Miniature Loop Heat Pipes—A Promising Means for Cooling Electronics

Yury F. Maydanik, Sergey V. Vershinin, Mikhail A. Korukov, and Jay M. Ochterbeck

Abstract—Loop heat pipes (LHPs) are highly ef?cient heat-transfer devices, which have considerable advantages over conventional heat pipes. Currently, miniature LHPs (MLHPs) with masses ranging from 10–20 g and ammonia and water as working ?uids have been developed and tested. The MLHPs are capable of transferring heat loads of 100–200 W for distances up to 300 mm in the temperature range 50–100 °C at any orientation in 1-g conditions. The thermal resistance for these conditions are in the range from 0.1 to 0.2 K/W. The devices possess mechanical ?exibility and are adaptable to different conditions of location and operation. Such characteristics of MLHPs open numerous prospects for use in cooling systems of electronics and computer systems.

Index Terms—Computer CPU cooling, heat-transfer device, loop heat pipe (LHP), thermal resistance.

NOMENCLATURE

R	Thermal resistance, K/W.
Q	Heat load (capacity), W.
T	Temperature, K.
ΔT	Temperature difference, K.
S	Surface, cm^2.
	Greek symbols
α	Heat-transfer coef?cient, W/m^2 K.
φ	Angle of inclination, deg.
	Subscripts
e	Evaporator.
c	Condenser.
v	Vapor.
l	Liquid.
a	Ambient.
tot	Total.
cool	Cooling.

I. INTRODUCTION

HEAT pipes, as highly ef?cient heat-transfer devices, have been used very effectively for many years in the cooling systems of electronics, including personal computer equipment [1]–[3]. At the same time, the inherent progress in the electronics ?eld, connected with a considerable increase of the heat dissipated from functional elements, results in new requirements that cannot be satis?ed by conventional heat pipes. These requirements mainly refer to a needed considerable increase in the capacity of heat-transfer devices used in cooling systems coupled with a simultaneous decrease in thermal resistance. To meet these challenging needs, signi?cant promise can be found in the potential of loop heat pipes (LHPs), which possess numerous advantages over conventional heat pipes. Such advantages include:

— a much higher capacity at comparable dimensions;
— ef?cient operation at any orientation in 1-g conditions at heat-transfer distances up to several meters;
— a lower thermal resistance;
— increased ?exibility in packaging, location and operation.

The LHP was invented and developed in the former Soviet Union [4]. The main aim of the initial development was to create a heat-transfer device possessing all the main functional advantages of a conventional heat pipe, but at the same time capable of operating ef?ciently at any orientation in 1-g conditions at heat-transfer distances over 1 m. Later, in 1985, an advanced modi?cation of this device was patented in the USA under the name "Heat Transfer Apparatus" [5]. The current name, LHP, has been used since 1989 [6]. Under this name the device became known outside the former Soviet Union after the ?rst open demonstration of LHPs at the 8th International Heat Pipe Conference, Minsk, Russia, 1990.

At present, most LHPs are used widely in space engineering [7]–[9], where many LHPs are successfully employed in thermoregulation systems of spacecraft constructed in Russia, the USA, Europe, and China. The advantages of LHPs over conventional heat pipes are most evident when dealing with high heat transfer loads over considerable distances, especially if heat transfer is required in an adverse tilt situation (i.e., evaporator above condenser). As an example, an LHP typically may be capable of transferring against gravity 1000 W over a distance of 4.5 m, or similarly 21 m long transferring about 2000W in a horizontal position [10]. In this example, the diameter of the lines along which heat is transferred is 6–8 mm, and the thermal resistance of LHPs does not exceed 0.05 K/W. These technical speci?cations corresponded fairly well to the problems which till recently were regarded as the most typical ones. Therefore, the tendency existed for development of these devices to be directed toward increasing the heat-transfer capacity, which may be viewed as being a product of the maximum heat transfer capacity and the heat-transfer distance. At the same time, this early tendency resulted in the appearance of a stereotypical idea that with a decrease in the LHP length and other characteristic

Manuscript received October 8, 2004; revised March 11, 2005. This work was recommended for publication by Associate Editor R. J. Culham upon evaluation of the reviewers' comments.

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Digital Object Identi?er 10.1109/TCAPT.2005.848487

dimensions, the advantages over conventional heat pipes were largely leveled. Another inaccuracy consisted of the assumption that an appropriate "envelope/structure — ?uid" combination as "copper — water," which is widely used for conventional heat pipes, would be problematic for LHP applications owing to the speci?c features of LHP design and mode of operation.

However, future heat transport requirements arising in connection with the problem of cooling of electronics and computer equipment made it necessary to look for new approaches to expand potential applications of LHPs. Efforts directed to LHP miniaturization and increased ef?ciency yielded results that now make it possible to regard LHPs as a quite promising means in the solution of electronics thermal control systems [11].

At the Institute of Thermal Physics various specimens of miniature loop heat pipes (MLHPs) with a mass of 10–20 g have been developed and tested. They are equipped with cylindrical evaporators 5–6 mm in diameter, transport lines for vapor and liquid with diameters of 2–2.5 mm, and are capable of transferring a heat load over distances up to 300 mm. If the operating conditions do not require MLHPs operation at any gravitational orientation, i.e., are limited to slopes within $\pm 10^{\circ}$ from the horizontal plane, the heat-transfer distance can be considerably increased. At nominal heat loads, which are about 70% of the maximum capacity, the thermal resistance is in the range from 0.1 to 0.2 K/W. The devices also possess mechanical ?exibility and are capable of being easily con?gured and integrated in the cooling system.

II. LHP Design and Operation

LHP operation is based on the same main physical principles that are typical of a conventional heat pipe, including the use of capillary forces for pumping the working ?uid, transportation of heat by the latent heat of evaporation and high heat transfer coef?cients during evaporation and condensation. In an LHP, however, the realization of these processesis organized more ef?ciently. For this statement to be supported, one should examine the schematic diagram of a typical LHP device presented in Fig. 1. The main features that differentiate the layout of an LHP from that of an ordinary heat pipe are separate smooth-walled lines intended for the vapor and liquid ?ows, which have a comparatively small diameter, and local positioning of the wick only in the evaporator, whose active zone is in contact with the heat source.

Such a scheme makes it possible to use very ?ne-pored capillary structures (wicks) with pore sizes of 1–10 microns, which are capable of creating high capillary heads, and porosities of 60–70%. At the same time, the small pore wicks do not result in a high hydraulic resistance, as the distance covered by the liquid ?ow in the capillary-porous material of the wick does not usually exceed a few millimeters, regardless of the evaporator active-zone length. Sintered metal powders of nickel, titanium, stainless steel and copper are most widely used at present for making LHP wicks.

The motion of vapor and liquid in the adiabatic (transport) section of an LHP proceeds along separate lines. Since in this case the vapor and liquid ?ows have no viscous interaction with each other, and the ?ow is along smooth walled tubing, the LHP hydraulic resistance in the transport section is also signi?cantly reduced (and can be insigni?cant in many cases). This makes it possible to use tubes for vapor and liquid with relatively small diameters, where values depending on the device length and required capacity may vary in the range from 1 to 10 mm. Also, the tubing can easily be bent and integrated into any required con?guration.

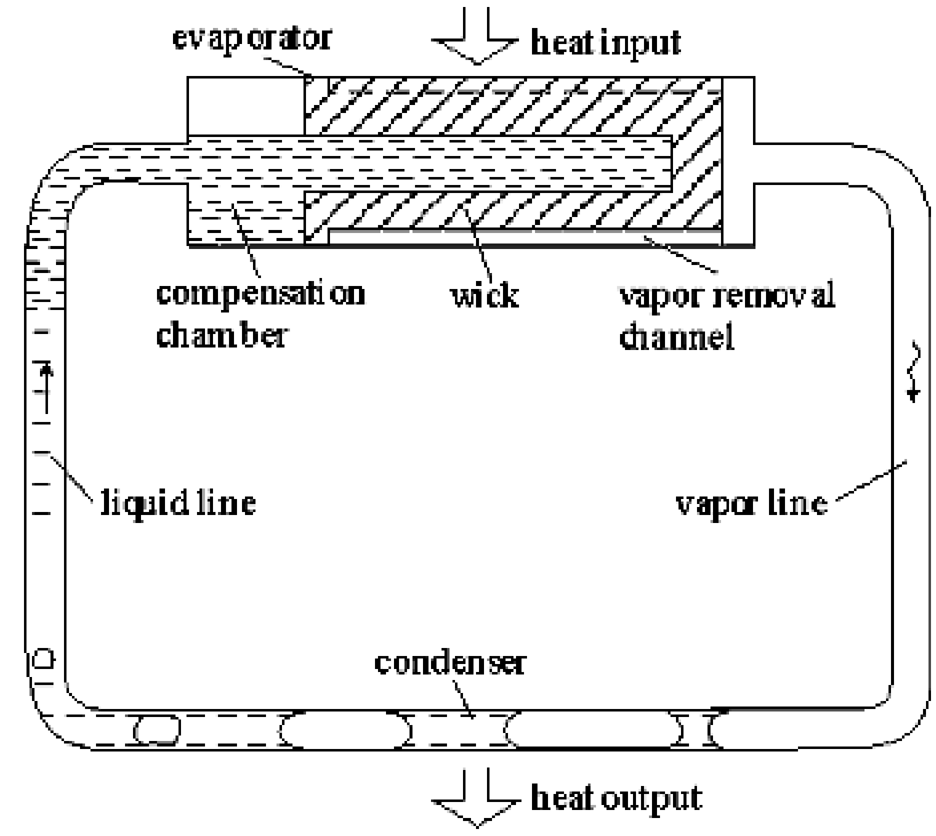

Fig. 1. Principal scheme of an LHP.

Another very important factor that results in a higher ef?ciency of LHPs is the possibility for the evaporation zone to use the "inverted menisci" principle. Such an arrangement of the evaporation zone makes it possible to place the liquid evaporating surface as close to the evaporator heated wall as possible, which considerably decreases its thermal resistance.

As for the condensation process, the LHP ?exibility allows a wide choice of different variants of the condenser design and shape, which corresponds best to the heat removal conditions. Fig. 2 shows the main types of available condensers for LHP use. The condenser of type "a" is the simplest and may be used for compact LHPs with relatively low capacities, where this condenser type usually has external ?ns. The condenser of type "b" is a ?at tubular coil joined to a plate of suf?ciently large dimensions, which may be smooth or ?nned. The condenser of collector type "c" has well-developed inner and outer surfaces and may be used for quite powerful LHPs. The version of the condenser "d" made in the form of a concentric tube heat exchanger is compact and has a comparatively simple design. It is best suited to cases when the means of cooling is the forced convection of a liquid or gas. In this case both its outer and its inner surface may be cooled. Condensers of types "a" and "c" to a greater extent correspond to the conditions of cooling under forced convection. A condenser of type "b" may be used both for natural and for forced gas convection, and also for radiation heat exchange or heat rejection to a thermal mass or cold plate.

III. Development of Miniature LHPs

Cooling systems for electronics and computer equipment, including portable computers, require compact and ef?cient

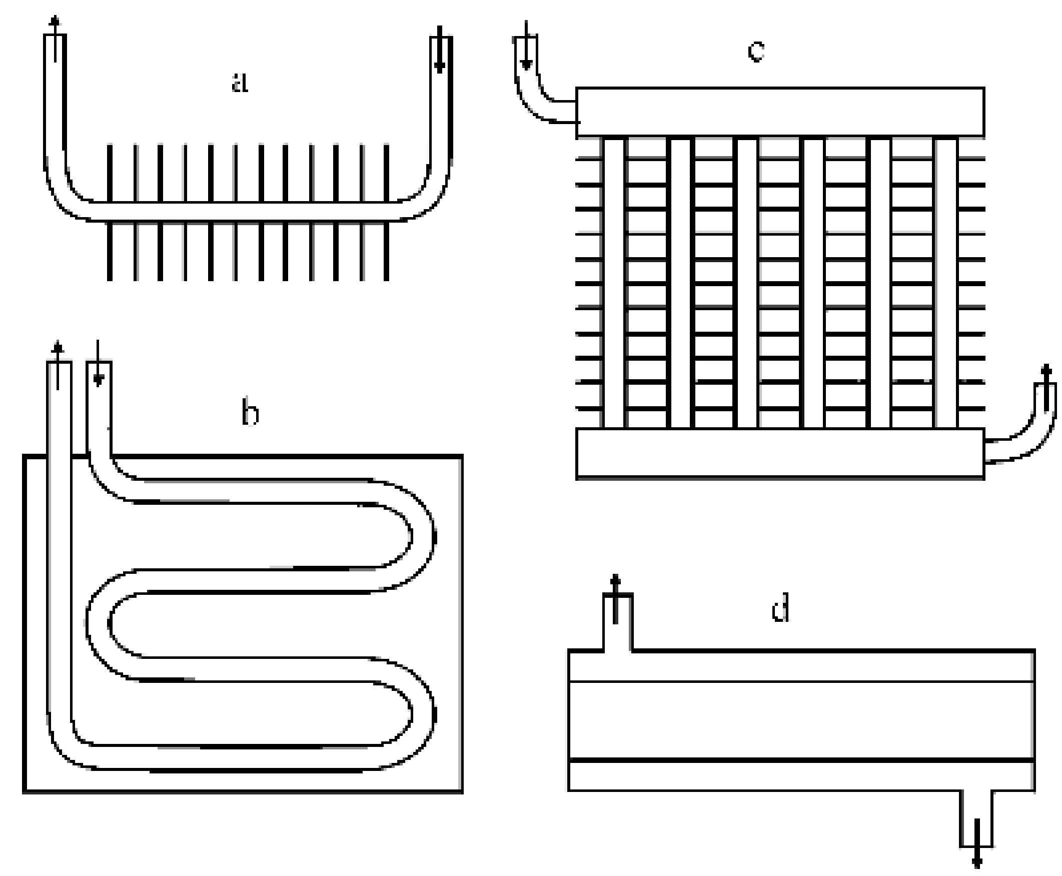

Fig. 2. Different variants of LHP condensers.

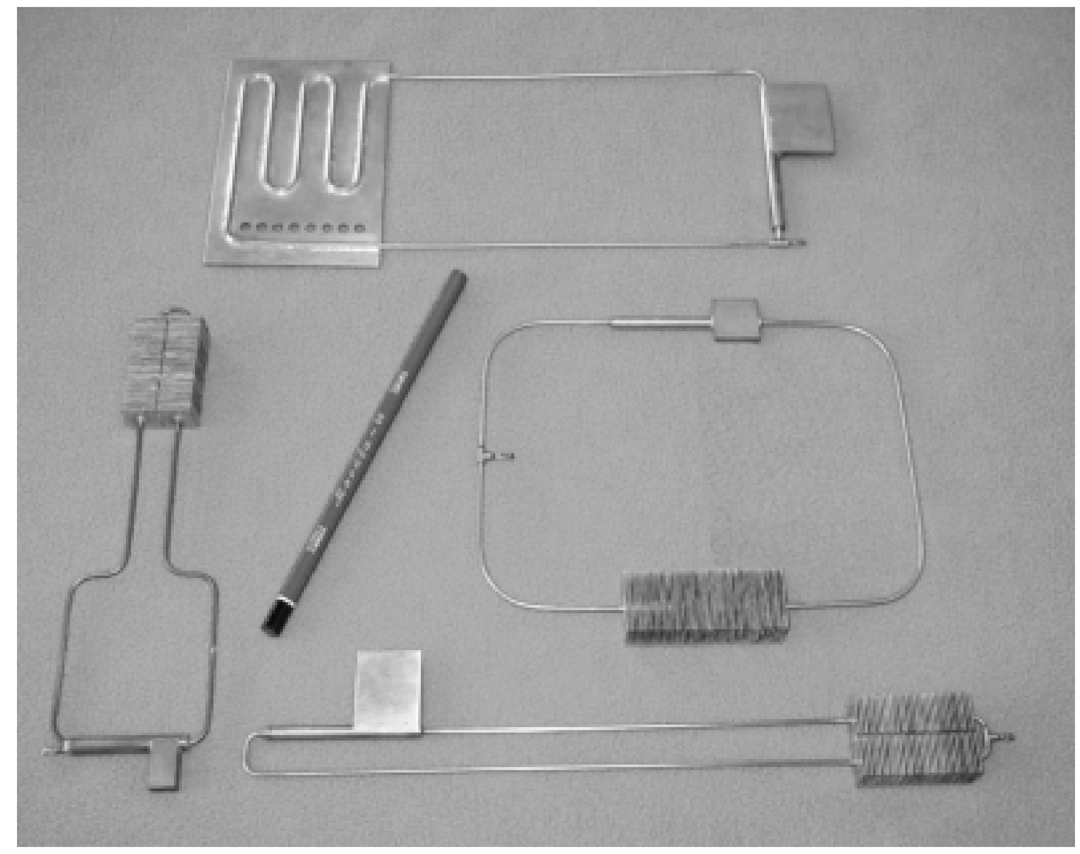

Fig. 3. General view of MLHPs.

heat-transfer devices. Along these lines, LHPs may be regarded as miniature if the diameter of the evaporator does not exceed 8 mm and the diameter of the vapor and the liquid lines are below 3 mm. In this case the length of the evaporator active zone to which a heat load may be applied is usually in the range from 10 to 50 mm, and the total length of the heat-transportation zone is, as a rule, no more than 500 mm. In developing miniature LHPs it is necessary to solve two important problems. The ?rst is connected with the necessity of decreasing the evaporator diameter. In this case one can observe the corresponding decrease of the thickness of the wick separating the absorbing and evaporating surfaces. Such a decrease is accompanied by an increase in parasitic heat ?ows through the wick. A negative consequence of parasitic heat leaks is the increase of the operating temperature and the minimal value of the start-up heat load.

The other problem concerns the decrease of thermal resistance. The reason becomes clear from analysis of the thermal resistance for a heat pipe, which takes the form

$$R = \frac{\Delta T}{Q} = \frac{1}{\alpha_e S_e} + \frac{1}{\alpha_c S_c} \tag{1}$$

where the ?rst component determines the thermal resistance of the evaporator, the second that of the condenser. The active-zone area S_e of a miniature evaporator should correspond to the dimensions of the heat-load source, and therefore is quite limited. The active surface of the condenser S_C, which corresponds to the conditions of heat removal, may be much larger. When it is considered that the value of the heat-transfer coef?cient during condensation α_C is suf?ciently large, the contribution of the condenser component, whose value is usually within 0.05 K/W, to the MLHP thermal resistance is not prevalent. Therefore, the only means of decreasing the thermal resistance is the greatest possible increase of the intensity of heat exchange in the evaporator determined by the value of α_e.

Compact heat-transfer devices with a capacity of 100–150 W and a thermal resistance of about 0.1 K/W will be required in the near future. It means that for the evaporators of MLHPs with an active-zone surface of 2–4 cm^2 the value of ae has to on the order of 50000–100 000 W/m^2 K. At ?rst glance, obtaining such operating characteristics would be seen as extremely dif?cult. However, results of MLHPs tests conducted at the Institute of Thermal Physics and Clemson University make it possible to show that this required range is indeed obtainable.

IV. Test Specimens and Methods

More than ten different variants of MLHPs with cylindrical evaporators have been manufactured and tested. The external view of some experimental specimens is given in Fig. 3. Table 1 gives the main design parameters of two MLHPs, which demonstrated some of the better operating characteristics during evaluation.

The evaporators of the MLHPs were equipped with rectangular interfaces made of copper, whose ?at surface measured 20 × 20 mm and was intended for the mounting a heat-load simulator. A copper cylinder with a resistance heater at the side surface and a rectangular face, with dimensions corresponding to the interface surface, was used as the heat load simulator.

The condensers had ?ns measuring 15 × 25 mm, with thicknesses varying from 0.3 to 0.4 mm. The condenser cooling was accomplished by air ?ow with a temperature of $22 \pm 2\,^\circ$C and circulated by a fan. The volumetric ?ow rate of air was equal to 0.64 m^3/min. The heat load supplied to the evaporators was measured by an alternating-current wattmeter and increased stepwise with a step of 5–10 W beginning with 1 W. The heat transport capacity was considered to reach a maximum if the temperature at the evaporator surface began to exceed 100 °C.

The temperature was measured at several MLHP points using copper-constantan (T-type) thermocouples with bead diameters of 0.2 mm. The location of the thermocouples is given in Fig. 4. The thermocouple measurements were recorded and processed using a data acquisition unit HP 34970 A.

Calculation of the heat-transfer coef?cient in the evaporator was made by the following relation:

$$\alpha_e = \frac{Q}{S_e \Delta T_e} \tag{2}$$

TABLE I
MAIN DESIGN PARAMETERS OF MLHPS

Parameter	Value	
	MLHP 1	MLHP 2
Evaporator diameter, mm	5	6
Evaporator active-zone length, mm	20	20
Evaporator active-zone surface, cm^2	1.57	1.88
Evaporator interface surface, cm^2	4	4
Vapor line length, mm	180	242
Liquid line length, mm	230	217
Vapor / Liquid lines diameter, mm	2	2,5
Condenser length, mm	62	62
Condenser fins surface, cm^2	375	412.5
Body material	SS	copper
Wick material	titanium	copper
Working fluid	ammonia	water

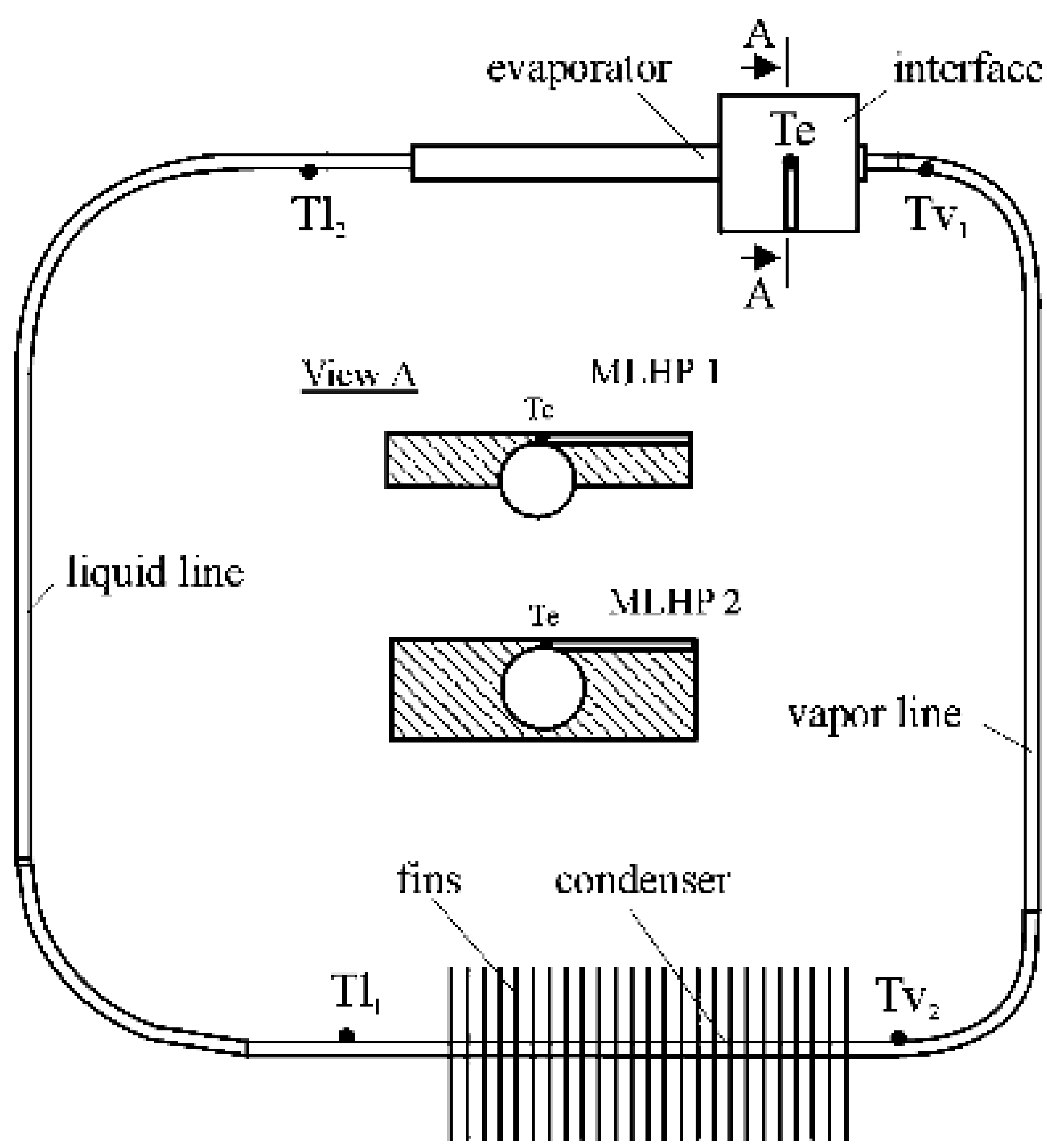

Fig. 4. Thermocouple locations.

where $\Delta T_e = T_e - T_{v1}$. For calculating thermal resistances the following relations were used.

Evaporator thermal resistance

$$R_e = \frac{T_e - T_{v1}}{Q}, \tag{3}$$

MLHP thermal resistance

$$R = \frac{T_e - T_{l1}}{Q}. \tag{4}$$

Total thermal resistance

$$R_{tot} = \frac{T_e - T_a}{Q} \tag{5}$$

It should be noted that usually for calculating the inherent thermal resistance of a heat pipe, R, the average temperature of the evaporator active-zone surface and the average temperature of the condenser active-zone surface are used. In this case the evaporator temperature was measured only at a single point situated at the direct center. For this purpose a special groove was made at the interface surface. Its depth allowed a thermocouple to be in immediate contact with the evaporator wall. The temperature of the condenser surface was not measured, as it was dif?cult to locate thermocouples because of the thin external ?ns. Instead, measurements were made of the temperature at the surface of the liquid line T_{l1} at the exit from the condenser. This substitution is quite accurate, especially at high heat loads when the condenser is fully two-phase, and its average temperature approaches the liquid temperature at the condenser exit.

V. TEST RESULTS

The critical operating characteristics of an MLHP which determine its ef?ciency are the values of the nominal and the maximum transport capacities, the evaporator temperature, the internal and the total thermal resistance. By the nominal capacity it is meant that the value of the heat load which corresponds to the thermal resistance reaches its minimum value. The heat-transfer coef?cient in the evaporator, which is the most important element of an LHP, is also quite convenient for analysis. In accordance with this the main aim of the tests consisted in obtaining the above-mentioned operating characteristics for two miniature LHPs quite similar in size, but made of different materials and ?lled with different working ?uids.

Fig. 5 gives a typical heat-load dependence of the evaporator temperature obtained in a horizontal position of the MHLPs characterized by the slope $\varphi = 0$ deg. From this ?gure it follows that at heat loads less than 55 W the temperature of the ammonia MLHP 1 has lower values than that of the water MLHP 2. Therefore, it should be expected that its heat-exchange intensity in the evaporator is higher, and the thermal resistance lower, at least in the same range of heat loads.

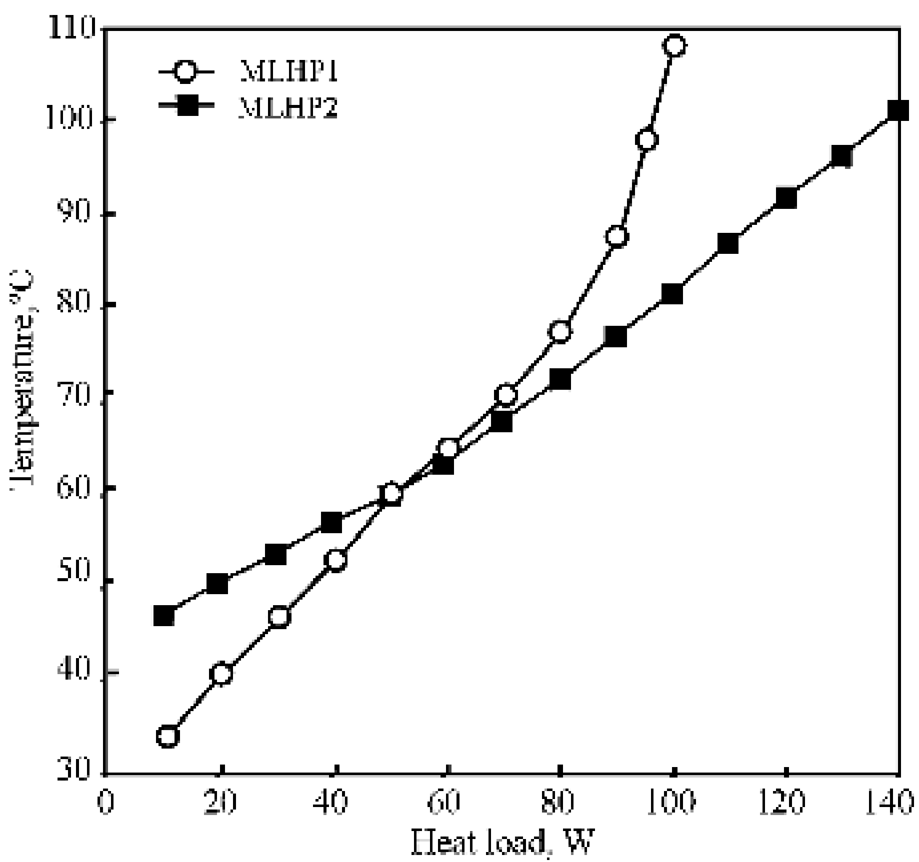

Fig. 5. MLHP evaporator temperature versus heat load.

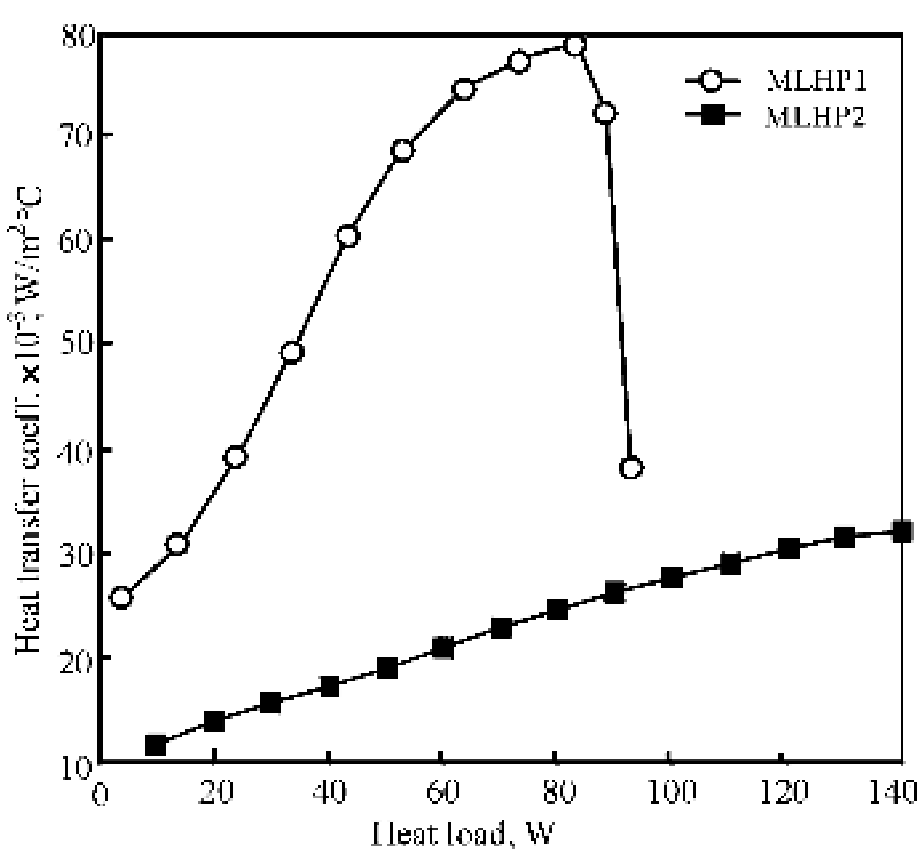

Fig. 6. Heat transfer coef?cient versus heat load.

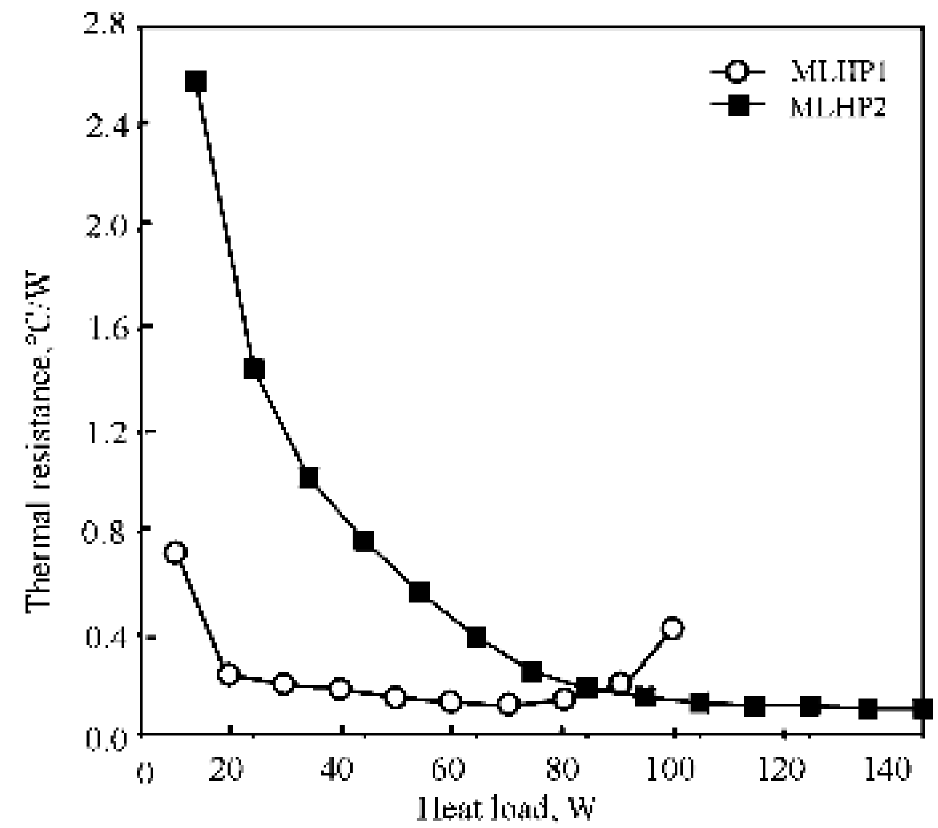

Fig. 7. MLHP thermal resistance versus heat load.

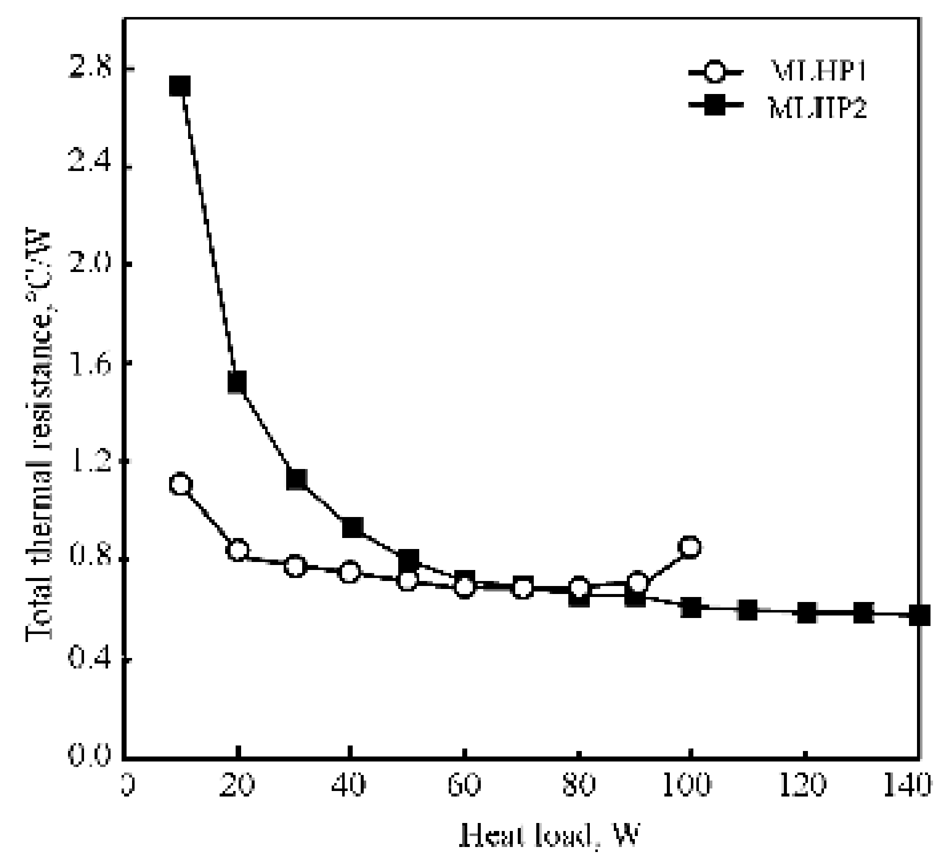

Fig. 8. MLHP total thermal resistance versus heat load.

Figs. 6 and 7 present results which con?rm this assumption. Here it can be seen that the heat transfer coef?cient in the evaporator of the ammonia MLHP 1, which reaches a maximum value of 78 000 W/m^{2}K, is much higher in the whole range of heat loads. As for the thermal resistance of MLHP 1, it increased up to the heat load of 85 W, and its minimum value of 0.12 K/W is attained at a capacity of 70 W and an at evaporator temperature of 70 °C. At the same time the advantage of the water MLHP 2 manifests itself at higher loads. The minimum value of its thermal resistance is 0.1 K/W at a capacity of 130 W and an evaporator temperature of 96.3 °C. Despite the fact that the value of the heat-exchange coef?cient of MLHP2, equal to 31 700 W/m^{2}K, proves to be lower than that of the ammonia MLHP 1, a decrease in the thermal resistance becomes possible at the expense of the increased surface of the evaporator active zone and a more developed condenser surface.

When comparing the value of the total thermal resistance R_{tot}, the heat-load dependence of which is shown in Fig. 8, it can be noted that the differences in its minimum values of 0.68 K/W for MLHP 1 and 0.58 K/W for MLHP 2 are more noticeable than the differences in the values of R, 0.12 K/W and 0.10 K/W, respectively.

The results presented in Figs. 5–8 show that the capacity of 140 W achieved by MLHP 2 is not limited in the test conditions under discussion. The plots of the dependences $T_e = f(Q)$, $R = f(Q)$ and $\alpha_e = f(Q)$ for this device have a monotonic characteristic and shows no evidence of critical phenomena. At the same time for MLHP 1 such a crisis is evident even at a heat load of 90 W, when the heat-transfer coef?cient begins to decrease abruptly, and the evaporator temperature and the thermal resistance increase. The reason for this may be particularly from a heat-exchange crisis in the evaporator or limited dimensions of the surface of the condenser, which does not allow rejection of a higher heat load in the given cooling conditions.

For a complete "disclosure" of the potential capabilities of the MLHP 1 evaporator, the intensity of cooling the condenser was considerably increased by its immersion in a water bath. The

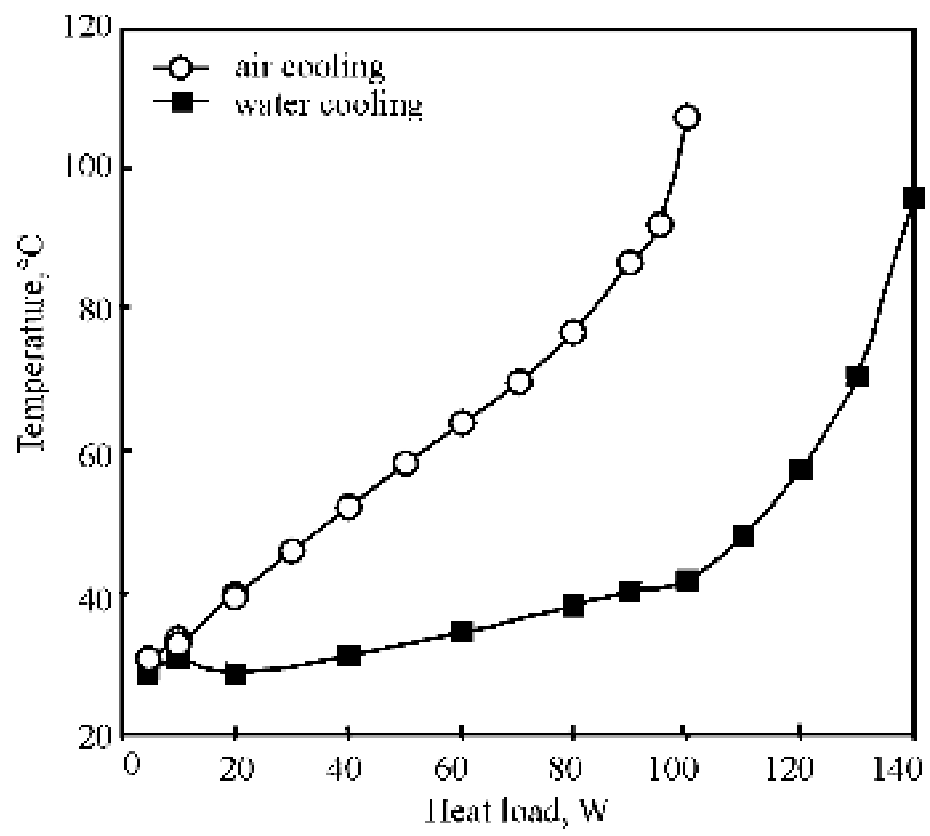

Fig. 9. MLHP 1 evaporator temperature versus heat load with different methods of the condenser cooling.

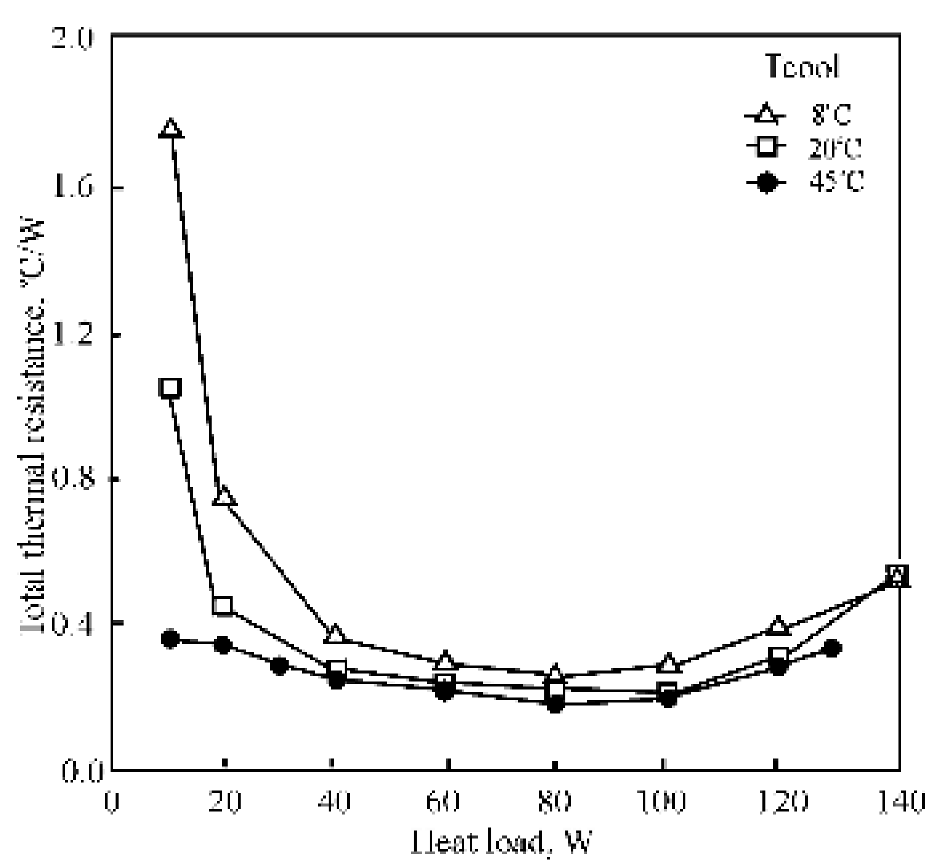

Fig. 10. MLHP 1 total thermal resistance versus heat load at different condenser coolant temperatures.

plots presented in Fig. 9 make it possible to compare the dependences $T_e = f(Q)$ obtained for MLHP 1 under different conditions of cooling the condenser and the same temperature of the cooling medium (equal to 20 °C). The obtained results show that the device maximum capacity increases up to 140 W, and the level of operating temperature of the evaporator decreases considerably. In this case the nominal value of the heat load reaches 100 W. Similar experiments conducted for MLHP 2 made it possible to increase its capacity up to 210 W.

Fig. 10 shows the results of another series of MLHP 1 tests in the form of the dependence $R_{tot} = f(Q)$ when the condenser was cooled with water with temperatures of 8 °C, 20 °C, and 45 °C. These results make it clear that the MLHP 1 total thermal resistance decreases considerably at more intensive condenser cooling in the entire range of heat loads. Thus, in particular, at a nominal capacity of 100 W the value of R_{tot} is within 0.2–0.3 K/W as the condenser cooling temperature changes from 8 °C to 45 °C. It also should be mentioned that the evaporator temperature in these conditions decreases respectively to 36.3 °C and 64.6 °C, and at a cooling temperature of 20 °C it is equal to 41.9 °C.

VI. CONCLUSION

1) Different variants of miniature LHPs with cylindrical (5 and 6 mm in diameter) evaporators made from stainless steel and copper have been developed and tested. Two experimental specimens with ammonia (MLHP 1) and water (MLHP 2) as working ?uids have been used to illustrate the test results.
2) Tests using forced air cooling of condensers at a temperature of 20 °C have shown that the nominal heat load is 70 W for MLHP 1 and 130 W for MLHP 2. The corresponding minimum values of the internal thermal resistance of the devices are 0.12 K/W and 0.10 K/W, and the total thermal resistance "evaporator—air" is respectively 0.68 K/W and 0.58 K/W. The maximum capacity limited by the evaporator temperature of 100 °C is equal to 95 W for MLHP 1 and 140 W for MLHP 2.
3) Using MLHP 1 as an example, it is shown that with more intensive water cooling of the condenser at temperatures of 8 °C, 20 °C, and 45 °C the device maximum capacity increases up to 140 W. At a nominal heat load of 100 W the evaporator temperatures were equal to 36.3 °C, 41.9 °C, and 64.6 °C, respectively, and the minimum value of the total thermal resistance in this case is within 0.20–0.30 K/W.
4) The obtained results make it possible to regard miniature LHPs as an ef?cient and promising means for cooling electronics and computer equipment.

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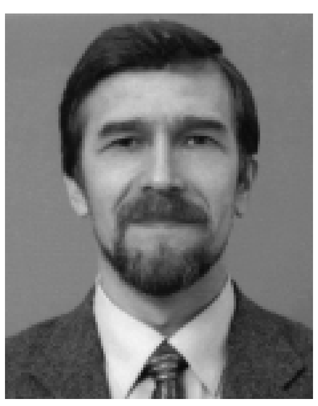

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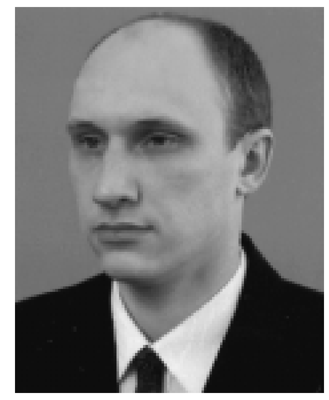

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