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Enhanced Boiling Heat Transfer on Surfaces Covered with Microstructural Mesh Coatings

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Abstract

The paper deals with the issue of pool boiling heat transfer enhancement. It presents the experimental test results of distilled water and ethanol boiling at ambient pressure on heater surfaces covered with microstructural porous copper coatings. The coatings are made of 1 to 5 layers of fine metal meshes sintered together to form a porous structure. The obtained values of the heat transfer coefficient from these surfaces were compared to the smooth surface test results. A considerable enhancement of heat transfer due to the application of the analyzed additional coatings was recorded, especially in the area of low superheat values.

Keywords: boiling heat transfer, enhancement, porous coatings;

1. Introduction

Boiling heat transfer is a phase – change phenomenon. It is highly efficient in dissipating significant heat fluxes at low temperature differences. Its applications in engineering and technology are wide, and they cover areas of refrigeration, electronics cooling, heat pipes, among many other usages. In the design of power engineering and energy systems, there is currently a growing need to search for more effective methods of the removal of heat. One of the techniques to intensify the heat exchange in phase change processes (boiling, condensation) is the application of additional coatings on heat exchangers. In the case of boiling, such structures could be made in the form of metal wire meshes, fibrous layers, flame spraying coatings and others. They might considerably enhance boiling heat transfer. Compared to the smooth surface, without any coating, heat flux values for the same superheat could be even several times higher. Up to now there exists no efficient model of boiling heat transfer on structural coatings that could successfully predict the performance of a coated surface based on the physical and chemical parameters of the system. Similarly, data on the impact of structural parameters of the coatings found in literature is sometimes incongruent or even contradictory. Although it is generally stated that the application of microstructures enhances heat transfer, some scientific reports indicate the opposite effect for some coatings. Moreover, typically data available in literature focuses on isothermal surfaces, while non – isothermal heaters in the form of a fin are very rarely considered. In the present work an analysis of the thermal performance of a non – isothermal heater covered with up to five mesh layers will be done.

Tsay et al. [1] presented experimental results of water boiling on a horizontal smooth and rough surface covered with a single stainless-steel mesh layer. The authors reported that the application of a single mesh layer led to enhanced heat transfer compared to the smooth reference surface at superheats (which is a difference between the surface temperature and the saturation temperature) above 6 K. The visualization studies enabled to conclude that the number of bubbles grown on the heater covered with the coating was higher than on the smooth reference surface. Brausch and Kew [2] provided test results of water boiling at ambient pressure on vertical surfaces with stainless steel mesh wicks. The porous coatings had one, three and five layers of mesh. The application of a single layer of mesh led to increased heat fluxes in relation to the smooth surface at low superheats. While at higher superheats more heat was dissipated from the smooth surface. This phenomenon was explained by the authors by the coalescence of vapor bubbles within the porous structure, which creates local dry out and the creation of an insulating film. This film decreases the heat transfer coefficient. Additional mesh layers increased thermal resistance of the microstructure and reduced the value of the heat transfer coefficient. Gerlach and Joshi [3] investigated boiling of PF 5060 dielectric liquid at atmospheric pressure on a horizontal surface with single copper and bronze meshes attached to a copper block by soldering. The tests were conducted for the cases when the mesh was open to a liquid pool and when the structure was covered with a plate and bubbles exited sideways. The application of the porous coating as it was open to the pool

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generally led to an increase in heat flux in comparison with the smooth surface. However, when the meshes were covered with the plate, a lower heat transfer than in pool boiling was recorded. The authors state that the results published by others might be contradictory, possibly due to different clamping methods as well as variation in spacing and contact at the mesh to the heater surface. Franco et al. [4] conducted experimental tests of pool boiling of the dielectric refrigerant R141b. The heating surface was covered with copper, aluminum, brass and stainless-steel mesh structures. It was found that mesh layers of small aperture enhance heat transfer comparing to the smooth surface for small heat fluxes. According to the authors, heat transfer is influenced by the structure height (number of layers). On the one hand, they increase active nucleation sites density, but on the other the hydraulic resistance of vapour flow rises. However, according to the experimental investigations by Rannenberg and Beer [5] on R11 and R113 boiling on surfaces covered with two to nine layers of mesh, there is no apparent impact of the number of layers on heat transfer. Similar findings were presented by Smirnov and Afanasiev [6].

Wire mesh structures can be used as an internal coating in heat pipes. Wong and Kao [7] analyzed performance of heat pipes with a two – layered mesh wick covering under water evaporation/boiling conditions. It was found that the coarse mesh nucleate boiling was absent at low heat loads while the heat was dissipated by surface evaporation on the water–vapor menisci within the wires. The fine mesh was reported to provide more nucleation sites. Investigations of multi-layer copper meshes sintered to the heat pipe surface were also performed by Liou et al. [8] with water as the working fluid. A development regarding the use of meshes has been recently proposed by Pastuszko [9], who investigated structures made of microfins and meshes sintered to them for further boiling heat transfer enhancement. The works of the author also confirm significant possibilities of boiling heat transfer enhancement with the use of microstructural coatings (for example [10, 11]).

It needs to be noted that the performance of microstructure coated heaters is tested on the isothermal surfaces. However, the use of the thermovision technique enables to perform tests on the non-isothermal surfaces of fins (which are more often found in practical applications). This measuring technique is only used at Kielce University of Technology.

2. Material and method

The tests have been designed to analyze the boiling performance of heaters covered with one to five mesh layers made of pure copper. The meshes have been joined together with the base surface (in form of the fin) using the sintering technology in order to produce durable bonds. The sintering temperature was ca. 900 ^{o}C and it occurred in the reduction atmosphere to prevent oxidation of the samples. The research has been done with two boiling agents: distilled water and ethanol (99.8% purity) under ambient pressure. The coatings have been located on the fin's surface of 4 mm thickness, 12 mm height and 90 mm length. The porosity of microstructures ranged from 51% to 73%, while their height from 0.4 mm to 1.25 mm.

The experimental analysis has been performed on the stand presented in Fig. 1. The main element of the experimental set-up is a copper fin placed within the wall of the vessel. On one side it is in contact with the boiling liquid. This side of the fin was earlier covered with the porous microstructure. On the other side the fin is open to the atmosphere and observed with the long-wave (8 – 14 μm) thermovision camera. In order to ensure correct temperature recordings, the fin is painted black on this side. Before the actual measurements, the emissivity of the black paint was determined on a separate experimental stand and amounted to 0.97. The stand consisted of the electrically heated surface covered with the investigated paint and observed with the thermovision camera. The temperature of the heater was determined with the K-type thermocouple located within the surface. The measurements were done with increasing heat flux at different temperatures (the temperature ranges reflected those encountered during the actual measurements on the fin). The discussion of the measurement errors during infrared testing can be found in the work by Orzechowski and Orman [12]. Considering the accuracy of the temperature readings, the errors of determination of the heat transfer coefficient using the described technique have been assessed to amount to 8%.

Heat is supplied to the base of the fin with an electric heater. Consequently, a temperature gradient along the element is created. The thermovision camera is applied to record the temperature along the fin. The measured temperature distribution is then used to determine local values of the heat transfer coefficient. An auxiliary heater (a resistance wire) is installed at the bottom of the vessel to maintain stable pool boiling conditions. The boiling process takes place on the inner side of the fin inside the vessel. This part is covered with the microstructure. A detailed description of the apparatus and the testing method used in the present study has been given in [13].

The obtained temperature distribution along the fin is numerically differentiated to determine local values of the heat transfer coefficient according to the method presented by Orzechowski [13]. Based on the experimental reports of pool boiling heat transfer on microstructure coated surfaces (e.g. [5, 15], it has been assumed that the heat transfer coefficient (α) between the fin's surface and the liquid in the vessel depends exponentially on superheat (Θ) according to the following equation:

$$\alpha = a\,\theta^{n} \tag{1}$$

where a, n are constants. Their experimental determination leads to a formula for the boiling curve as a dependence of the heat transfer coefficient or heat flux. The values of n have been found to differ depending on the kind of the microstructural coating and the range of superheat under investigation (e.g. [5. 15]). This concept of the exponential dependence is based on experimental observations of pool boiling heat transfer phenomenon.

Having considered (1) and the simplifying assumptions for one - dimensional heat conduction, a formula for temperature distribution along the fin is produced:

$$\frac{d^2\theta}{dx^2} = m^2\,\theta^{n+1} \tag{2}$$

This issue was analyzed by Ünal in [16].

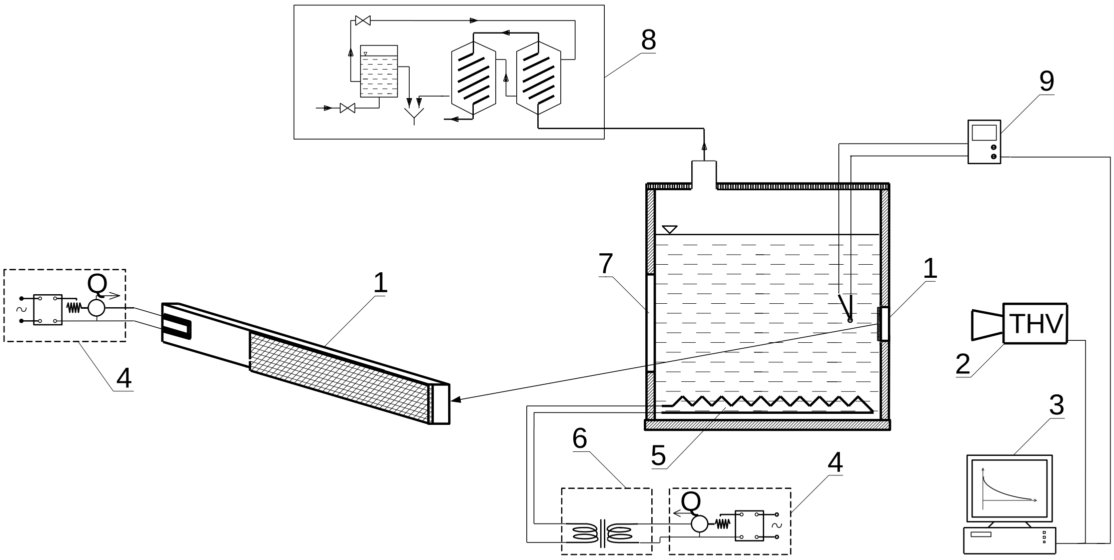

Figure 1. Schematic of the experimental set - up:

1 – copper fin with the porous microstructure, 2 – thermovision camera, 3 – data acquisition unit, 4 – autotransformer, 5 – auxiliary heater, 6 – electrical current separation unit, 7 – observation window, 8 – cooling and condensate retrieval unit, 9 – temperature measuring device [14]

After differentiation of equation (2) the superheat gradient in logarithmic coordinates is calculated:

$$\ln\left(\frac{d\theta}{dx}\right)^2 = \ln\left(\frac{2m^2}{n+2}\right) + (n+2)\ln\theta \tag{3}$$

where $n \neq 2$ and m^2 is defined in the following form:

$$m^2 = \frac{aP}{\lambda F} \tag{4}$$

In the above equation P and F are the circumference and surface area of the fin, respectively. λ is the thermal conductivity of the material of the fin. For fins of considerable length, as in the analyzed case, no heat transfer at the tip can be assumed. In such a case an integration constant in (3) was 0.

The temperature distribution along the fin obtained with the thermovision camera and its numerical differentiation enabled to determine the constants: a and n from (3). It is done by applying the linear fitting to research data. Consequently, with the use of equation (1) it is possible to draw boiling curves which are a function of the heat transfer coefficient or heat flux vs. wall superheat. In order to more precisely determine the heat transfer coefficients for ethanol the measurement results have been analyzed assuming the non - linear dependence for the heat transfer coefficient (as proposed by Orzechowski [17]. The applied data acquisition technique based on thermovision measurements for the analyses of non – isothermal surfaces in the boiling mode is used only at Kielce University of Technology [18].

3. Results and discussion

The tests have been conducted for distilled water and ethanol as boiling liquids at atmospheric pressure. Before the measurements the surface of the fin was covered with the special black paint of high emissivity. This side of the heater was observed with the thermovision camera and temperature distributions were recorded. The inside part of the fin was located inside the vessel were boiling took place. The recorded temperature gradients along the x axis of the horizontally located fin made it possible to provide data on thermal performance of the heaters with different porous coatings. Fig. 2a and 2b present the test results for two fluids and different numbers of layers as the dependence of the superheat gradient vs. superheat.

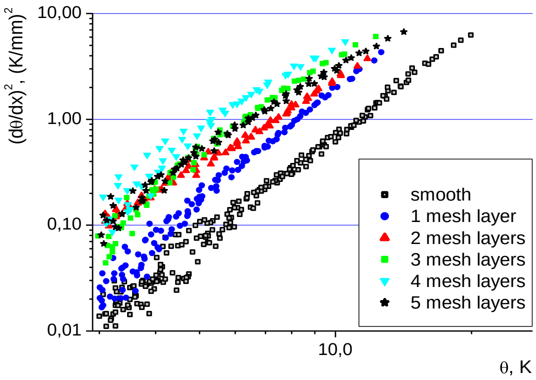

Figure 2a. Superheat gradient vs. wall superheat for distilled water: 1 – 5 mesh layers and the smooth surface test results.

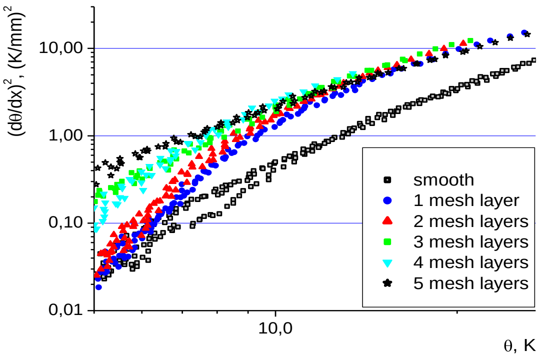

Figure 2b. Superheat gradient vs. wall superheat for ethanol: 1 – 5 mesh layers and the smooth surface test results.

The analysis of the above figures indicate that the performance of the meshed surfaces improves as the number of meshes increases. The best performance for distilled water has been recorded for the four mesh layers, while in the case of ethanol the most advantageous has been the coating that consisted of five meshes up to the superheat value of ca. 8 K, and four and three meshes for the higher superheats.

Based on the above test results, it is possible to draw boiling curves, which more precisely show the heat transfer performance of each kind of surface. The results for wall superheat values in the range of 5 to 12 K have been given in Fig. 3a and 3b for distilled water and ethanol, respectively.

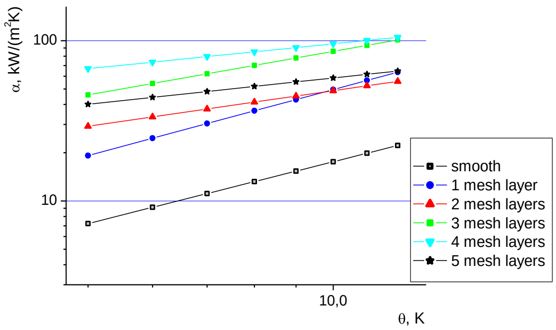

Figure 3a. Heat transfer coefficient for distilled water for different mesh layers and wall superheat 5 – 12 K.

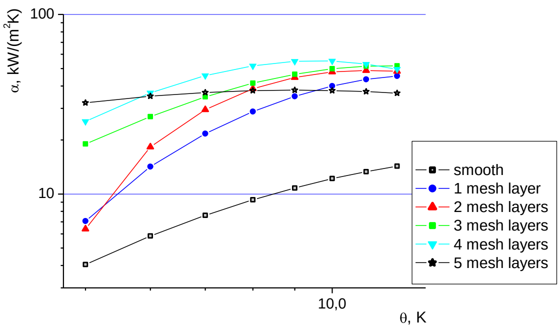

Figure 3b. Heat transfer coefficient for ethanol for different mesh layers and wall superheat 5 – 12 K.

The presented results indicate that the application of mesh layers enhances boiling heat transfer in comparison with the smooth surface. The heat transfer coefficient for the same wall superheat for the surface with the additional layers has been much higher than for the reference surface without the meshes. In can be noted that, generally, the heat transfer performance is improved as the number of mesh layers increases up to four – which seems to be the optimal number – especially for distilled water. Adding the fifth coating results in lowering the value of the heat transfer coefficient. It can be explained by higher hydraulic resistance of the flow of the created vapour out of the microstructure and of fresh liquid flow inside the microstructure for vaporization, so worse conditions of mass transfer within the coating.

In the case of ethanol at low superheats, the 5-layer coating provided the highest enhancement. It might be related to increased density of nucleation site for this working fluid already active at low superheats. At higher wall superheats vapor could be permanently present within the structure which might result in smaller heat fluxes dissipated from the fin. It needs to be noted that the two fluids considered in this paper have different surface wetting properties (surface tension for water is a few times higher), which might be a factor considerably influencing boiling heat transfer in microstructural coatings.

The impact of the number of mesh layers for two values of superheat (5 K and 12 K) has been presented in Fig. 4 in order to precisely determine the enhancement produced by the application of the mesh microstructures. The ratio of the heat transfer coefficient for the microstructure coated surface (α_m) and the smooth surface without any coating (α_s) is considered as a dependence of the number of mesh layers. The results have been presented for both distilled water and ethanol.

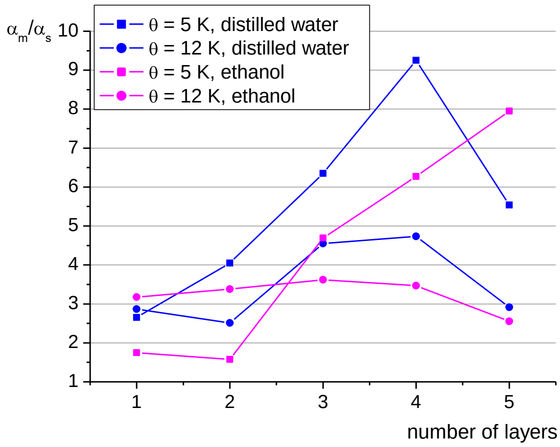

Figure 4. The ratio of heat transfer coefficient for the microstructure coated surface (α_m) and the smooth surface (α_s).

As can be seen in Fig. 4, the value of the heat transfer coefficient can be over nine times higher in comparison with the smooth surface if the additional mesh layers are added onto the heat exchanging surface for water boiling. In the case of ethanol about eight times higher values have been recorded. Generally, more significant enhancement has been observed for low superheats of 5 K than for 12 K. This phenomenon might be explained by a higher number of active nucleation sites for microstructures, which are activated at low superheats. In the case of the smooth surface the number of nucleation sites increases with temperature, while this growth is not so rapid for the structural coatings. That could be the reason why the enhancement for the higher superheat of 12 K might not be so significant as at low temperature differences.

The investigation of pool boiling on the non-isothermal surface in the presented experimental set-up has its limitations. Thus, in order to more precisely consider the impact of the number of mesh layers on the thermal performance of the samples, the boiling curves have been determined on the isothermal experimental facility, described in detail by the author in [10]. The samples have been located horizontally, which as opposed to the vertical location in the case of the non-isothermal testing set – up. The boiling performance of the samples has been presented in the form of heat flux dependence on wall superheat in Figs. 5a and b for distilled water and ethanol, respectively. The mesh layers had the porosity in the range of 64 – 70%.

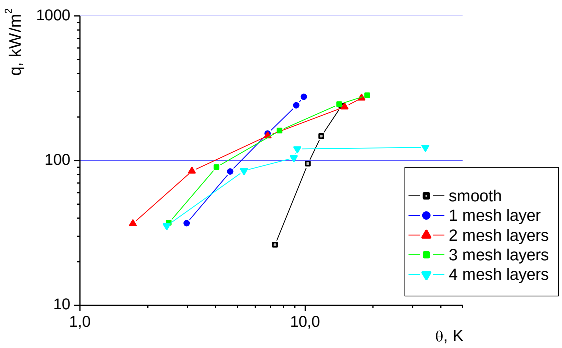

Figure 5a. Heat flux for distilled water for different mesh layers.

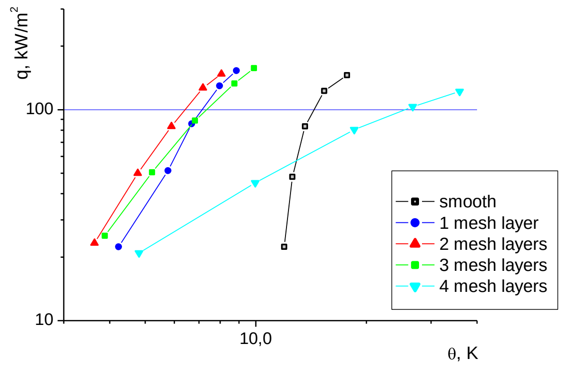

Figure 5b. Heat flux for ethanol for different mesh layers.

The analysis of the above figures confirms that there is an optimal number of meshes, which provides the best performance for the certain boiling liquid and the considered superheat range. The porous layers with the highest number of meshes (four in figures 5a and b) might produce the unfavorable effect of blocking the flow of liquid and vapor, thus, limiting the transferred heat fluxes. In the case of water boiling, the application of the 4-mesh layer led to a rapid transition to film boiling, while in the case of ethanol, the microstructure transferred heat in the nucleate boiling mode, but its performance was significantly reduced (even worse than the smooth surface without any coating). This might be related to the fact that water vapor bubbles are larger, and the highly packed structure produced the blocking effect, while in the case of ethanol the bubbles are smaller and the two-phase flow was still possible, although hampered.

4. Conclusions

The application of microstructural coatings enhances boiling heat transfer for the analyzed mesh layers located on the analyzed fin. The heat transfer coefficient can be several times higher if such additional coatings are used. The four-layer structure has been most efficient for water boiling, producing over nine times higher value of the heat transfer coefficient in comparison to the smooth surface. In the case of a different kind of mesh and the isothermal heating, four mesh layers proved to be least effective. This phenomenon proves that the thermal performance of phase change heat exchangers is significantly affected by the geometrical parameters of the coatings as well as the properties of the working fluid. Thus, the design of such heat exchangers for industrial applications should consider several parameters.

The application of an increasing number of mesh layers generally leads to elevated heat transfer coefficients. However, a considerable number of meshes could result in the opposite effect, namely the reduction of the dissipated heat flux. It seems to be related to difficulties in mass transport as vapor needs to flow out of the structure and liquid into it. Some vapor could permanently remain in the voids of the multi-layered coatings, especially at high superheats, thus, reducing the heat transfer coefficient in relation to surfaces with a lower number of meshes.

More tests in this field could help to produce a reliable model of boiling heat transfer on microstructure covered surfaces and provide information for the design of efficient phase change heat exchangers.

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