3). Parameter R_z is an arithmetic average of absolute height of five the highest roughness' peaks and height of five the deepest valleys in the range of sampling length l (Figure 4).

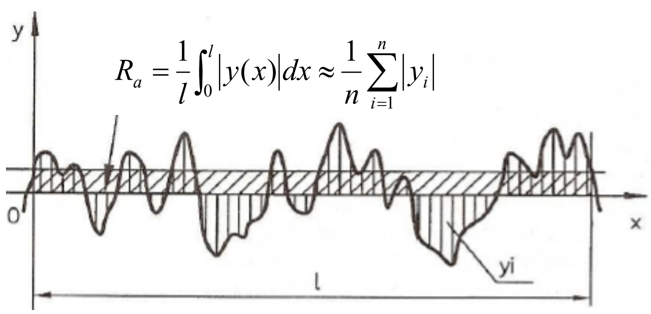

Figure 3 Graphical interpretation of parameter R_a

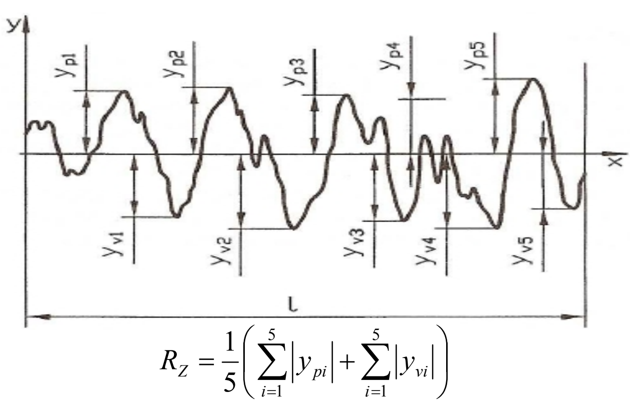

Figure 4 Graphical interpretation of parameter R_z

The mechanical working – abrasive blasting caused six times higher roughness expressed by parameter R_a and about five times higher roughness expressed by parameter R_z.

EXPERIMENT

The experimental investigations of plate heat exchangers were carried out on a dedicated facility for testing of heat exchangers, Figure 5.

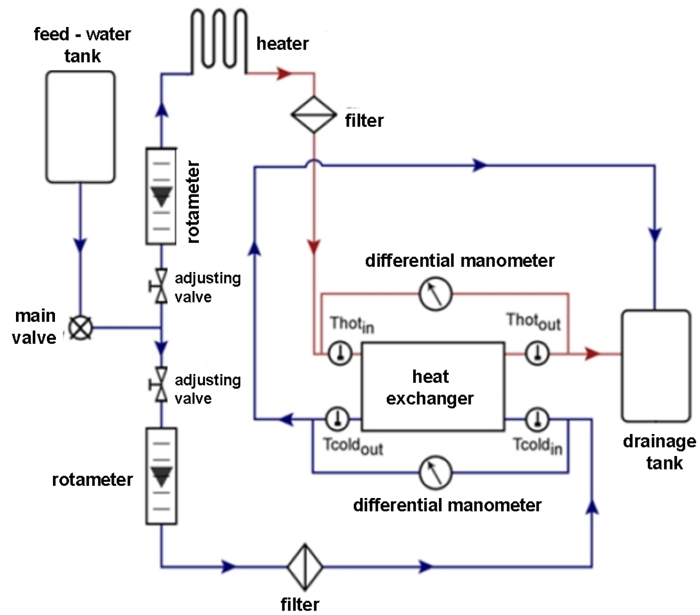

Figure 5 Scheme of experimental facility

The 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 co-current flow of working media. The fluid flow rates were measured by rotameters with the accuracy of ±3 l/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. The pressure drop was measured by mercury manometers with accuracy of ±2 mmHg. Thermocouples of J-type were used to measure temperature in four points i.e. at the inlet and outlet of heat exchanger's cold side and at the inlet and outlet of heat exchanger's hot side. Prior to experiments all thermocouples were calibrated to yield the accuracy of measurements of ± 0.5 ^{o}C. The reference temperature for thermocouples measurements was equal to 0 ^{o}C.

During experiments the following parameters were measured: the hot water temperature at the inlet (T_{h-in}) and at the outlet (T_{h-out}) of heat exchanger, the cold water temperature at the inlet (T_{c-in}) and at the outlet (T_{c-out}) of heat exchanger, the pressure drop connected with the hot water flow (ΔP_h), the pressure drop connected with the cold water flow (ΔP_c), the volumetric flow rate of hot water ($\dot{V}_h$) and the volumetric flow rate of cold water ($\dot{V}_c$). The volumetric flow rate of hot/cold water was varied in the range from 100 to 500 l/h. The water supply pressure was 4 bar. On the basis of measurement results the heat flux (q), the Logarithmic Mean Temperature Difference in the heat exchanger ($LMTD$) and the overall heat transfer coefficient (k) were calculated. The overall heat transfer coefficient was determined with the Peclet law based on the rate of heat transfer, taken up by hot water and the heat transfer area equal to 0.468 m^2.

DETERMINATION OF HEAT TRANSFER DATA

The experimental investigations of heat exchangers require determination of mean heat transfer coefficients on both sides of the wall separating fluids exchanging heat. Usually that requires installation of thermocouples for measurements of wall temperature separating two fluids. If the heat exchanger has a large number of tubes and 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 put up together. Such difficulties can be alleviated if the Wilson's method [12] is applied. The method is very simple and can be applied to the analysis of different types of heat exchangers [13]. A simple and efficient Wilson method in a version similar to the original one was applied in determination of heat transfer coefficient. The classical Wilson method, as well as its modifications, requires only determination of the overall thermal resistance in the heat exchanger. From the Wilson's method an accurate energy balance, based on measurement of flow rates of fluids exchanging heat and their mean temperatures at inlet and outlet of the heat exchanger is obtained.

The thermal balance of heat exchanger can be presented in the form:

$$\dot{Q} = k \cdot LMTD \cdot A = \dot{m}_h \Delta h_h = \dot{m}_c \Delta h_c \tag{1}$$

where: $LMTD$ - logarithmic mean temperature difference, A – heat transfer surface, whereas overall heat transfer coefficient can be described as:

$$k = \left(\frac{1}{\alpha_h} + \frac{\delta}{\lambda} + \frac{1}{\alpha_c} \right)^{-1} \tag{2}$$

where: α_h and α_c are heat transfer coefficients for respective mass flow rates; δ is a thickness of a wall separating two fluids, whereas λ its thermal conductivity.

The mean wall temperature can be determined from a relation:

$$LMTD = \frac{\left(T_{h\_in} - T_{c\_out}\right) - \left(T_{h\_out} - T_{c\_in}\right)}{\ln \dfrac{\left(T_{h\_in} - T_{c\_out}\right)}{\left(T_{h\_out} - T_{c\_in}\right)}} \tag{3}$$

That is especially important in the case of finned tubes where determination of a mean value of wall temperature is difficult basing on local measurements.

Assuming that heat transfer is primarily governed by flow velocities of both fluids, then simple relations can be written:

for $\dot{m}_c$ = const. and $\dot{m}_h$ = var there is:

$$\alpha_c = const\text{ , } \alpha_h = C_h w_h^n \tag{4}$$

for $\dot{m}_h$ = const. and $\dot{m}_c$ = var there is:

$$\alpha_h = const\text{ , } \alpha_c = C_c w_c^n \tag{5}$$

where w_h and w_c are respective flow velocities; n is coefficient depending on the character of heat transfer, for example in case of turbulent flow inside tubes n=0.8, whereas in case of a laminar one, n=0.5.

For heating medium following relation can be formulated:

$$\frac{1}{k} = \left(\frac{1}{\alpha_c} + \frac{\delta}{\lambda} \right) + C_h w_h^{-n} \tag{6}$$

or: $$\frac{1}{k} = C_3 + C_h w_h^{-n} \tag{7}$$

where: $$C_3 = \frac{1}{\alpha_c} + \frac{\delta}{\lambda} \tag{8}$$

for a series where $\dot{m}_c$ = const. Assuming new variables, i.e. $x = w_h^{-n}$ and $y = 1/k$ a linear relation is obtained:

$$y = C_3 + C_h x \tag{9}$$

For cooling side analogical relations can be derived.

The heat transfer coefficient calculations by Wilson's method were conducted for the plate thickness of 0.5 mm. The plate material (the stainless steel) has the thermal conductivity λ equal to 15 W/(m·K). For the hot fluid the straight line was plotted and its equation has form $\boldsymbol{y = C_3 + C_h\, x}$ (Figure 6), where $C_h = 63\times10^{-6}$ and $C_3 = 29\times10^{-5}$ (modified) and $C_h = 31\times10^{-6}$ and $C_3 = 39\times10^{-5}$ (commercial).

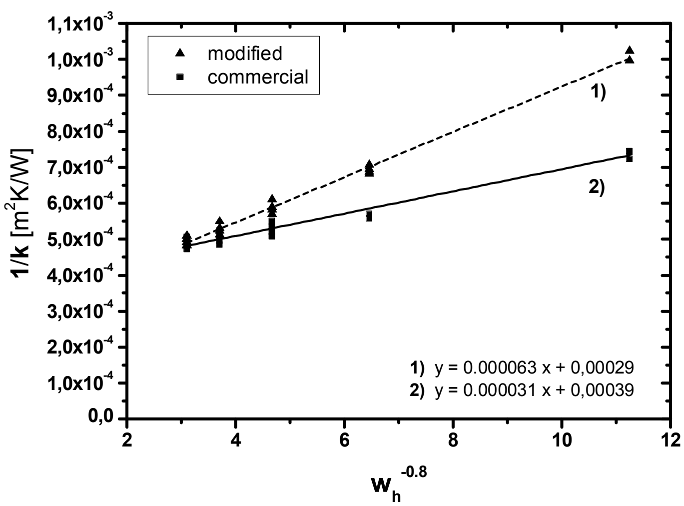

Figure 6 Experimental points and linear regression for $\dot{V}_h$ = 500 l/h

In Tables 3 and 4 the summary of heat transfer coefficient values obtained in the cold and hot passes are presented.

Table 3 Heat transfer coefficient on the "cold side"

	$\dot{V}_h$ [l/h]	100 ÷ 500				
	$\dot{V}_c$ [l/h]	100	200	300	400	500
commercial	α_c [W/(m^2·K)]	1384	2120	2788	3429	3851
modified		1970	3736	4511	5245	6148

Table 4 Heat transfer coefficient on the "hot side"

	$\dot{V}_c$ [l/h]	100 ÷ 500				
	$\dot{V}_h$ [l/h]	100	200	300	400	500
commercial	α_h [W/(m^2·K)]	932	1516	2046	2655	2836
modified		1240	2055	2676	3628	3822

RESULTS OF MEASUREMENTS

The exemplary comparison of studied heat exchangers' thermal characteristics are shown below. 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 measurements' series.

The presented below graphs were constructed at following conditions: temperature of hot water at the heat exchanger inlet was 80°C, temperature of cold water at the heat exchanger's inlet was 10.5°C, volumetric flow rate of cold water was equal to 500 l/h, volumetric flow rate of hot water was varied in the range from 100 to 500 l/h.

Distribution of LMTD versus the volumetric flow rate of hot water (heating medium) is presented in Figure 7. The pressure drop as a function of volumetric flow rate applied in the experiment is presented for hot and cold passes respectively in Figure 8 and Figure 9.

Analysis of presented flow and thermal characteristics shows that in the heat exchanger with modified surface (larger roughness) increasing volume flow rate of working fluids causes distinct increase of the LMTD and simultaneous of the flow resistance increase.

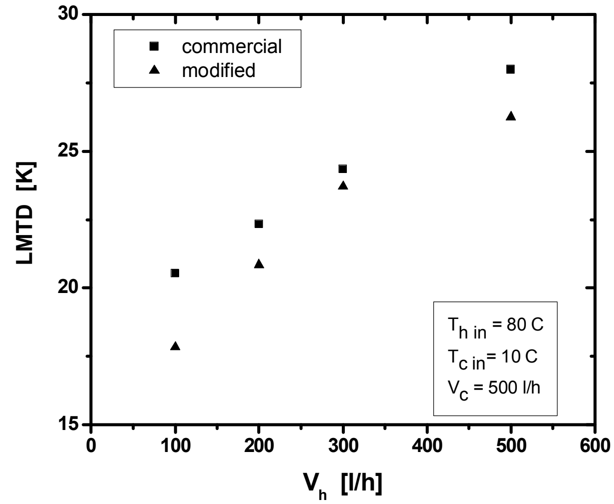

Figure 7 LMTD versus volumetric flow rate of hot water

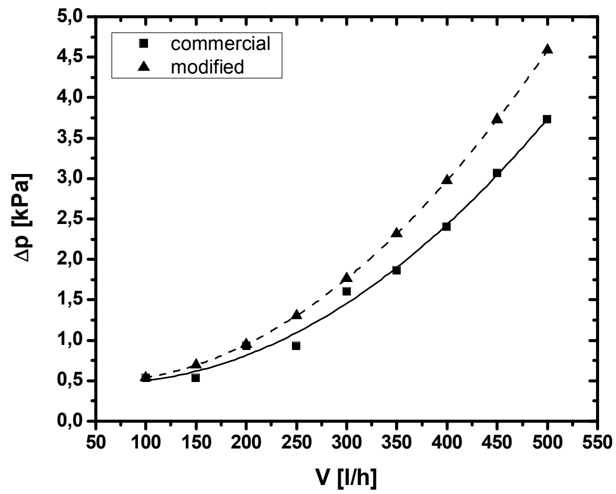

Figure 8 Flow characteristics of hot circuit

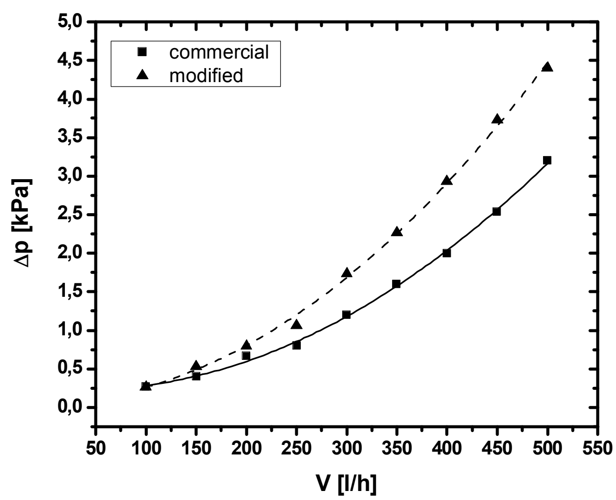

Figure 9 Flow characteristics of cold circuit

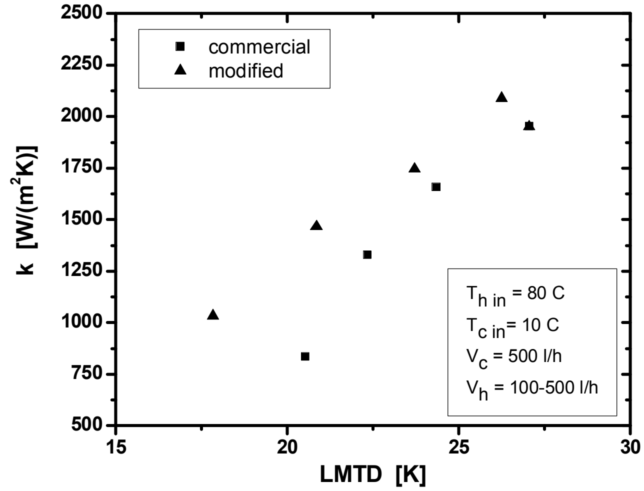

Figure 10 Overall heat transfer coefficient versus LMTD

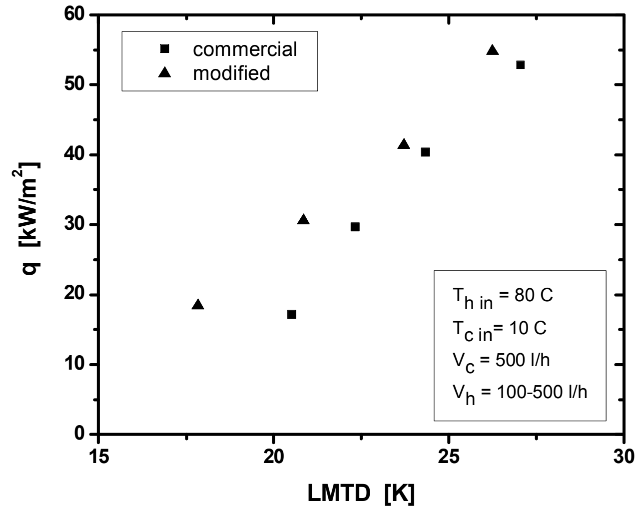

Figure 11 Heat flux density versus LMTD

In Figure 10 the distribution of overall heat transfer coefficient of the heat exchangers in relation to the LMTD is presented. It can be seen that in the heat exchanger with modified surface it was possible to obtain higher values of overall heat transfer coefficients. In Figure 11 the characteristics of heat flux density versus the Logarithmic Mean Temperature Difference (LMTD) for two kinds of heat exchanger are shown. The increase of heat flux density with the increase of LMTD can be observed.

The higher values of heat transfer coefficient indicates that the higher values of heat flux could be transferred by the modified heat exchanger in comparison with the commercially available one.

CONCLUSIONS

The systematic experimental investigations of two heat exchangers: normal one and with modified heat transfer surface were described. On the basis of presented in the paper results it can be seen that larger roughness of heat transfer surface leads to an increase of heat transfer coefficient on the side of cooling water (increase about $30 \div 35\%$) and simultaneously to an increase of flow resistance (to 30% when the volumetric flow rate is equal to 500 l/h). On the side of heating water it was found that the heat transfer coefficient increased about 25%, while the flow resistance of about 22% (the volumetric flow rate of 500 l/h). These values indicate that the higher values of heat flux could be transferred by the heat exchanger with modified heat transfer surface.

Presented results are promising and show the perspective for further heat transfer enhancement. Described model of heat exchanger will be submitted to the patent procedure.

ACKNOWLEDGEMENTS

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