

USE OF A NOVEL NANO- AND MICRO-POROUS STRUCTURE FOR ENHANCED BOILING IN A PLATE HEAT EXCHANGER

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Abstract: Presented research is an experimental study of the performance of a standard plate heat exchanger evaporator with a novel nano- and micro-porous copper structure, used to enhance the boiling heat transfer mechanism in the refrigerant channel. The enhancement structure had a porosity of 95%, a pore density of 75 pores/mm² and a thickness of about 250 µm. Flow boiling tests were conducted at heat fluxes ranging between 4.5 and 17 kW/m² with R134A as refrigerant. Based on water-to-water experiments to determine the water side heat transfer coefficient, the plate heat exchanger with the enhancement structure displayed up to 10 times improved heat transfer coefficient in the refrigerant channel. The overall performance of the heat exchanger was improved by over 100%. This significant boiling enhancement is in agreement with previous pool boiling experiments and confirms that the enhancement structure may be used to enhance the performance of plate heat exchangers.

Key Words: *enhanced boiling, plate heat exchanger, nano and micro technology, R134A, flow boiling.*

1 INTRODUCTION

Energy efficiency, miniaturization, product reliability and safety and economic advantages are the major benefits of improving boiling heat transfer in various heat exchangers. Hence, an extensive research and development effort has been undertaken in the area of enhanced boiling heat transfer over the last couple of decades. Several informative summaries have been published (Thome 1990) and (Webb 1994), which report on the boiling performance of various micro-scale enhancement structures. The primary industrial usage of these micro-sized boiling enhancement structures have been in so called shell and tube heat exchangers for large refrigeration applications (Webb 2004). At the same time, over the past 20 years, compact plate heat exchanges have replaced traditional shell-and-tube heat exchangers, since the former are more energy and space efficient and are cheaper to produce. But, apart from using macro enhancement structures, often in the form of a herringbone pattern, few attempts to enhance compact plate heat exchangers with micro-size enhancement structures have been reported on.

Matsushima and Uchida have pointed out the relatively low heat transfer coefficient on the refrigerant side, compared to the single-phase water side, as a major drawback of plate heat exchangers. In an attempt to overcome this large thermal resistance on the boiling side, a brazed plate heat exchanger with pyramid-like structures was developed and tested in R22. The structure features were 1.5 mm in height, hence not in the micro-sized region, but the evaporation heat transfer coefficients were estimated to 1.5-2 times than those of regular herringbone-type plates (Matsushima and Uchida 2002). Longo et al. applied 50-200 μm sized, also tetrahedron like, surface features to a plate heat exchanger with herringbone macro-scale corrugation, resulting in a 40% increase of the boiling heat transfer coefficient in R22 (Longo et al. 2004). This enhancement was larger than the increase in heat transfer area, suggesting a real improvement of the evaporation and/or boiling heat transfer mechanism. Since both of these pyramid-like structures are fabricated through mechanical deformation, the possibilities to control sub-micron features are few. Hence, without re-entrant cavities and large area enhancements, the boiling enhancement potential of these aforementioned structures is rather limited.

The importance of nano-sized surface structures for the boiling performance of a surface has been shown (Honda 2002) and (Vemuri and Kim 2005). Recent developments within nano- and micro- technologies have made possible the creation of well defined three dimensional connected porous network structures (Davis 2002), which, so far, have mainly been used in applications such as catalysis molecular sieves, fuel cells, sorption and separation (Stein 2003). These developments have opened up new possibilities to structure high performing boiling surfaces with well defined micro- and sub-micron topology, using methods more precise than mechanical deformation.

Presented research is an experimental study of the performance of a gasket plate heat exchanger evaporator with a novel nano- and micro-porous structure applied to the refrigerant channel. The surface structure was manufactured with an electrodeposition method where metallic nano-particles are dendritically connected into an ordered micro-porous structure using hydrogen bubble evolution as a dynamic masking template. The manufacturing technique provides ample room to control both the micro- and the nano-sized features of the surface structure. The flow boiling tests were conducted in R134A with heat fluxes ranging from 4.5-17 kW/m^2 .

2 EXPERIMENTAL APPARATUS AND PROCEDURE

The schematics of the experimental test set-up are described in Figure 1. The system facilitates independent control of;

- The refrigerant in- and outlet quality of the test object, by regulating the refrigerant pump speed and the pre-heater power.
- The flow rate on the water side of the test object, with the frequency controlled water pump.
- The sub-cooling of the refrigerant by controlling the temperature of the cooling water into the sub-cooler.
- The overall heat transferred in the test object with the condenser power and the electrical heater on the water side of the test object.

The vital components of the test set-up are; the refrigerant pump is from Ismatec (MZP-Z), pressure transducers from Druck (abs. PDCR 960, diff. PDCR 2110), refrigerant mass flow meter from Micro motion (D series) and two water flow meters from Siemens (Sitrans fm intermag 2). The T-type thermocouple were connected to a logger from Hewlett-Packard (HP 34970A).

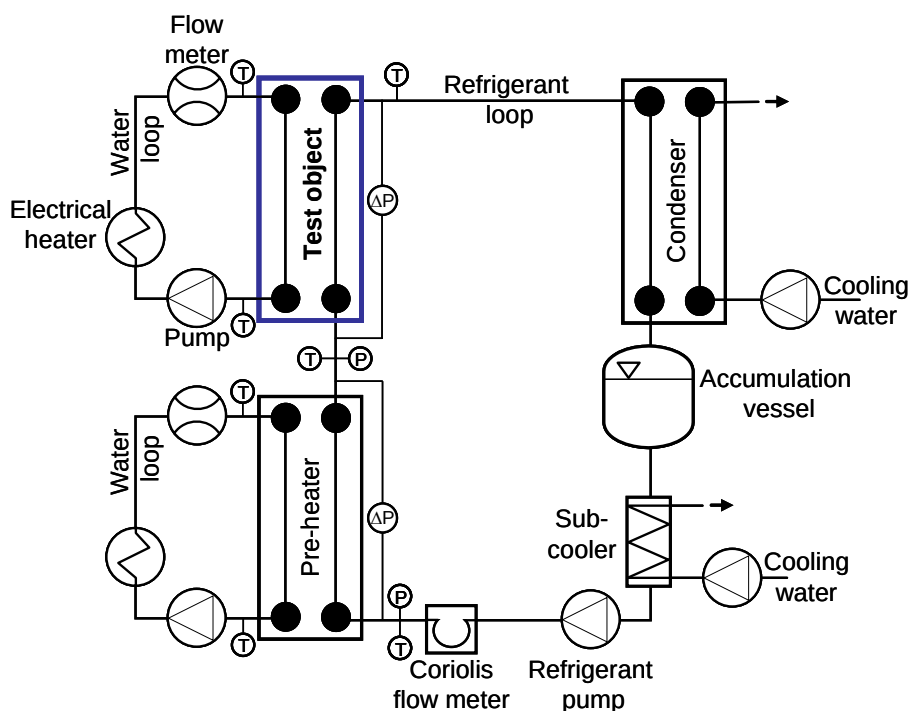


Figure 1: Experimental set-up for thermal tests of plate heat exchanger.

2.1 Test conditions

R134a was used as refrigerant for all tests at a pressure varying between 3-4 bar, depending of overall performance of the heat exchanger and the heat flux. Tests were carried out under oil free conditions. The water side flow rate through the test object corresponded to a temperature drop from 12-7 °C, which further corresponded to a Reynolds number of 90 at 4.5 kW/m². The incoming vapor quality was set to 5-15% and outgoing vapor quality was ≤100%. The flow direction of the water and the refrigerant was counter current. Results from preliminary tests showed that difference in inlet quality between 5 and 15% had a negligible effect on the heat transfer coefficient on the refrigerant side. The outgoing vapor quality was controlled with the refrigerant pump to near 100% by decreasing the refrigerant flow until the measured superheat was 0°C. All the components of the experimental set-up, but the test object itself, were insulated with 10 mm thick foam insulation. The heat flux was varied between 4.5 and 14 kW/m² for the standard plates (higher heat flux could not be tested at the time due to limitations in the test rig) and 4.5 and 17 kW/m² for the enhanced heat exchanger.

2.2 Data reduction

The overall heat transfer coefficient U of the heat exchanger was defined as the ratio between the exchanged heat flow rate, Q , and the logarithmic temperature difference, ΔT_{ln} , and the nominal heat transfer area, A_{tot} :

$$U = \frac{Q}{A_{tot} \cdot \Delta T_{ln}} \quad (\text{Eq. 1})$$

where Q is calculated from a heat balance on the water side according to:

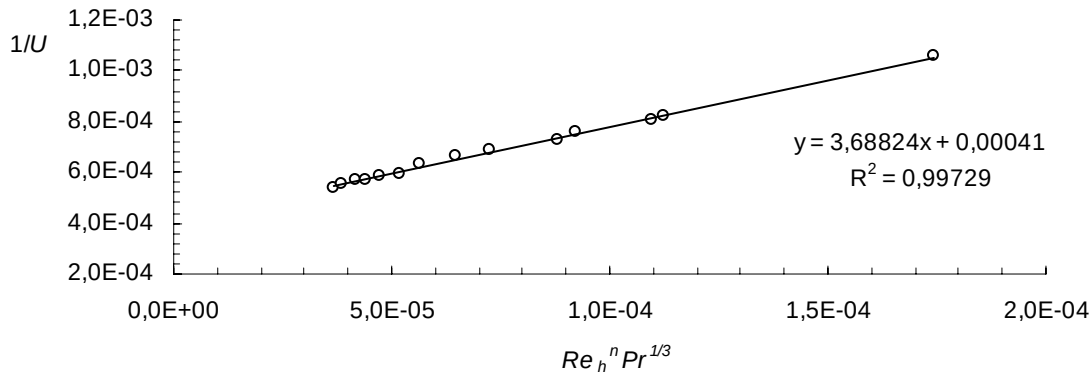


Figure 2: Modified Wilson plot results for calibration of water side heat transfer coefficient.

$$Q = \dot{m}_w \cdot c_{pw} \cdot (T_{w_in} - T_{w_out}) \quad (\text{Eq. 2})$$

where $\dot{m}_w$ is the water mass flow rate, c_{pw} the specific heat capacity of water and T_{win} and T_{wout} are the in- and outgoing water temperatures respectively. The logarithmic temperature difference was calculated based on the saturated refrigerant temperature, T_{sat_out} , at the outlet of the heat exchanger, according to:

$$\Delta T_{ln} = \frac{(T_{w_in} - T_{sat_out}) - (T_{w_out} - T_{sat_out})}{\ln\left(\frac{T_{w_in} - T_{sat_out}}{T_{w_out} - T_{sat_out}}\right)} \quad (\text{Eq. 3})$$

It can also be shown that the overall heat transfer coefficient, U , may be calculated using:

$$U = \frac{1}{\frac{1}{h_w} + \frac{x}{\lambda_{plate}} + \frac{1}{h_{ref}}} \quad (\text{Eq. 4})$$

In order to evaluate the effect that the enhanced surface structure would have on the heat transfer performance on the refrigerant side, the heat transfer coefficient on the water side was first tested using a modified Wilson plot method. The plates in the heat exchanger were configured to form two channels. Water-to-water tests were carried out where the heat transfer coefficient was kept constant on in one channel, while the water flow in the other channel was varied between Reynolds number 94 and 1070. The general heat transfer correlation for in the water channel was assumed to follow:

$$Nu_w = \frac{h_w d_{hw}}{\lambda} = C_1 Re_h^n Pr^{1/3} \quad (\text{Eq. 5})$$

where the hydraulic diameter of the water channel was defined as $d_{hw} = 2B/\phi$. A best fitting procedure, as seen in Figure 2, was used on the experimental data and the constants were determined to $C_1=0.271$ and $n=0.668$. The heat balance between the hot and the cold side during the Wilson plot tests was within $\pm 3\%$. Figure 3 shows the single phase heat transfer coefficient as a function of the Reynolds number (the Prandtl number is the same in each point) for presented heat exchanger (Furberg et al.) as well as for a selection of plate heat

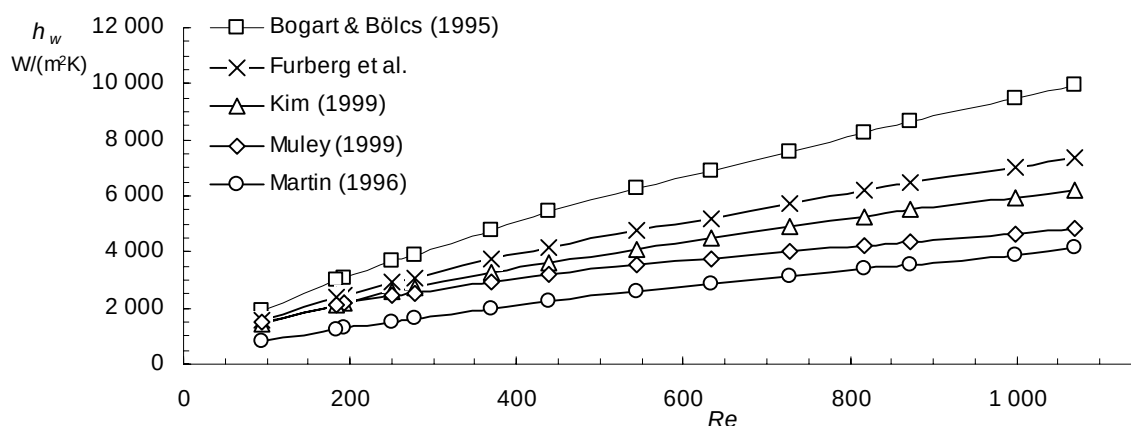


Figure 3: Heat transfer coefficient on the water side as a function of Reynolds number.

exchanger correlations. Kim's correlation under predicts the data with about 10 to 15%, but is still the most accurate correlation out of the selected ones.

2.3 Uncertainty estimates

An uncertainty analysis, according to the Kline and McClintock approach (Kline and McClintock 1953), of the overall heat transfer coefficient (the U-value) has been made. The overall heat transfer coefficient, based on the heat transfer plate surface area A_{tot} , is a function of 4 independent variables: water volume flow ($\dot{V}$), temperature difference between incoming and outgoing fluids (ΔT), specific heat of water (c_{pw}), water density (ρ_w). Table 1 presents the uncertainty of each variable and for three different test cases at low and high heat flux.

As evident from table 1, the accuracy of the temperature measurement had the major influence on the calculated U-value as shown in Table 1. The uncertainty interval for the temperature difference (ΔT) has been estimated to 0.1°C (20:1 odds). Measurements of the enhanced heat exchanger at low heat flux and high flow rates resulted in the lowest temperature differences ΔT and therefore come with the largest overall uncertainty of about 5%. Heat gain to the heat exchanger during the tests was estimated using natural convection correlations (Incropera and DeWitt 2002) and was between 1-5% of the transferred heat. The U-values presented in this work have not been adjusted for the heat gain.

2.4 Test object

As illustrated in Figure 4, the plate pack of the tested plate heat exchanger was assembled between two frame plates and compressed by tightening bolts. The plates were fitted with a

Table 1: Uncertainty contribution to the U-value from different variables.

	Standard		Enhanced	
$\dot{q}$ [kW/m ²]	4.6	13.9	4.5	16.9
U [kW/(m ² K)]	0.8	1.4	1.3	3.4
Volume flow, $\dot{V}$	0.5%	0.5%	0.5%	0.5%
Property data (c_{pw} , ρ_w)	0.2%	0.2%	0.2%	0.2%
Temperature differences, ΔT	3.6%	2.9%	4.7%	3.8%
Combined uncertainty	2.6%	2.3%	3.4%	2.7%
Estimated heat gain	5.5%	1.8%	5.6%	1.5%

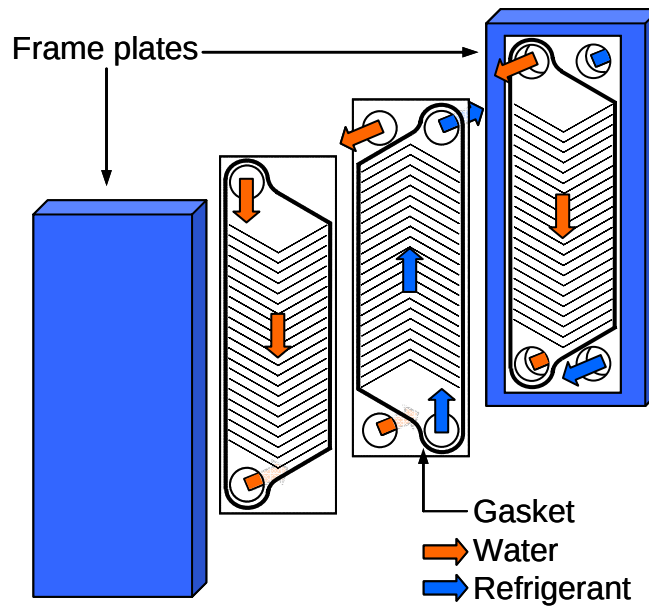


Figure 4: Heat exchanger with gasket seals.

gasket which seals the channels and directs the fluids into alternate channels. One advantage of this set up is that the plates can easily be replaced with plates that have been enhanced with the novel nano- and micro-porous structure. Four plates were used, thus forming two water channels and one refrigerant channel in the center. A few important geometrical characteristics of the stainless steel plates are found in table 2.

2.5 The enhancement structure

The novel nano- and micro-porous enhancement structure was recently shown to enhance pool boiling heat transfer in R134A with over 15 times compared to a plain, machined copper surface (Furberg 2006). As shown in Figure 5a and b, the novel surface structure was manufactured on the surface of two herringbone patterned heat exchanger plates in contact with the refrigerant. The surface structure was fabricated with an electrodeposition method where metallic copper nano-particles are dendritically connected into an ordered micro-porous structure using hydrogen bubble evolution as a dynamic masking template. The manufacturing technique has been shown to provide ample room to control both the micro- and the nano-sized features of the surface structure (Furberg et al. 2006), hence opening up the possibility to change the geometrical dimensions of the structure to suit different refrigerants. As seen in Figure 5c, the micro-sized pores have an average top layer pore

Table 2: Geometrical characteristics of heat exchanger plates

Fluid flow plate length, L , [mm]	357
Fluid flow plate length, W , [mm]	100
Nominal heat transfer area, A_{tot} , [m ²]	0.032
Macro-corrugation pattern	Herringbone
Surface enlargement factor, ϕ , (A_{tot}/A_{proj})	1.15
Corrugation angle, β , [degrees]	60
Corrugation pitch, P , [mm]	10.0
Corrugation pressing depth, B , [mm]	2.4
Plate thickness, δ , [mm]	0.5
Hydraulic diameter of water channel, d_{hw} , [mm]	4.2

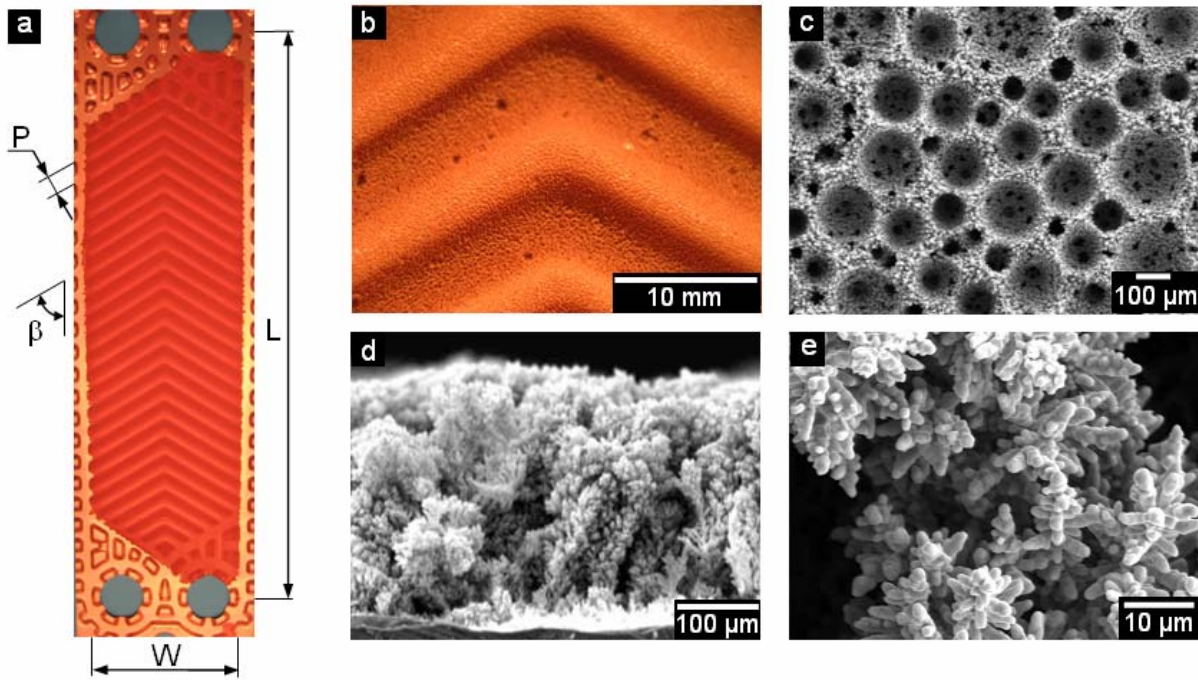


Figure 5: a. Heat exchanger plate with enhanced surface structure fabricated on the refrigerant side. **b.** Close-up view of enhanced heat exchanger plate. **c** and **e.** SEM picture of the enhancement structure. **d.** SEM side-view picture of the enhancement structure.

diameter of 105 μm with a pore density of approximately 75 pores/ mm^2 . The thickness of the surface layer, $x_{\text{structure}}$, measured approximately 250 μm , as seen in Figure 5e. The sub-micron copper particles were connected in a continuous dendritic pattern is shown in Figure 5e. The overall volume porosity, ϕ , of the surface structure was about 95% defined as:

$$\phi = \frac{m_{\text{structure}}}{\rho_{\text{cu}} \cdot x_{\text{structure}} \cdot A_{\text{tot}}} \quad (\text{Eq. 6})$$

where $m_{\text{structure}}$ is the mass of the fabricated surface structure on the plate and ρ_{cu} is the density of copper.

3 EXPERIMENTAL RESULTS AND ANALYSIS

The enhanced surface structure in the refrigerant channel significantly improved the performance of the heat exchanger. As seen in Figure 6, compared to the standard heat exchanger, the overall U-value of the enhanced heat exchanger is increased with 68-108% (at 4.5-14 kW/m^2). The theoretical limit, where the heat transfer coefficient on the refrigerant side approaches infinity, is also shown in Figure 6.

Based on the water side heat transfer constants, C_1 and n , in equation 2, the heat transfer coefficient on the refrigerant side, h_{ref} , was calculated, using equation 5. As seen in Figure 7, the enhanced refrigerant heat transfer coefficient was as high as 22 $\text{kW}/(\text{m}^2\text{K})$ at a heat flux of 17 kW/m^2 . This corresponds to an enhancement effect of up to 10 times on the refrigerant side. Considering the low refrigerant mass flux of 2.4 $\text{kg}/(\text{m}^2\text{s})$ at 5 kW/m^2 , the boiling mechanism is expected to be similar to that of pool boiling on a vertical surface. This is confirmed by previous pool boiling tests of the enhancement structure (Furberg et al. 2006) which are in fairly good agreement with presented results. The enhancement effect may be attributed to three main characteristics of the structure. Firstly, the pores form channels that penetrate the whole structure and are directed normally to the surface with increasing diameters, 105 μm large at the top layer. The high density and the shape of these channels

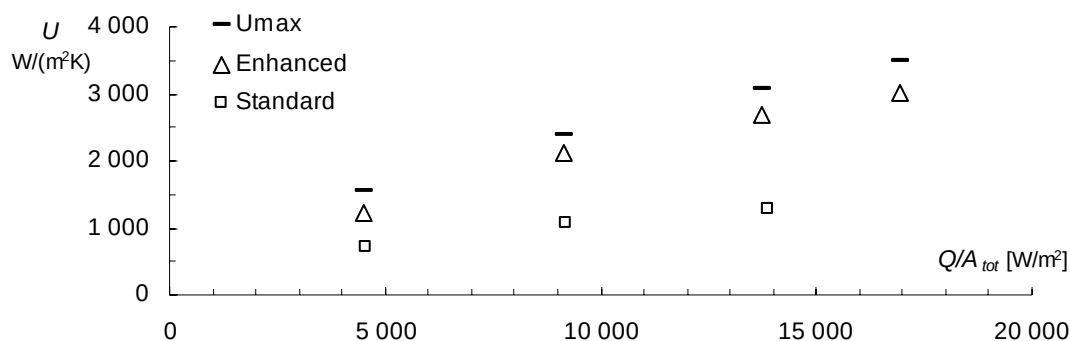


Figure 6: Overall U-value as a function of heat flux for standard and enhanced heat exchangers.

ensure that the vapor produced, during evaporation inside the structure, can quickly be released with low resistance from the dendritic structure. Secondly, the high porosity of the interconnected network effectively traps vapor and facilitates liquid transport within the structure and allow for communication between active nucleation sites. Thirdly, the dendritically shaped wall structure, with its large surface area, might facilitate formations of thin liquid films with high evaporation rate and long three-phase-lines with high heat transfer rates. The presence of lubrication oil in the system might affect these enhancing characteristics of the porous structure, depending of the type of oil and refrigerant used.

From Figure 7 it is also clear that in the standard heat exchanger, with the Reynolds numbers used in these tests ($Re=90$ at 4.6 kW/m^2), the thermal resistance is about the same on the water and the refrigerant side. In the enhanced heat exchanger the thermal resistance on the refrigerant side is up to 6 times smaller, which means that an increased Reynolds number on the water side would have a rather large positive effect on the overall U-value. The pressure drop in the refrigerant channel was not affected by the porous structure, but remained at about 30-40 mbar.

4 CONCLUSIONS

The heat transfer coefficient on the refrigerant side of a herringbone patterned plate heat exchanger evaporator was enhanced up to 10 times, in oil free R134a, by a novel nano- and micro-porous copper structure. As a result, the overall heat transfer coefficient was enhanced with over 100% at a heat flux of 14 kW/m^2 . This significant boiling enhancement is in agreement with previous pool boiling experiments and confirms that the enhancement structure may be used to enhance the performance of plate heat exchangers.

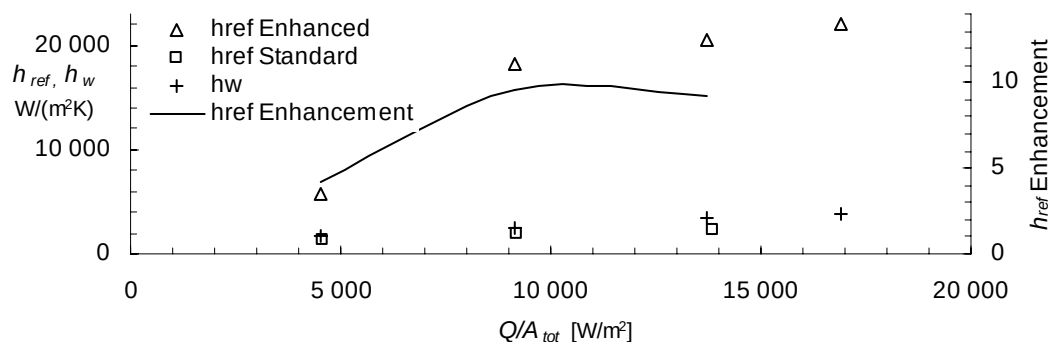


Figure 7: Heat transfer coefficient on the water side and refrigerant side as a function of heat flux.

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