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Review

Loop heat pipes

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Abstract

Loop heat pipes (LHPs) are two-phase heat-transfer devices with capillary pumping of a working fluid. They possess all the main advantages of conventional heat pipes, but owing to the original design and special properties of the capillary structure are capable of transferring heat efficiency for distances up to several meters at any orientation in the gravity field, or to several tens of meters in a horizontal position. Besides, the LHP conception allows a wide variety of different design embodiments, which essentially extends the sphere of functional possibilities and practical application of these devices. The paper is a review of developments, results of theoretical analysis and tests of LHPs performed at the Institute of Thermal Physics and some other organizations. It gives examples of successful application of these highly efficient devices in space technology and electronics.

Keywords: Heat-transfer device; Loop heat pipe; Thermoregulation system; Cooling; Space technology; Electronics

Contents

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1. Introduction

The history of development of loop heat pipes (LHPs) originates from 1972, when the first such device with a length of 1.2 m and a capacity of about 1 kW, with water as a working fluid, was created and tested successfully by the Russian scientists Gerasimov and Maydanik from the Ural Polytechnical Institute [1,2].

The appearance of LHPs was a response to the challenge connected with the acute demand of aerospace technology for highly efficient heat-transfer devices with all the main advantages of conventional heat pipes [3], but at the same tune much less sensitive to the change of orientation in the gravity field. The problem here lies in the fact that conventional heat pipes, in which the capillary structure (wick) is situated along the whole length, abruptly decrease their heat-transfer capacity when the evaporation zone is above the condensation one. First of all, it refers to low-temperature devices, where use is made of working fluids with a relatively low value of the surface-tension coefficient. This circumstance is connected with the fact that the maximum value of the capillary head providing the circulation of a working fluid in a heat pipe is directly proportional to this coefficient and inversely proportional to the effective pore radius of the wick. To compensate for the additional pressure losses during the liquid motion to the evaporation zone against the gravity forces, it is necessary to increase the capillary head. It is evident that in this case it can be done only at the expense of decreasing the effective pore radius of the wick. However, here one can observe an increase in the hydraulic resistance of the latter approximately proportional to the square of the pore radius. As a result of this contradiction, attempts to create a heat pipe of sufficient length capable of operating efficiently against gravity forces do not meet with success. Even when the working fluid is water, which is the "strongest" working fluid in the low-temperature range, heat pipes 0.5 m long decrease their capacity almost by an order when it is necessary to transfer heat from above downwards at a vertical position of the device [4]. Numerous well-known variants of solution of this problem by using various additional means for reinforcement or even replacement of the capillary mechanism of pumping of the working fluid lead only to an inadequate loss of these or those valuable properties typical of heat pipes [5–11]. Therefore, they have not gained extensive distribution and practical implementation.

The LHP conception to a considerable extent makes it possible not only to overcome this drawback of conventional heat pipes, but also to obtain some additional advantages, remaining in the framework of the capillary mechanism and using all its advantages. It includes the following main principles:

- the use of fine-pored wicks;
- maximum decrease in the distance of the liquid motion in the wick;
- organization of effective heat exchange during the evaporation and condensation of a working fluid;
- maximum decrease in pressure losses in the transportation (adiabatic) section.

For realization of this conception special capillary–porous materials were created on the basis of sintered nickel, titanium and copper powders with an effective pore radius of 0.7–15 μm and a porosity of 55–75%. Such materials possess a sufficient strength, which allows complex mechanical processing in making wicks, are capable of creating a very high capillary pressure with the use of low-temperature working fluids and are chemically compatible with most of them.

Minimization of the distance for the liquid motion in the capillary structure limited, as a rule, by several millimeters is achieved at the expense of the wick design, whose extent corresponds to the dimensions of the evaporation zone and does not depend on the total length of the device. The motion of vapor and liquid flows in such a wick proceeds mainly in the radial direction and has a counter character at which the evaporating menisci are inverted towards the wall being heated.

The evaporation zone, which is formed here by a ramified system of vapor-removal channels located at the "wall-wick" boundary ensures an effective heat exchange even during evaporation from a fine-pored capillary structure. Depending on the wick material and the kind of the working fluid, the intensity of heat exchange in such a zone may reach values from 10,000 to 100,000 W/m^2 K.

The design and the dimensions of the condensation zone in LHPs may be quite different, which makes it possible to easily adapt it to the conditions of heat exchange with an external heat sink. Condensation here takes place, as a rule, at a smooth surface, in constrained conditions and is of film character. Where it is possible to organize a film suction, the intensity of heat exchange during condensation may be considerably increased.

A decrease in pressure losses in the adiabatic section of LHPs is ensured by the fact that for the motion of a working fluid here use is made of separate smooth-walled pipe-lines, which exclude both the thermal and the viscous interaction between counter flows of vapor and liquid.

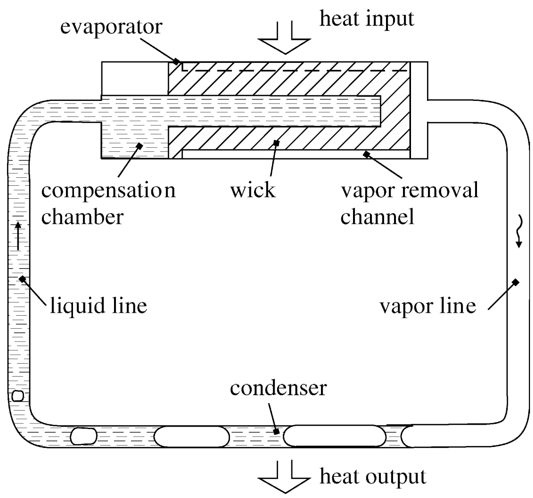


Fig. 1. Principal scheme of an LHP.

Fig. 1 shows the schematic diagram in which the LHP conception is realized. Besides attaining the main aim formulated above, it allows wide variety for different design embodiments and, at the same time, extends considerably the sphere of functional potentialities of two-phase heat-transfer devices with capillary pumping of a working fluid. At present it forms the basis for large and powerful LHPs operating efficiently at any orientation in 1-g conditions, and also flexible, ramified, controllable, reversible, miniature and other devices, many of which have already found application in space technology and electronics and have good prospects for application in some other spheres. The wide variety of designs and functional indications of LHPs known to-day makes it possible to perform their classification, which is given in Table 1.

The paper presents the fundamentals of the theory of LHPs, various examples of their designs classified in the table, and also the results of tests and actual application of these highly efficient heat-transfer devices.

Table 1
Classification of LHPs

LHP design	LHP dimensions	Evaporator shape	Evaporator design
• Conventional (diode) • Reversible • Flexible • Ramified	• Miniature • All the rest	• Cylindrical • Flat disk-shaped • Flat rectangular	• One butt-end compensation chamber • Two butt-end compensation chambers • Coaxial
Condenser design	**Number of evaporators and condensers**	**Temperature range**	**Operating-temperature control**
• Pipe-in-pipe • Flat coil • Collector	• One • Two and more	• Cryogenic • Low-temperature • High-temperature	• Without active control • With active control

2. LHP theory

Theoretical analysis of different aspects of operation of loop heat pipes is presented in numerous publications [12–21]. Below are given some basic elements of this theory.

The operation of LHPs is based on the same physical processes as those used in conventional heat pipes. However, they are organized in quite a different way. First of all, it concerns the functions of the wick, which plays here a more complicated role. To determine these functions, one can use the scheme presented in Fig. 2.

In the absence of a heat load the free surface of the working fluid in an LHP is at a certain level A-A located in the liquid line and the evaporator as in communicating vessels. The wick in this case is saturated with a liquid, and the vapor line and the condenser are completely filled. If a heat load is supplied to the evaporator, the liquid begins to evaporate from the wick both in the evaporation zone and in the compensation chamber. Since the wick possesses quite a definite thermal resistance, the temperature and the pressure of vapor in the evaporation zone, which is closer to the evaporator wall to be heated, become higher than in the compensation chamber [13]. The wick in this case serves as a "thermal lock". At the same time hotter vapor cannot penetrate into the compensation chamber through the saturated wick owing to the capillary forces which hold the liquid in it. Here another function of the wick manifests itself, that of "a hydraulic lock". The arising pressure difference causes displacement of the working fluid from the vapor line and the condenser and filling of the compensation chamber. In this case three interfaces may exist in the LHP simultaneously: in the vapor zone, in the condenser and in the compensation chamber. In the last two cases these interfaces may move depending on the value of the heat load.

For analyzing the LHP operation it is convenient to use the diagram of the working cycle with respect to the saturation line of the working fluid in *P*–*T* coordinates, which in the idealized form is given in Fig. 3.

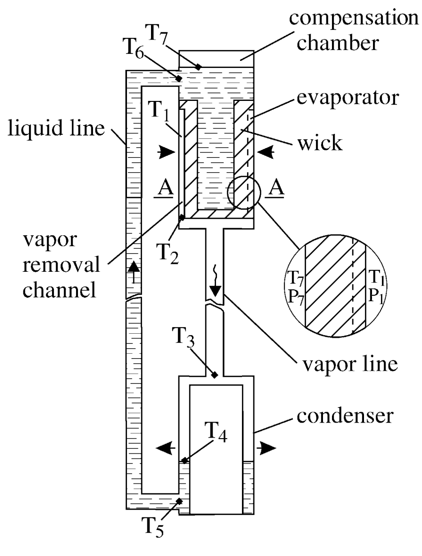


Fig. 2. Analytical LHP scheme.

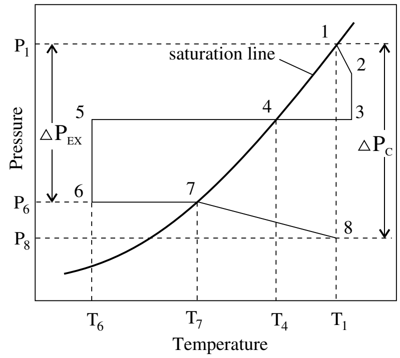


Fig. 3. Diagram of the LHP working cycle.

The point 1 on the saturation line determines the vapor parameters P_1, T_1 above the evaporating surface of the wick menisci in the evaporation zone, and the section 1–2 corresponds to the vapor motion in the system of vapor-removal channels into the vapor line. Since the vapor motion here proceeds along the hot wall of the evaporator, a decrease in its pressure is accompanied by a slight superheat. The vapor motion in the vapor line (section 2–3) ideally may be considered close to isothermal. Pressure losses in the LHP condenser are usually negligible. The working fluid here is condensed and in section 4–5 supercooled. Further its motion along the liquid line in the diagram is shown as isothermal, though in many actual cases it may be accompanied by considerable heating or cooling owing to the heat exchange with the surrounding medium. As a result, a liquid with parameters T_6, P_6 enters the compensation chamber. Simultaneously here comes part of the heat flow supplied to the evaporator, at the expense of which the working fluid is heated to the temperature T_7. The section 7–8 corresponds to the liquid filtration through the wick into the evaporation zone. On this way the liquid may prove to be superheated, but its boiling-up does not take place owing to the short duration of its being in such a state. The point 8 determines the state of the working fluid in the vicinity of the evaporating menisci, and the pressure drop ΔP_{1-8} corresponds to the value of total pressure losses in all the sections of the working-fluid circulation.

In accordance with this, the first condition of LHP serviceability is determined in the same way as for conventional heat pipes and may be written as

$$\Delta P_c \geqslant \Delta P_v + \Delta P_l + \Delta P_g. \tag{1}$$

ΔP_c is the capillary pressure created in the wick; ΔP_v, ΔP_l are pressure losses during the motion of the working fluid in the vapor and the liquid phase; ΔP_g are pressure losses caused by the hydrostatic resistance of a liquid column.

The values of ΔP_v and ΔP_l are determined here in accordance with the well-known equations [3] for the motion of vapor and liquid in different LHP sections, including the capillary structure. The value of ΔP_g may be determined by the formula:

$$\Delta P_g = (\rho_l - \rho_v)g\ell \sin\varphi, \tag{2}$$

where ρ_l, ρ_v are the liquid and the vapor densities; g is the free fall acceleration; ℓ is the LHP effective length; φ is the LHP slope with a horizontal plane.

During operation in the "antigravitational regime", if the value of ℓ is sufficiently high, and the value of φ is close or equal to 90°, the hydrostatic resistance ΔP_g becomes essentially prevailing. It may be compensated only at the expense of a high capillary head created by a fine-pored wick.

The second condition of serviceability, characteristic only of LHPs, is the necessity of creating a sufficient temperature and pressure drop between the evaporating surface of the wick and the compensation chamber. This pressure drop is to be equal to the value of ΔP_{EX}, which is equal to the sum of pressure losses in all the sections of circulation of the working fluid except the wick. This condition, with the use of the designations in Fig. 3, may be written as follows:

$$\left.\frac{\partial P}{\partial T}\right|_{\overline{T}_v} \Delta T_{1\text{–}7} = \Delta P_{EX}, \tag{3}$$

where dP/dT is the derivative determined by the slope of the saturation line at the point with temperature $\overline{T}_v$, average between T_1 and T_7.

In accordance with the condition (3) the quantity $\Delta T_{1\text{–}7}$ may be regarded as the motive temperature head required for displacing the liquid into the compensation chamber during an LHP start-up.

From analysis of the diagram it also follows that the working fluid should be sufficiently supercooled for preventing its boiling-up in the liquid line as a result of pressure losses and heating owing to some external heat inflows. The value of supercooling may be determined by the relative:

$$\left.\frac{\partial P}{\partial T}\right|_{T_v} \Delta T_{4\text{–}6} \geqslant \Delta P_{5\text{–}6}. \tag{4}$$

This condition may be regarded as the third condition of LHP serviceability.

It should be noted that the presence of a certain number of vapor bubbles in the liquid line, which can freely penetrate into the compensation chamber, does not interfere with the normal operation of the device, but only leads to an increase in the operating temperature. The boiling of a working fluid in the compensation chamber is not critical for LHPs either.

The design, the means of operation and the conditions of serviceability of LHPs discussed above determine quite a specific kind of operating characteristics of these devices, which are usually presented as the dependence of the vapor temperature on the heat flow $T_v = f(Q)$. In [22] it is shown that this dependence is determined by the relation:

$$T_v = T_0 + (T_{cc} - T_0)(R_2/R_1)^{CpQ/2\pi\ell_w K_w L}, \tag{5}$$

where T_0 is the temperature of the liquid that enters the compensation chamber; T_{cc} is the temperature of the compensation chamber; R_2, R_1 are the inner and the outer radii of the wick; ℓ_w is the length of the evaporation zone; k_w is the wick thermal conductivity; L is the latent heat of vaporization.

A typical kind of operating characteristics obtained by calculation and by experiment for a water and ammonia LHPs is presented in Fig. 4. Here one can see that the dependences have an ambiguous character. At first, with increasing heat load, the vapor temperature decreases, and then there appears a section of its quasi-stabilization. It is connected with the simultaneous action of three main factors: liberation of the condenser at the expense of gradual liquid displacement into the compensation chamber, lowering of the temperature of the latter with increasing mass flow rate of the cold liquid that enters it and the increase of the heat-exchange intensity

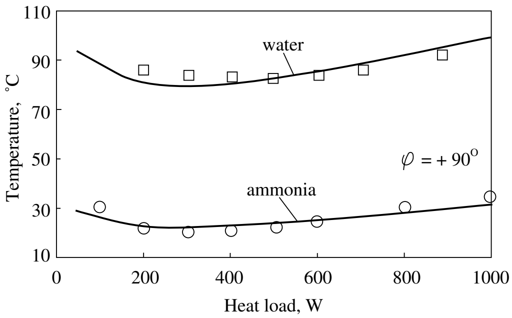


Fig. 4. Vapor temperature VS heat load for ammonia and water LHPs (——— calculation; □, ○ experiment).

in the evaporation zone. In this range of heat loads LHPs operate in the regime with a variable conductance. Once the compensation chamber has got completely filled with the working fluid, and the condensation surface stops changing, the regime of operation with a constant conductance sets in. The behavior of the dependence $T_v = f(Q)$ obtains here the character typical of conventional heat pipes.

3. Materials and working fluids for LHPs

The main structural material for making elements of the LHP body is stainless steel, which is amenable to different kinds of vacuum-hermetic welding and possesses a sufficiently high strength. The drawback of stainless steel is its relatively low thermal conductivity. More seldom use is made of more heat-conducting, but considerably less durable aluminium alloys. Copper is also quite a promising material for use in LHPs.

Sintered nickel and titanium powders are most widely distributed as wicks. They make it possible to obtain capillary structures with quite a small pore size, possess a high strength and are compatible with many low-temperature working fluids [23–26]. Table 2 gives the main characteristics of capillary structures sintered from metal powders. There are also successful examples of using cheaper polymeric, ceramic, copper and stainless steel porous materials.

The choice of working fluids for LHPs is realized on the basis of the same criteria as for conventional heat pipes. In this case, however, one should take into account the value of dP/dT, which is to be sufficiently high to ensure the device start-up at a minimum temperature difference between the evaporation zone and the compensation chamber in accordance with the condition

Table 2
Characteristics of capillary structures

Material	Porosity, %	Effective pore radius, μm	Permeability, $\times 10^{13}$, m^2	Thermal conductivity, W/m K
Nickel	60–75	0.7–10	0.2–20	5–10
Titanium	55–70	3–10	4–18	0.6–1.5
Copper	55–75	3–15	–	–

(3). With allowance made for it, the most efficient working fluid for LHPs in the temperature range from $-20\,°C$ to $+80\,°C$ is ammonia. The advantages of water manifest themselves at higher temperatures and reach the maximum at 100–150 °C. When the working fluid is water, one should bear in mind that freezing is impermissible because of the danger of disturbing the device tightness. Neon, oxygen, nitrogen, ethane, propane, propylene, freon 152A, freon 11, *n*-pentane, acetone and toluene were also successfully used as working fluids for LHPs.

4. Development and test results

4.1. Large and powerful LHPs

The advantages of LHPs are best manifested at a large capacity and heat-transfer distance. This especially concerns the cases when it is necessary to ensure the efficient operation of the device at any orientation in the gravity field. The most suitable design for this purpose is that of an LHP with two compensation chambers situated at the evaporator butt-ends [27]. As an example Fig. 5a presents the scheme of a flexible ammonia LHP 2 m long made of stainless steel [28]. The evaporator 24 mm in diameter with an active-zone length of 200 mm was equipped with a nickel wick with a break-down pore radius of 1.1 μm. The condenser 310 mm in length and 24 mm in diameter was made in the form of a pipe-in-pipe heat exchanger, whose inner and outer surface were cooled by running water at a temperature of $17 \pm 1\,°C$. The vapor and the liquid line 6 mm in diameter had flexible inserts 500 mm long.

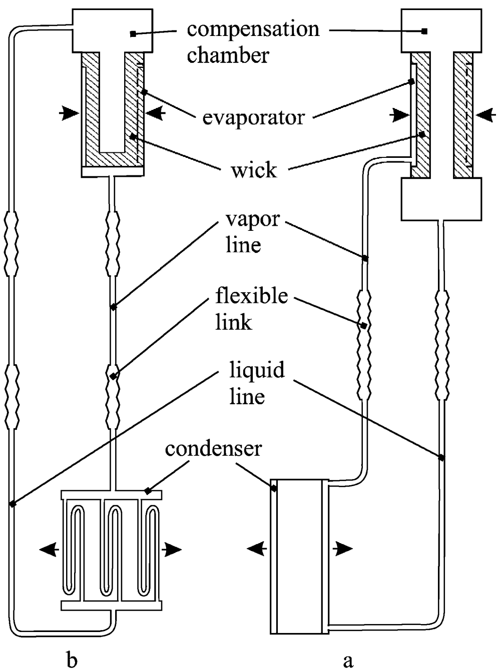


Fig. 5. Scheme of flexible LHPs with a high heat-transfer capacity.

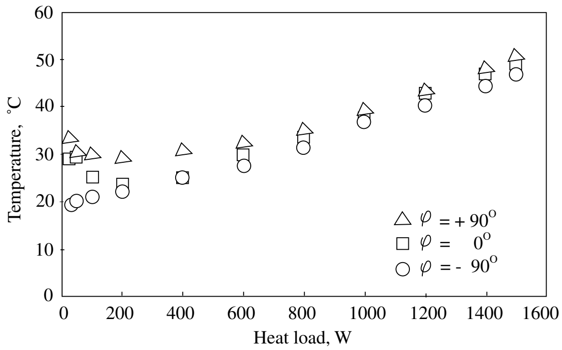


Fig. 6. Evaporator temperature VS heat load of the ammonia LHP 2m long.

The results of tests of the LHP are presented in Fig. 6 as heat-load dependences of the temperature of the evaporator wall at different orientations of the device in 1-g conditions. It can be seen that in the range of heat loads from 500 to 1500W, when the regime of LHP operation with a constant conductance is established, the evaporator temperature at different orientations varies quite moderately. In this case there is no evidence of crisis even at the maximum capacity achieved under these conditions of tests. At relatively low heat loads the temperature difference is more considerable, especially when operating in the "gravitational" regime with a slope $\varphi = -90°$, when the evaporator is below the condenser. In this case the conditions for replenishment of the evaporation zone are the most favorable. It should also be mentioned that this LHP is not the longest of all developed and tested successfully in the "antigravitational" regime. Presented in [29] are the results of development of an ammonia LHP 5.2m long, which transferred downwards a heat flow from 200 to 1000W. It was equipped with an evaporator 24mm in diameter with an active-zone length of 300mm and had a vapor and a liquid line 6mm in diameter. A heat exchanger of the collector type with a well-developed heat-exchange surface was used here as a condenser. The scheme of the devices is given in Fig. 5b.

The high capillary pressure that the wicks are capable of creating may be used not only for operation against the gravity forces but also for increasing the heat-transfer distance at a horizontal orientation. Such an example, in particular, is an ammonia LHP about 21m in length with a vapor and a liquid line 8mm and 6mm in diameter, which transferred 1.7kW at a temperature from 40 to 60°C [30]. The external view of the device is shown in Fig. 7. Such operating characteristics are not limiting for ammonia LHPs. If the working fluid is water in the temperature range that is optimal for it, the heat-transfer capacity of LHPs may be considerably increased.

4.2. Controllable LHPs

As it has been mentioned above, the temperature of LHPs may remain quasi-stable in a certain range of heat-load variation. It refers not only to the vapor temperature, but also to the temperature of the evaporator wall which is related to it through the heat-transfer coefficient. Thus, we can speak about the autocontrollability of LHPs as the maintenance of the temperature close to a certain level is realized automatically without any active external action, and also without the use of an inert gas, as in the case of gas-controlled heat pipes. The range of heat loads in which this

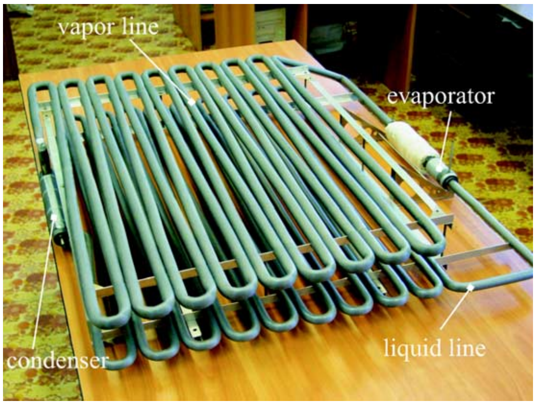


Fig. 7. General view of the ammonia LHP 21 m long.

property of LHPs manifests itself depends on the dimensions of the compensation chamber and the condenser, and also on the intensity of cooling of the latter. The larger they are, the wider the range of autocontrol [31]. However, as a rule, they are severely limited by the operating conditions. Besides, the condenser temperature may change over a wide range. In such cases active control over the LHP operating temperature is possible [22,32,33].

From Eq. (5) it follows that the vapor temperature in an LHP depends on the temperature of the compensation chamber and the temperature of the working fluid that enters it. It means that with the help of a controllable thermal action on the compensation chamber or the liquid line one can maintain the LHP operating temperature at a given level under changes of both the heat load and the condenser temperature. Such a thermal action may be both positive and negative, and its value does not usually exceed 5–10% of the heat flow transferred by the LHP.

As an example, Fig. 8 shows the scheme of active control of a small ammonia LHP, its external view and test results. It is shown that as the heat load ranges from 6 to 10 W and the condenser cooling temperature from −10 °C to +20 °C active control makes it possible to maintain the temperature of the evaporator interface at the level of 42 °C with an accuracy of ±0.1 °C.

4.3. LHPs with a high heat flux

The evaporation zone in LHPs is organized, as a rule, on the basis of the principle of "inverted menisci" [34]. In accordance with this the evaporation of a working fluid takes place in a relatively thin layer of the wick immediately at the hot wall of the evaporator, towards which the evaporating menisci are turned, and the heat flow is directed towards the liquid flow. To organize effective heat exchange here, it is necessary to create a special system of vapor-removal channels located at the wall-wick boundary. A system of such channels may be one-, two- or three-step. In the first, the simplest case vapor-removal channels are situated only on the wick or on the evaporator wall. A two-step system presupposes the presence of small azimuthal grooves on the evaporator wall and larger longitudinal channels on the wick. In the three-step variant the so-called biporous layer is added here. It is a fine-pored wick penetrated by communicating chains of larger pores. A typical scheme of such an evaporation zone of LHPs and a mathematical model of heat exchange are

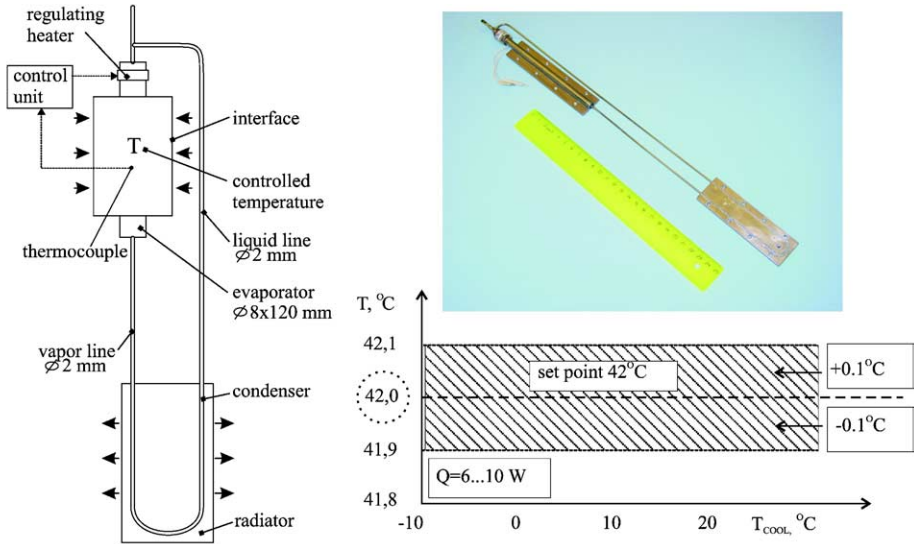


Fig. 8. Scheme and general view of the LHP with an active temperature control.

presented in papers [35,36]. In some cases a two-step evaporation zone including only a biporous layer and longitudinal channels is also quite efficient [37].

The developed system of vapor-removal channels in LHPs makes it possible to achieve very high parameters characterizing heat exchange in the evaporation zone. Thus, in particular, an ammonia LHP with a three-step evaporation zone, whose external view and scheme are shown in Fig. 9, demonstrated a record value of the heat flux maximum equal to 130 W/cm^2 at any positions of the device in the gravity field. Maximum values of the heat-exchange coefficients at the wall-wick boundary ranged between 33,000–44,000 W/cm^2 K and were achieved, respectively, at heat fluxes of 10 and 30 W/cm^2. The LHP effective length was equal to 300 mm, the evaporator diameter to 20 mm, and the heat load was supplied with the help of a "thermal wedge" at a surface

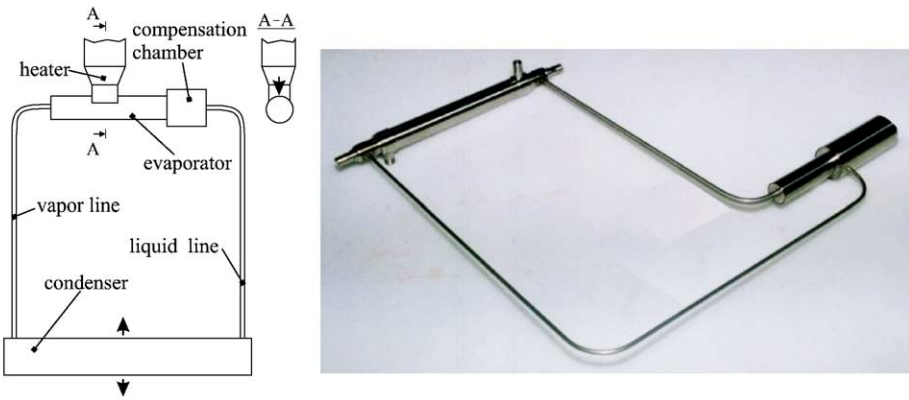


Fig. 9. Scheme and general view of the LHP with a high flux in the evaporation zone.

of 4.25 cm^2 [35]. Another LHP of similar dimensions and configuration, but having a two-step evaporation zone with a biporous wick demonstrated in the horizontal position a maximum heat flux of about 78 W/cm^2. The maximum value of the heat-exchange coefficient was 100,000 W/m^2 K in the heat-load range 1–2 W/cm^2 [37]. At the maximum heat flux the heat-exchange coefficients of both LHPs were approximately the same and equal to 15,000–18,000 W/m^2 K.

In the first case an advantage in the value of the heat flux was achieved at the expense of using a fine-pored wick and a more developed system of vapor-removal channels. In the second case an advantage in the heat-transfer coefficient was obtained owing to the use of a large-pored wick, which created more favorable conditions for heat exchange in the evaporation zone, but at the same time ensured an efficient operation of the device only at its horizontal orientation.

4.4. Ramified LHPs

The LHP principle makes it easy to create ramified heat-transfer devices including different numbers of evaporators and condensers situated in different ways with respect to each other [38–43]. Such devices are regarded, in particular, as a very promising means for thermoregulation systems of spacecraft, which makes it possible to reduce the mass of a system and make it more compact. The most typical cases when the use of such devices may be considered justified are the following:

- the heat-load source has a large contact surface and/or capacity. In this case several parallel evaporators are joined by a common "cold plate", on which the heat load is located;
- several heat-load sources are situated at a sufficiently large distance from one another. In this case each of them is in thermal contact with a separate evaporator;
- there are several heat sinks removed from one another. In this case the LHP has a corresponding number of condensers having their own heat sink.

Some variants of the lay-out diagrams of ramified LHPs are shown in Fig. 10. In particular, one of them (Fig. 10a) was used for developing an LHP 1 m long with ammonia as a working fluid.

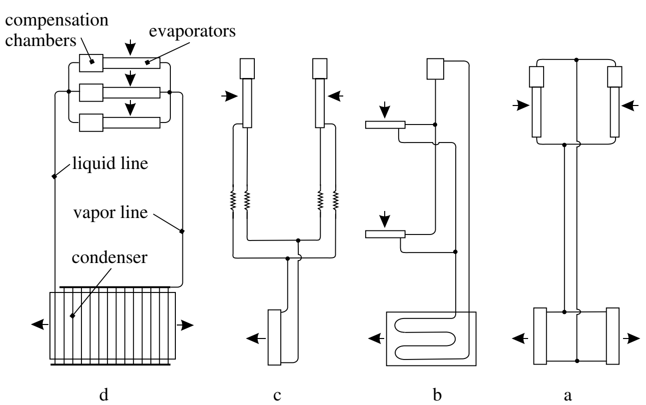


Fig. 10. Scheme of ramified LHP.

The device was equipped with two similar evaporators 24 mm in diameter and two condensers of the "pipe-in-pipe" type situated in parallel and symmetrically about the vapor line of diameter 4 mm. LHP tests demonstrated a stable operation at different orientations of the device, including the "antigravitational" regime, symmetric and asymmetric heat-load distribution between the evaporators. The maximum total capacity achieved on trials was 1100–1400 W depending on the LHP orientation. The maximum capacity of one evaporator with a zero heat load on the other was, respectively, 900 W and 1100 W. The device also functioned steadily at different temperatures of cooling the condensers when the difference did not exceed 70 °C.

4.5. Reversible LHPs

As distinct from a conventional heat pipe, an LHP is a "natural" thermal diode. It means that it can transfer heat only in one direction from the evaporator to the condenser. At the same time quite attractive is the possibility to use the advantages of LHPs also in cases requiring the reverse of a heat flow, if the heat source and the heat sink change places. Such a problem exists, for instance, for television communication satellites in a geostationary orbit. They require an efficient thermal bond between the north and the south radiators, whose illuminance changes from time to time, and there arises a necessity of additional heating of one of them.

In this connection special reversible loop heat pipes have been developed (RLHPs). They have two similar evaporators capable of acting as condensers during the reverse of a heat flow. Fig. 11 shows the scheme of such a device. The paper [44], in particular, presents the results of development and tests of an RLHP with freon-12 as a working fluid, which demonstrated the ability to transfer 60 W in both directions at a temperature drop of 6 °C. Another paper [45] gives the results of development and tests of a 2-m ammonia RLHP which transferred up to 900 W at a temperature drop of 31 °C. Here it is also shown that the evaporator provided with azimuthal grooves at the inner surface of the body quite efficiently fulfils the functions of a condenser. Vapor penetrating into the longitudinal vapor-removal channels located at the wick surface near the evaporator wall is freely distributed along the azimuthal grooves, where it is cooled and condensed. The forming liquid film is sucked off by the wick, as a result of which the thermal resistance of the condenser decreases.

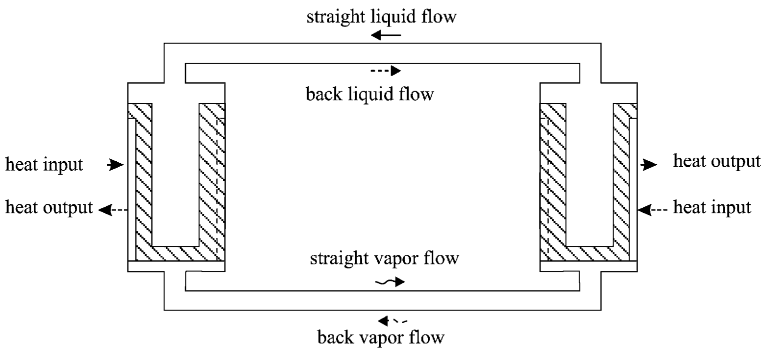


Fig. 11. Reversible LHP scheme.

4.6. LHPs with flat evaporators

Most often objects to be cooled have a flat thermocontact surface. Therefore, to ensure a thermal contact with a cylindrical evaporator, one needs a special thermal interface, which is a "cylinder-plane" reducer located at the surface of the evaporator active zone. Despite the fact that such interfaces are usually made of heat-conductivity materials, such as aluminium or copper, they possess a sufficiently high thermal resistance of their own, which increases the total thermal resistance and the mass of an LHP. For conventional heat pipes an immediate contact with a flat heat source is not a special problem as it is quite easy to make them flat or flat-oval. In this case their thickness is not a critical parameter. The possibilities of LHPs for decreasing the evaporator thickness are very limited. The reason is that the evaporator, as a rule, is combined in one body with the compensation chamber, and the thickness of the wick that separates the evaporation zone and the compensation chamber cannot be too small. Besides, the flat walls of the evaporator have to be sufficiently thick to resist the vapor internal pressure, which in most working fluids used in LHPs may be quite high.

One of the variants of solving this problem is the use of flat disk-shaped evaporators in LHPs [46]. The scheme and the external view of such evaporators 10 and 13 mm thick, whose thermocontact surface is made in the form of a flange 45 mm in diameter for fixing the heat source, are shown in Fig. 12. The papers [47,48] contain the results of development of ammonia LHPs 0.86 m and 1 m long with a vapor and a liquid line 2 mm in diameter equipped with such evaporators of stainless steel. On trials the devices demonstrated serviceability at any orientations in 1-g conditions. The maximum capacity was, respectively, 90–110 W and 120–160 W depending on the orientation, and the value of the minimum thermal resistance 0.30 K/W and 0.42 K/W. The external view of LHPs with flat disk-shaped evaporators is shown in Fig. 13.

The paper [49] presents the results of testing another ammonia LHP with an effective length of about 0.5 m equipped with a flat evaporator 14 mm thick, whose body was made of an aluminum alloy, and the wick of a polymeric material. The mass of such an evaporator did not exceed 13 g. The maximum capacity achieved in testing the device was 50 W, and the thermal resistance 0.5 K/W.

4.7. Miniature LHPs

At present conventional heat pipes are most widely used for cooling electronic devices and computers. At the same time, the tendency to a constant increase of heat load on the functional

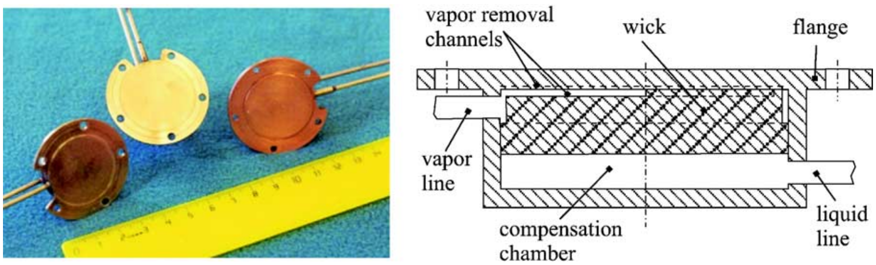


Fig. 12. Scheme and external view of flat disk-shaped evaporators.

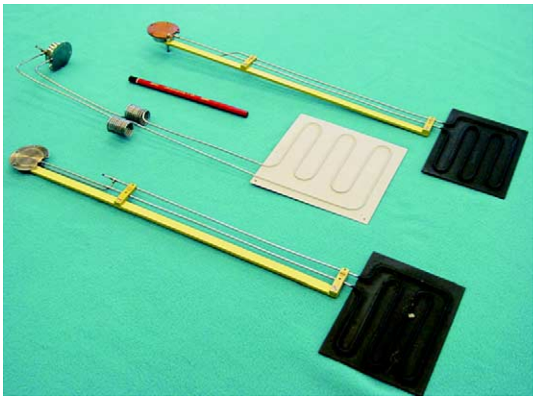


Fig. 13. External view of LHPs with flat evaporators.

components of this equipment with a simultaneous decrease of its mass and dimensions creates a situation when heat pipes can no longer cope with heat removal. New, more efficient and sufficiently miniature heat-transfer devices are required.

It has been mentioned above that the advantages of LHPs are most pronounced at large dimensions and capacities of these devices. However, new challenges make us evaluate differently possibilities for developing the LHP principle directed at the creation of miniature and more efficient heat-transfer devices. Two main problems arise on this way. The first of them is connected with the decrease of the evaporator diameter, which causes a corresponding decrease in the thickness of the wick separating its evaporating and absorbing surfaces. A thin wick gives no way of creating the temperature and pressure drop required, according to condition (3), for the start-up and operation of the device. This necessitates an appropriate adaptation of the evaporator design, which should not be beyond the scope of the LHP base conception. The use of wicks with a low thermal conductivity for obtaining the required temperature drop does not solve the problem in full measure, as it hinders the solution of the second problem. This is the problem of decreasing the thermal resistance of miniature LHPs having by definition quite limited heat-exchange surfaces in the evaporation and condensation zones. First of all, it concerns evaporators, which are the main critical components of LHPs. A decrease in thermal resistance here is possible only at the expense of maximum heat-exchange intensification during the evaporation of a working fluid from a sufficiently heat-conducting capillary structure.

The use of a modified conception makes it possible to create efficient LHPs with an evaporator diameter of 6 mm and less [50]. The external view of such devices is shown in Fig. 14. The paper [51] presents the results of development and tests of miniature LHPs made of stainless steel and copper with ammonia and water, respectively, as working fluids. The ammonia LHP had an evaporator of diameter 5 mm with a titanium wick and lines for vapor and liquid 2 mm in diameter. The water LHP was equipped with an evaporator of diameter 6 mm and lines of diameter 2.5 mm. The effective length of the devices was about 300 mm. Each of them had a condenser 62 mm long provided with finning, whose total surface was about 400 cm^2. To cool the condensers, use was made of a fan creating an air flow rate of 0.64 m^3/min at a temperature of 22 ± 2 °C.

Tests showed that the maximum capacity of the ammonia LHP was 95 W at an evaporator-wall temperature of 93 °C. The maximum capacity for the water LHP was not achieved, but at the

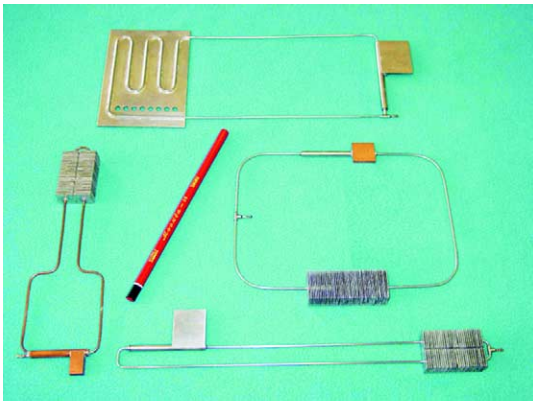


Fig. 14. Miniature LHPs external view.

same temperature it was equal to 130 W. The minimum values of thermal resistance of the LHP equal to 0.12 K/W and 0.1 K/W were obtained at heat loads of 70 W and 130 W, respectively. It should be noted that the ammonia LHP demonstrated a higher value of the heat-transfer coefficient in the evaporator, which reached 78,000 $W/m^2 K$ at a heat-flow density of 21.2 W/cm^2 at the surface of an interface with an area of 4 cm^2. For the water LHP these values were, respectively, 31,700 $W/m^2 K$ and 35 W/cm^2. It should be added that in this case immediately at the surface of the evaporator's active zone the heat-flow density was much higher. For the ammonia LHP it was 44.5 W/cm^2, and for the water one 69.1 W/cm^2.

5. Application of LHPs

5.1. Thermoregulation systems of spacecraft

At present the main area of application of loop heat pipes is space technology. The first flight experiment in conditions of microgravitation was conducted in 1989 aboard the Russian spacecraft "Gorizont" [31]. The experimental module included an LHP with three parallel evaporators 24 mm in diameter and a condenser of the collector type joined to the panel of a radiator. The vapor and the liquid line 6 mm in diameter had a length of about 0.6 m. The evaporators were combined with a common plate measuring 100×100 mm, on which heat-load simulators of capacity 40, 80 and 120 W were located. Freon-11 was used as a working fluid. The external view and the scheme of the experimental module are shown in Fig. 15. The main aim of the experiment—to demonstrate the serviceability of LHP in space conditions—was achieved. At the same time, problems arose at the start-up when the spacecraft found itself in the shadow of the Earth, and the temperature became too low for freon-11. Therefore for the next flight experiment, which was carried out in the same year aboard the spacecraft "Granat", use was made of an LHP with propylene as a working fluid [52]. The telemetric data that came from the board in the period from 1989 till 1996 pointed to the normal operation of the device [53].

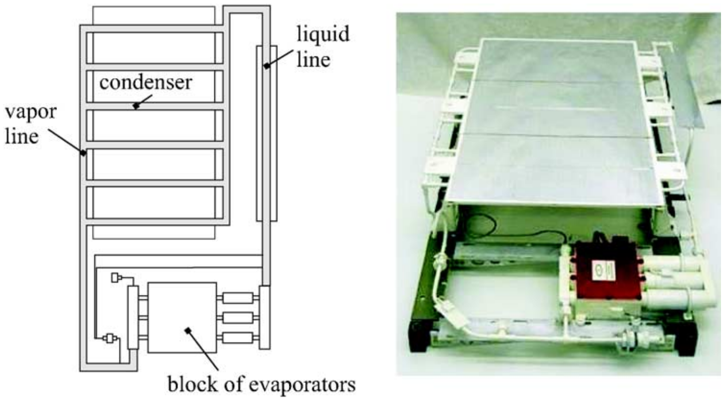


Fig. 15. Scheme and external view of the first LHP flight experiment.

The first actual application of LHPs took place in 1994 aboard the Russian satellite "Obzor", where two ammonia and one propylene LHPs were installed in the thermoregulation system of the unit of optical instruments [54]. The scheme of the thermoregulation system is given in Fig. 16.

At present LHPs are finding ever widening use in space, where several tens of such devices are already successfully used [55,56]. In particular, a unique thermoregulation system including six LHPs was installed aboard the Russian spacecraft "Mars-96". The same number of LHPs is used in the cooling system of nickel–cadmium batteries on a Chinese meteorological satellite FY-IC. Sixty two ammonia LHPs are installed on six American satellites Hughes-702. On another American spacecraft ICESar, launched in 2003, use is made of two propylene LHPs with active control for cooling the Geoscience Laser Altimetry System (GLAS).

In conditions of microgravitation the ability of LHPs to operate in the "antigravitational" regime is not required, but during the ground-based preparation of spacecraft this quality is necessary in many cases. Besides, in creating space-purpose thermoregulation systems use is made of such advantages of these devices as a high heat-transfer capacity, a low thermal resistance, mechanical flexibility and excellent adaptability to various conditions of arrangement.

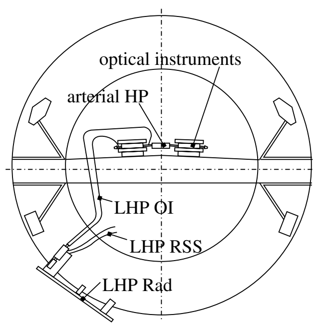


Fig. 16. Thermal control system scheme of the spacecraft "Obzor".

5.2. Cooling of electronics and computers

Electronics and computers are quite a promising sphere of LHPs application. Here they can supplement or replace conventional heat pipes and create favorable conditions for further development of this sphere of technology.

The first actual application of LHPs in electronics dates back to the end of the seventies, when they were used as a duplicate system for cooling of a unit of powerful transistors. The devices were made of stainless steel and equipped with evaporators 22 cm in diameter with a nickel wick. The condensers had the form of a flat coil joined to the base of a ribbed radiator. The nominal capacity of LHPs with acetone as a working fluid was 30 W. The external view of the unit with an LHP is shown in Fig. 17.

The next step on the road of using LHPs in this direction was the creation of cooling systems for quantum-electronic converters, whose total capacity was 10 W. The problem of cooling lay in the fact that the temperature of the converters had to be maintained at a level of no more than +30 °C at an ambient temperature of +25 °C. The use of any forced convection was excluded. The problem was successfully solved with the help of an LHP with an evaporator 10 mm in diameter and lines for vapor and liquid 3 mm in diameter, which had a rather complicated configuration. As in the first case, the condenser was made in the form of a flat coil joined to a ribbed radiator. The working fluid was freon-152A. The scheme of the device is shown in Fig. 18.

Computer technology, in particular PC "Notebook", is a new sphere of LHPs application, which was revealed owing to the appearance of miniature and fairly efficient devices [57,58]. The first experience in this direction was obtained in 2001, when a number of compact coolers for CPU with a mass of about 50 g were created on the basis of LHPs. They dissipated a heat flow at the level of 25–30 W. The scheme and the external view of such coolers are shown in Fig. 19. At present a cooler on the basis of a copper-water LHP with an evaporator diameter of 6 mm is being tested. It is intended for the PC "Notebook" with a processor Athlon XP, which at a maximum loading dissipates about 70 W. By predictions of experts, in the near future one can expect the appearance of more powerful processors dissipating 100 W and more, for which LHPs may prove to be the only alternative.

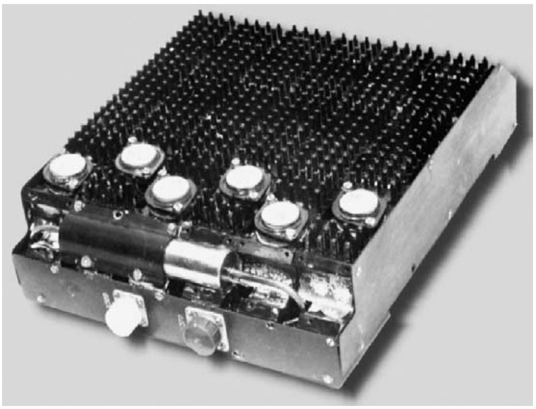


Fig. 17. General view of the electronic block with an LHP.

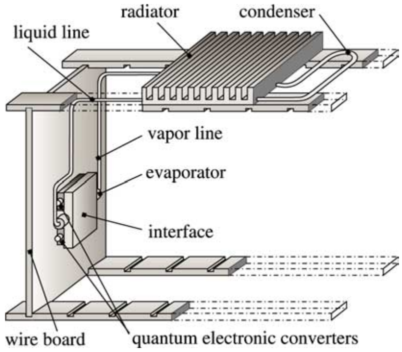
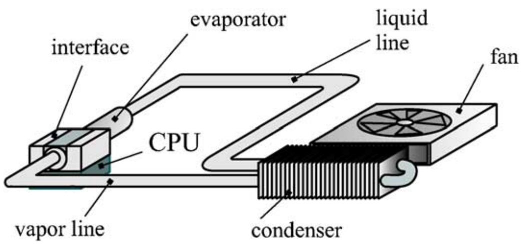


Fig. 18. Scheme of cooling quantum-electronic converters with an LHP.



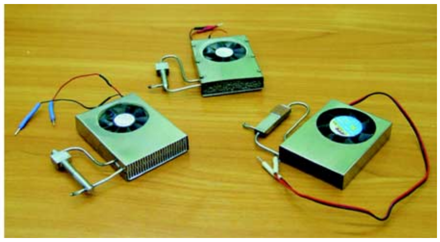


Fig. 19. Scheme and general view of CPU coolers with an LHP.

6. Conclusion

Loop heat pipes are highly efficient heat-transfer devices capable of transferring considerable heat flows for great distances at any orientation in 1-g and 0-g conditions. On their basis it is possible to create ramified, reversible, controllable systems for heat-transfer possessing mechanical flexibility and high adaptability to various operating conditions. A new generation of these devices—miniature heat pipes—can solve the problem of cooling of promising electronics and computer equipment.

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