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# TWO-PHASE HEAT TRANSFER IN LIQUID POOL AND ANNULAR MINI-CHANNEL WITH POROUS CYLINDRICAL WALL

# DIVFĀZU SILTUMAPMAIŅA ŠĶIDRUMA TILPUMĀ UN NEREGULĀRAS FORMAS MIKROKANĀLOS AR PORAINĀM CILINDRISKĀM SIENIŅĀM

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## Introduction

The phenomenona of phase transitions is widely used in the heat exchange equipment of different designation. Heat exchange with evaporation and boiling realizes in power plants, refrigeration engineering, chemical equipment, units of electronics. In particular, this process finds use in the vaporizers of adsorptive heat pumps. These heat engines are very promising due to the topical character of the energy-saving and ecology problems, therefore in recent years the work on creation of non-compression heat pumps are conducted intensively. Such devices can use the surrounding (water basins, ground) and outflow low-grade heat, they are characterized by the low energy consumption, noiselessness of work, ecological safety and other merits. The creation of the nonelectric heat engines, which work without harmful ejections, is the important direction of the power engineering development; that's why the considerable attention is attended to the problems of study and design of adsorptive heat pumps. It should be noted that the adsorptive heat pump is the unique device, which makes it possible to use alternative energy sources for the heat and cold production without consumption of electric power.

The problems, connected with development of adsorptive heat pumps, are studied in the Republic of Belarus, by Russia Federation, Japan, Poland and other countries with a high level of science of and technology development [1-4].

Evaporative heat exchangers are used also in the industrial production of liquefied natural gas, whose cooling is provided for boiling of hydrocarbons or their mixtures in the tube space of heat exchanger-vaporizers. At regasification the process of evaporation is realized as well. Liquefaction and regasification of natural gas require the high power inputs and significant expenditures for the operation of installations, in this connection is the search for the ways of the increasing of the heat exchange equipment effectiveness is the vital problem.

The development of microelectronics and optoelectronics is characterized by increase of the devices effectiveness in combination with the miniaturization (high density of layout). Under the conditions a limited space of modular constructions the use of active cooling is often impossible, while the heat rejection arrangements in the passive cooling systems can be integral in the electronic assembly [5, 6]. Therefore the problems of the passive methods creation are extremely urgent. For the heat removal under the confined conditions the heat pipes are successfully adapted as the alternative to liquid cooling. In the Luikov Heat and Mass Transfer Institute of the National Academy of Sciences of Belarus (Minsk) the experimental study of heat exchange special features in thin capillary coatings of the heat-releasing objects surfaces are conducted. The general goal of this investigation is to increase the heat transfer efficiency of the evaporative heat exchange apparatuses.

**Experiments and results**

Experiments are carried out on the experimental set-up, its design is described in [7, 8] the main parts of which are the test vessel, insulated chamber with temperature controlled liquid circuit, cooling machine (refrigerator), thermostats, condenser liquid loop, temperature control system, vacuum pump, liquid feed system. The test samples represent the single copper tubes (outer diameter 20 mm, length 100 mm) with copper sintered porous coating. They are placed horizontally in a test vessel filled by the working fluid. Experiments showed that the sample with particle diameters 63-100 μm (mean particle diameter 82 μm), mean pore hydraulic diameter 24.5 μm, porosity ~50% thickness 0.3-0.8 mm has the best heat engineering characteristics. A heat flow to the heat releasing component (tube) was supplied by the cartridge electrical heater. There is the condenser placed in the upper part of the vessel. To prevent the heat losses between the test vessel and ambient medium the temperature inside a thermally-controlled chamber was maintained equal to the vessel temperature. All experimental data were obtained for the steady state working conditions. In order to verify the reliability of the experimental data, a series of experiments with boiling on plain stainless steel and copper tubes were carried out.

Two sets of experiment were realized to compare the hydrodynamic and the heat transfer of the tested sample placed in the liquid pool (macro-scale heat transfer) and inside the system of mini-/micro-channels. To visualize the hydrodynamic of the process in micro-scale system the transparent glass cylinders with different thickness of the annular mini channel (from 0.1 mm to 1.8 mm) were placed over the heat releasing copper tube, Figure 1. Propane (R290) was chosen as the working fluid due to its good thermodynamic properties, low cost, availability, compatibility with constructional materials, environmental friendship. The propane saturation temperature was $T_s=20\,°C$ ($p_s=8.4$ bar). Temperature measurement was carried out by copper-constantan thermocouples, sensor signals were sent to the measuring complex and computer.

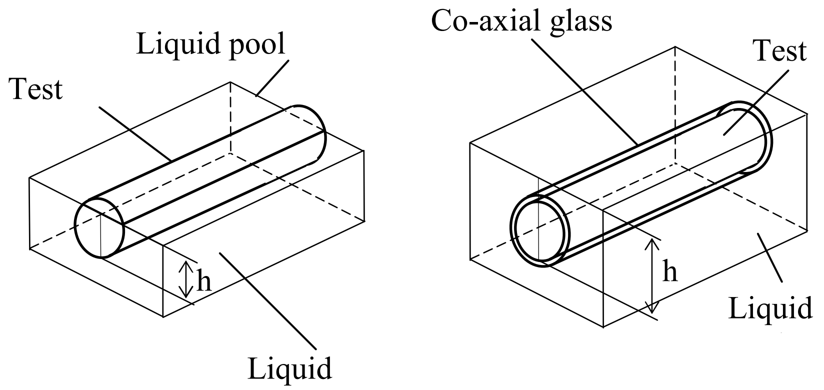


Figure 1. The sample placed into the liquid pool with the height h (left) and the sample
with co-axial transparent glass cylinder over it (right)

Horizontal porous tube disposed inside the transparent glass cylinder with annular mini-channel has all particularities of micro-scale and mini-scale effects of the heat transfer. The size of mini channels need to be close to the capillary constant κ

$$\kappa = \left[2\sigma / \left(\rho_l - \rho_v\right) g\right]^{1/2}. \tag{1}$$

In this case the influence of capillary forces on the process is substantial.

**Heat transfer in the liquid pool**

The horizontal heat releasing tube with porous coating was immersed in the liquid pool. The heat transfer of this tube and liquid was studied at different height h of liquid interface – from the 70 mm above the sample lower generatrix down to h=5 mm (Figure 1). The heat transfer coefficient was calculated as

$$\alpha = q/\Delta T = q/\left(T_w - T_l\right). \tag{2}$$

During the experiments it was concluded that the main contribution to the heat transfer phenomena in the liquid pool brings the evaporation of the liquid inside the metal sintered powder wick, Figure 2. The heat transfer efficiency at vaporization in the sintered powder wick with open pores is up 6-8 times as much as a liquid boiling heat transfer on smooth tube [9], for the heat flux range up to $q=100$ kW/m$^2$. To analyze the peculiarities of the heat transfer in sintered powder wick the experiments were carried out at various levels of liquid pool. Following the definition of the efficient boiling surface, to keep the temperature difference ($\Delta T=T_w-T_{l\infty}$) constant, all the cavities (macro-pores) of the porous coating must have the same radii; it means mono-cavity-pattern [10]. The thermodynamic wall superheat corresponding to this cavity pattern would be the same for the whole surface. All the macro-pores are expected to become activated simultaneously at the same superheat. As the vapor production mainly occurs at the three phase line (TPL), the wall superheat would remain unchanged whereas the heat flux would rise.

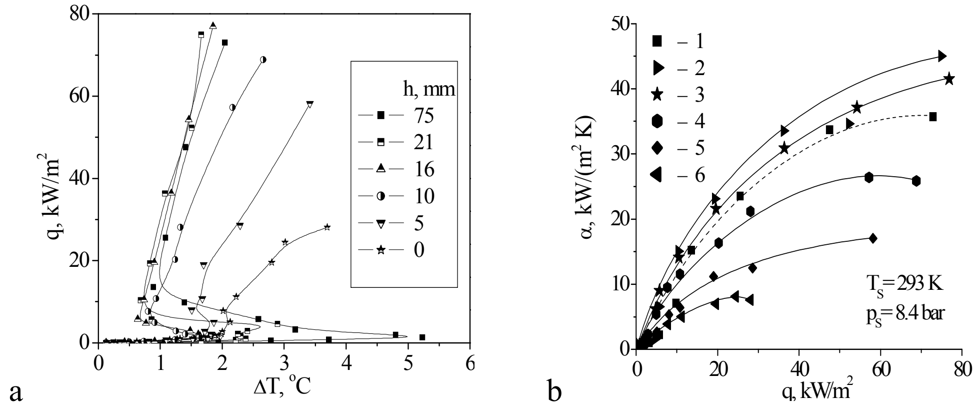


Figure 2. Heat flux versus the temperature drop ΔT (a) and heat transfer coefficient (b) of the sintered powder porous tube versus heat flux for different liquid interface height h: 1-6 – h=70, 20, 15, 10, 5, 0 mm respectively

It means a rise in the heat flux would be possible without increasing the wall temperature, Figure 2. For such case a bubble formation is possible at a very short time. A vapor rest in a pore is able to grow immediately after a bubble break-off from a heating surface. A distribution of pores on a heating surface is such that one pore is near the neighboring pores. This pore distribution would prevent a heat transfer from a wall to a liquid and a heat would be consumed almost completely by vapor generation in pores. So a copper sintered powder wick was performed following the above mentioned recommendations.

In the case of partly flooded sample a heat transfer coefficient was typical for a heat pipe wick evaporation zone and was higher than one on the same sample completely immersed in a liquid pool. A liquid column was a reason of an additional hydraulic and thermal resistance to vapor exhaust. The experimental data, obtained on flooded (h=70 and 20 mm) or partially flooded (h=15, 10, 5 mm) sample show us this difference (Figure 2). For low heat fluxes an influence of a liquid interface height h above a porous coating is important at h less than 2 mm. Decrease of h by a quarter of cylinder diameter promotes to increase of the mean heat transfer coefficient at low and moderate heat fluxes $q<100$ kW/m$^2$. Measuring of temperature heads between a heat releasing surface and liquid pool temperatures $\Delta T = T_w - T_l$ in various zones of the porous tube shows that it goes on due to ΔT decrease on the unflooded part of a tube.

More efficient heat transfer on this part of a sample is evident. It can be explained in the following way. There is not boiling but evaporative heat transfer mechanism occurs in a sintered powder porous media with capillary transport of liquid from a liquid pool to a heated zone. Hydrodynamic conditions for vapor release through macro pores of a sample are better for the flood-free part of a surface to compare with completely flooded sample. A direct heat transfer would be evaporation at an existing vapor-liquid interface which interacts with heating surface – three-phase line (TPL, liquid, solid, wall). This phenomenon is acting at micro-pore outlets contacting with macro-pore vapor channel of a porous structure.

## Vaporization in a porous body

Sintered powder wick can be considered as a system with open micro- and macro-pores (Figure 3). Micro-pores are used as capillary channels for liquid transport to zones of vaporization (meniscus). Macro pores represent channels for vapor transfer. The vapor is generated on surfaces of meniscuses in orifices of micro-pores. The high intensity of a vapor generation (evaporation) occurs in the zone II of meniscus. The thickness of a liquid film in the zone I is close to the size of molecular sorption film and there are not favourable conditions for vaporization. In the zone III a liquid film is thick, so its thermal resistance is higher than in zone II. There are a great number of such meniscuses over a volume of porous media, so the total area of evaporation is very large.

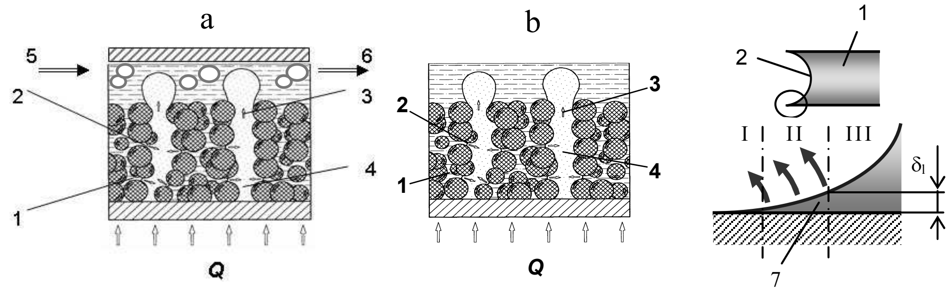


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Figure 3. Schematic of the copper sintered powder porous structure placed in mini-channel (a) and liquid pool (b). 1 – micro-pore, 2 – meniscus, 3 – macro-pore, 4 – vapor bubble, 5 – liquid flow in the mini-channel, 6 – two phase flow in the mini channel, 7 – zone of evaporation Q – heat flow

It means a rise in the heat flux would be possible without increasing the wall temperature, Figure 2. For such case the bubble formation is possible at a very short time. A vapor rest in the pore is able to grow immediately after the bubble break-off from the heating surface. The distribution of the pores on the heating surface is such that one pore is near the neighboring pores. This pore distribution would prevent the heat transfer from the wall to the liquid and the heat would be consumed almost completely by vapor generation in the pores. So the copper sintered powder wick was performed following the above mentioned recommendations. A test sample was completely immersed in the liquid pool, or was partially flooded. Figure 4 shows the data of heat transfer coefficient for various positions of the sample inside the experimental set-up at fixed height of liquid interface (h=75 mm): 1) porous sample is in the liquid pool, 2) porous sample is between vertical plates (flank mini-gap), 3) the porous sample is inside the glass annular mini-channel, 4) the sample – horizontal plain tube is disposed in the liquid pool.

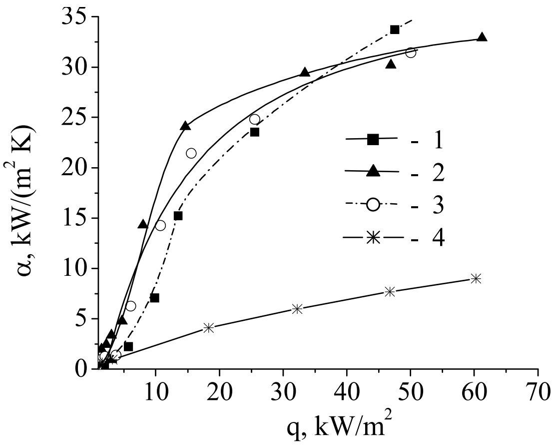


Figure 4. Heat transfer of the horizontal porous tube in the liquid pool, annular mini-channel and flank mini-gap (h=75 mm): 1 – liquid pool, 2 – annular mini-channel, 3 – flank mini-gaps;
4 – plane tube in the pool

There are two limitations for heat transfer intensity in such porous body: hydrodynamic ability of porous coating to transport liquid and finite number of vaporization zones (curvilinear meniscuses in orifices of micro-pores). The heat transfer intensity depends on the curvature K of meniscuses. While the curvature K doesn't exceed some value $K_{max}$ the capillary suction of liquid is good. When curvature of meniscus rises to $K>K_{max}$ the drainage of heated surface begins. On reaching the certain quantity of heat flux $q_{max}$, meniscuses move inside micro pores, the meniscus curvature K increases and exceed $K_{max}$. Consequently, a heat exchange surface disposed above the liquid interface doesn't get sufficient amount of liquid, "dry spots" appear and then spreads to all over the surface. The liquid interface is lower; the heat transfer coefficient is decreasing.

**Evaporation heat transfer in annular mini-channel**

An approximate physical criterion for the macro-to-micro scale threshold diameter [11] is based on the confinement effects of a bubble within a channel. According to them, for hydraulic diameters lower than $D_{th}$, the macroscopic laws are not suitable to predict either the heat transfer coefficient or flow pattern transitions, where $D_{th}$ is given by:

$$D_{th} = \left[4\sigma/g\left(\rho_l - \rho_v\right)\right]^{1/2}. \tag{3}$$

Usually this hydraulic diameter is less 3 mm for mini-channels and is less 200 μm for micro channels [12]. In our experiments the sample is disposed inside the transparent glass tube with annular mini-channel 0.8-1.8 mm (Figure 1). Visual observing of the process testified that vapor bubbles movement in annular mini-channel has a complicated character. Vapor and liquid move not only perpendicularly to the surface of heat releasing component but also ensure the swing two-phase flow along a tube axe from the input (bottom) toward the outlet (upper part) of the glass tube before enter the liquid pool, Figures. 5, 6.

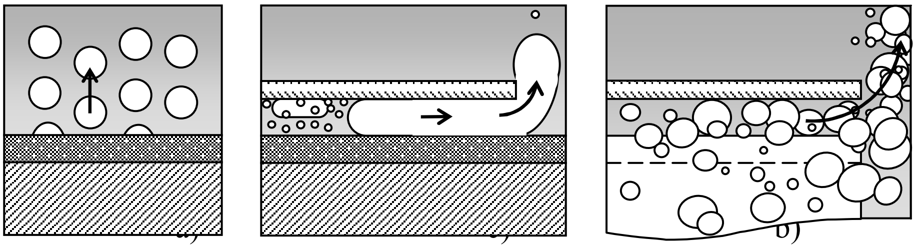

Figure 5. Vapor bubbles movement in liquid pool (a) and annular mini-channel (b, c), $q_b < q_c$.

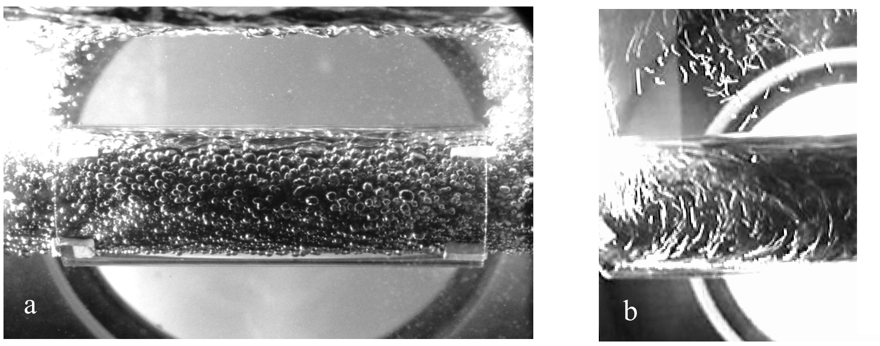


Figure 6. Visualization of the hydrodynamic and heat transfer of heat releasing tube
with porous coating, disposed inside the annular mini-channel, propane, q=34 kW/m$^2$:
a – freeze frame, b – the dynamics of the vapor bubbles movement

At low and moderate heat fluxes (less than 100 kW/m$^2$) the micro heat pipe effect is available in the pores accompanied with two-phase forced convection in the annular mini-channel (porous sample inside the glass tube). This coupled mode of heat transfer increases the value of total heat transfer coefficient near 10 times to compare with boiling heat transfer on the plane tube in the liquid pool. A micro scale effect of heat transfer enhancement took place inside the porous body and a mini-scale effect was ensured due to circular mini-channel between the sample and glass tube. To the boiling heat transfer coefficient the convective heat transfer coefficient is added, such as at the flow boiling through conventional channel or small diameter tubes, considered in [13]:

$$\alpha_{Total} = S\alpha_b + F\alpha_c, \tag{4}$$

where S and F account for influence of a vapor flow rate on a forced convection and a vapor quality on a heat transfer. So the total heat transfer intensity at moderate heat loads was higher than in a liquid pool on the same sample. But in our case the reason of the forced convection was not a mechanical pumping, bubbles can be considered as the micro pumps to move vapor and liquid. At the high heat fluxes the difficulties for the vapor phase exhaust from annular mini-channel limited a growth of the heat transfer intensity.

To the boiling heat transfer coefficient the convective heat transfer coefficient is added, that is why the total heat transfer intensity at moderate heat loads was higher than in a liquid pool. At the high heat fluxes the difficulties for the vapor phase exhaust from annular mini-channel limited a growth of the heat transfer intensity. Heat transfer coefficient was found to be dependent on a liquid/vapor interface position inside the glass tube: it was the most high, when the interface position inside the glass tube was accompanied with a partial flooding of the sample. The mean heat transfer coefficient for sample inserted inside a glass tube was higher to compare with pool and confined space between two vertical plates.

Thus the of heat transfer intensity at the moderate heat flux values in the annular mini-channel is higher than on the same sample in a liquid pool. However, it is important to note that in our case the positive effect is avhieved without active regulation and control of the flow in a system. The forced convection is caused not by the mechanical pumping of liquid, but the role of micro pump is performed by the vapor bubbles, which force liquid and vapor phases to be moved to the output from the annular mini-channel. The heat transfer intensity depends on the liquid level high h inside the glass cylinder, Figure 7.

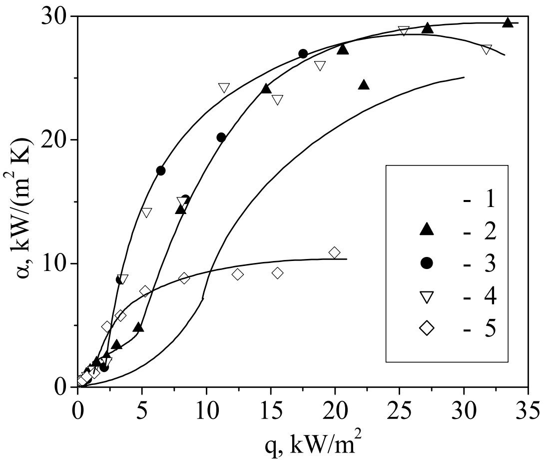


Figure 7. Heat transfer of porous tube placed in a liquid pool and annular mini-channel with different liquid level height: 1 – liquid pool, 2-5 – annular mini-channel, h=75, 20, 15, 10 mm

A lowering of the liquid level h under the conditions of an annular mini-channel to a quarter of the sample diameter contributed to an increase in the averages heat transfer coefficients over the surface, in this case a reduction of superheatings of the nonflooded part of surface was noted. With further lowering of a liquid level the system of porous coating capillaries did not provided for the sufficiency amount of liquid phase to the places of vaporization, it led to the decrease of the heat transfer intensity.

## Conclusions

1. Two-phase heat transfer in the annular mini-gap in combination with the stable process of vaporization in pores of coating of the heat dissipating pipe create conditions for the forced convection appearence without the application of additional energy, that contributes to an increase of the heat transfer intensity. The availability of an annular mini-channel promotes to a significant (2.5-3 times) increase of the heat transfer coefficients in comparison with the process in a liquid pool.

2. Reducing the size of cooling system we increase its efficiency, improve system performance by adding micro scale function (micro heat pipe effect) to macro scale engineering application.

3. Two-phase heat transfer in annular mini-channel with stable propane bubble generation on the thin (0.3 mm) porous surface of heat releasing tube stimulates Marangoni convection and 2-3 times enhance heat transfer to compare with the same tube heat transfer in the liquid pool without additional power supplying (no mechanical pumping).

4. A totality of the micro heat pipes phenomenon inside the porous structure and the forced two-phase convection flow in the annular mini-channel can be considered as the effective mechanism, which can be applied in the evaporative heat exchange systems, such as cooling systems of electronics components, vaporizers of adsorptive heat pumps, etc.

**Nomenclature**

D – diameter, F – coefficient, g – gravitational acceleration, h – height of liquid level, K – curvature, L – length, p – pressure, Q – heat flow, q – heat flux, S – coefficient, T – temperature. Greek: $\alpha$ – heat transfer coefficient, $\kappa$ – capillary constant, $\rho$ – density, $\sigma$ – surface tension coefficient. Subscripts: b– boiling, c – convection, l – liquid, s – saturation condition, th – threshold, v – vapor, w – wall.

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***Vasiļjevs L.L., Žuravļovs A.S., Šapovalovs A.V., Draguns L.A., Divfāžu siltumaptmaiņa šķidruma tilpumā un neregulāras formas mikrokanālos ar porainām cilindriskām sieniņām***

*Darbā aprakstīti eksperimentālu siltumapmaiņas pētījumi propāna iztvaikošanas procesiem horizontālas caurules porainajā pārklājumā un gadījumā, kad šo cauruli apskalo divfāžu plūsma radiālā minikanālā ar platumu līdz 2 mm. Tika pētīti pilnībā vai daļēji šķidrumā iegremdēti paraugi, gan arī paraugi ierobežotā tilpumā. Porainajā pārklājuma materiālā un šaurajos cilindriskas formas kanālos ar ierobežotu tilpumu tika novēroti gan mikroskopisku, gan minimērogu efekti. Eksperimentu rezultāti rāda, ka šāda kombinācija ir labvēlīga siltumapmaiņas intensifikācijai. Siltuma slodzēm līdz 50 kW/m2 gredzenveida cilindrisku kanālu izmantošana ļauj palielināt siltumapmaiņas intensitāti 2,5 – 3 reizes salīdzinājumā ar attiecīgajiem procesiem šķidruma tilpumā.*

***Vasiliev L.L., Zhuravlyov A.S., Shapovalov A.V., Dragun L.A. Two-Phase Heat Transfer in Liquid Pool and Annular Mini-Channel with Porous Cylindrical Wall***

*The results of experimental investigation of heat transfer at propane evaporation in the porous coating of the horizontal tube and at its two-phase flow streamline at the conditions of the annular mini-channel with width till 2 mm. The data were obtained on a flooded or partially flooded sample in a liquid pool or in confined space. A micro scale effect took place inside the porous body and a mini-scale effect was ensured due to annular mini-channel. Experimental results show, that such combination is favorable for the heat transfer enhancement. The availability of annular mini-channel significantly promotes to intense heat transfer (up to 2.5 - 3 times as high) at heat fluxes up to 50 kW/m$^2$, as compared with process in a liquid pool.*

***Васильев Л.Л., Журавлёв А.С., Шаповалов А.В., Драгун Л.А. Двухфазовая теплопередача в жидком бассейне и кольцевом мини-канале с пористой цилиндрической стенкой.***

*Представлены результаты экспериментального исследования теплообмена при испарении пропана в пористом покрытии горизонтальной трубы и при ее обтекании двухфазным потоком в условиях кольцевого мини-зазора шириной до 2 мм. Данные получены на затопленном и частично затопленном образце в объеме жидкости и в стесненном пространстве. Внутри пористого покрытия имели место микромасштабные эффекты, в кольцевом мини-зазоре – эффекты мини-масштаба. Результаты экспериментов показывают, что такое сочетание благоприятно для теплообмена. При тепловых нагрузках до 50 кВт/м$^2$ наличие кольцевого мини-канала способствует значительному (в 2,5 - 3раза) повышению интенсивности теплообмена по сравнению с процессом в объеме жидкости.*