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# Low-noise cooling system for PC on the base of loop heat pipes

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**Abstract**

The problem of current importance connected with a wide use of personal computers (PC) and a rapid growth of their performance is a decrease in the noise level created at the operation of cooling system fans. One of the possible ways of solving this problem may be the creation of passive or semi-passive systems on the base of loop heat pipes (LHPs) in which the heat sink is an external radiator cooled by natural and/or forced air convection. The paper presents the results of development and tests of several variants of such systems, which are capable of sustaining an operating temperature of 72–78 °C on the heat source thermal interface which dissipates 100 W at an ambient temperature of 22 °C. It is also shown that the use of additional means of active cooling in combination with LHPs allows to increase the value of dissipated heat up to 180 W and to decrease the system thermal resistance down to 0.29 °C/W.


*Keywords:* Computer cooling; Loop heat pipe; Passive CPU cooling system

## 1. Introduction

For the time being the most widespread means of cooling components of personal computers (PC) which are under high thermal load, first of all such as central processors (CPU), graphic processors etc., is forced air convection created by fans. As with the growth of PC performance energy consumption and the corresponding value of dissipated heat that must be removed, as a rule, increase, the number of fans in the cooling system constantly rises. At present their amount in the computer may reach 6 units or more. These tendencies inevitably lead to an increase in the noise level reaching 40–60 dB which exerts quite an adverse effect on the user. That is why combating the noise created during PC operation is very urgent.

One of the possible ways of solving the mentioned problem is based on the conception of passive cooling with the use of natural air convection. Since for natural convection the intensity of heat exchange is approximately by an order of magnitude lower than for forced convection, a necessary condition is a considerable increase of the heat sink surface and also its sufficiently high temperature and isothermality. Quite a logical step on this way is, for example, the use of the system-block body, which may be made in the form of an integral radiator (heat sink), and heat pipes, which ensure a good thermal connection between the components to be cooled and such a radiator. It is precisely such a design that has been successfully realized lately [1].

It is necessary to mention that heat pipes as heat-transfer devices with a superhigh thermal conductivity are widely used in active cooling systems of desktop and mobile computers that have a fan and a heat sink with a finned surface in their structure [2–8]. However, such a cooling system, besides the fact that it is a source of noise, has one more essential disadvantage. It is connected with the fact that coolers are located inside the computer body where the air temperature may be considerably higher than the ambient temperature. This results in the necessity to use coolers that have quite a low thermal resistance, which makes it possible to sustain an acceptable operating temperature on the thermal interface of the object to be cooled. In particular, the maximum value of this temperature for

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Intel CPUs is usually set by the value 70 °C and for AMD CPUs it is equal to 80 °C. The thermal resistance of a cooler may be calculated by the equation:

$$R_{ca} = \frac{T_c - T_a}{Q}, \quad \text{where} \tag{1}$$

$T_c$ is the temperature of a CPU thermal interface, °C
$T_a$ is the air temperature inside the computer, °C
$Q$ is the dissipated heat, W.

If we take $Q = 100$ W, which is typical of processors with a clock speed about 3 Gz, then at the temperature $T_a = 45$ °C the value of $R_{ca}$ should be equal to 0.25 °C/W for Intel processors and to 0.35 °C/W for AMD processors. For more fast-operating modern processors, which may dissipate up to 140 W, the value should be even lower. The creation of such coolers is a very difficult task. Its solution is achieved, in particular, at the expense of using several heat pipes in one cooler and also at the expense of increasing the heat sink surface and the fan efficiency.

If as a heat sink one may use an external radiator, which may be made, for example, in the form of a finned wall of a system-block body, then instead of the temperature $T_a = 45$ °C in Eq. (1) one can take an ambient temperature the maximum value of which is usually 35 °C. Then the thermal resistance of a cooler may allowed to be at the level of 0.35–0.45 °C/W. In the majority of cases when a PC is employed at an ambient temperature of 20–25 °C such coolers make it possible to decrease the CPU operating temperature down to 55–65 °C or to increase its fast-operation.

The use of loop heat pipes [9–12] as a thermal connection between the heat source and the heat sink opens new opportunities for the increase of effectiveness of both passive and active electronics cooling systems including computers of various types. This statement is based on the advantages of LHPs over conventional heat pipes. Among them is a considerably higher heat-transfer capacity, lower thermal resistance, small sensitivity to the position in the gravity field, wider opportunities for various design embodiments and also high adaptability to different location and operation conditions.

The paper presents the results of development and tests of four variants of desktop PC CPU cooling systems created on the base of copper–water loop heat pipes, in which a system-block body sidewall with finning was used as a heat sink. In the passive regime, the systems ensured an operating temperature on the thermal contact surface of a heat source equal to 72–78 °C at dissipation of a heat flow of 100 W, which corresponded to thermal resistances of 0.50–0.56 °C/W. It was shown that additional use of active cooling means, which include thermoelectric module (TEM) and/or forced convection, allows increasing the maximum value of dissipated heat up to 180 W at an operating temperature of 75 °C and decreasing the thermal resistance down to 0.29 °C/W.

## 2. LHP schematic diagram and operation

According to the functional definition a loop heat pipe is a hermetic two-phase heat-transfer device that operates on the closed evaporation–condensation cycle with the use of capillary forces for the working fluid circulation. From this definition it follows that the operation of loop and conventional heat pipes is based on the same physical processes. Both devices use the high latent heat of vaporization of liquids, which makes it possible to ensure an intensive heat removal in the evaporation zone and to have a relatively small mass flow of a working fluid inside the device. To pump the working fluid both of them also use the capillary pressure created by a special capillary structure (wick) due to which these devices do not contain mechanically moving parts and do not consume additional energy. However the LHP schematic diagram and the characteristic features of its operation allow a more effective realization of these processes. Fig. 1 presents the LHP schematic diagram.

The device includes an evaporator supplied with a special wick and a condenser that are connected by separate smooth-walled pipelines for vapor and liquid transport and whose diameters are usually 2–4 mm. The evaporator may have a cylindrical, flat-rectangular, flat-oval or disk-shaped form and the condenser form, size and design may vary widely depending on the heat sink characteristics.

As vapor and liquid pipelines have a considerably small diameter and do not contain a capillary structure they may be easily bent taking practically any form necessary for the installation of an LHP in the cooling system. Due to the fact that the wick length is limited only by the evaporator the distance which liquid covers moving in the wick is extremely small. That is why the wick does not create a high hydraulic resistance even if it has a small pore size which is necessary for the creation of a high capillary pressure. The evaporator is combined with a compensation chamber that serves for the accumulation of the working fluid displaced from the vapor line and the condenser when the device is operating.

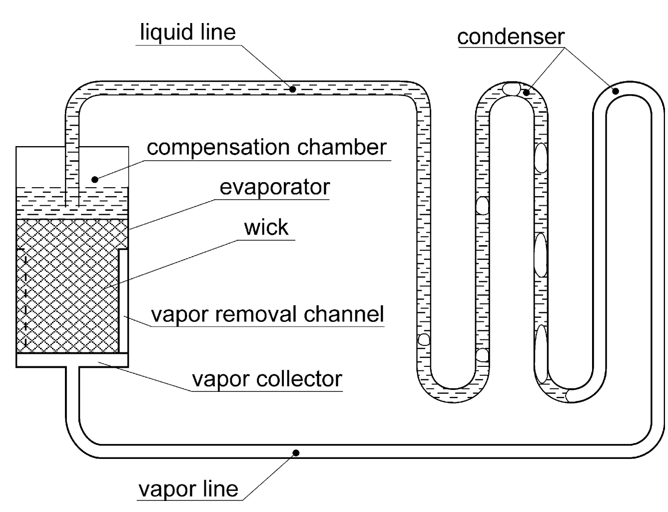



Fig. 1. LHP schematic diagram.

The basic condition of LHP operation is completely analogous to that for a conventional heat pipe:

$$\Delta P_c \geqslant \Delta P_v + \Delta P_l + \Delta P_g, \quad \text{where} \tag{2}$$

$\Delta P_c$ is the capillary pressure created by the wick, Pa
$\Delta P_v$ are pressure losses in the vapor phase, Pa
$\Delta P_l$ are pressure losses in the liquid phase, Pa
$\Delta P_g$ is the hydrostatic pressure head, Pa.

The hydrostatic pressure head value is calculated by:

$$\Delta P_g = (\rho_l - \rho_v) g L \sin\varphi, \quad \text{where} \tag{3}$$

$\rho_l$ is the liquid density, kg/m$^3$
$\rho_v$ is the vapor density, kg/m$^3$
$g$ is the gravitational acceleration, m/s$^2$
$L$ is the LHP effective length, m
$\varphi$ is the LHP slope to the horizontal plane, grad.

The value of $\Delta P_g$ may become prevailing at $\sin\varphi \to 1$, when the evaporator is located above the condenser if the LHP length is considerably great. However due to the fact that LHPs use wicks capable of creating a high capillary head the value of $\Delta P_g$ has no fatal influence even if the length of an LHP reaches 1 m and more. As for the values of $\Delta P_l$ and $\Delta P_v$, they are defined by the known equations for liquid and vapor flows in smooth-walled cylindrical pipelines.

There is also a second compulsory condition of LHP operation that may be defined as follows:

$$\frac{\partial P}{\partial T}\Big|_{\overline{T_v}}(T_{v1} - T_{v2}) = \sum \Delta P_{ext}, \quad \text{where} \tag{4}$$

$\partial P/\partial T$ is a derivative which characterizes the slope of a working fluid saturation line at the temperature $\overline{T_v}(\bar{T}_v = \frac{T_{v1}+T_{v2}}{2})$, Pa/°C
$T_{v1}$ is the vapor temperature above the evaporating surface of the wick menisci, °C
$T_{v2}$ is the vapor temperature above the liquid–vapor interface in the compensation chamber, °C
$\sum \Delta P_{ext}$ are the total pressure losses in the area of a working fluid circulation between the evaporation zone and the compensation chamber, Pa.

Eq. (4) expresses the condition that is necessary for the creation of a definite temperature difference and the corresponding vapor pressure difference between the wick evaporating surface in the evaporator heating zone and the "vapor–liquid" phase boundary in the compensation chamber. The fulfillment of this condition makes it possible to replace liquid from the vapor line and the condenser to the compensation chamber at the LHP start-up and operation. It follows from this condition that there is a connection between the vapor temperature in the compensation chamber and the LHP operating temperature. It also means a very important circumstance consisting in the fact that by controlling the vapor temperature in the compensation chamber one may control the LHP operating temperature while there is a phase boundary in the compensation chamber [13,14].

There are also additional conditions of LHP operation that in the simplest way may be represented as follows:

$$V_{wf} = V_w + V_{ll} + V_{cc} \tag{5}$$

$$V_{cc} = V_{vl} + V_{cond}, \quad \text{where} \tag{6}$$

$V_{wf}$ – the volume of the LHP working fluid, $V_w$ – the volume of the working fluid that the wick may absorb, $V_{ll}$ – the volume of the liquid line, $V_{cc}$ – the volume of the compensation chamber, $V_{vl}$ – the volume of the vapor line, $V_{cond}$ – the volume of the condenser.

The conditions (5) and (6) are not as compulsory and strict as the two main conditions (2) and (4). Nevertheless, their fulfillment guarantees a steady contact of the working fluid with the wick and ensures the readiness of the device for a reliable start-up in any situation including the situation when the evaporator is located above the condenser.

A more detailed calculation of a working fluid volume in LHP and CC volume is presented in papers [15,16].

## 3. Description of experimental cooling systems

The systems presented below were developed for the location in PC bodies like midi-tower of two size types with sidewall sizes (width × height) 410 × 410 mm and 460 × 410 mm. The system configuration was determined by the component layout inside the body, the chosen condenser design and by the ease of its installation.

The general view of the first variant of a cooling system (CS #1) is shown in Fig. 2. The evaporator (1) with a rectangular cross section was connected with a condenser of the collector type (4) by means of vapor (2) and liquid (3) lines and was equipped with a copper thermal interface (5) on the side of the contact surface. The condenser collectors (6 and 7) and the bunch of parallel channels (8) were

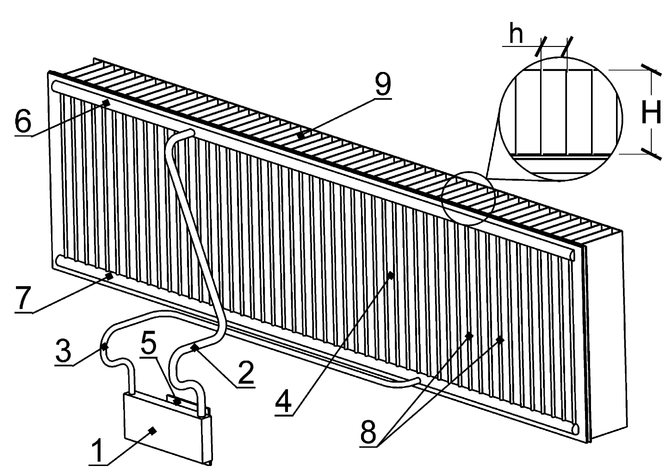



Fig. 2. General view of the cooling system #1: 1, evaporator; 2, vapor line; 3, liquid line; 4, condenser; 5, thermal interface; 6, vapor collector; 7, liquid collector; 8, condenser channels; 9, fins.

formed between two thin copper plates. Fins (9) were located on the outer flat condenser plate.

The cooling system CS #2 is presented in Fig. 3. It was made on the basis of an LHP with a cylindrical evaporator (1) equipped with a rectangular copper interface (4). The vapor line (2) and the liquid line (3) had diameters of 4 mm and 3 mm, respectively. The condenser (5) was glued to the aluminum radiator (6). The radiator finning was made of corrugated aluminum profile with rectangular corrugations.

The general view of the cooling system CS #3 is presented in Fig. 4. The evaporator (1) had a flat-oval section with a contact surface measuring 32 × 32 mm, this allowed to avoid a special thermal interface from the design. The vapor line (2), liquid line (3) and condenser (4) were made of a tubing with a diameter of 3 mm. The condenser (4) in the form of a tubing loop was mechanically pressed to the finned radiator (5). The pressure was effected with the help of an aluminum plate and bolts that are not shown in the picture. The cooling system CS #4 differed from the cooling system CS #3 only by the diameter of the loop tubing that was increased to 4 mm.

The general view of the system CS #2 installed in a PC is shown in Fig. 5. The constructional characteristics of the systems are presented in Table 1.

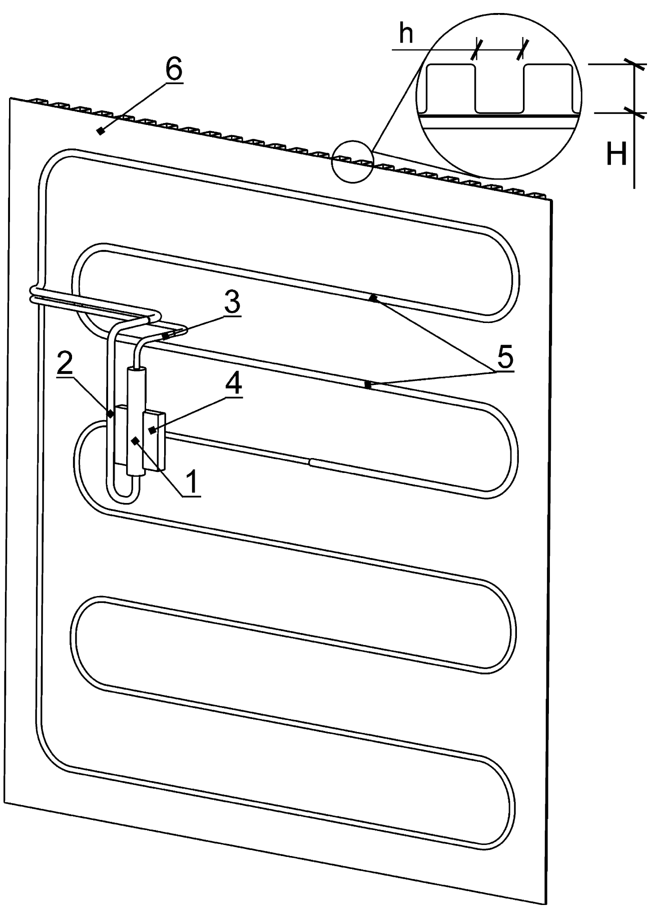



Fig. 3. General view of the cooling system #2: 1, evaporator; 2, vapor line; 3, liquid line; 4, thermal interface; 5, condenser channels; 6, radiator.

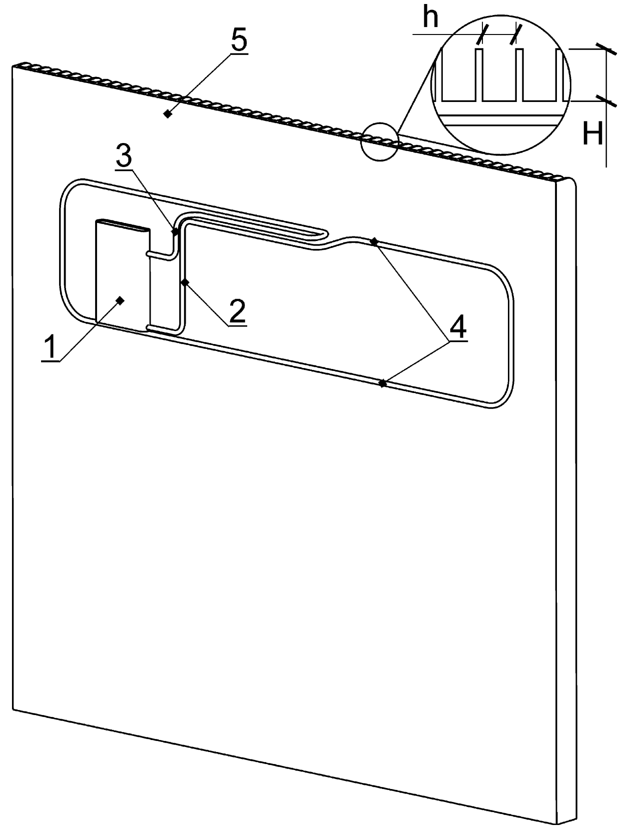
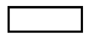
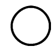
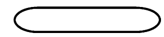



Fig. 4. General view of the cooling system #3 and #4: 1, evaporator; 2, vapor line; 3, liquid line; 4, condenser channels; 5, radiator.


Fig. 5. General view of the cooling system CS #2 in the DC structure.

Table 1
Parameters of cooling systems

| Parameter | CS #1 | CS #2 | CS #3 | CS #4 |
|---|---|---|---|---|
| LHP constructional material | Copper | | | |
| Working fluid | Water | | | |
| Evaporator characteristics | | | | |
| Section form |  |  |  | |
| Section size (mm) | 25 × 7 | ∅10 | 42 × 7 | |
| Body length (mm) | 61 | 61 | 64 | |
| Active zone length (mm) | 32 | 32 | 32 | |
| Thermal interface sizes | | | | |
| Length × width × thickness (mm) | 32 × 32 × 1 | 32 × 32 × 6 | – | |
| Pipelines | | | | |
| Vapor line (OD × wall, mm) | 4 × 0.5 | 4 × 0.5 | 3 × 0.5 | 4 × 0.5 |
| Liquid line (OD × wall, mm) | 3 × 0.5 | 3 × 0.5 | 3 × 0.5 | 4 × 0.5 |
| Radiator characteristics | | | | |
| Material | Copper | Aluminum | Aluminum | |
| Sizes (mm) | 410 × 110 × 35 | 400 × 400 × 9 | 410 × 460 × 18 | |
| Fin height ($H$[a]) (mm) | 30 | 8 | 15 | |
| Distance between fins ($h$[a]) (mm) | 8 | 8 | 7 | |
| Fin thickness (mm) | 0.4 | 0.5 | 1.5 | |
| Finned surface area ($m^2$) | 0.50 | 0.54 | 0.85 | |
| System total mass (kg) | 1.60 | 0.85 | 3.02 | |

[a] Positions are pointed in Figs. 2–4.

## 4. Test results

### 4.1. Testing technique

All the tests were conducted in the normal environment with radiator cooling by means of natural or forced convection at an ambient temperature of 22 ± 2 °C. The heat load was supplied from a special ohmic heater with controllable electric power playing the role of a CPU heat simulator.

The main task of the simulating tests was the determination of the system temperature characteristics depending on the value of the heat flux (heat load) dissipated by the simulator. The temperature measuring was realized by copper-constantan thermocouples located on the simulator contact surface, compensation chamber, vapor line and condenser line. Besides, 5–8 measuring points were located on the condenser surface.

In every experiment the time dependence of the temperature field was taken at a stepped increase of the heat load from a minimum of 5–10 W to a maximum of 120–180 W with steps of 10–20 W. The maximum heat load was limited by the temperature, which was not to exceed 80–90 °C. At every new heat load value a time period of 30–45 min was allowed for establishing a stationary regime of operation.

Together with the simulation tests the systems CS #1 and CS #2 were also tested in the PC structure. The system CS #1 was tested with a CPU Intel P4/2.8 GHz (Motherboard – Soltec, Socket 478, Intel 845 Series). The system CS #2 was installed and has been exploited up to now in a PC with a CPU AMD Athlon XP2500+, 1.833 GHz (Motherboard – ASUS 7V8X-X/LAN, Socket A). During testing the Software like CPUburn was used at which the CPUs dissipated maximum heat of about 70 W.

With the purpose of increasing the cooling system efficiency simulation tests with additional cooling of the LHP compensation chamber were conducted. The experiments were realized with CS #3 during which two methods of compensation chamber cooling were employed. The first method, which was first applied in paper [17], is based on the use of a Thermal Electric Module (TEM). It is shown in Fig. 6. In this variant the TEM was pressed to the compensation chamber surface with its "cold" side. Thereby its "hot" side was cooled via the thermal link that channeled heat to the active evaporator zone. For a better cooling of the TEM "hot" side an additional radiator blown by a low-noise fan was used. In the experiments the TEM cooling efficiency amounted to about 3–4 W at a power consumption of 7 W.

The second method of the compensation chamber cooling was realized with the use of a two-phase thermosyphon with a diameter of 6 mm (Fig. 7). The thermosyphon thermal interface was installed on the compensation chamber and the thermosyphon condensation zone equipped with finning was blown by the fan.

### 4.2. Discussion of test results

Generalized results for all the systems obtained during the bench tests are given in Fig. 8 in the form of CPU simulator temperature versus heat load. The CPU thermal

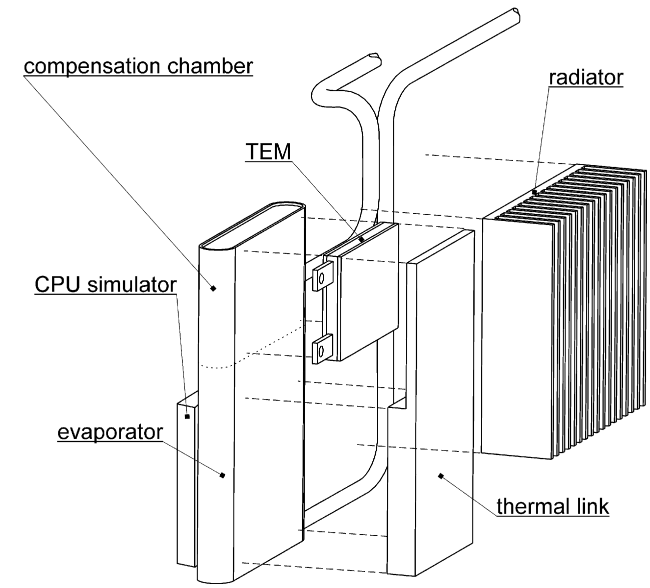



Fig. 6. Schematic of compensation chamber cooling by TEM.

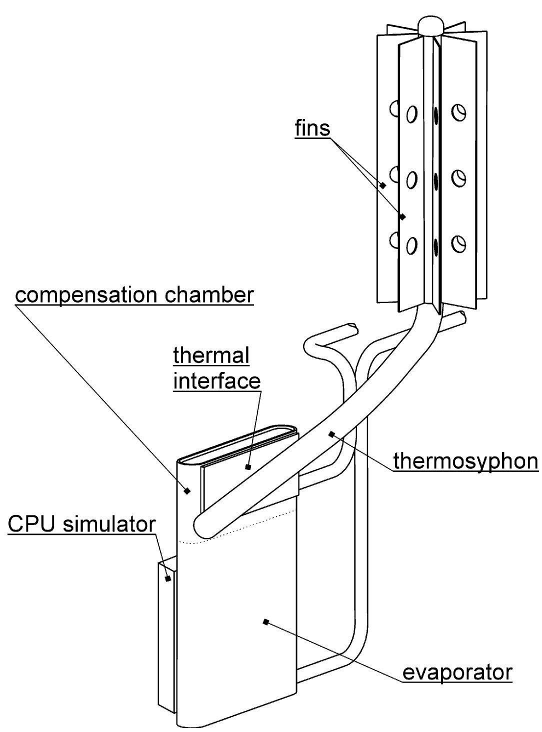



Fig. 7. Schematic of compensation chamber cooling by thermosyphon.

contact surface temperatures are also presented. They were received in real conditions with the use of CS #1 and CS #2 when the maximum value of heat dissipated by the processors was 68 W. The insignificant CPU temperature overshoot in comparison with the same value of dissipated heat that was obtained in the tests with a simulator is explained by a certain difference in the ambient temperature. One can see that in the range from 20 to 100 W CS #1 is the most effective because it maintains a lower temperature on the cooled object. It was caused both by the

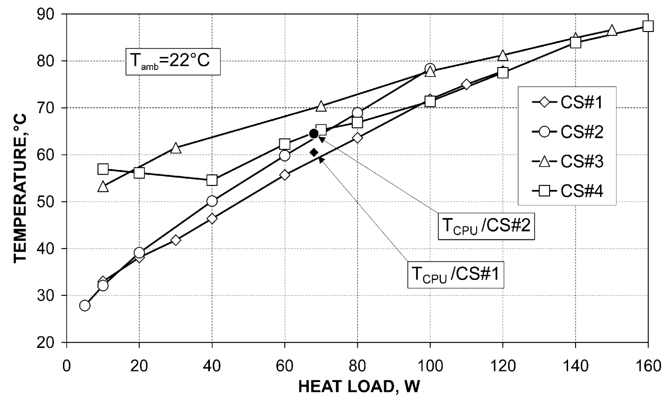



Fig. 8. Dependence of the CPU simulator temperature on heat load during the tests with natural convection.

condenser design, which provides good radiator isothermality, and by a more favorable position regarding the evaporator. At the same time such a system is more complex for execution and there are some limitations concerning its location in the computer. Successive efforts that were directed to the simplification of the cooling system have led to the creation of CS #4, which demonstrated results very close to the best ones at heat loads up to 90 W. The maximum value of heat load that the system demonstrated was 160 W.

As has already been mentioned above, the LHP operation is accompanied by the redistribution of the working fluid between the condenser and the compensation chamber. As the heat load increases, a gradual release of liquid from the condenser and a filling of the compensation chamber occur. A very important here is the possibility to reach such a state when the condenser is released of liquid to the maximum and condensation finishes at its exit. In this case its operation is the most effective because this particular LHP state allows reaching the minimum operating temperature at the given conditions. However one cannot always get this state because of parasitic heat leakage into the compensation chamber. This heat leakage does not allow decreasing the vapor operating temperature in the LHP and the corresponding pressure to such a degree that a complete condenser release will be possible. An obvious solution for this problem is an additional compensation chamber cooling.

Fig. 9 presents the results of experiments with CS #3 when different variants of compensation chamber cooling were used, including the use of a thermoelectric module and a two-phase thermosyphon, the radiators of which were blown by fans with a flow speed of 1.5–2.0 m/s. One can see that in both cases the effect was considerable in the whole operating range. In the heat load interval 60–140 W the temperature decrease was 10–15 °C. The heat flux removed from the compensation chamber was 10% of the heat load applied to the evaporator.

The results of a similar experiment with CS #4 are given in Fig. 10. Here the main radiator was blown beginning with a heat load of 160 W. One can see that upon this

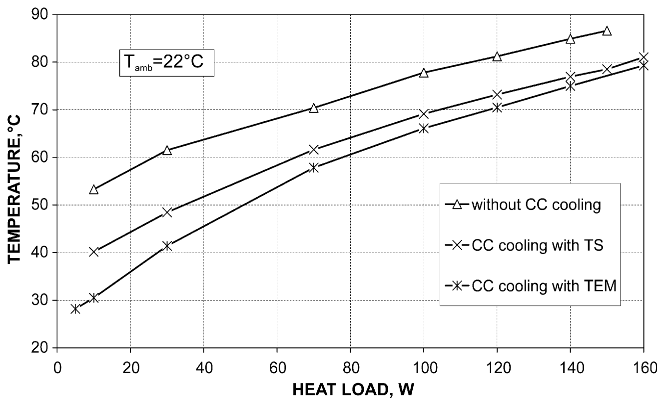



Fig. 9. The results of an experiment with additional cooling of the evaporator compensation chamber of CS #3.

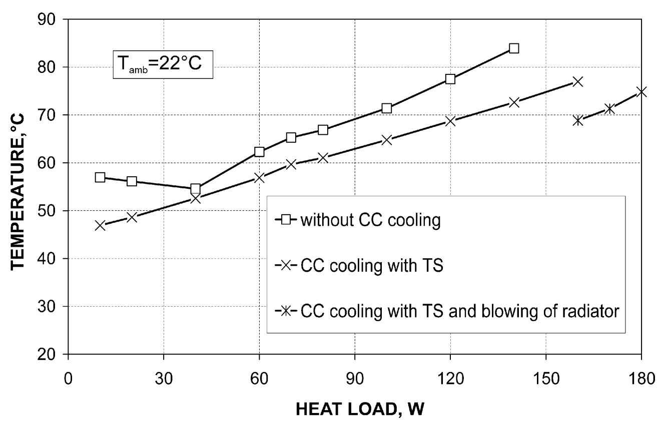



Fig. 10. The results of an experiment with additional cooling of the evaporator compensation chamber of CS #4 and the main radiator blowing.

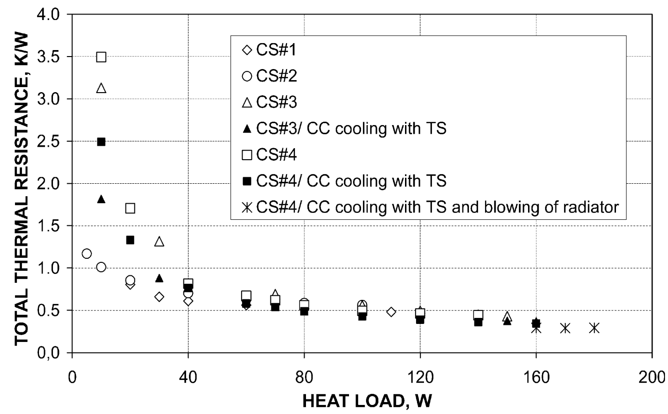



Fig. 11. Dependences of total thermal resistance on heat load.

the maximum value of the dissipated heat reached 180 W at a simulator temperature about 75 °C. Further heat load increase was limited only by the simulating heater possibilities.

It was also pointed out above that an important characteristic reflecting the cooling system efficiency is the total thermal resistance. In Fig. 11 the dependences of the total thermal resistance on the heat load are shown for all the tested cooling systems obtained at different conditions.

Such dependences are typical for LHPs, viz. with heat load increase the system thermal resistance decreases. This process is caused to a considerable degree by the redistribution of the working fluid between condenser and compensation chamber, which results in a release of liquid from the condenser as the heat load increases. This is also facilitated by an increase in the intensity of the heat exchange of the radiator with the environment at forced convection.

One can also see in Fig. 11 that when the main radiator is cooled by means of natural convection the system thermal resistance is 0.5–0.6 K/W at heat loads of 80–100 W. At additional compensation chamber cooling the thermal resistance decreases to 0.4–0.5 K/W. The minimum value of 0.34 K/W was reached by the CS #4 at a heat load of 160 W. At additional main radiator blowing the thermal resistance of the same system CS #4 decreased to 0.3–0.29 K/W at heat loads of 160–180 W.

## 5. Conclusions

The use of loop heat pipes makes it possible to create low-noise and effective systems for cooling of PC components under high thermal load in which the external radiator is the main heat sink, which is combined with the system-block body and cooled by means of natural or forced air convection.

For natural convection a cooling system with a condenser of the collector type is the most effective. It has a thermal resistance of about 0.5 K/W and is able to dissipate up to 80–100 W at an acceptable temperature level ($\leqslant$70 °C).

An additional active compensation chamber cooling considerably increases the system efficiency. At a heat load of 160 W its thermal resistance decreases to 0.34 K/W. The use of forced air convection for the main radiator cooling allows to increase the heat dissipation up to 180 W for a still acceptable operating temperature level (75 °C) and to decrease the system thermal resistance to 0.29 K/W.

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