Available online at www.sciencedirect.com
ScienceDirect
Applied Thermal Engineering 28 (2008) 266–273
Applied Thermal Engineering
www.elsevier.com/locate/apthermeng

Micro and miniature heat pipes – Electronic component coolers

L.L. Vasiliev *

Luikov Heat and Mass Transfer Institute, National Academy of Sciences, P. Brovka 15, 220072 Minsk, Belarus

Received 15 December 2005; accepted 12 February 2006
Available online 17 April 2006

Abstract

The time of beginning of heat pipe science was near 40 years ago with first heat pipe definition and prediction of most simple cases. Micro and miniature heat pipes have received considerable attention in the past decade. The interest stems from the possibility of achieving the extremely high heat fluxes near 1000 W/cm^2, needed for future generation electronics cooling application. Now at the computer age some changes of basic equations are performed, more powerful predicting methods are available with increasing awareness of the complexity of heat pipes and new heat pipe generations. But even today heat pipes are still not completely understood and solution strategies still contain significant simplifications. Micro and miniature heat pipes have some additional complications due to its small size. A short review on the micro and miniature heat pipes is presented.

Keywords: Micro heat pipes; Miniature heat pipes; Heat transfer; Phase change; Evaporation; Condensation; Confined space

1. Introduction

Micro (MHP) and miniature heat pipes (mHP) [1] are small scale devices that are used to cool microelectronic chips. Microchannels in MHP are fluid flow channels with small hydraulic diameters. The hydraulic diameter of MHPs is on the order of 10–500 μm, the hydraulic diameter of mHPs is on the order 2–4 mm. Smaller channels application is desirable because of two reasons: (i) higher heat transfer coefficient, and (ii) higher heat transfer surface area per unit flow volume. Actually new cooling techniques are being attempted to dissipate fluxes in electronic components in order of 100 up to 1000 W/cm^2. Recently high-performance miniature heat pipe panels were designed and manufactured in the Luikov Institute, Belarus, Fig. 1.

Besides electronic cooling, there are many other applications, where MHPs may be useful. For example, MHPs are interesting to be used in implanted neural stimulators, sensors and pumps, electronic wrist watches, active transponders, self-powered temperature displays, temperature warning systems.

MHPs are promising to cool and heat some biological micro-objects. So there is a real necessity to improve heat pipe parameters. In heat pipes basic phenomena and equations are related with liquid–vapor interface, heat transport between the outside and the interface (“radial” heat transfer), vapor flow and liquid flow. There is a strong interaction between basic phenomena in heat pipes, Fig. 2. Feedbacks may cause instabilities, such as waves, flooding, performance jumps. Following Busse [2] basic equations are related to vapor flow in the MHP channel, liquid flow in the capillary structure, interface position between the vapor and liquid (mechanical equilibrium yields interface curvature K), radial heat transfer, vapor flow limit, capillary limit. MHPs and mHPs are sensitive to the surplus liquid inside. Surplus liquid tends to be accumulated at the wet point defined by $K = K_{\min}$. Sometimes the wet point is disposed not at the end of the heat pipe and there could be deterioration of the radial heat transfer coefficient. Interface instability is the reason of the liquid accumulation in the condenser and leads to dry-out spots arrival in the evaporator. While traditionally heat pipe

* Fax: +7 375 17 284 21 33.
E-mail addresses: lvasil@hmti.ac.by, lvasil@ns1.hmti.ac.by

1359-4311/$ - see front matter © 2006 Elsevier Ltd. All rights reserved.
doi:10.1016/j.applthermaleng.2006.02.023

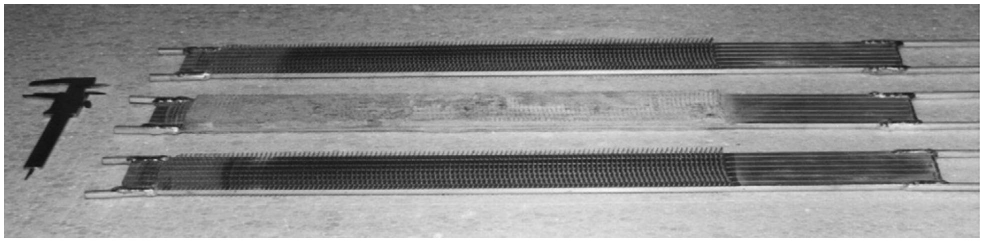

Fig. 1. Flat aluminium heat pipe panels for semiconductor component cooling.

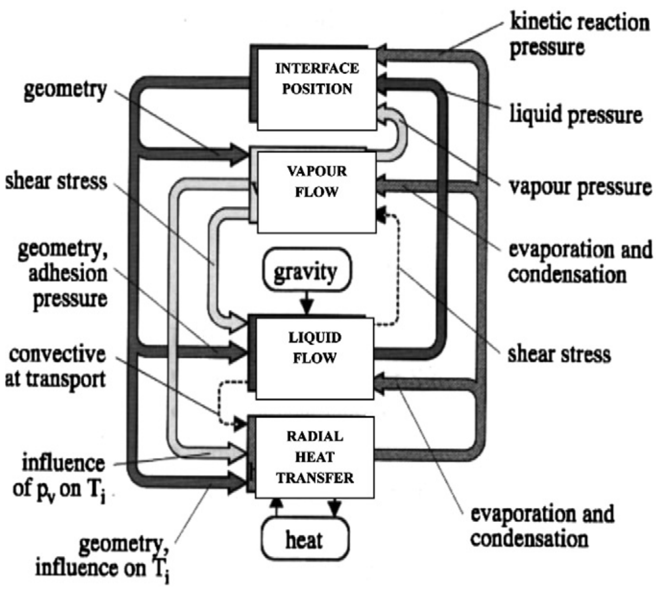

Fig. 2. Interaction between basic phenomena in heat pipes.

condenser resistance is seemed small and often neglected in MHP/mHP the detailed tests revealed substantial temperature drop across the length of the condenser. Potential sources of this temperature drop may be non-condensable gases, the surplus liquid and constrained vapor space. The resulting change of the vapor stress on the interface tends to increase the deformation of the interface.

2. Micro heat pipes

Micro heat pipe phenomena is often available in nature. For example, there is an analogy between micro heat pipe operation and functioning of a sweat gland [3]. Open-type mini/micro heat pipes are suggested in [4,5], as a system of thermal control of biological objects and drying technology. Some theoretical models capable to predict the effects of the thin film region on the evaporating and condensing heat transfer have been developed, particularly for triangular and trapezoidal-grooved MHPs, in order to determine the maximum evaporation heat transfer through the thin film region [9–11]. The detailed theoretical analysis of capillary flow, the heat transfer in the condenser, evaporator and macro region (Fig. 3) is presented in [10,12]. In all above mentioned references related to MHP 1D theoretical analysis is available with emphasizes on one microchannel hydrodynamic and heat transfer: Most of investigators focus on the capillary heat transport capability because the fundamental phenomena that govern the operation of MHPs, arise from the difference in the capillary pressure across the liquid–vapor interface in the evaporator and condenser zones. The experimental data on silicon micro heat pipe arrays filled with methanol or water were published in [11]. Recently a review paper on MHP/mHP for the cooling of electronic devices was published in [12]. Some new MHP designs are presented in the literature mostly related with an increasing of the surface of the evaporation and condensation and vapor pressure drops decreasing in the vapor channels (heat pipe spreaders, flat plate micro heat pipes, etc.) [13]. Analysis of the applicability of different grooved MHPs shows, that there are some advantages of this heat pipe design (simple geometry of the microchannel, low cost of fabrication, using etching technology in silicon chip) and drawbacks such as sensitivity to the presence of non-condensable gases in the vapor channel, the strong liquid–vapor interfacial shear stress,

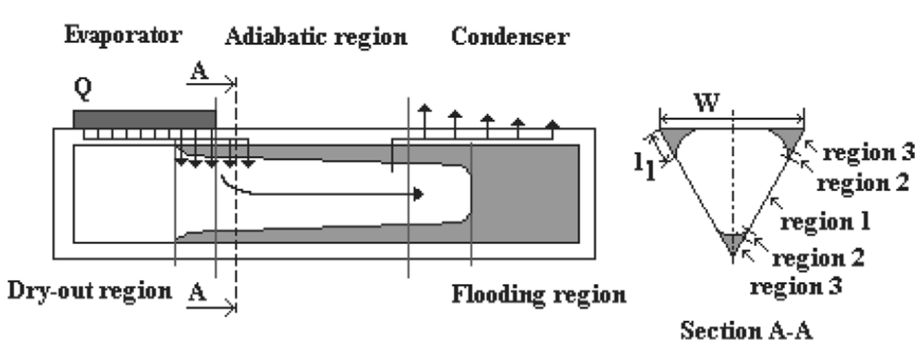

Fig. 3. Micro heat pipe with triangular capillary microchannels [12].

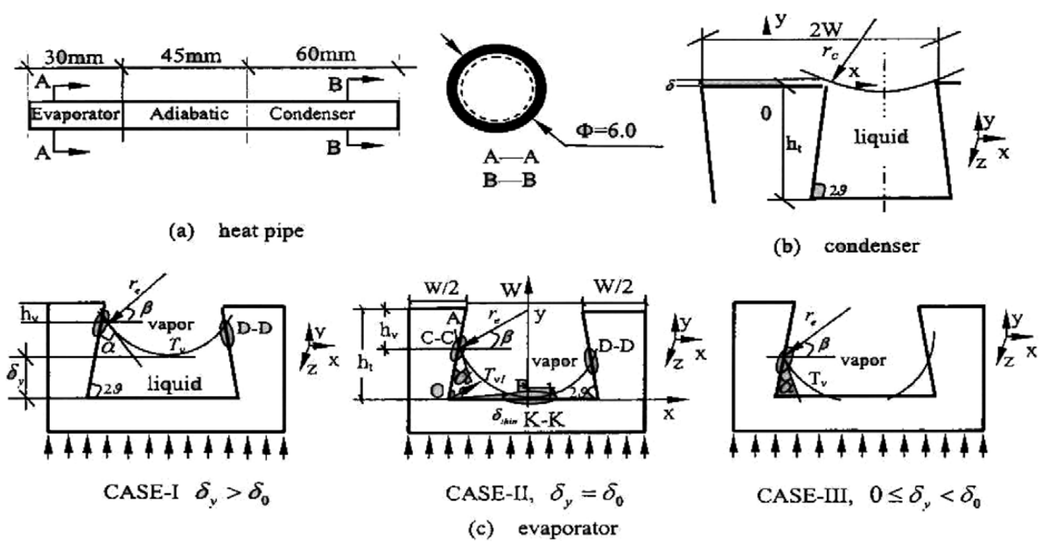

Fig. 4. Schematic of trapezoidal grooves (evaporator, adiabatic zone and condenser) for different liquid flow profiles and thin film region distributions [10].

dry-out effects with liquid accumulation in the vapor channel and hot spots arrival, low heat transfer output due to the low surface of the evaporation. More sophisticated trapezoidal grooves give the possibility to enhance heat transfer in MHP [10] and increase the surface of heat transfer to compare with triangular grooves (Fig. 4).

2.1. Micro heat pipe with sintered powder wick inside

Some new possibilities to enhance heat and mass transfer in evaporators of MHP, mHP, mini heat pumps and refrigerators covered by capillary–porous coatings of heat releasing components are based on micro heat pipe phenomena [6]. The most efficient improvement of the MHP/mHP parameters can be obtained, if the surface of the evaporation and condensation zones would be dramatically increased, applying capillary–porous coating. An example of such heat transfer enhancement by increasing the surface of heat transfer is the design of capillary–porous MHP element (Figs. 5–7), made from the copper sintered powder. The heat transfer enhancement 3–4 times more is achieved to compare with grooved surface heat transfer. For example, for copper sintered powder structure disposed on the surface of horizontal copper tube and propane as a working fluid the evaporative heat transfer coefficient is 8 times as high as boiling heat transfer coefficient on the same diameter smooth tube at heat flux up to $q = 10^4$ W/m^2, and 6 times at $q > 10^4$ W/m^2. In these micro/mini-evaporators the liquid evaporation mostly is realized near the interline and intrinsic meniscus region on the micropore outlets. A liquid is supplied to zones of vaporization by capillary force. The vapor is generated on the annular surfaces of meniscuses in orifices of micropores. This vapor goes out through macropores (Fig. 5), served as vapor channel. The heat transfer is similar to the heat transfer in the evaporator of micro heat pipe. In

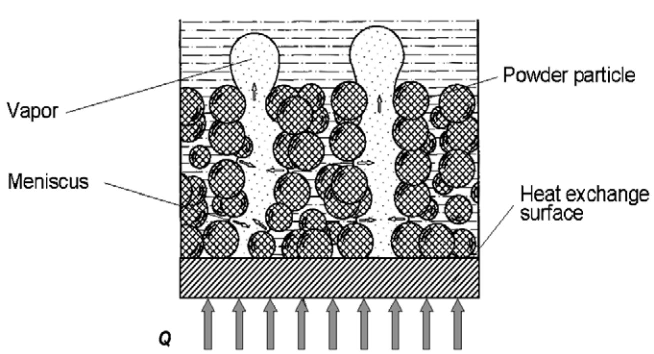

Fig. 5. Micro heat pipe phenomena available in the capillary–porous structure, the element of the MHP evaporator [4–6].

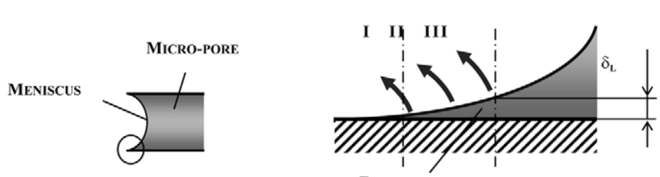

Fig. 6. Schematic of the micropore and the three zones of the evaporation of the meniscus.

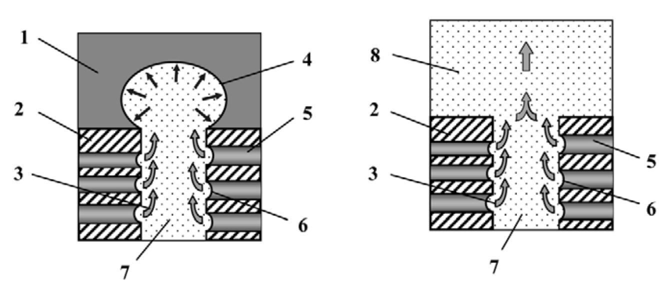

Fig. 7. The elementary sintered powder MHP evaporator unit containing some micropores connecting with one macropore.

contrast to the conventional MHPs with polygon, triangular, trapezoidal-grooved capillary system, MHPs with micro and macropores have a complex shape. The elementary microevaporator is considered as near cylindrical with diameter of the order of some microns, but the number of such evaporators inside MHP porous structure (some micropores connecting with one macropore) is many times more to compare with conventional MHPs. It means the total surface of the evaporation is many time more also. The three zones of meniscus profile inside the micropore outlet to macropore includes zone 1 – non-evaporating region, where disjoining pressure is dominant; zone II (transient zone), where disjoining pressure is in the same order as the capillary pressure and the zone III (meniscus region), where capillary force is dominant.

This profile is almost stable and the curvature of the interface is equal $-K = 2/r_{mp}$, where r_{mp} is hydraulic radius of micropore. The most intensive evaporation rate (70–80%) occurs in zones II and III with mean thickness δ_l. The macropore in such wick of the MHP evaporator can be considered as macropore of closed type, when the macropore outlet has a bubble shape in the liquid pool (condenser). There is a macropore open type, when the vapor on the macropore outlet is going to the MHP condenser. The unit of the micro evaporator, Fig. 7, is placed into the liquid pool 1. This unit has a heat loaded solid porous material 2, which has some micropores 5 and macropore 7 inside. Micropores 5 are filled with the liquid. The liquid is sucking from the pool. The meniscus of the interface liquid–vapor is a capillary pump situated between the micropore 5 outlets and the macropore 7 entrance. During the evaporation the vapor flow 3 leaving from the part of the meniscus is going to the macropore, which serves as a vapor channel. The vapor bubble 5 is used as a condenser. Such micro heat pipe system is considered as a variable conduction micro heat pipe due to the pressure constancy inside and variable geometry of the bubble type condenser.

For the case of the open type micro heat pipe the liquid pool 1 is situated outside of the heat transfer system. The pool is connected with the porous wick by the liquid pipe. The vapor flow 8 from the macropore outlet is moving to the condenser disposed also outside of the heat loaded part of system.

This model of heat transfer is very useful to analyze the drying phenomena of different capillary porous materials. It is known, that, during the constant rate and falling rate of drying procedure there is a certain zone inside of porous media of intensive heat transfer with evaporation. This is a zone, where micropores saturated with liquid are contacting with macropores served as a vapor channels.

2.2. Micro-loop heat pipes

Such types of capillary–porous wicks of MHP evaporators are also applicable for micro-loop heat pipes, Fig. 8. MHPs actually are mostly interesting to be implemented directly in the silicon, or Al_2O_3/Al chip. Compared to other materials, silicon provides several advantages. It has a good heat conductivity (150 W/(m K)), and permits to obtain much smaller devices than other metals because of the etching process accuracy. Moreover, as the MHD can be machined in the core of the chip, the thermo-mechanical constraints are lower compared to other materials. Its thermal expansion coefficient is 7 times lower than that of copper and 10 times lower than that of aluminium. In the design of MHPs a number of heat transfer limitations should be taken in consideration [7,8].

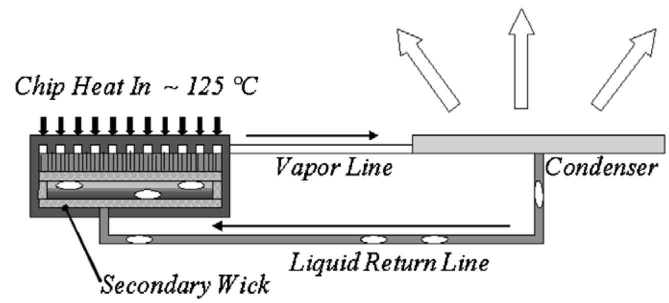

Fig. 8. Micro-loop heat pipe – the electronic microchip cooler [14].

2.3. Sorption micro/mini heat pipe

Sorption micro heat pipe includes the advantages of conventional heat pipes and sorption machines in one unit. The major advantage is its ability to ensure the convective two-phase heat transfer through capillary–porous wick under the pressure drop due to sorbent action inside the heat pipe. In the sorption heat pipe the same working fluid is used as a sorbate and as a heat transfer media. The sorption heat pipe includes some basic phenomena interacting with each other: (1) in the sorbent bed there is a vapor flow (two phase flow) with kinetic reaction rate and pressure, vapor pressure, geometry, conductive and convective heat transport with radial heat transfer; (2) in the condenser and evaporator there is a vapor flow, liquid flow, interface position, radial heat transfer with kinetic reaction pressure, liquid pressure, vapor pressure, condensation and evaporation, shear stress, geometry, adhesion pressure, convective heat transport, radial heat transfer under the influence of the gravity field. Very important feature – cryogenic sorption heat pipe (hydrogen, oxygen, and nitrogen) has no needs to be protected against super pressure influence at room temperatures, because the pressure inside is regulated by the sorption structure and basically is low (Fig. 9).

For loop heat pipe the maximum pressure rise due to the surface tension effects in the wick can be evaluated by Laplace equation:

$$(p_c)_{max} = 2\sigma/r_c, \tag{1}$$

where σ is a surface tension of working fluid and r_c, the effective capillary radius of the wick.

In the real LHP design capillary pressure drop ΔP_c depends on some LHP parameters and need to be

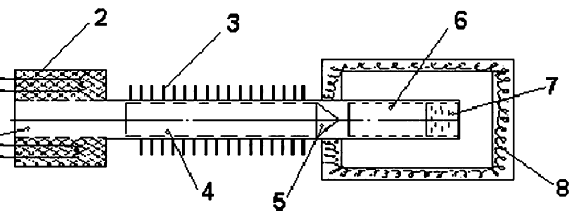

Fig. 9. Micro/mini-sorption heat pipe: 1 – sorption canister; 2 – sorbent material; 3 – micro-fins; 4 – evaporator; 5 – porous valve; 6 – second evaporator/condenser; 7 – liquid accumulator; 8 – thermal insulation.

$$\Delta P_c \geqslant \Delta P_v + \Delta P_l + \Delta P_w + \Delta P_g, \tag{2}$$

where ΔP_v and ΔP_l are the pressure drop in the vapor and liquid lines; ΔP_w, the pressure drop in the wick pores and $\Delta P_g = \rho_l g L_{eff} \sin\theta$, pressure drop due to the gravity field action. In real devices this pressure head is less 1 b.

For sorption heat pipe the maximum pressure rise is determined by the vapor pressure difference in the evaporator and adsorber following the Clausius–Clapeyron equation:

$$dLnP/d(1/T) = -L/R, \quad \text{or} \; -\Delta H/R. \tag{3}$$

For such fluids as ammonia the pressure drop in the sorption heat pipe could be near 10 b, it is 10 times more to compare with conventional heat pipe.

3. Miniature heat pipes

Actually a lot of miniature heat pipes with different wick structures are fabricated by some leading companies. Miniature heat pipes are promising in the applications in fuel cells thermal control and electronic components cooling. The typical example of mHP application for the electronic components cooling is shown in Fig. 10 [15]. Let us consider mHP with diameter 4 mm and the length 200 mm designed in the Luikov Institute. The maximum heat transport capability of the mHP is governed by several limiting factors which ought to be considered when designing a heat pipe. There are five primary heat pipe heat transport limitations: viscous, sonic, capillary pumping, entrainment or flooding and boiling. For the low temperature heat pipes (for example, miniature copper/water heat pipe) the most important are capillary pumping and boiling limits. In some cases the flooding limit (in condenser zone) is also important. Two main properties of the wick are the pore size and the permeability. The pore size (radius) determines the fluid pumping pressure (capillary head) of the wick. The permeability is responsible for frictional losses of the fluid as it flows through the wick. Actually there are several types of the wick structures available including metal sintered powder; fine fiber bundle, axially grooves, screen mesh.

Metal sintered powder wicks have a high fluid pumping pressure (mHP can work against gravity field), low effective thermal resistance (high effective thermal conductivity), can be partially dried (still working efficiently), boiling crisis is smooth between Q_{max1} and Q_{max2}, but have low fluid permeability (the pressure losses are relatively high). Have a good reliability (wettability) after the crisis phenomena in the evaporator.

Grooves as a wick have a large pore radius and high permeability (the pressure losses are low), but its pumping head (fluid pumping pressure) is low (mHP cannot be used against the gravity field), it can't be functioning with partially dried evaporator zone. Boiling crisis is sharp at Q_{max}, grooves structures have a bed reliability (wettability) after the crisis phenomena (dry-out).

Fine fiber bundle wicks have a good capillary pumping head, but have low permeability and high effective thermal resistance (low thermal conductivity) across the wick. They have the low heat transfer coefficient between the mHP envelope and the wick. They cannot be used with partial

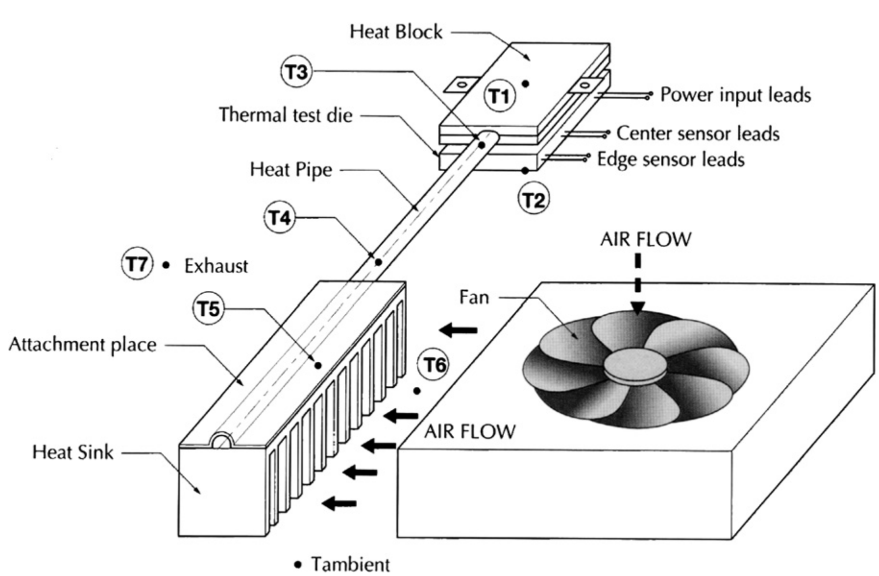

Fig. 10. mHP application for the electronic components cooling, made by Fujikura Ltd. [15].

wick drying; the boiling crisis (dry-out) is sharp. They have the bad wettability after the evaporator dry-out.

Screen mesh wicks have a moderate capillary pumping head, but have low permeability and high effective thermal resistance. They do not have so good wettability after the evaporator dry-out. The thermal and hydraulic parameter of the wick is determined through the experimental measurement of

1. capillary height (through which the equivalent porous radius can be evaluated);
2. liquid hydraulic head (through which the liquid pressure drop in the wick is determined);
3. wick permeability (found from the hydraulic head – Darcy's law);
4. heat flux;
5. wick mass flow rate (to calculate the wick two-phase pressure drop);
6. wick porosity (to determine the thermal conductivity of the wick saturated with liquid).

Comparative analysis of flattened mHPs available in the market with the same dimensions, but different wick structures such as copper sintered powder, wire bundle and screen, Fig. 11, testifies that for water as a working fluid the copper sintered powder wick is the most effective with the point of view of Q_{max} transfer. The effective thermal resistance (or thermal conductivity) of heat pipe is one of the important parameters and is not constant but a function of a large number of variables, such as heat pipe geometry, evaporator and condenser length, wick structure and working fluid. The total thermal resistance of a heat pipe is the sum of resistances due to conduction through the wall (heat pipe envelope) and the wick, evaporation or boiling, axial vapor flow, condensation, and conduction losses through the wick in the condenser and heat pipe wall. The detailed thermal analysis of different heat pipes is rather complicated, but now, following the data of Fig. 11 it is clear, that a heat pipe with a metal sintered powder wick is the most efficient in its function in any position of heat pipe in the gravity field with good heat input capability. Sintered powder wick because of the close particle to particle spacing, generate very high pumping capabilities as compared to more conventional grooves or mesh screen wicks.

The heat flux depends on the distance between the condenser and evaporator zone, the wall superheat and the liquid subcooling, the thermal contact between the heater and wick and the superficial boundary conditions of the wick.

Let us consider the miniature cylindrical heat pipe with the length l, the condenser length l_c, evaporator length l_e, and the transport zone length l_t (effective transport zone length l_{ef}). Heat pipe is inclined to the horizon on the angle $\varphi > 0$ (evaporator is disposed above to the condenser), the wick cross-section square is S (heat pipe outer diameter D_p and inner diameter D_{ch}). The sintered powder wick is saturated with liquid. The liquid at the temperature T has a density ρ_l, surface tension σ, dynamic viscosity μ_l and the

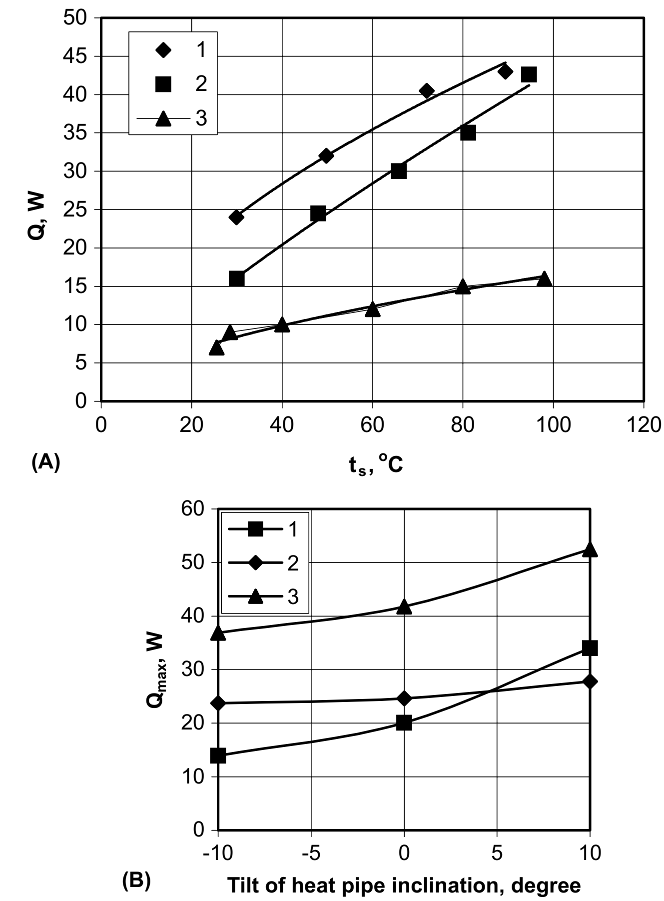

Fig. 11. (A) Q_{max} as a function of heat pipe temperature. (B) Q_{max} as a function of the heat pipe inclination for three flattened mHPs with different wicks: 1 – screen layers, 2 – wire bundle, 3 – copper sintered powder.

latent heat of evaporation L. The vapor has the density ρ_v, viscosity μ_v. The angle of the wick wetting is θ.

The following assumptions are adopted:

1. Wick parameters are constant along the heat pipe.
2. Evaporation of the liquid is on the surface of the evaporator.
3. The heat flux in the evaporator and in the condenser is constant.
4. There is a saturated vapor in the transport zone, and its temperature is T_s.
5. The liquid and the vapor motion is described by the Navier–Stokes set of equations, valid for the non-compressible fluid.
6. No heat sources and heat sinks are available in the vapor media.
7. The liquid movement inside the porous wick is followed by the Darcy law.
8. The friction forces on the vapor–liquid interface in negligibly small to compare with the friction forces inside the wick.
9. The hydrodynamic and heat transfer are considered as 1D model.

Sintered powder wick has an additional advantage over screen wick, it has relatively high thermal conductivity. But sintered powder wick needs to be optimized to ensure the high heat flux performance capability. The problem of the wick structure optimization is related with structural porous wick parameters: the particle size and its form, wick porosity, specific surface of porous wick, pore diameter.

The capillary pressure, which we need to calculate $Q_{\max}$ is equal:

$$\Delta p_c = \Delta p_v + \Delta p_l + \Delta p_g, \tag{4}$$

where Δp_c, Δp_l, Δp_v and Δp_g – pressure drop due to the capillary, liquid, vapor, and gravity forces.

The capillary drop is described by Eq. (4). The liquid pressure drop (Darcy law) is

$$\Delta p_l = \frac{Q\mu_l l_{ef}}{\rho_l L S \xi d_0^v}. \tag{5}$$

The vapor pressure drop is determined by the Poiselle equation:

$$\Delta p_v = \frac{128 Q\mu_v l_{ef}}{\pi D_{ch}^4 \rho_v L}. \tag{6}$$

The gravity pressure drop is equal to

$$\Delta p_g = \rho_l g l \sin\varphi, \tag{7}$$

where g is the gravity constant.

Following this analysis Q depends on two capillary structure parameters – the mean hydraulic pore diameter and the inner diameter of the porous wick. To find the $Q_{\max}$ we need to analyze Eq. (8) for the extreme function finding. Due to the temperature dependence of the thermo-physical properties of the working fluid the maximum heat flow $Q_{\max}$ in the heat pipe transport zone will be different for different saturated vapor temperatures T_{sat}. For different angles of heat pipe inclination to the horizon $Q_{\max}$ must be determined at the worst situation with the point of view of the heat transfer, when the heat pipe evaporator is situated above the heat pipe condenser, vertical (inverted) heat pipe disposition.

$$Q = \frac{\pi L}{4 l_{ef}} \frac{\frac{4\sigma\cos\vartheta}{d_0} - \rho g l \sin\varphi}{\frac{\mu_l}{\rho_l (D_p^2 - D_{ch}^2)\xi d_0^v} + \frac{32\mu_v}{D^4 \rho_v}}. \tag{8}$$

The cylindrical mHP developed in the Luikov Institute, Minsk [16] has dimensions: $L = 200$ mm, $L_e = 70$ mm, $L_c = 85$ mm, $L_a = 45$ mm. Outer diameter $D_p = 4$ mm, mHP wall thickness – 0.2 mm, diameter of the vapor channel $D_{ch} = 2$ mm, size of the copper powder particles – <100 μm, $Q_{\max} = 50$ W. Following the analysis of $Q_{\max}$ for different cylindrical mHPs, Fig. 12, it is clear that copper sintered powder as a wick has some advantages to compare with another wicks in the field of gravity, when the evaporator is disposed above the condenser. Advanced copper sintered powder wick (curve 1, Fig. 12) has some advantages to compare with such wicks as mesh structure, grooves, fiber bundle.

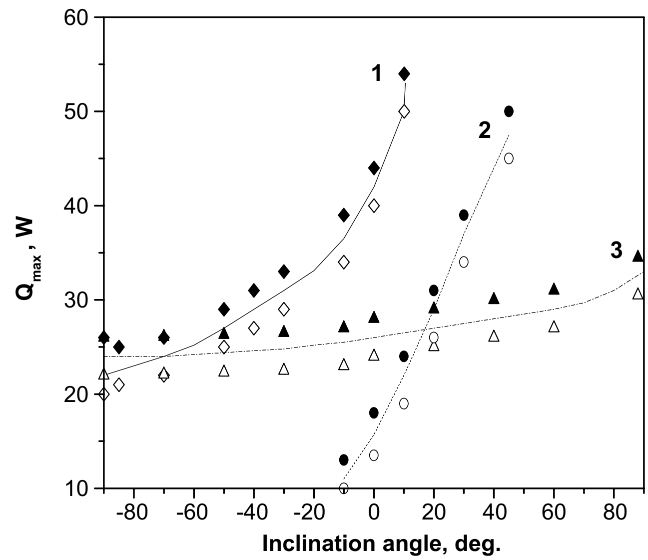

Fig. 12. $Q_{\max}$ as a function of the heat pipe inclination for cylindrical mHP: 1 – advanced mHP with sintered copper powder; 2 – mHP with longitudinal grooves; 3 – conventional mHP with sintered copper powder [16].

4. Conclusion

The existing technologies of MHP/mHP production must be significantly improved in order to face the new challenges in electronic and fuel cells cooling. The heat transfer limit of MHP/mHP ought to be increased by optimizing the geometric and operating parameters. Thermal modeling is a powerful way to predict the performance and the temperature response of MHP/mHP. Unfortunately most of developed thermal models are 1D models and empirical correlations are employed to determine fraction factor of vapor flow.

In order to predict the heat transfer limit and temperature distribution of MHP/mHP, a comprehensive 3D model that includes heat transfer in liquid and vapor must be developed.

In order to find MHP/mHP commercial application in microelectronic cooling it must compete with other cooling methods, such as forced convection, impingement and two phase direct cooling in areas such as manufacturing cost and reliability.

Optimization of copper sintered powder wick in miniature copper/water heat pipes with outer diameter 4 mm and length 200 mm is a good challenge to improve the mHP parameters.

Analysis of the experimental data for a new optimized miniature heat pipe with copper sintered powder wick proves the possibility to use such heat pipes independently of its orientation with the maximum heat transport capability near 50 W.

Theoretical simulation of mHPs with different wick structures (sintered powder, mesh structure, wire bundle) is an efficient tool to perform the comparisons of mHP efficiency. Experimental verification of mHP parameters proves validity of the simulation software.

References

[1] T.M. Cotter, Principles and prospects of micro heat pipes, in: Proceedings of the 5th International Heat Pipe Conference, Tsukuba, Japan, 1984, pp. 328–335.

[2] C.A. Busse. Heat pipe science, in: Proceedings of the 8th International Heat Pipe Conference, Beijing, China, 14–18 September 1992, pp. 3– 8.

[3] P.D. Dunn, D.A. Reay, Heat Pipes, Pergamon Press, 1976, pp. 258–260.

[4] L.L. Vasiliev, Open-type miniature heat pipes, Journal of Engineering Physics and Thermophysics 65 (1) (1993) 625–631.

[5] V.G. Reutskii, L.L. Vasiliev, Doklady Akademii Nauk Minsk 24 (11) (1981) 1033–1036.

[6] L. Vasiliev, A. Zhuravlyov, A. Shapovalov, Comparative analysis of heat transfer efficiency in evaporators of loop thermosyphons and heat pipes, Preprints of the 13th International Heat Pipe Conference, Shanghai, China, 21–25 September 2004, pp. 52–59.

[7] H. Ma, G.P. Peterson, Experimental investigation of the maximum heat transport in triangular grooves, Journal of Heat Transfer 118 (1996) 740–745.

[8] R. Hopkins, A. Faghri, D. Khrustalev, Flat miniature heat pipes with micro capillary grooves, Journal of Heat Transfer 121 (1999) 102–109.

[9] S.J. Kim, J.K. Seo, K.H. Do, Analytical and experimental investigation on the operational characteristics and the thermal optimization of a miniature heat pipe with a grooved wick structure, International Journal of Heat and Mass Transfer 46 (2003) 2051–2063.

[10] A. Jiao, R. Riegler, H. Ma. Groove geometry effects on thin film evaporation and heat transport capability in grooved heat pipe, in: Preprints of the 13th International Heat Pipe Conference, Shanghai, China, 21–25 September 2004, pp. 44–51.

[11] B. Badran, F. Gerner, P. Ramada, T. Henderson, K. Baker, Experimental results for low-temperature silicon micromashined micro heat pipe arrays using water and methanol as working fluids, Experimental Heat Transfer 10 (1997) 253–272.

[12] M. Lallemand, F. Lefevre, Micro/mini heat pipes for the cooling of electronic devices, in: Preprints of the 13th International Heat Pipe Conference, Shanghai, China, 21–25 September, 2004, pp. 12–23.

[13] M. Katsuta, Y. Homma, N. Hosova, T. Shino, J. Sotani, Y. Rimura, Y. Nakamura, Heat transfer characteristics in flat plate micro heat pipe, in: Proceedings of the 7th International Heat Pipes Symposium, Jeju, Korea, 12–16 October 2003, pp. 103–108.

[14] D. Cytrynowicz, M. Hamdan, P. Meddis, A. Shuia, H.T. Henderson, F.M. Gerner, E. Golliher, MEMS loop heat pipe based on coherent porous silicon technology, in: Space Technology & Applications International Forum (STAIF-2002), February 3–6, 2002, Institute for Space and Nuclear Power Studies, University of Mexico.

[15] P. Ektummakij, V. Kumthonkittikun, H. Kuriyama, K. Mashiko, M. Mochizuki, Y. Saito, T. Nguyen, New composite wick heat pipe for cooling personal computers, in: Preprints of the 13th International Heat Pipe Conference, Shanghai, China, 21–25 September 2004, pp. 263–268.

[16] L.L. Vasiliev, A.A. Antukh, V.V. Maziuk, A.G. Kulakov, M.I. Rabetsky, L.L. Vasiliev Jr., O.S. Min, Miniature heat pipes experimental analysis and software development, in: Proceedings of the 12th International Heat Pipe Conference, Moscow, 2002, pp. 329–335.