EXPERIMENTAL INVESTIGATION OF PROPANE BOILING IN POROUS STRUCTURES

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Abstract

The experimental results of propane pool boiling on surfaces with porous coatings are presented. Porous coatings were performed by electric arc gas-thermal spraying (stainless steel) or sintered of metal powder particles (copper). Heat transfer enhancement in the evaporators of vapor-dynamic thermosyphons and loop heat pipes (LHP) charged with propane is discussed. Al_2O_3 ceramic as alternative to the metal sintered powder wick was tested. The experimental data for the heat transfer with propane pool boiling on horizontal smooth and porous tubes inside vapor-dynamic thermosyphon and the LHP evaporator were obtained for the saturation temperature T_s=–10÷+30 °C and heat input q=0.1-120 kW/m^2 (q=0.02-8 kW/м^2 for LHP evaporator). The propane boiling heat transfer intensity dependence on a saturation pressure, porous layer thickness and porosity were studied. Heat transfer enhancement up to 3-5 times was recorded with the heat transfer coefficient up to $30{\cdot}10^3$ W/(m^2 K).
The descriptions of experimental set-up, procedure of propane pool boiling heat transfer investigation are presented.
The results obtained can be applied in the chemical industry, refrigerator engineering, electronics, etc.

KEYWORDS

Heat transfer; Mass transfer; Pool boiling; Hydrocarbon; Propane; Porous media

INTRODUCTION

Evaporation-cooled heat exchange devices are widely used in the industry. Their main qualities are compactness, low material capacity and relatively small level of energy consumption.
Because of environmental concerns (ozone depletion, global warming) in accordance with the Montreal Protocol of 1992, a number of states have undertaken to cease applications using freons (chlorfluorocarbons CFC and hydrochlorfluorocarbons HCFC), and propane equally with other hydrocarbons is considered as an alternative to aforementioned working fluids. Interest in the research of hydrocarbon boiling heat transfer deals with refrigeration engineering, chemical industry, electronics, etc.

The problem of modern electronic components cooling often related with heat pipe, or two phase closed thermosyphon application. The most common approach to achieving meaningful reduction on electronic equipment mass is to reduce system volume. Reduced volume is achieved by increasing electronic packaging density which is accomplished by incorporating new coolers (heat pipes, thermosyphons).

The main benefits of heat pipe electronic coolers are such as: 1) high heat transfer intensity due to the evaporation and condensation in a closed and small space (near isothermal heat pipe operation), there is a possibility to raise the temperature of fins on the heat pipe condenser; 2) heat pipe thermal resistance is a weak function of its length, the temperature drop is mostly related with heat pipe evaporator and condenser thermal resistance; 3) heat pipe coolers are flexible in the setting of heat releasing components in relation to the ultimate cooling systems.

Heat releasing elements can be disposed inside the heat pipe evaporator [1, 2], Fig. 1, or to be in the thermal contact with the evaporator outer surface [3]. There is another possibility to put heat releasing elements inside the evaporator of a thermosyphon [4], Fig. 2. For such a case electronic components are immersed in the pool of dielectric fluid [5]. Its surface can be smooth or covered with a thin layer of capillary porous structure. One of the purposes of submerging electronic components with porous covering in a dielectric fluid is to initiate a boiling heat transfer at low heat fluxes and reduce a boundary layer thermal resistance of a low thermal conductivity dielectric fluid (a boundary layer over a smooth surface can be often overheated). Porous coatings reduce a liquid-solid surface energy. As a result we have reduction in bubble departure diameter. We have also the bubble size reduction and the active site density increase. A heat transfer enhancement exist when a favorable balance between an increase in site density and reduction in bubble size occurs.

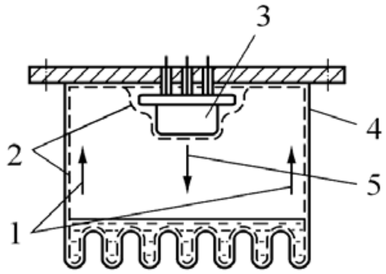

Fig. 1. Heat pipe with the electronic component inside wrapped by the sintered powder wick, Luikov Institute invention [1]: 1 – liquid flow, 2 – sintered powder wick, 3 – electronic component, 4 – heat pipe envelope, 5 – vapor flow.

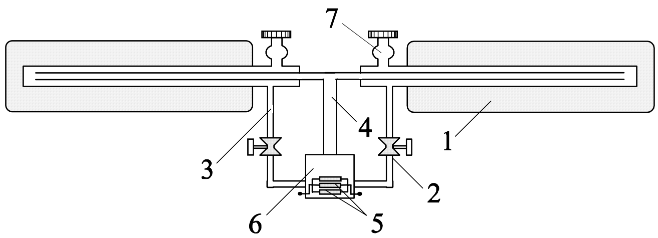

Fig. 2. Schematic diagram of the vapour-dynamic thermosyphon [5] with two condensers: 1 – adsorber, 2 – valve, 3 – liquid channel, 4 – vapour channel, 5 – heat releasing tubes with porous coating, 6 – evaporator, 7 – small volume and gap for non-condensed gas.

Therefore experimental data for boiling heat transfer of dielectric fluids on smooth and porous surfaces are essential. In further experiments a systematic study of the critical heat flux for propane as a nature friendly dielectric fluid will be investigated also.

Actually some hydrocarbons (ethane, methane, ethylene, propane, propylene, propadiene, butane, et.) are discussed as a vacant fluids for low temperature electronic equipment cooling. One of the most convenient (no noise, no vibrations) cooling systems is a loop heat pipe cooler (LHP) [7-9]. Cryogenic LHP electronic coolers need to be protected against super pressure influence at room temperatures. A combination of LHP electronic components cooler with solid sorption gas store can solve this problem [5].

The general goal of this work is an investigation of the heat transfer enhancement with propane boiling inside the vapor-dynamic thermosyphon on different porous structures and the heat transfer enhancement with propane evaporation in the LHP evaporator made from porous dielectric (AL_2O_3 ceramic).

EXPERIMENTAL SET-UP

On Fig. 3 a schematic of the apparatus is shown that was used to measure the pool boiling heat transfer on tubular elements (smooth and with porous coatings).

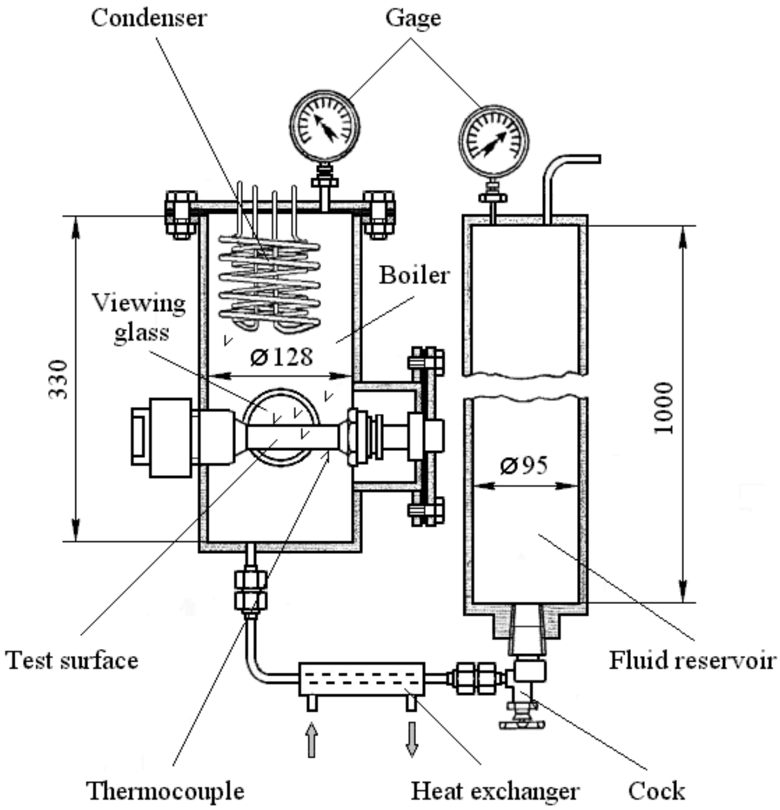

Fig. 3. Experimental set-up for hydrocarbon fluid boiling investigation.

The experimental set-up for boiling heat transfer investigation consists of test vessel (thermosyphon boiler), insulated chamber with cooling circuit and thermal control system, cooling machine (refrigerator), thermostats, condenser liquid loop, temperature control system, vacuum pump, liquid feed system, optical system for visual observation.

The test vessel has dimensions:

outer diameter:	160 mm,
inner diameter:	152 mm,
length:	200 mm.

A stainless steel condenser liquid loop is installed on top of the test vessel. To set the saturation temperature an electrical heater is wrapped around the outer surface of the vessel. Through stainless steel pipes welded to the bottom and the top of the vessel the experimental set-up is connected with the vacuum pump and the liquid feed system.

The vessel is disposed inside a temperature-controlled chamber, the air temperature of latter is held equal to the saturation temperature of the boiling liquid, controlled by a cooling or heating system.

The experimental set-up was used to measure the liquid saturation temperature (T_s), the average pool boiling heat flux (q), the wall temperature (T_w) of the test vessel, the wall temperature of tubes (smooth and with porous coatings). The temperature in the liquid and the vapor inside the test vessel was measured by copper-constantan thermocouples. The tubular specimen was made of stainless steel and covered with flame-sprayed stainless steel porous layers. The test vessel was charged with propane from the purger, giving a liquid height of approximately 70 mm above the test surface.

The test vessel was visible through three quartz windows. The vapour generated by liquid boiling was condensed by tubular condenser of a cooling machine and returned as a liquid to the pool by gravity forces.

To provide for a saturated liquid pool stable, the mass of liquid in the pool was large compared to mass of liquid evaporated. The lack of temperature difference between thermocouples placed on the test surface and the well-insulated test chamber walls essentially eliminates temperature errors due to radiation. The copper-constantan thermocouples and the data acquisition system were calibrated against a glass-rod standard platinum resistance thermometer.

Liquid boiling is realized on the outer surface of the test unit, the latter is heated by the electric heater or vapor-dynamic thermosyphon condenser. The test unit represent stainless steel smooth tube and tube with stainless steel porous layer on its surfaces (tube outer diameter 20 mm, length 100 mm).

The porous layers on tube surface were produced by gas-thermal spraying. These porous layers were performed as uniform structures with small microroughness distribution in any direction of the surface. An electronic scanning microscope was used to do the porous layer parameters analysis. The surface temperature of a tube was measured by copper-constantan thermocouples disposed in small narrow channels and covered by copper brackets.

The boiling heat transfer coefficient α was calculated using relationship typical for the smooth tube outer surface F_{sm}:

$$\alpha = \frac{q}{\Delta T} = \frac{Q}{F_{sm}(T_w - T_s)} \,, \tag{1}$$

where: q – heat flux, Q – heat input, T_w – temperature of outer tube surface, T_s – saturated liquid temperature. The temperature difference T_w–T_s was measured directly by four thermocouples, one junction of which was on the tube wall, the other was inserted in the liquid pool. Inaccuracy of heat transfer coefficient determination didn't exceed 20 %.

EXPERIMENTAL RESULTS

All pool boiling tests were carried out in the temperature range –10 °C to +40 °C, that corresponds to the saturation pressure range approximately p=3.45-13.8 Bar ($p^*=p/p_c$=0.081-0.323, p_c=42.64 Bar – critical pressure of propane). The heat flux was varied from 0.1 kW/m^2 to 100 kW/m^2 to simulate typical operating conditions of a heat pipe based electronic cooler.

Pool Boiling on Tubes with Porous Surface

All experiments were carried out with increasing and decreasing heat flux to investigate the boiling hysteresis.

During the experiments with propane pool boiling on tubes with porous coating, three heat transfer modes were checked: free convection, non-developed nuclear boiling and developed boiling (Fig. 4). Our experimental data obtained on smooth tubes were compared with the data of Gorenflo [6] and are in good agreement, what testify the validity of the experiments. The boiling curves have different slope characterized by the index "n" in the dependence $\alpha=Cq^n$. This slope and the length of a region of nuclear boiling is changing depending on the kind of evaporation surface, saturation pressure and the direction of heat flux change. The nuclear pool boiling is arriving at temperature heads of 0.1-0.3 °C.

The pressure increase intensify the boiling heat transfer. On the smooth polished tube, with increasing heat flux the free convection was observed in a wide range of heat fluxes and temperature heads (up to 4-5 kW/m^2 and 6 °C). This process is reducing with increase of the saturation pressure. The pressure increase initiate the boiling nuclear centers activation, but at high heat loads the most number of potential centers of nucleation are already active.

The first centers of nucleation arrived on the lower tube generatrix. After the heat load was changing in opposite direction, the nuclear boiling was extended into the region, which was initially occupied by free convection. Even for very small heat fluxes (q<0.3 kW/m^2) widely spaced centers of nucleation with very low detachment frequency were observed on the lower generatrix.

On porous tubes the first centers of nucleation are noticed on the upper part of tube. Here bubbles of very small size and high detachment frequency were generated and departed in the form of chains. Increase of the heat load led to a gradual covering of the overall upper tube surface by centers of nucleation and the first centers of nucleation on the lower generatrix sprang up. At this regime of nuclear boiling a reduction of the average temperature head and considerable intensification of heat transfer (up to three times on the samples with porous coating) were noticed. At further increases of the heat load, the compactness of nucleation centers and bubble detachment frequency on the lower generatrix increased, and the departure diameters of bubbles expanded. With the next heat flux increase, the temperature head began to grow also. In the range q=1-30 kW/m^2 experimental data can be generalized in the form $\alpha=Cq^n$, the index "n" is equal to 0.5-0.7. In the region q>20-30 kW/m^2, the nuclear boiling on tubes with porous coatings is similar to the nuclear boiling on smooth tubes. A strong screening of the tube surface by vapor bubbles was observed and the heat transfer intensity reduced. During a propane nuclear boiling on the tubes with porous coating the heat-flux hysteresis phenomenon was observed.

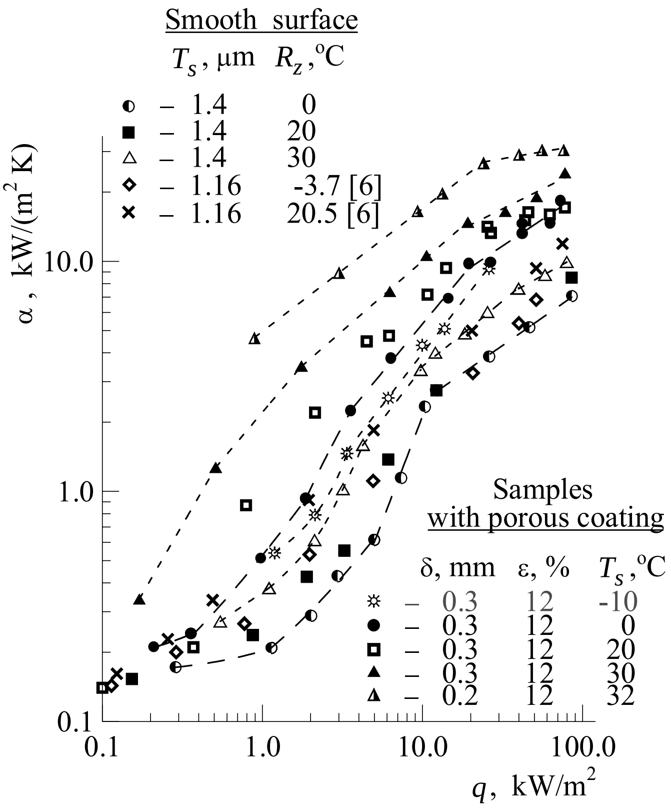

Fig. 4. Propane pool boiling heat transfer intensity on smooth tubes and tubes with porous coating.

An application of metal porous covering obtained by gas-thermal spraying technology allows one to increase significantly (3-5 times as high at q<8 kW/m^2, and 2.5-3 times as high at q>8 kW/m^2) the heat transfer intensity for propane boiling on horizontal tubes. In the region q>2 kW/m^2 the heat transfer intensity on tubes with porous coatings is comparable to, and even exceeds the heat transfer intensity with boiling on tubes with structured surfaces, published by Gorenflo et al. [6], who experimented with regular surface micro-geometry, such as GEWA-T-x (T-shaped microfins with high 1.04 mm, step 1.35 mm, gap 0.23 mm, tube diameter at the bottom of fin 12.48 mm).

Heat flux with propane pool boiling as a function of temperature head ($\Delta T = T_w - T_s$) is shown on Figs. 5, 6 for different thickness of a wick and different wick porosity.

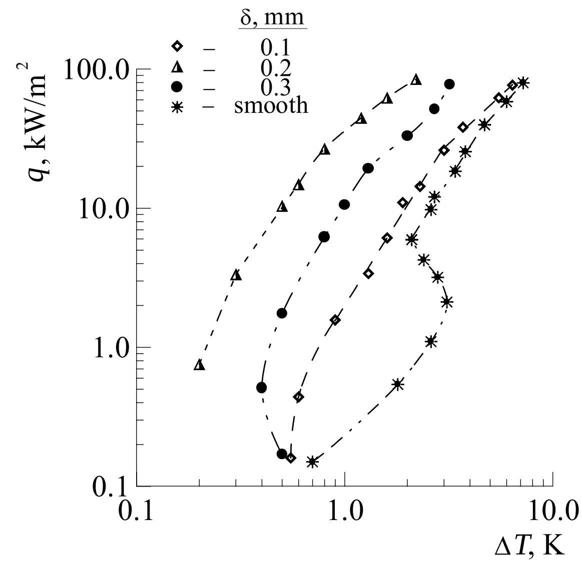

Fig. 5. Heat flux as a function of temperature head for different porous structures for propane pool boiling (wick porosity ε=12 %) at T_s=30 °C, p^*=0.254, δ – porous structure thickness.

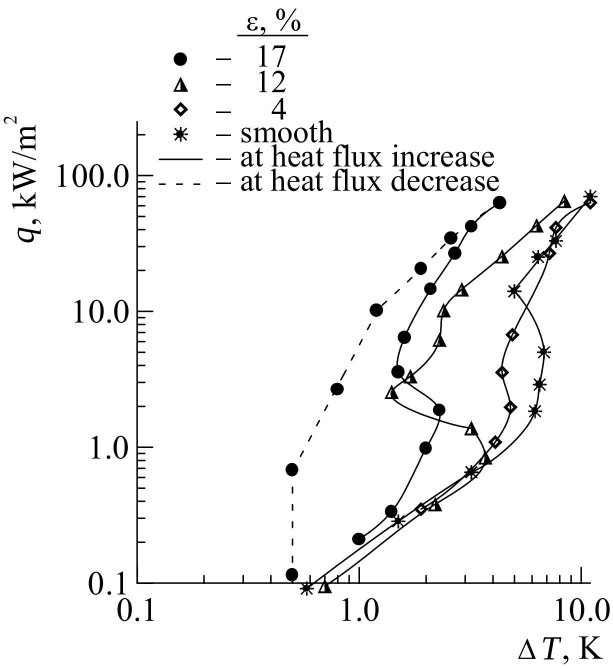

Fig. 6. Heat flux versus the temperature head for propane pool boiling on horizontal porous tubes with different porosity, δ=0.3 mm, T_s=0 °C (p^*=0.111).

On the samples with identical porosity (ε ~12 %) the maximum heat transfer was reached on the tube with porous covering thickness 0.2 mm (Fig. 5). A diminution of porosity from 17 to 4% (Fig. 6) promoted a significant decrease of the heat transfer intensity. On further heat flux increase, the heat transfer intensity was on the same level, and at the some regimes definitely lower, than at boiling on smooth polished surfaces.

The increase and decrease of the coating thickness in such a case lead to a certain decrease of heat transfer intensity. An ability of structure for supplying and holding of liquid at the surface-side zone is

decreasing due to reduction of a number of the small diameter pores. As a thickness of this porous layer increase the conditioning of vapor evacuation becomes worse. The heat transfer intensity with propane boiling on the tube with a thickness of porous coating 0.1 mm was similar to the heat transfer intensity on a smooth surface.
These data are useful for the further applications of heat generating porous tubes inside the vapor-dynamic thermosyphon with some condensers (Fig. 2). Liquid channels 3 (Fig. 2) ensure the liquid subcooling before this liquid enters the evaporator. Special valves 2 (Fig. 2) guarantee the intermittent liquid feeding of the evaporator.

Loop Heat Pipe Evaporators with Internal and External Heat Load

Another alternative to this vapor-dynamic thermosyphon is a loop heat pipe (LHP), Fig. 7.

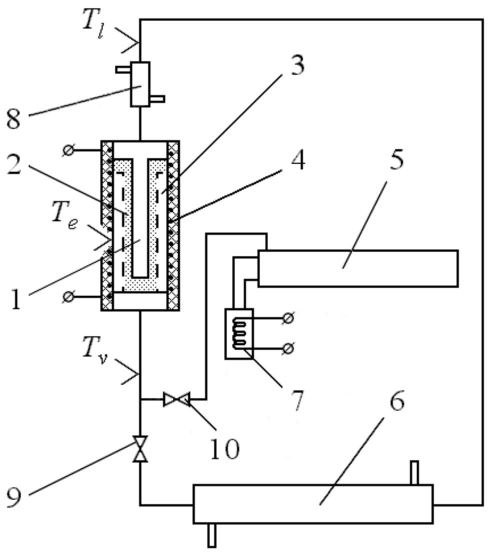

Fig. 7. Experimental set-up composed with loop heat pipe (LHP) and solid sorption cooler (SSC): 1 – liquid accumulator, 2 – sintered powder wick, 3 – vapor channels, 4 – electric heater, 5 – solid sorption gas store SSC, 6 – LHP condenser, 7 – SSC heater, 8 – LHP subcooler, 9 – valve, 10 – pressure regulated valve, T_l – liquid temperature, T_e – evaporator wall temperature, T_v – vapor outlet temperature.

A wick structure of a loop heat pipe (LHP) is done from AL_2O_3 ceramic as dielectric structure convenient to be used with hydrocarbons for electronic components cooling (Fig. 8).

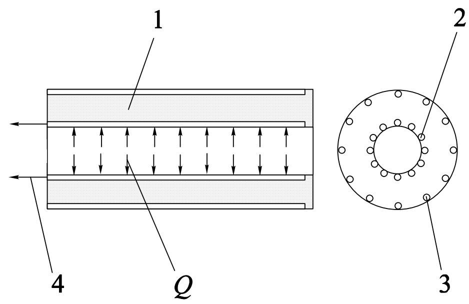

Fig. 8. A wick structure of a loop heat pipe: 1 – Al_2O_3 ceramic porous wick, 2 – vapour channels, 3 – liquid artery, 4 – vapour output into the propane pool

The samples were immersed into the propane pool, liquid suction was ensuring by the capillary forces, which guarantee the uniform liquid distribution in the porous structure of the wick. Acquisition of heat generated in the cylindrical electric cartridge heater provides the motivating force in the LHP evaporator, as a capillary pump. Liquid flows through the liquid channels in the capillary structure and the radial liquid flow occurs through the wick to the electric heater wall and evaporation occurs in the region of the wick near the wall of the heater. The resultant vapor then is moving along the axial vapor channels inside the wick to the liquid pool. The direct contact between the electric heater wall and the evaporating liquid inside the wick with vapor channels establishes liquid-vapor meniscus near channels surface. This produces heat transfer coefficient two or more times greater than those obtained for conventional grooved heat pipe evaporators, where a heat flux go through the liquid layer. The pumping head required for a wick is produced by its surface tension forces. Menisci are formed, which adjust naturally to provide the exact pumping head required to match the flow losses associated with applied heat load. It should be noted that the fluid flow problem is of significant importance in the LHP evaporator and combined fluid flow and heat transfer analysis are necessary. In the evaporator No. 1 four vapor channels (d=2 mm) were situated along the heater near its surface in the wick. There were no liquid channels in the porous structure and liquid from the pool is sucked by capillary forces of a wick. In the evaporator No. 2 eight vapor channels (d=3 mm) were situated near the heater surface and eight liquid channels were disposed along the heater near the sample surface. For this case we have liquid channels to stimulate the liquid entrance inside the evaporator. The Photo of evaporator No. 2, immersed in a propane pool, is shown in Fig. 9.

Fig. 9. Al_2O_3 porous wick with 8 vapour channels and 8 liquid channels is immersed in a propane pool and heated with an electric heater inside the porous structure. ("inverted" meniscus of evaporation), T_s=303 K, q=1.2 kW/m^2.

The wick porosity is 50 %, pore diameter 0.15-0.20 mm. The wick thermal resistance R as a function of the temperature head between the electric heater wall and the vapor output ($\Delta T=T_w-T_v$) is shown on Fig. 10 (T_v is the temperature of vapor output into the propane pool). The thermal resistance of the evaporator No.1 with four vapor channels is definitely higher with comparison to the thermal resistance of the evaporator No. 2. It means, that the number of vapor channels near the heat releasing surface influence on this thermal resistance. At low heat flux (q=0,03-0,3 kW/m^2) there is no boiling inside the wick and heat dissipation is ensured by the thermal conductivity and convection inside the porous structure. At the range of the elevated heat flux (q=0.3-10 kW/m^2) there is an intensive increasing of centres of nucleation inside the porous media, and the thermal resistance of the wick is steadily reducing. At the heat flux q=20 kW/m^2 porous structure near the heating surface is completely saturated with the vapor and the thermal resistance of porous wick is rapidly increasing.

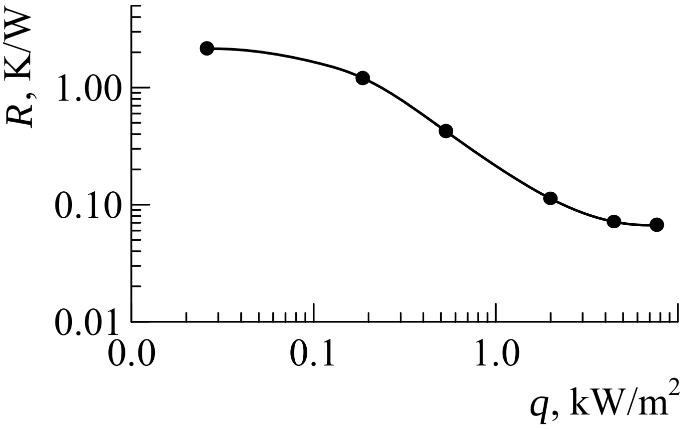

Fig. 10. Thermal resistanse of the Al_2O_3 porous wick evaporator with 4 vapour channels as a function of heat load from the inner surface (electric heater).

The heat flux as a function of the temperature head ($\Delta T = T_w - T_v$) for two samples of Al_2O_3 wick is shown on Fig. 11. Analysing the behaviour of this curve we could conclude, that in the heat flux range 0.2–2.0 kW/m^2 for the evaporator No. 1 and No. 2 heat transfer inside the porous structure is realised as micro heat pipe phenomena (evaporation-condensation in the pores with capillary sucked liquid), and there is a negligibly small temperature drop inside the wick. At the heat flux q>2 kW/ m^2 some of pores near the heated surface of the evaporator become saturated with vapor, and its number is increasing, when the heat flux is increasing.

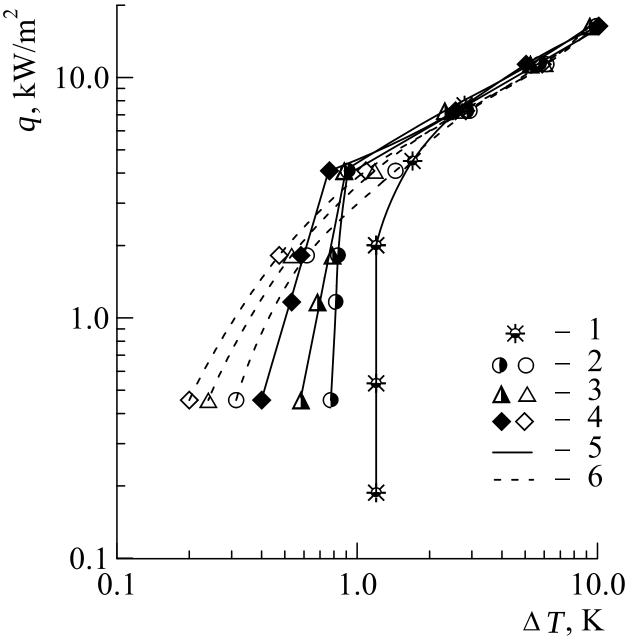

Fig. 11. Al_2O_3 evaporator heat flux as a function of the temperature head $\Delta T = T_w - T_v$. Propane boiling inside a porous wick, T_s=30 °C. (1 – 4 vapour channels, d=2 mm,
2 – 8 vapour channels and 8 liquid channels, d=3 mm,).

The heat transfer intensity during the propane boiling inside Al_2O_3 porous media is shown on Fig. 12. The heat transfer coefficient with propane boiling inside the wick is rapidly increasing from 300 to 6000 W/(m^2 K) in the heat flux range 0.2-20 kW/m^2 for the evaporator No. 2 (8 vapor channels) and from 150 to 3000 W/m^2 K for the evaporator No. 1 (4 vapor channels).

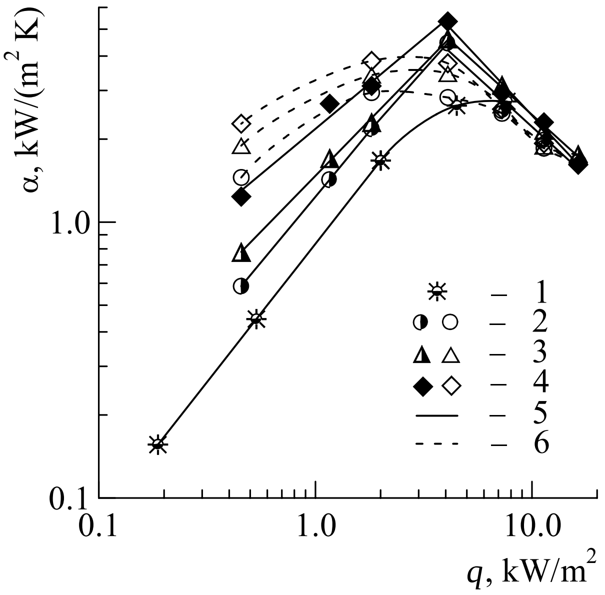

Fig. 12. Al_2O_3 porous wick evaporator heat transfer coefficient as a function of heat flux. Propane boiling inside a porous wick, T_s=30 ^{0}C. (1 – 4 vapour channels, d = 2 mm, 2 – 8 vapour channels and 8 liquid channels, d=3 mm).

Dielectric porous wick is a good combination with hydrocarbon (propane) as a working fluid for LHP. This wick can be used in direct contact with the electronic component (Fig. 1) to stimulate its cooling by evaporation. The experiments were performed at the liquid pool temperature T_s=30 °C in the heat load interval q=0.02-30 kW/m^2. From the experimental data it is clear, that the geometry and number of vapor channels, pore characteristics have a strong influence on hydraulic resistance, thermal resistance, heat transfer intensity and critical heat flux in this structure. For our case the wick structure of the evaporators need to be optimized with the point of view of its geometric parameters and the surface of a thermal contact between the heat releasing elements and porous media need to be increased.

CONCLUSION

Electronic components can be efficiently cooled, if they are disposed inside the evaporator (immersed in the liquid) of the vapor-dynamic thermosyphon, or loop heat pipe (LHP).
Hydrocarbon (propane) fluids are beneficial as a working media for low temperature heat pipes to cool the electronic components and could be used instead of CFC and HCFC fluids.
Heat transfer intensification (3-5 times, q=30 kW/m^2) was observed during the experiments with propane nuclear boiling on the horizontal tubes with porous metal covering .
Al_2O_3 ceramic was preliminarily tested as a wick for LHP and is considered as a vacant alternative to the metal sintered powder wick.

Nomenclature

C – constant coefficient, unitless; F – surface square, m^2; n – index, unitless; p – pressure, bar; $p^*=p/p_c$ – reduced pressure, unitless; Q – supplied heat flow, kW; q – heat flux, kW/m^2; R_z – roughness, μm; T – temperature, K, °C; ΔT – wall superheat, K, °C; α – heat transfer coefficient, kW/(m^2 K); δ – porous layer thickness, mm; ε – porosity, %.
Subscripts: c – critical; e – evaporator; s – saturation; sm – smooth; v – vapor; w – wall.

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