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Boiling on fins with wire screen of variable effective conductivity

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Abstract. The high scale of integration of modern equipment used for medical, military and other purposes puts heavy demands as regards the removal of great heat fluxes. This can be achieved only in exchangers that apply the phase change phenomena. Among many methods to improve boiling heat transfer, the wire mesh covering demonstrates some advantages due to the possibilities of designing the desired microstructure parameters, availability on the market, and low cost. The wire mesh microstucture with specified geometrical parameters produces anisotropy in conductivity. The different arrangement of the mesh layers relative to the direction of the heat flux is a cause of the change of temperature distribution within the layer. The consequence is a respective change in the discharge conditions of the gas phase and liquid feed. The experiments were conducted on fins covered with a single layer of copper mesh with lumen of 38 % and boiling FC-72 at ambient pressure. Compared with the smooth surface, the wire mesh structures yield an increase in the heat transfer rate at boiling. It is also shown that nucleate boiling is initiated at lower wall superheat. Formulas for longitudinal and perpendicular thermal conductivity are given for different mesh structure arrangements.

1 Introduction

These Modern heat exchangers used in electronics, cooling and chemical industry need to have compact structure and remove large amounts of heat. The machines generate so much heat that simple surface enhancement with ribs stopped to provide a sufficient solution long ago. It was found that phase change processes offer new possibilities as regards the construction of high-efficiency heat exchangers. An additional, yet very desirable feature they have is the fact that they maintain almost constant temperature. That prevents overheating that is unwanted, and also allows setting the optimum technological parameters. Therefore, exchangers of that type have gained increasing popularity [1-4].

In the mid 1960s, in studies by Haley and Westwoter [5], it was shown that boiling heat transfer coefficient is related to the fin temperature along its height, and can be described by the boiling curve. However, when fins are packed in the exchanger, different behaviour of those can be observed, which depends on packing density and fin geometry [6].

To dissipate increasingly high heat fluxes, higher wall superheats are required, which at too large values at the base, leads to the quick development of film boiling. That poses a threat of the surface burn-out [7]. One of the ways to prevent such phenomena is to deposit an additional insulation layer with low heat conductivity, or to make fins that have variable cross-section along the height [8]. Both options offer a limited possibility of increasing the amount of dissipated heat. It becomes necessary to deposit additional layers that enhance heat transfer surface. Here, the following techniques can be distinguished:

- various structures obtained by pressing or bending ribs [9], [10];
- modifying of the heating surface by increasing its roughness mechanically or technologically [11], [12];
- imposing additional laminar-composite layers with preset parameters [13], [14].

The application of many proposed solutions is limited because of technology requirements and the costs. The other factor that has to be taken into account is the repeatability due to the heat transfer conditions. Mesh structures constitute a specific group. Their main advantage is that they are relatively readily available on the market, and can be made from different materials, and have different geometrical parameters to suit the purposes of heat transfer. This surface enhancement can be provided with desirable geometry to meet the anticipated operating conditions. Such structures allow constructing layers with selected parameters, such as pore diameters and the coating thickness. Wire mesh packing increases efficiency of heat exchangers, as well as the heat flux and heat transfer coefficient, while simultaneously lowering the temperature of the heating surface [15]. Heat dissipation from such systems is a result of complex processes that occur inside the layer. The construction factors that mostly affect the process include porosity, mesh size and the wire thickness, which was discussed in, e.g. [16], [17] and many other studies.

The design of mesh layers that have desirable parameters is relatively simple. That involves the deposition of successive mesh layers onto one another.

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Depending on the needs, those are pressed against the base [19], soldered or sintered. The conductivity of such structures depends on their arrangement, and can vary in value in different directions [18]. That also refers to spatially extended structures [19]. The magnitude of the thermal conductivity coefficient is also affected by the technology used to deposit layers onto the base. In the process, contacts of different quality, between the structure skeleton elements are formed. They are decisive for the internal transport of the heat. In study [20], the example of square-shaped and diamond-shaped wire screens was used to present the methodology for calculating volumetric porosity, specific surface area, and in-plane thermal conductivity. The dependences given in the study concern the wire mesh screens that are cold compressed. In study [21], the existing models of computing the effective conductivity of mesh screens were classified into two groups. The first category includes those dependences, on the basis of which relatively high values are obtained, which results from strong contact between wires. The other group comprises those formulas which produced low effective conductivity due to weak contacts. It was demonstrated that, under the assumption of isotropy, divergence in calculations amounts to two orders of magnitude. A new theoretical model was proposed to determine effective thermal conductivity taking into account contact conditions between wires. The model, however, does not account for the lengths of wire-to-wire or wire-to-base contacts that formed in the sintering process. The contacts depend on the process parameters, whereas their surface characteristics is different from that resulting exclusively from the layer geometry [22], [23].

Although the formulas for calculating mesh conductivity differ much in various studies, they some of them indicate variable conductivity in the direction longitudinal and perpendicular to the surface. If mesh layers are deposited on the fin, a part of heat is also transported parallel to the surface. Consequently, when calculating heat transfer in such systems, it is not possible to disregard the anisotropic properties of mesh screens. The aim of this study is to analyze the effect of the geometry of the mesh structures on the heat transfer process.

2 Experimental facility

Investigations into heat transfer on the fin with mesh covering were conducted using the facility shown, in a form of diagram, in Fig. 1. The facility was installed in a large volume vessel in such a way so that one side with the mesh cover could be in contact with the boiling liquid, whereas the other side is exposed to ambient atmosphere. Such fixing of the sample is necessary because of the adopted measurement procedure that is based on the analysis of the thermal field measured with the thermal imaging camera.

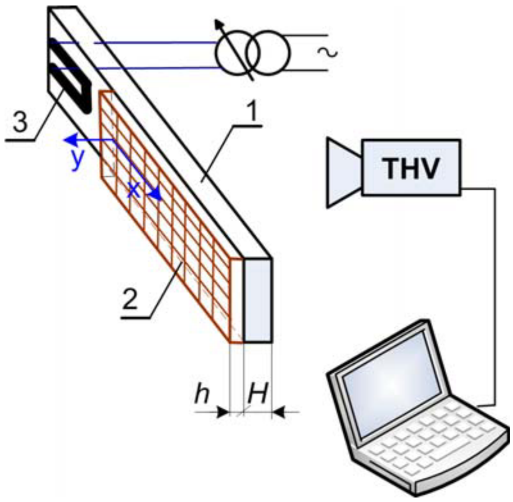

Fig. 1. Diagram of the test facility: 1 - examined element, 2 - mesh structure, 3 - main heater.

A majority of liquids used in boiling exchangers is almost entirely opaque to infrared radiation of thermal imaging cameras. Consequently, it is not possible to directly observe the surface on which heat transfer occurs. Therefore, the only possible measurement to be taken is the one on the side in contact with atmosphere. In this study, the investigations concerned a long and thin copper fin that had the thickness of H, selected in such a way so that, under measurement conditions, the number $Bi<0.1$. In such conditions, it is possible to assume one-dimensional model of heat transfer in the fin [24]. Otherwise, a substantial temperature drop can occur in the mesh layer [25], depending on its thickness h (see Fig.1). The phenomenon is also dependent on the type of contact with the base and the geometry of the arrangement.

3 Thermal conductivity of the mesh structure - physical model

The market offers various meshes, which depending on the needs, are made from different materials and have different weave. In heat transfer applications, plain-weave copper meshes are most frequently used. They are deposited on the surface, one onto the other, and compressed or sintered, depending on the technology. Due to the process repeatability, it is assumed that meshes are symmetrically arranged relative to heat flow direction. The geometric arrangement of either squares or diamonds is found, as shown in Fig. 2. A repeatable unit cell in the orthogonal projection is also marked in the figure.

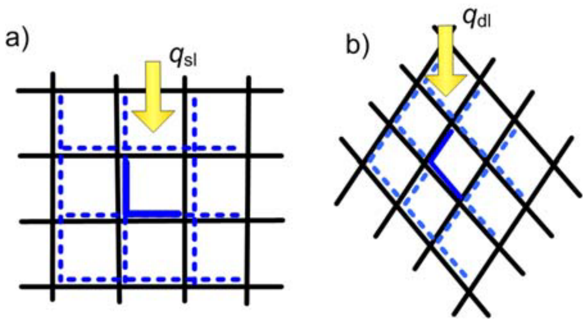

Fig. 2. Structure arrangement relative to the longitudinally flowing heat flux: a) square-shaped screen relative to q_{sl}, b) diamond-shaped screen relative to q_{dl}.

The symmetric structure of diamond-shaped wire screen with a constant opening width w and wire diameter d can be characterised by the weave angle β (see Fig. 3). Dimensions a and b, which determine the size of the repeatable unit cell of the mesh, result from the geometric quantities given above.

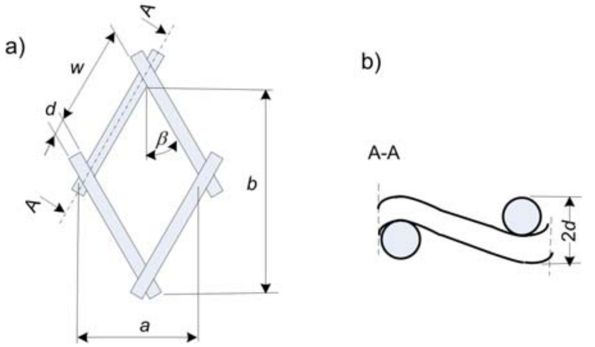

Fig. 3. Geometry of a unit cell: plain view and A-A section.

For diamond-shaped plain-weave wire mesh shown in Figs. 2b and 3, respectively, the values of the thermal conductivity coefficients can be computed from the following dependences:

- a ratio of the mean value of the thermal conductivity coefficient in the perpendicular direction λ_{dp} to the mesh material conductivity coefficient λ_n:

$$\frac{\lambda_{dp}}{\lambda_n} = \frac{\delta(\delta + d)}{(w+d)^2 \sin 2\beta} \tag{1}$$

- in the longitudinal direction:

$$\frac{\lambda_{dl}}{\lambda_n} = \frac{\delta^2}{d + 2w\sin\beta} \cdot \left(\frac{1}{d + 2w\cos\beta} + \frac{d + 2w\cos\beta}{2d(w+d)} \right) \tag{2}$$

where $\delta \approx 0.9d$.

For square-shaped wire mesh, Fig 2a, the formula for the mean value of the thermal conductivity coefficient in the longitudinal direction λ_{sl} is as follows:

$$\frac{\lambda_{sl}}{\lambda_n} = \frac{\delta^2}{d(d+w)} \tag{3}$$

The mean value of the thermal conductivity coefficient of the structure in the perpendicular direction λ_{sv} is calculated in the same way in both cases, namely from formula (1), which for square-shaped mesh takes on the following form:

$$\frac{\lambda_{sv}}{\lambda_n} = \frac{\delta(\delta + d)}{(w+d)^2} \tag{4}$$

All the dependences above were derived while disregarding the thermal conductivity of vapour that fills the structure, and of thin liquid film that feeds structure. Both quantities are negligibly small compared with the mesh material conductivity.

4 Results and discussion

Figure 4 shows an exemplary result of investigations into boiling heat transfer on a long fin, with Fluorinert FC-72 as a working medium.

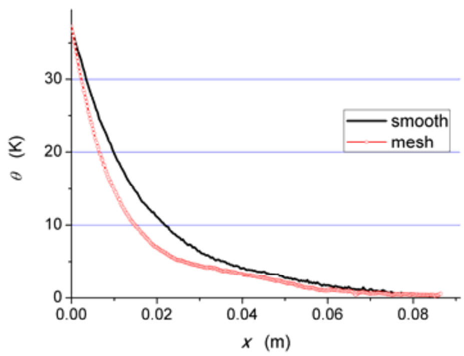

Fig. 4. Wall superheat distribution for smooth and single-layered copper specimens.

The local boiling curve for the copper fin, 3 mm in thickness, was determined in accordance with the procedure described in study [24]. The technique utilises fin vertical surfaces with wire mesh structures. Mesh with lumen of 38 %, made of 99.9% copper wire, 0.2 mm in diameter, was applied. The experiment was conducted for refrigerant boiling at ambient pressure. The measurement results yield the temperature distribution along the fin length. Each specimen was measured at least three times, at different values of electric power supplied to the main heater. In each case, the other parameters remained unchanged and constant. Local values of heat flux were determined on the basis of the results of measurements carried out on specimens having a smooth surface and with a single layer of superimposed mesh structure. The values are correspondingly shown in Fig. 5.

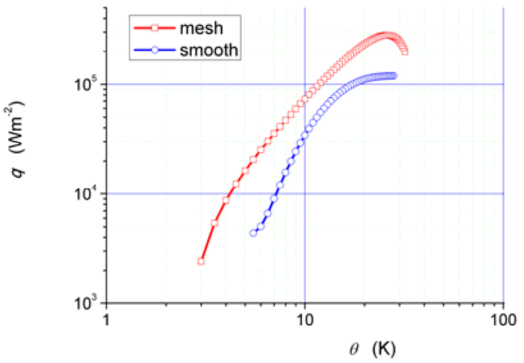

Fig. 5. Boiling curve defined locally for smooth and single-layered copper specimens.

Compared with the smooth surface, the wire mesh structure produces an increase in the heat transfer rate at boiling. From Fig. 5, it is clear that nucleate boiling is initiated at lower superheat. The results of measurements show that the imposition of mesh onto the surface reduces, by ~2÷3 K, the initiating point of nucleate boiling as compared with the smooth one.

Mesh structures are characterised by conductivity anisotropy, which was discussed in Section 3. Figure 6 shows a change in the mean value of the thermal conductivity coefficient in the longitudinal direction (the value decreases) and the perpendicular one (the graph line is symmetrical with respect to the angle β=45^o) for diamond-shaped mesh and different weave angles. With a decrease in β, the structure becomes more close-packed, and the change in the dimensions of the balanced unit cell leads to an increase in the heat transfer coefficient.

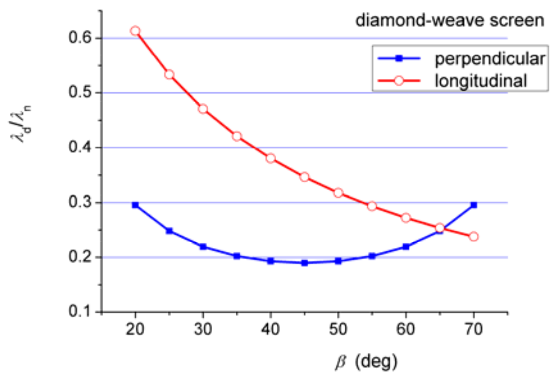

Fig. 6. Conductivity coefficient in the longitudinal and perpendicular directions in diamond-weave screens.

Substantial changes in the amount of heat dissipated by the surface with a single sintered layer compared with the smooth surface results from complex process taking place inside the structure. The conductivity model is one of many computational tools available for boiling heat transfer on enhance surfaces. For this model, it is necessary to precisely give the effective conductivity, which is very difficult. The literature provides a lot of dependences, however, the results of calculations obtained with those formulas often differ by as much as a few orders of magnitude [21].

For the structure arranged as shown in Fig. 2a, thermal conductivity in the longitudinal direction, calculated acc. formula (3) is 72.2 W/mK. The value calculated acc. (4), in the direction perpendicular to the fin surface is 102.6 W/mK. Figure 7 presents the distribution of superheat inside a single layer of copper mesh, which was obtained on the basis of the values quoted above. Computations were conducted numerically acc. to the procedure discussed in [25]. For the sake of comparison, superheat distributions are shown for the same distances from the fin base, under the assumption of isotropy, and the effective conductivity of the mesh equal to 10.9 W/mK.

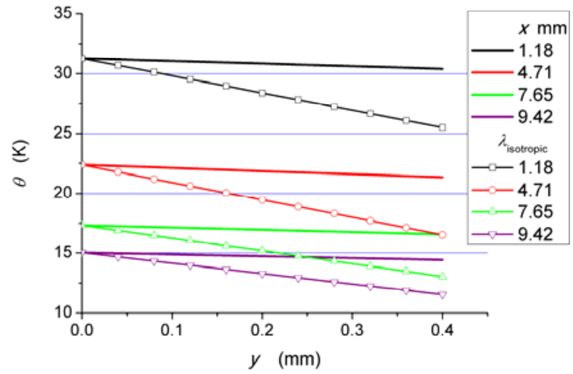

Fig. 7. Superheat distribution in the mesh layer in selected cross-sections.

5 Conclusions

The result of the investigations into FC-72 boiling at ambient pressure comes in the form of the boiling curve, i.e. heat flux change with wall superheat θ, as presented in Fig. 5. The experiments were conducted on fins covered with a single layer of copper mesh with lumen of 38 %. Compared with the smooth surface, the wire mesh structures yield an increase in the heat transfer rate at boiling. It is also shown that nucleate boiling is initiated at lower wall superheat.

Mesh structures are widely available, which makes them an option of choice when the surfaces of boiling heat exchangers are enhanced. Unlike coverings of other types, e.g. metal – porous ones, they are characterised by different thermal conductivity. That feature must be considered on individual basis, while taking into account the geometry of the finned surfaces. If the calculations are made for incomplete data on the structure properties, the results produced will considerably differ from the real ones. As shown in Fig. 7, neglecting the mesh anisotropic properties results in up to 5 K difference in the fin external surface superheat, which directly affects the amount of dissipated heat.

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