AN ADVANCED MINIATURE COPPER HEAT PIPES DEVELOPMENT FOR COOLING SYSTEM OF MOBILE PC PLATFORM

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Abstract

At present days the miniature heat pipe with sintered copper powder inside are widely used for cooling of mobile PC. This paper presents an overview of the copper miniature heat pipes with wick structure (MMHP) studies that has been undergone in Porous Media Laboratory, Minsk, Belarus and Asian Vital Components Ltd., Beijing, China for its applications in cooling systems of high power CPU for the present and future designs of PC mobile platforms. Three different, most attractive for the CPU cooling system, sizes of MHPs, namely those 4, 5 and 6 mm outer diameters and 200 and 300 mm in lengths, have been developed in this study. De-ionized water is the working fluid for all heat pipes. All these MHP designs are compared in terms of maximum heat flux capability and equivalent thermal conductivity (thermal resistance) at different vapor temperatures at the adiabatic MHPs zone and orientations in space (inclination angle). A new design of 5mm outer diameter miniature copper heat pipe with advanced biporous wick is suggested as a promising candidate for present and future cooling systems of high power mobile PC platforms.

KEYWORDS

Electronic cooling, heat pipe, evaporation, porous media.

1. INTRODUCTION

The modern trend in microelectronic and optoelectronic devices is to increase the level of integration by minimizing the device size (high density packaging) and increasing the performance of the device (higher frequency). This results in an increase in both power dissipation and power density on the device with prediction to dissipate local heat flux more then 300 W/cm^2. Such high heat flux presents a serious challenge to existing thermal management techniques to ensure device performance and reliability while maintaining acceptable temperatures, especially for computer processor [1].

In the past the concept of a high heat flux evaporators was considered for different electronic devices such as laser mirrors and radio frequency antennae for Tokomak reactor to impact energy, via an induction field to the confined plasma. In such case peak heat flux of 300 W/cm$_2$ was dissipated by pumping fluids through sintered powder wick structures [2]. However, application to future generations of PC mobile platform will require more uniform surface temperatures and higher peak fluxes than are available today. With electronic packages becoming more dense and powerful, traditional methods of thermal energy removal are reaching their limits. The removal of very large heat fluxes is becoming a barrier to the technology roadmaps for microprocessors, power electronic modules, and many other applications incorporating microelectronic or optoelectronic devices. Spray cooling is a promising candidate to address the thermal concerns of system requiring high heat flux removal in a compact volume, as demonstrated by its successful application in the Crag XI vector supercomputers and numerous patents that have been granted for various spray cooling applications. A review of the literature reveals that very limited data are available for spray cooling in a closed system. For use of spray cooling in electronic packaging, the system will have to be closed [3].

Another possibility to dissipate high heat flux is heat pipes, which offer a better alternative to liquid cooling provided they can handle the heat flux and power requirement [4-7]. The first implementation of a heat pipe in a notebook computer occurred in 1994 despite the fact that the first working heat pipe was built in 1963 [8]. Earlier notebook computers relied on simple metallic heat sinks, but by the late 1990 s approximately 60 % of notebook computers used heat pipes in thermal management solutions [8]. It is now common to find heat pipes in different configurations used in thermal solutions for portable computers. Almost every current notebook computer uses a heat pipe in its thermal management design. Most of these heat pipe-heat sinks use a heat pipe with outer

diameters between 3 and 6 mm that carries the power from the CPU to a large aluminum plate. The heat is then conducted into the EMI shield of the keyboard, through the keyboard, and into the ambient air by natural convection and radiation. Most recent designs incorporate a heat sink/fan/vent with the use of a heat pipe. This concept permits the notebook designer to locate the CPU independent of the heat sink. This allows the notebook designer to design the most effective heat exchanger and an optimal airflow path. The optimal design helps to reduce airflow requirements and noise. Most applications use an aluminum evaporator block (heat input section), the heat pipe (heat transport section), and aluminum fins (heat sink section).Up to present days, copper tubing, copper sintered powder and water as a working fluid is the briliant combination of the materials used to make heat pipes for electronic components cooling system application due to the material's chemical compatibility, high cooling capability and manufacturability.

The main distinction of heat pipes, besides working fluid and envelope material, is the wick structure. There are several types of wick structures: screen, grooves, felt, and sintered powder. Sintered powder metal wicks offer several advantages over other wick structures. An emerging advantage of the sintered powder wick is its ability to handle high heat fluxes with usually low thermal resistance. Since sintered powder wicks are generaly more then 60 % porous, there is accordingly a large surface area for evaporation.

Another advantage of a heat pipe with a sintered powder wick especially for the cooling system of mobile PC platform is that it can work in any orientation, including against gravity (i.e., the heat source above the cooling source). The power transport capacity of the heat pipe will typically decrease as the angle of operation against gravity increases. Since groove and screen mesh wicks have very limited capillary force capability, they typically cannot overcome significant gravitational forces, and dry out generally occurs. Additionally, since a sintered powder wick is integral with the heat pipe envelope, and the fluid charge is only enough to saturate the wick, the heat pipe can be subjected to freeze/thaw cycles with no degradation in performance. Moreover, the heat pipe including the sintered powder wick structure can be bent and/or flattened in different shapes. The above attributes make the sintered powder wick the optimal structure for many thermal management solutions.

There are generally at least five physical phenomena that will limit, and in some cases catastrophically limit, a heat pipe's ability to transfer heat. They are commonly known as the sonic limit, the capillary limit, the viscous limit, the entrainment limit, and the boiling limit. Boiling and capillary limits are most restricting factors for the maximum heat flux that can be removed by heat pipe with sintered powder and water inside it. Capillary limit occurs if the wick capillary pressure is lower than the sum of the pressure drops along the liquid circulation path. Boiling limit occurs when the vapor generated at hot spots of the wick is trapped due to geometry limitation. It means, that the cooling capability of the heat pipe strongly depends on the balance between capillary pressure and hydraulic resistance of the wick in both (longitudinal and cross) direction and an ideal heat pipe wick should have simultaneusly the high capillary pressure head and wick permeability. In turn the capillary pressure head and wick permiability is the function of the pore distribution inside of the wick and finnaly the copper powder size and shapes.

One of the possible way to extend the capillary and boiling limits and thus to increase the maximum heat flux, which could be transferred by the heat pipe with fix geometrical size, is to apply so called non regular sintered wick structure. One recent effort made by Semenic and all. Proves, that sintered copper wick with non regular biporous wick is able to dissipate up to 500 W/cm_2 using water as a working fluid [9].

In this paper a summary of the work devoted to the R&D of 4, 5 and 6mm in outer diameter miniature heat pipes with regular and non regular wick structures is presented.

2. THEORETICAL ANALISYS

MHP wick is a key element in MHP design, because the driving force for the working fluid circulation is the capillary pressure developed at the liquid vapor interfaces of the menisci in porous wick. For the successful operation of the heat pipe the MHP wick has to provide capillary head, which is equal or exceeds all pressure losses in the system.

$$\Delta p_c \geq (\Delta p_{sum}) \geq \Delta p_l + \Delta p_v \pm \Delta p_{gr} + \Delta p_{phc} \quad (1)$$

where Δp_c is the capillary pressure losses which are the function of liquid surface tension forces, pore radii r_{pr} and solid surface wettability:

$$\Delta p_c = \frac{2\sigma\cos\theta}{r_{pr}}. \qquad)$$

It is important to note that the pressure loss due to phase change Δp_{phc} is not significant for such kind of MMHP and could be not taking into account. Thus the pressure balance equation ought to be expressed as:

$$\Delta p_c \geq \Delta p_l + \Delta p_v + \Delta p_{||} + \Delta p_+ + \Delta p_\delta \qquad (3)$$

where Δp_l and Δp_v frictional pressure losses along liquid and vapor paths, $\Delta p_{||}$ and Δp_+ pressure losses along and across of the heat pipe due to action of the gravity force and Δp_δ – pressure losses due to working fluid trapping by the wick in the MHP condenser.

The pressure losses in the MHP vapor path is evaluated by the Poiseuille equation :

$$\Delta p_v = \left(\frac{C(f_v \mathrm{Re}_v)\mu_v}{2(r_{h,v})^2 A_v \rho_v h_{fg}} \right) L_{eff} Q \qquad (4)$$

where $L_{eff} = 0.5L_e + L_a + 0.5L_c$ and values C and $f_v Re_v$ are depended on the vapor flow regimes.

Following the Darcy equation the pressure losses in the MHP liquid path could be found as:

$$\Delta p_l = \left(\frac{\mu_l}{K A_w h_{fg} \rho_l} \right) L_{eff} Q, \qquad (5)$$

where K represents the wick permeability. Hydrostatic pressure head across and along the MHP could be accordantly expressed as:

$$\Delta p_+ = \rho_l g\, d_v \cos\varphi,\ \Delta p_{||} = \rho_l g\, L \sin\varphi. \qquad (6)$$

Capillary force by action of which the working fluid trapped in the wick on the MHP condenser zone could be determinate as:

$$\Delta p_\delta = 2\sigma \cos\theta / r_{h,\delta}. \qquad (7)$$

As it was mentioned above, the most strict limitation in the MMHP cooling capability is the capillary limit:

$$Q_c = \frac{\Delta p_c - \Delta p_+ + \Delta p_{||} - \Delta p_\delta}{\Delta p_v / Q + \Delta p_l / Q} \qquad (8)$$

and, finally ,by taking into account the equations (2,3-7), the equation (8) could be expressed in next form:

$$Q_c = \frac{\left(\frac{1}{r_c} - \frac{1}{r_{h,\delta}} \right) 2\sigma\cos\theta - \rho_l g d_v \cos\varphi + \rho_l g L \sin\varphi}{\left(\frac{C(f_v \mathrm{Re}_v)\mu_v}{2(r_{h,v})^2 A_v \rho_v h_{fg}} + \frac{\mu_l}{K A_w h_{fg} \rho_l} \right) L_{eff}} \qquad (9)$$

3. MHP POROUS STRUCTURE OPTIMIZATION

The wick provides a means for the flow of liquid from the condenser to the evaporator section of the MHP. It also provides capillary pressure head for successful MHP operation in any space orientations. The wick structure also has an impact on the radial temperature drop at the MHP evaporator zone between the inner MHP hot envelope surface and the liquid-vapor surface. Thus, an ideal MHP wick should have high level of permeability in radial and axial directions as well with high capillary pressure difference between MHP evaporator and condenser zones. Good mechanical contact between internal surface of the MHP envelope and wick as well as the high level of a thermal conductivity also play an important role to minimize the temperature drop in a MHP radial direction. Generally speak, there are three possible ways to increase the wick permeability of a single material with still essential capillary pressure difference inside of MHP: to vary the porosity and hydraulic pore sizes with increasing from evaporator to the MHP condenser by applying copper particles with different size; to use the special kind of copper powder with so called dendridic form (Fig.1) ; to build the bi-porous wick structure which consist of the macro and micro pores. Last two solutions are more attractive to be applied for the MHP mass production due to the complicity to find an optimal temperature for sintered process of different sizes copper particles.

To satisfy above mentioned requirements, three types of wick structures have been developed. Two homogenous wicks where round and dendridic shapes copper powders were used as a raw material and bi-porous one.

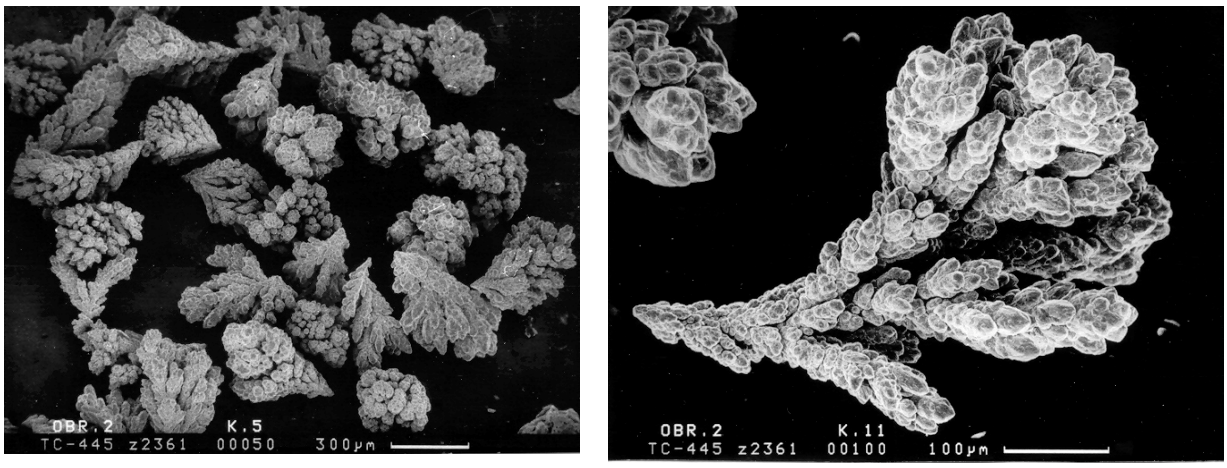

Fig. 1. Photo of the copper powder with dendrite shape [10]

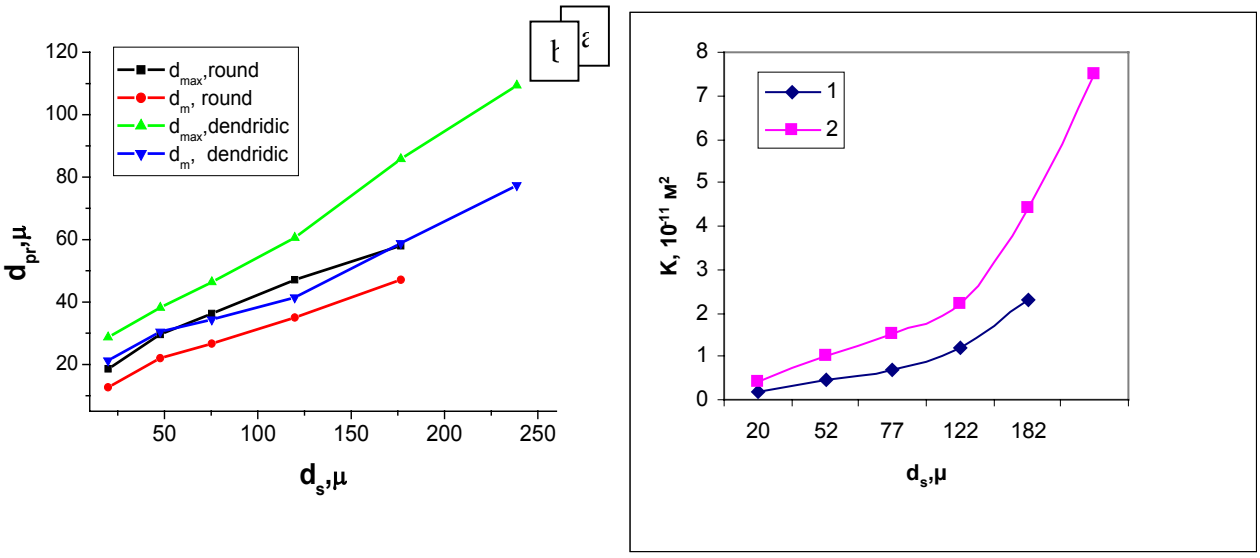

Fig. 2 Wick structure parameters vs. copper particles shape and sizes. a – maximum and mean pore sizes of the wick vs. different copper particles shape and sizes; b – permeability of the wick with round shape copper particles (curve 1) and dendrite (curve 2) [10]

As ones can see, by using the different sizes of copper particles as a raw material the wick structural characteristics and first of all wick permeability could be changed in very wide range (Fig. 2). Moreover the shape of copper particles plays also an important role for the working fluid circulation inside of the MHP. Thus finally the MHP cooling capability, at given outer diameter, with homogenous wick depends on three parameters: size of copper particles, its shape and porous media thickness inside of the MMHP.

4. EXPERIMENTAL APPARATUS and PROCEDURES

The experimental set-up (Fig. 3) to determine the miniature heat pipe parameters has been developed. The general goal of the experimental work were to determine the temperature distribution along the heat pipe for different heat loads, estimate the heat pipe MHP maximum heat flux in any position, evaluate the dependency between a heat pipe thermal resistance and heat dissipation.

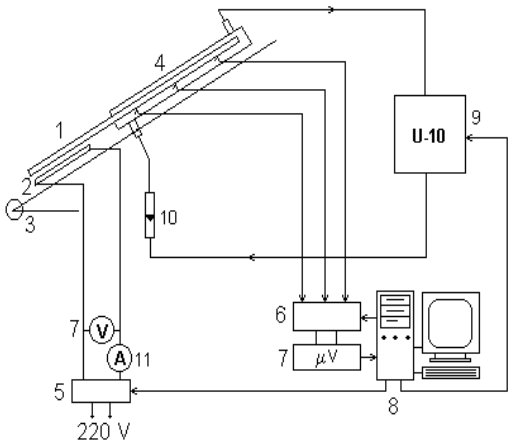

Fig. 3. Experimental set-up schematic: 1 – heat pipe disposed inside the reinforced polymer chamber, 2 – electric cartridge heater, 3 – the platform for the heat pipe inclination, 4 – liquid cooling jacket, 5 – electric source of energy, 6 – electric switch unit with thermocouples, 7 – electric voltmeter, 8 – computer, 9 – thermostat, 10 – water rate recorder, 11- ampere meter

The HP evaporator was heated by electric heaters, which are controlled by a DC power supply, while heat removal was performed by circulation of liquid through a co-axial liquid heat exchanger (cooling jacket) in the condenser (Fig.4).

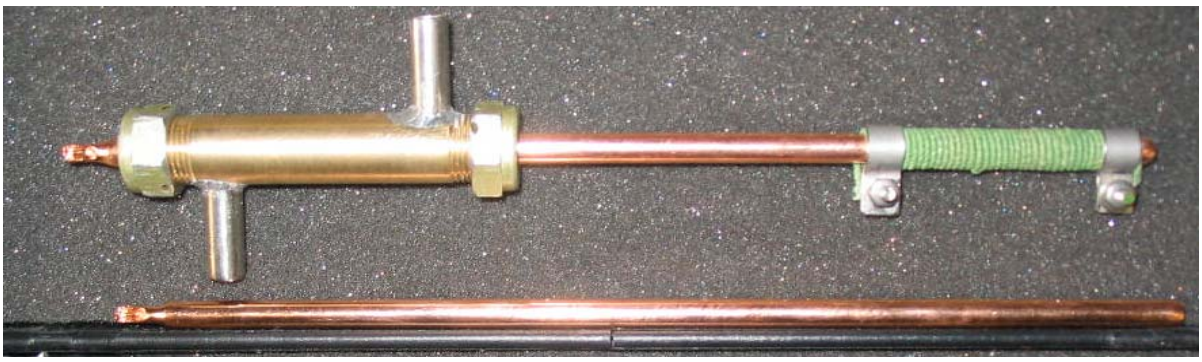

Fig. 4. Photo of the MHP and with electrical heater on the MHP evaporator zone and liquid heat exchanger on the condenser zone

The recirculating thermostat was used to provide continues supply of cooling water in the heat exchanger at the constant temperature and flow rate. The temperature stability of thermostat was controlled within the accuracy of ± 10 ºC. The heat load to the evaporator was supplied following a computer's program. The MHP operation was monitored by 7 thermocouples T-type (Fig. 5).

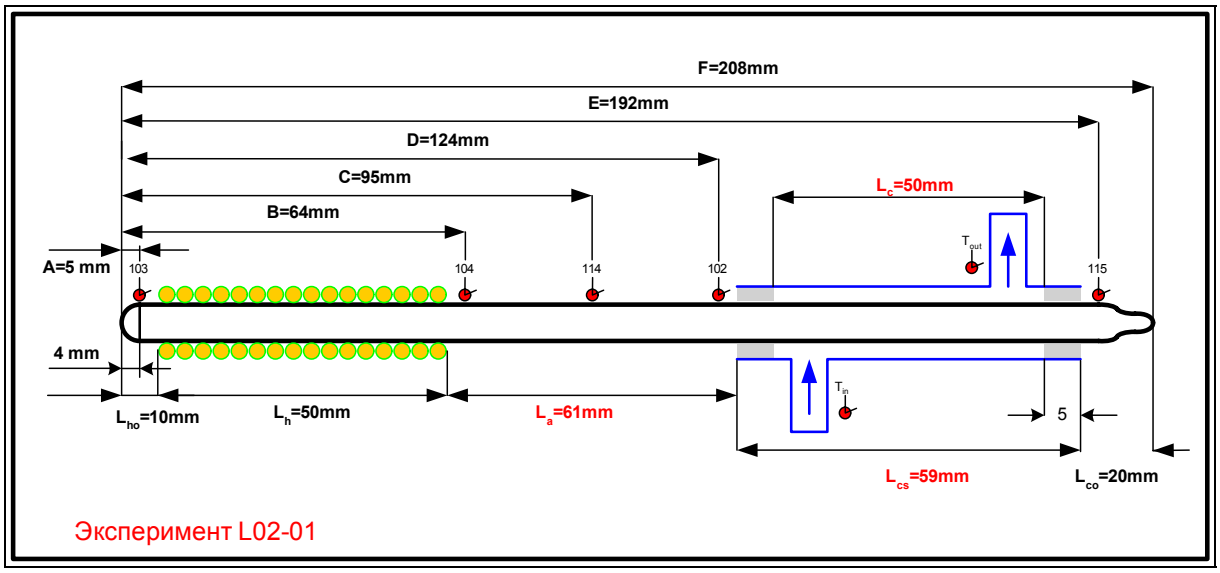

Fig. 5. Scheme of the thermocouple location on the MHP. Experiment L02-01

All measured temperatures were transmitted to the computer through Data Acquisition Switch Unit in real time and were recorded for its analysis every five seconds. Heat pipe tilt measurements were realized by a system of tilt regulation. To minimize the thermal losses from the MHP evaporator to the environment by the natural convection, the whole MHP surface was thermally insulated. During the tests MHP with all test rig was installed inside the climatic chamber.

RESULTS AND DISCUSSIONS

Fore types of MHP with outer diameter 4–6 mm and 200 mm length developed in LHMTI and AVC were investigated in this study. To reproduce the operation conditions close to real one for all three MHP samples the constant temperature of cooling water (40 ºC) in the condenser heat exchanger was maintained. The heat load value in the MHP evaporator was increased step by step during the tests. The maximum heat flux (Q_{max}) was determined, when the temperature in the evaporator zone started sharply to rise (not proportionally) to compare with the temperature rise in other MHP zones.

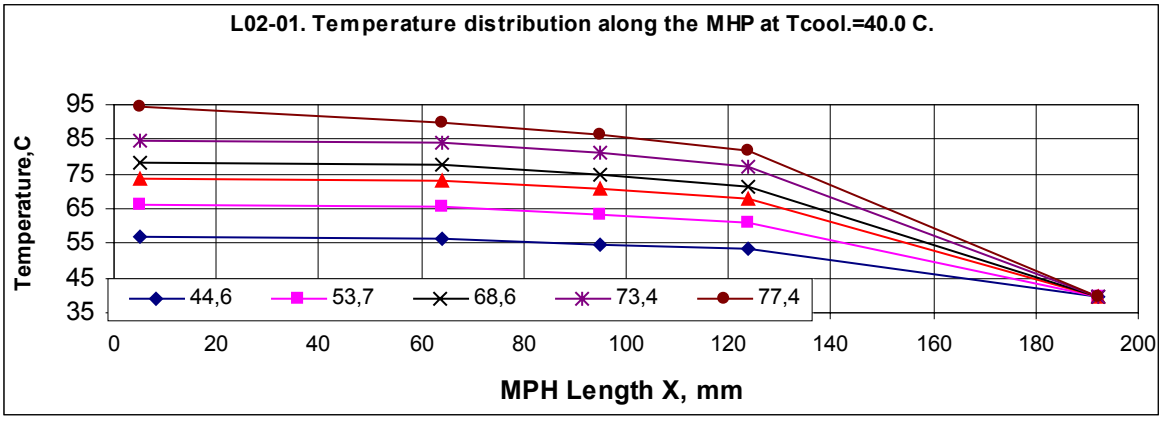

Fig. 6. Temperature distribution along the LHMTI MHP with 6 mm in outer diameter and homogenous porous structure made from 150-250 μ dendridic shape copper powder.

As one can see from Fig. 6, 9 the temperature rise in the evaporator zone of the MHP with homogenous wick structure was proportional to the evaporator heat load rise. Meanwhile for the MHP with bi-porous wick the situation was completely different. The temperature of the evaporator was near constant, with increasing of heat load. It was so, till the Q_{max} was achieved. When the heat load in

the evaporator exceeded the first critical value, the temperature started to rise sharply from its end, (Figs. 7, 9). The similar situation was indicated for AVC MHP too (Figs. 8–9).

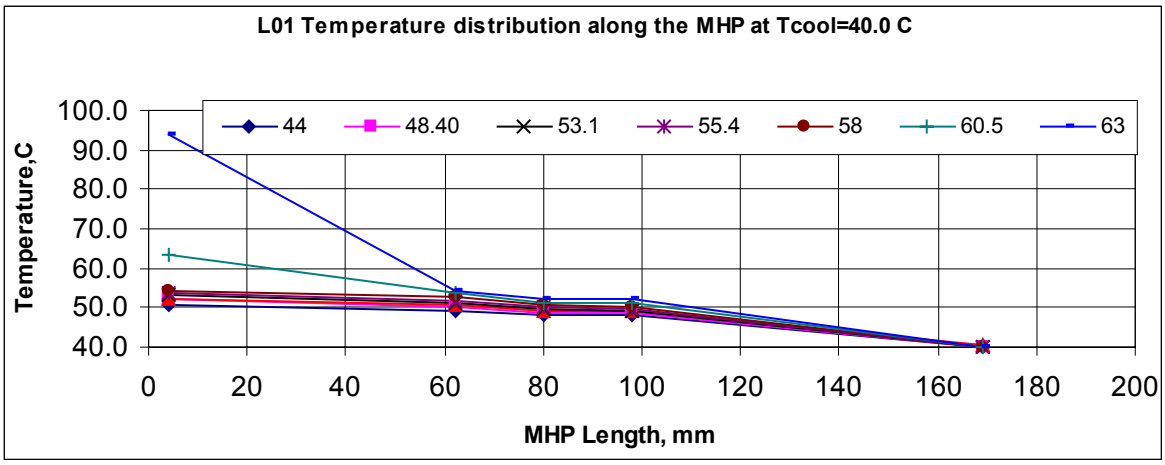

Fig. 7. Temperature distribution along the LHMTI MHP with 5 mm in outer diameter and bi- porous structure made from 150-250 μ dendrite shape copper powder

The main difference between MHP's with homogenous (uniform) porous structure and MHP's with bi-porous structure is the ratio of macro and micro pores in the wick. When the wick has homogenous pores the volumetric zone of two phase heat transfer in the evaporator is thin (micro pores are dominant) and the surface of the meniscus of the evaporation is limited. This surface of the evaporation (limited number of the meniscus) is roughly the same for different heat loads. When the heat flux is increasing the temperature of the evaporator is also increasing (Fig. 6). Therefore the heat transfer coefficient in the evaporator is near constant.

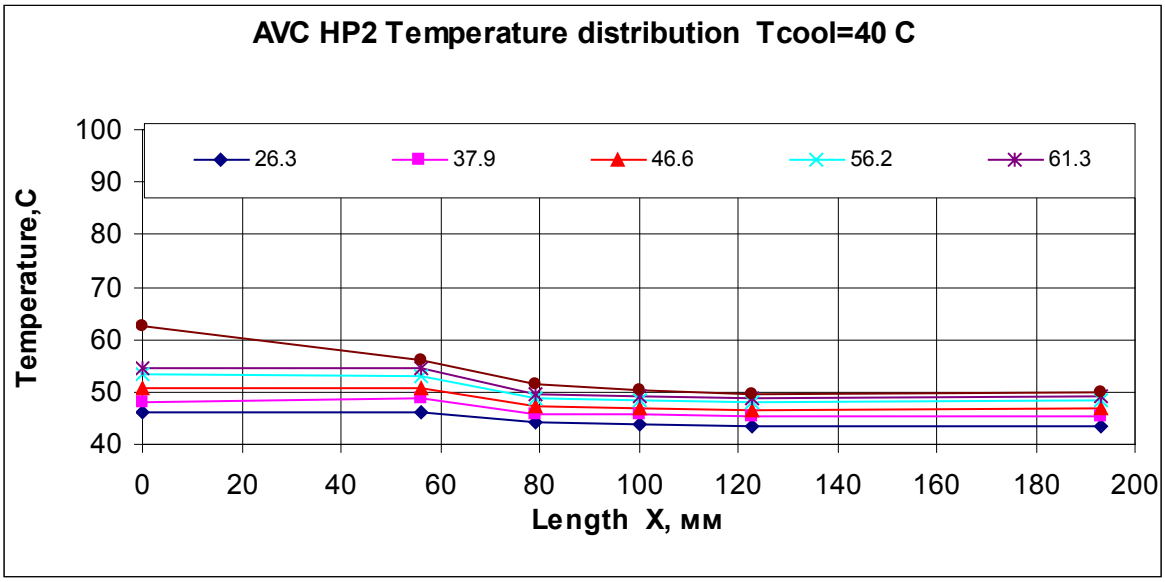

Fig. 8. Temperature distribution along the AVC MHP with 6 mm in outer diameter

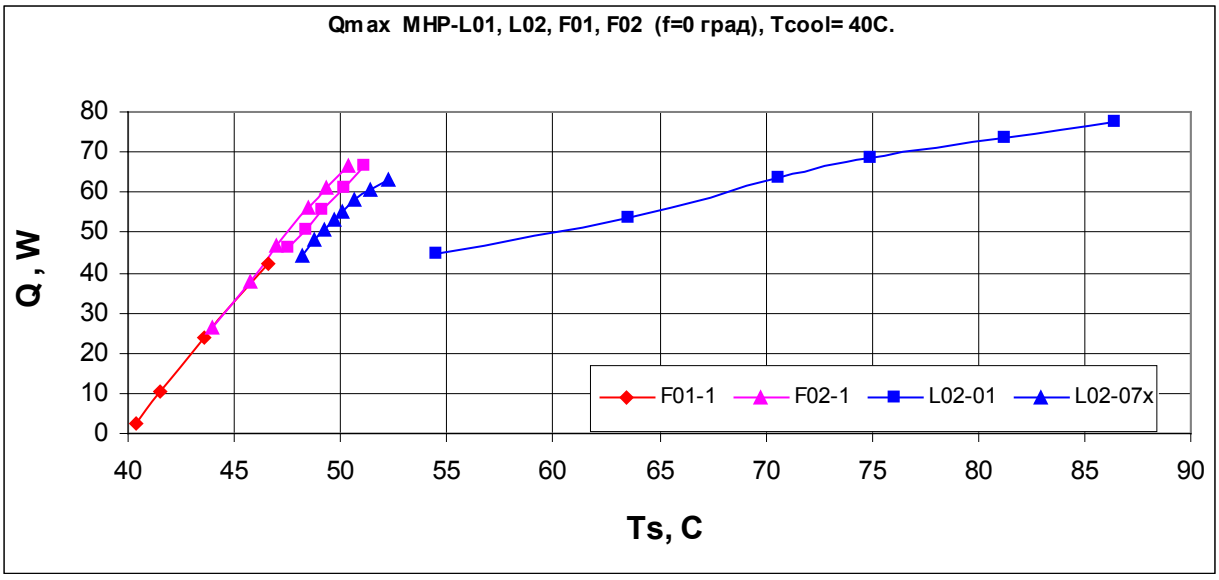

Fig. 9. Critical heat flux vs. vapor temperature in the MHPs adiabatic zone for the tested F01-1; F02-1 and L02-01 6mm in outer diameter , L02-07 5mm in outer diameter MHPs (F01 and F02 – AVC MHPs; L02-01 and L02-07 LHMTI MHPs)

For bi-porous wick the ratio between macro and micro pores is different. There is a constant ratio between micro and macro pores (for one macropore there are some micropores, depending on the wick thickness), Fig. 10. The number of active meniscus of the evaporation depends proportionally on the heat flux number. The heat flux is equal to N (the number of the meniscus), multiplied on latent heat of the evaporation $h_{lv,}$ α (local heat transfer coefficient) and the surface S of evaporation, for ΔT_{evap} = constant.

$$Q = N_{menuscus}\ h_{lv}\ \alpha\ S\ \Delta T\ . \qquad (10)$$

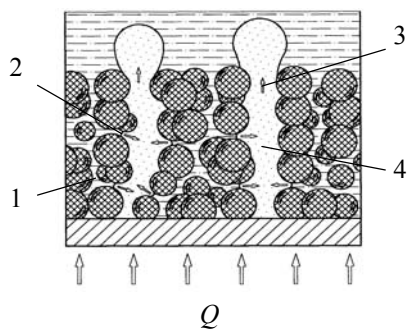

Fig. 10. Schematic of the copper sintered powder porous structure. 1 – micro-pore, 2 – meniscus, 3 – macro-pore, 4 – vapor bubble

It means that when the evaporator heat load is increasing the number of the meniscus of the interface micro pores and macropore (vapor channel) is increasing, but the temperature of the evaporation T_{sat} remains constant.

CONCLUSIONS

1. Three different, most attractive for the CPU cooling system, sizes of MHPs, 4mm, 5mm and 6mm outer diameters and 200mm lengths, have been developed and tested. With help of R&D MHPs in this study is possible successfully dissipate up to 80 W for 6mm and 60 W for 5mm single tubes at the MHP adiabatic temperature below 90 ^{0} C with total thermal resistance R_{tMHP} not more then 0,2 C/W .
2. It was found that sizes and shapes of raw copper particles played an essential role in final wick structural characteristics. By applying the copper particles with dendrite form the final wick permeability was increased in 4 times compare with wick made from the same size but round shape copper particles.
3. Bi-porous wick concept was suggested in this study. The pilot sample of the MHP with 5mm in outer diameter were designed, fabricated and tested. It was shown that MHP with bi-porous wick has an advantages to compare with the homogenous one especially when the temperature of the MHP should be not exceed 70 ^{0} C.

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