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Operating temperature and distribution of a working fluid in LHP

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Abstract

One of the main factors that influence the operating temperature of a loop heat pipe (LHP) is the distribution of a working fluid in the device. The paper presents the classification of LHP operating modes on the basis of the criterion of presence or absence of the working-fluid vapor phase in the compensation chamber (CC). It gives a description of method of calculating the LHP operating temperature for every operating mode and shows the characteristic features, advantages and disadvantages of every mode.

Keywords: Loop heat pipe; Operating temperature; Operating mode; Temperature hysteresis

1. Introduction

The operation of many technical objects is accompanied by the heat emission Q^{+}, which disturbs the thermal regime of their operation. This heat is harmful and should be removed. The conditions for heat removal are significantly complicated when the object to be cooled and the heat sink are situated at a considerable distance from each other. To ensure an efficient thermal link between them, it is necessary to use an auxiliary element with a high thermal conductivity. A loop heat pipe [1] may be used as such an element. Presented in Fig. 1 is the scheme of one possible version of LHP. The device contains an evaporator and a condenser connected by smooth-walled pipelines for separate motion of vapor and liquid, which have a relatively small diameter. The evaporator is equipped with a special wick and joined to a compensation chamber, which serves to receive the working fluid displaced from the vapor line and the condenser during the start-up and from the condenser in the process of operation of the device.

By its functional definition the LHP is a heat-transfer device operating on a closed evaporation-condensation cycle with the use of capillary forces for pumping a working fluid. Owing to this the device contains no mechanically mobile parts and consumes no additional energy. The heat transfer in the LHP is realized with the use of latent heat of vaporization, which ensures a relatively small mass flow rate of a working fluid inside the device and an intense heat exchange in the evaporation and the condensation zone. A special system of vapor-removal channels in the evaporation zone, intended for an organized vapor removal, makes the process of evaporation at the surface of a fine-pored capillary structure as efficient as possible.

Loop heat pipes have a high heat-transfer capacity $Q_{\mathrm{load}} \cdot L_{\mathrm{LHP}}$ and at the same time a low thermal resistance R_{LHP} [2], which is determined in the following way:

$$R_{\mathrm{LHP}} = \frac{T_{\mathrm{ev}} - T_{\mathrm{cond\_ext}}}{Q_{\mathrm{load}}}, \qquad (1)$$

where T_{ev} is the temperature at the external heat-receiving surface of the evaporator, $T_{\mathrm{cond\_ext}}$ is the temperature at the external heat-losing surface of the condenser, Q_{load} is the heat load transferred by an LHP, L_{LHP} is the effective length of heat transfer.

Fig. 2 shows the scheme of a system of thermoregulation for a heat-generating object organized on the basis of an LHP. The length L_{LHP} corresponds to the distance between

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Nomenclature

G	working-fluid mass flow rate, kg s^{-1}
L	length, m
P	pressure, Pa
Q	heat load, W
Q_{ch}	heat load at which the LHP operating modes change, W
R	thermal resistance, K W^{-1}
T	temperature, K
V	volume, m^3
Greek symbols	
α	heat-transfer coefficient, W m^{-2} K^{-1}
d	diameter, m
φ	slope, deg (°)
k	heat of vaporization, J kg^{-1}
g	free fall acceleration, m s^{-2}
μ	dynamic viscosity, Pa s
P	density, kg m^{-3}
Subscripts and superscripts	
cond	condenser
cc	compensation chamber
ev	evaporator
ext	external
int	internal
l	liquid
ll	liquid line
sink	heat sink or heat receiver
source	heat source
v	vapor
vc	vapor removal channel
vl	vapor line

the object to be cooled and the heat receiver. The temperature of the object being cooled T_{source} exceeds the temperature of the heat receiver T_{sink} by a certain value $\Delta T_\Sigma = T_{source} - T_{sink}$, which according to the diagram of distribution of temperatures in such a system (see Fig. 2) is determined by the contribution of three components:

$$\Delta T_\Sigma = \Delta T_{source} + \Delta T_{LHP} + \Delta T_{sink}, \quad (2)$$

viz. the temperature drops in the heat-supply zone $\Delta T_{source} = T_{source} - T_{ev}$ and in the zone of heat rejection

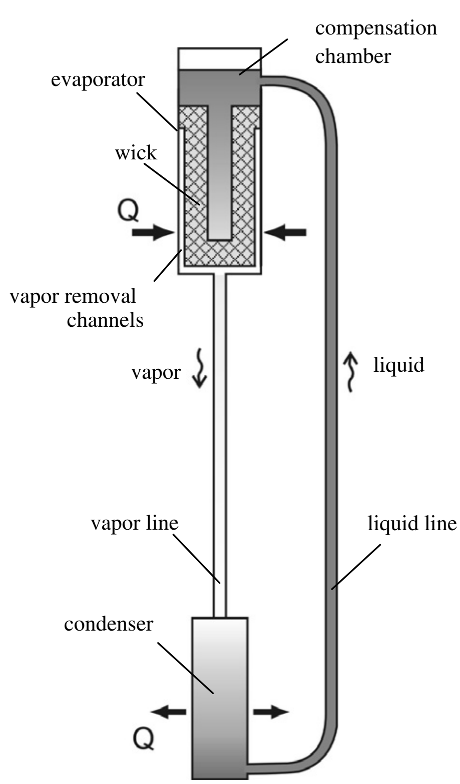

Fig. 1. Scheme of a loop heat pipe.

$\Delta T_{sink} = T_{cond\_ext} - T_{sink}$, and also the temperature drop ΔT_{LHP}, which is caused by the LHP thermal conductance:

$$\Delta T_{LHP} = T_{ev} - T_{cond\_ext}. \quad (3)$$

In this case, according to the data from Ref. [3], less than 50% of the total temperature difference ΔT_Σ falls to the share of ΔT_{LHP}. Therefore for increasing the efficiency of heat transfer in the thermal regulation system under consideration it is necessary to pay attention to each of the three components of the total temperature drop ΔT_Σ.

The temperature jump ΔT_{source} in the heat-supply zone is caused by the nonideal character of the thermal contact between the LHP evaporator and the object being cooled. The quality of the thermal contact joint between them depends on the means of joining used in a concrete thermoregulation system. If the thermal resistance of the contact zone is equal to R_{cont}, and the heat-release capacity of the object being cooled is equal to Q^+, one can write:

$$T_{source} - T_{ev} = Q^+ \cdot R_{cont}. \quad (4)$$

As for the temperature drop in the zone of heat-load rejection ΔT_{sink} (see Fig. 2), the relationship between the temperature at the external heat-losing surface of the condenser T_{cond_ext} and the temperature of the heat receiver ΔT_{sink} may be expressed by the following relation:

$$T_{cond\_ext} - T_{sink} = \frac{Q^-}{\alpha_{cond\_ext} \cdot S_{cond\_ext}}, \quad (5)$$

where S_{cond_ext} is the external heat-losing surface of the condenser, α_{cond_ext} is the coefficient of heat exchange at the external heat-losing surface of the condenser, Q^- is thermal capacity rejected. In most technical problems the heat receiver has a sufficiently high heat capacity, which is why with time its temperature remains unchanged

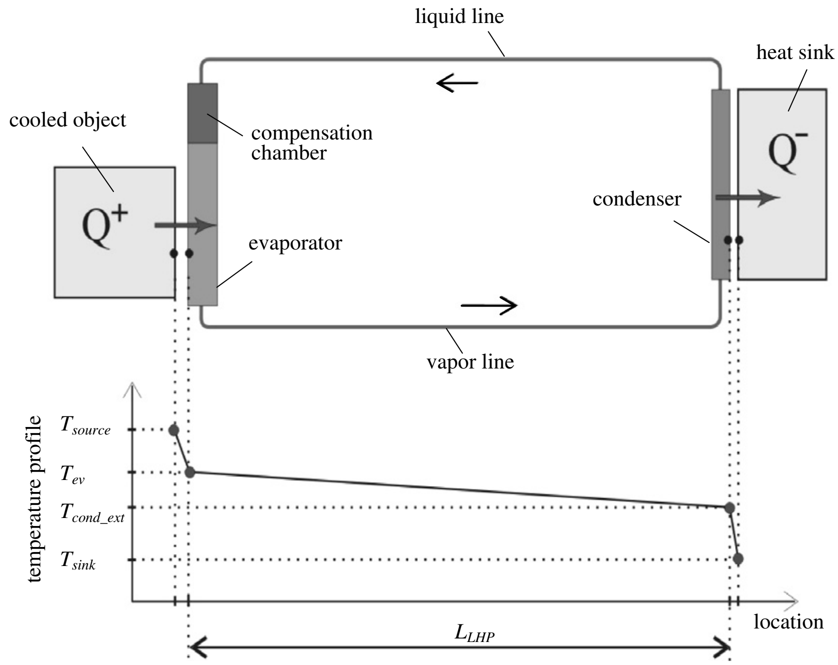

Fig. 2. Scheme of a thermoregulation system on the basis of LHP and diagram of temperature distribution.

$T_{\mathrm{sink}} = \mathrm{const}$. In the case that the heat exchange of the system under consideration with the surrounding medium is absent or reduced to a minimum we have

$$Q^{+} \approx Q^{-} \approx Q_{\mathrm{load}}. \tag{6}$$

Assuming this condition, later we will use Q_{load}.

The temperature of the object to be cooled, as the main quantity sought, may be determined by stages, through intermediate calculations of temperature values at the control points of the thermoregulation system under consideration according to the following computational scheme:

$$T_{\mathrm{sink}} \Rightarrow T_{\mathrm{cond\_ext}} \Rightarrow T_{\mathrm{ev}} \Rightarrow T_{\mathrm{source}}. \tag{7}$$

Usually the temperature of the heat sink T_{sink} is given. Then $T_{\mathrm{cond\_ext}}$ is determined by formula (5) with allowance for the conditions of the external heat rejection $S_{\mathrm{cond\_ext}} \cdot \alpha_{\mathrm{cond\_ext}}$ and the heat-release capacity on the object being cooled. Thereafter the temperature T_{ev} is determined, whose peculiarities of formation will be discussed later. And at the last stage of the main procedure of calculation formula (4) is used to determine the temperature level of the thermostatted object T_{source}.

According to the above scheme (7), the thermal state of the object being cooled is, finally, determined by the temperature T_{ev} at the external surface of the LHP evaporator, and therefore it is quite an important parameter of the thermodynamic system under consideration. Besides, the temperature T_{ev} characterizes the operating condition of the heat-transfer device itself, or, in other words, the loop heat pipe operating temperature. And, lastly, the LHP thermal resistance, which is one of the critical parameters that characterize the heat-transfer device efficiency, is also determined according to (1) through the value of T_{ev}.

In Ref. [4] it is noted that the LHP operating temperature depends on a great number of factors and parameters. Several of them are external with respect to the heat-transfer device. Among these are the temperature of the heat sink, the action of fields of mass forces, the relative positions of the heat source and the sink, and if there is heat exchange with the ambient medium, its temperature as well. Other factors and parameters are internal, i.e. they characterize the device itself and are determined by its design features. To the internal factors and parameters one should refer the thermophysical characteristics of the material and the working fluid in use, the quantity of the working fluid V_{ch} in the LHP, and also the distribution of the working-fluid liquid phase inside the heat-transfer device.

The role of the last-mentioned factor is caused by the following circumstances. First, by the fact that the LHP is a two-phase heat-transfer device. Second, by the presence of such a constituent in the LHP design as a compensation chamber. The third reason is the fact that the LHP is a passive heat-transfer system the provision of serviceability of which does not require any additional external thermal or mechanical actions controlling the distribution of the working-fluid liquid phase inside the LHP and regulating the filling of the CC (or hydroaccumulator reserve), as is done, for instance, in the CPL. At the same time it is known that the presence or the absence of the working-fluid phase in the CC, and also the position of the vapor–liquid interface in it, may have a profound effect on the value of the LHP operating temperature and the character of the behavior of the functional dependence $T_{\mathrm{ev}} = f(Q_{\mathrm{load}})$ [5–9]. From this it follows that this topic is of undoubted interest for investigation. However, no systematic analysis was previously made in this sphere. Therefore the aim of the present paper is to investigate the effect of the liquid distribution in the LHP on its operating temperature.

2. The main LHP operating modes

The compensation chamber is an important element in the LHP design. The CC is intended to accumulate the liquid which is driven out of the vapor line and the condenser during a start-up. An obligatory condition of a successful LHP start-up is a complete liberation of the vapor line from the liquid because only in this case the vapor that is generated in the evaporation zone can reach the condenser. At the same time a successful start-up and a stable operation of the device do not require a complete liberation of the condenser. Only such an amount of liquid is driven out of the condenser into the CC as is required for the formation of a condenser inner surface S_{cond_in} sufficient for the condensation of vapor, whose mass flow rate depends on the value of the heat load transferred Q_{load}:

$$G = \frac{Q_{load}}{k}, \tag{8}$$

where G is the mass flow rate of the working fluid, k is the heat vaporization. Therefore in the process of LHP operation under changes of heat load Q_{load} the extent of liberation of the condenser from the liquid may be different. Consequently, there may also be changes in the extent of the CC filling in the range from some minimum value at low Q_{load} to a complete filling at high Q_{load}.

Of considerable importance for the distribution of a working fluid in the LHP is the orientation of the device in the gravity field, or, in other words, the relative positions of the condenser and the evaporator to which the CC is joined. According to the generally accepted designation, the LHP slope $\varphi = +90^\circ$ corresponds to the vertical orientation of the device, with the evaporator located above the condenser. At the horizontal orientation with $\varphi = 0^\circ$ the evaporator and condenser are at the same level. At $\varphi = -90^\circ$ the device is oriented vertically, but the evaporator is under the condenser.

For the orientation of an LHP with slopes $\varphi \geqslant 0^\circ$ LHP operation with a fully-filled CC is not always possible, only at certain relations between the CC volume V_{cc} and the volume of the working-fluid charge V_{ch} [7]. At LHP orientations with slopes $\varphi < 0^\circ$ the conditions prove to be more favourable for the CC filling as in this case the evaporator is situated below the condenser and the liquid flowing down by gravity is poured into the compensation chamber. The physical principles that form the basis for the LHP operation allow the device to function both with a fully- and with a partially-filled CC. Therefore the presence or the absence of a vapor–liquid interface in the CC may be used as a criterion for classification of the LHP operating modes. According to such a classification there are two main modes of LHP operation: the mode of operation with a filled CC (the vapor–liquid interface is absent here); the mode of operation with an unfilled CC (the vapor–liquid interface is present). Investigations have shown that each mode has its peculiarities, and the operating temperature T_{ev} forms depending on the LHP operating mode.

3. Formation of the LHP operating temperature in the absence of a vapor–liquid interface in the compensation chamber

Let us examine the LHP operation in the mode when the compensation chamber is fully filled with a liquid. In this mode the surface that is free for vapor condensation in the condenser S_{cond_in} does not change and is determined by the amount of a working fluid in the device V_{ch}. Making use of the fact that both in the evaporation and in the condensation zone there are vapor–liquid interfaces, we can write down an equation which relates the temperature T_{cond} and the pressure P_{cond} of saturated vapor in the condensation zone to the temperature T_v and the pressure P_v of the saturated vapor in the evaporation zone:

$$\left.\frac{dP}{dT}\right|_{\overline{T}} \cdot (T_v - T_{cond}) = \Delta P_v, \tag{9}$$

where $\frac{dP}{dT}|_{\overline{T}}$ is the derivative that characterizes the slope of the saturation line at the temperature $\overline{T} = \frac{T_v - T_{cond}}{2}$, ΔP_v shows the pressure losses during the motion of vapor from the evaporator into the condenser, which are the result of the pressure drop on every ith section of the way along which the vapor is moving:

$$\Delta P_v = \sum_i \Delta P_i. \tag{10}$$

Writing down Eq. (10) in expanded form, we have:

$$\Delta P_v = \Delta P_{vc} + \Delta P_{vl} + \Delta P_{v\_cond}. \tag{11}$$

The terms in the right-hand side of Eq. (11) take into account the pressure drop in the vapor removal channels ΔP_{vc}, in the vapor line ΔP_{vl} and in the vapor section of the condenser ΔP_{v_cond}, respectively.

If the heat-transfer coefficient in the evaporation zone is taken equal to α_{ev}, the relation between the temperature T_{ev} and the vapor temperature in the evaporation zone T_v may be written as follows:

$$T_{ev} = T_v + \frac{Q_{load}}{\alpha_{ev} \cdot S_q}, \tag{12}$$

where S_q is the evaporator surface to which heat is supplied. Then with allowance for (9), (10) and (12) the expression for the temperature T_{ev} will look like:

$$T_{ev} = T_{cond} + \left.\frac{dT}{dP}\right|_{\overline{T}} \cdot \sum_i \Delta P_i + \frac{Q_{load}}{\alpha_{ev} \cdot S_q}. \tag{13}$$

Eq. (13) is the base one in calculating the LHP operating temperature depending on the heat load for the LHP operating mode with a filled compensation chamber.

Fig. 3 presents an experimental dependence of the operating temperature T_{ev} on the heat load Q_{load} obtained in testing a vertically-oriented LHP at $\varphi = -90^\circ$. The

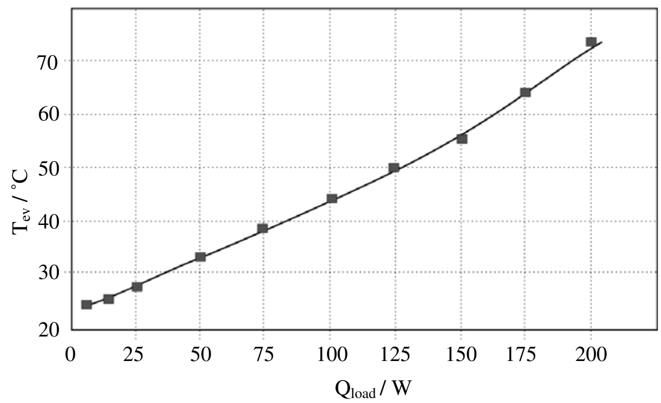

Fig. 3. Heat–load dependence of the operating temperature. The working fluid is ammonia, $\varphi = -90°$, $T_{sink} = 19\,°C$. (—) Calculation; (■) experiment.

condenser was above the evaporator, which created favourable conditions for liberation of the condenser from the liquid, which drained down into the CC. Consequently, it can be said with assurance that the LHP was functioning in the mode with a filled CC. The working fluid was ammonia. The temperature of the heat sink was equal to 19 °C. The heat-transfer length was 500 mm.

From the graph it is evident that the experimental curve had a near-linear shape. In the whole range of changing heat load the operating temperature increases with increasing Q_{load}. In the graph one can also see the calculated dependence $T_{ev} = f(Q_{load})$ obtained with the use of Eq. (13). It should be noted that one can observe here quite a good agreement between the results of calculation and experiment. In the whole range of changing heat load Q_{load} the calculation and the experimental curve behave identically, having in doing so almost a linear dependence. Such an operating mode is often called the mode of constant conductance.

It can be shown that it is precisely the linelike type of the operating curve $T_{ev} = f(Q_{load})$ that is characteristic of the operating mode with a filled CC. For this purpose we transform Eq. (13). We make use of relation (8), which relates the mass flow rate of the working fluid with the heat load Q_{load}, and present the pressure drop ΔP_i on every ith transportation section for a laminar and a noncompressible vapor flow in the following way:

$$\Delta P_i = W_i \cdot F_i \cdot Q_{load}, \tag{14}$$

where W_i is the complex taking into account the geometrical parameters of the transportation section, F_i is the complex taking into account the thermophysical parameters of the working fluid. For instance, when vapor is moving in a pipe of circular section, the W-complex and the F-complex have the form:

$$W = \frac{128 \cdot L}{\pi \cdot d^4}, \tag{15}$$

$$F = \frac{\mu_v}{\rho_v \cdot k}, \tag{16}$$

where L is the pipe length, d is the pipe diameter, ρ_v is the vapor density, μ_v is the vapor viscosity. With allowance for (14), Eq. (13) for the operating temperature of an LHP functioning with a filled CC may be written as follows:

$$T_{ev} = T_{cond} + \left(\left.\frac{dT}{dP}\right|_{\overline{T}} \cdot \sum_i W_i \cdot F_i + \frac{1}{\alpha_{ev} \cdot S_q} \right) \cdot Q_{load}. \tag{17}$$

Since the derivative dT/dP taken on the saturation line is also a thermophysical characteristic of the working fluid, it may be combined with the F-complex. The product of the F_i-complex by dT/dP will be denoted as Fn_i:

$$Fn_i = F_i \cdot \left.\frac{dT}{dP}\right|_{\overline{T}}. \tag{18}$$

Expression (17) for T_{ev} includes the vapor temperature in the condenser T_{cond}, which is related to the temperature of the heat sink T_{sink} by the following relation:

$$T_{cond} = T_{sink} + \left(\frac{1}{\alpha_{cond\_ext} \cdot S_{cond\_ext}} + R_{cond\_body} + \frac{1}{\alpha_{cond\_int} \cdot S_{cond\_int}} \right) \cdot Q_{load}, \tag{19}$$

where R_{cond_body} is the thermal resistance of the condenser wall, α_{cond_int} is the heat-exchange coefficient during vapor condensation in the condenser, S_{cond_int} is the internal surface of the condenser accessible to vapor condensation. With allowance for the two latter relations (18) and (19), Eq. (17) for T_{ev} will look like

$$T_{ev} = T_{sink} + \left(\frac{1}{\alpha_{cond\_ext} \cdot S_{cond\_ext}} + R_{cond\_body} + \frac{1}{\alpha_{cond\_int} \cdot S_{cond\_int}} + \sum_i W_i \cdot Fn_i + \frac{1}{\alpha_{ev} \cdot S_q} \right) \cdot Q_{load}. \tag{20}$$

Analyzing Eq. (20) we can note the following:

1. The curve $T_{ev} = f(Q_{load})$ described by Eq. (20) is a monotonically increasing function as $T_{sink} = \text{const}$ and the multiplier Q_{load} is always positive, because it includes only positive values

$$\frac{1}{\alpha_{cond\_ext} \cdot S_{cond\_ext}} + R_{cond\_body} + \frac{1}{\alpha_{cond\_int} \cdot S_{cond\_int}} + \sum_i W_i \cdot Fn_i + \frac{1}{\alpha_{ev} \cdot S_q} > 0. \tag{21}$$

2. The dependence of the operating temperature T_{ev} on the heat load Q_{load} can be strictly linear if and only if the expression in brackets is a constant. It is possible if the heat-exchange conditions remain unchanged or change only slightly:

$$a_{cond\_ext} \approx \text{const}, \quad a_{cond\_int} \approx \text{const}, \quad a_{ev} \approx \text{const},$$

and if the Fn-complex, which is composed of thermophysical characteristics of the working fluid, remains constant under changes of Q_{load}. But the thermophysical characteristics of the working fluid included in the *Fn*-complex depend on the temperature, which in its turn changes itself with changing heat load Q_{load}. In this case, the closer the LHP operating temperature is to the critical temperature T_{crit} of the working fluid, the more considerably the Fn-complex changes with changing temperature. Away from the critical temperature T_{crit} and under a small change of the value of heat load Q_{load} we may assume that $Fn(Q_{load}) \approx \text{const}$. In this case the functional dependence of the operating temperature T_{ev} on Q_{load} may be regarded in some approximation as a straight line. With weak nonlinearity the expression in brackets in Eq. (20) may be regarded as the coefficient R_{ev_sink}, which determines the slope of the operating curve $T_{ev} = f(Q_{load})$. The value of the slope R_{ev_sink} sets the tempo of increase of the operating temperature with increasing heat load and is the thermal resistance of a particular section of a thermoregulation system, which is between the control point T_{ev} and the control point T_{sink} (see Fig. 2):

$$R_{ev\_sink} = \frac{T_{ev} - T_{sink}}{Q_{load}}. \tag{22}$$

At the same time, it should be borne in mind that under rigorous treatment the dependence $T_{ev} = f(Q_{load})$ is always quasi-linear. In this case the deviation from the rectilinear dependence is expressed in the fact that with increasing Q_{load} the temperature curve $T_{ev} = f(Q_{load})$ will turn up. The wider the operating temperature range embraced, the more distinct the effect of nonconformity to linearity, which is confirmed, in particular, by the behavior of the dependence $T_{ev} = f(Q_{load})$ in Fig. 3.

4. Formation of the LHP operating temperature with a vapor–liquid interface in the compensation chamber

A different mechanism of formation of the vapor operating temperature T_v is realized in the LHP operating mode with a partially-filled CC. The main difference is the fact that the presence of the working-fluid vapor phase in the CC has a considerable effect on the vapor operating temperature T_v. With allowance for the relation between T_v and T_{ev}, which is expressed by Eq. (12), this effect can be extended to the value of the LHP operating temperature.

The operation of a heat-transfer device in the mode under consideration is possible only when the following relation is fulfilled:

$$\left.\frac{dP}{dT}\right|_{\overline{T}} \cdot (T_v - T_{cc}) = \Delta P_{ext}, \tag{23}$$

which means that for the circulation of a working fluid it is necessary to create a certain vapor temperature and pressure drop between the wick evaporating surface and the compensation chamber. The pressure drop should be equal to the value of ΔP_{ext}, which corresponds to the sum of the pressure losses in all the sections of the transportation line for the working fluid, with the exception of the wick. The pressure losses ΔP_{ext} may be written as the sum of the hydrostatic pressure ΔP_g, the pressure losses in the transportation section along which the vapor moves ΔP_v (11) and also in the section along which the liquid moves ΔP_l:

$$\Delta P_{ext} = \Delta P_l + \Delta P_v + \Delta P_g, \tag{24}$$

where

$$\Delta P_l = \Delta P_{l\_cond} + \Delta P_{ll} \tag{25}$$

and

$$\Delta P_g = \rho_l \cdot g \cdot L_{KTT} \cdot \sin\varphi. \tag{26}$$

Rearrangement of Eq. (23) with allowance for Eq. (24) gives an expression for the vapor temperature T_v:

$$T_v = T_{cc} + (\Delta P_v + \Delta P_l + \Delta P_g) \cdot \left.\frac{dT}{dP}\right|_{\overline{T}}. \tag{27}$$

Then the equation for the temperature at the external surface of the evaporator of an LHP operating in the mode with a partially-filled CC will look like:

$$T_{ev} = T_{cc} + (\Delta P_v + \Delta P_l + \Delta P_g) \cdot \left.\frac{dT}{dP}\right|_{\overline{T}} + \frac{Q_{load}}{\alpha_{ev} \cdot S_q}. \tag{28}$$

The dependence of the operating temperature T_{ev} on the heat load Q_{load} for the mode of LHP operation with an unfilled CC is U-shaped, owing to which this mode is often called the mode of variable thermal conductance [6]. It can be observed when an LHP operates at low heat loads and the device orientation is determined by slopes $\varphi \geqslant 0°$. The specific U-shaped behavior of the temperature curve is expressed in the fact that with increasing heat load the LHP operating temperature first decreases, and then begins to increase. On the basis of experimental results it has been concluded [9,10] that the lower is the condenser cooling temperature T_{sink} as compared with the temperature of the ambient medium T_{amb}, the more pronounced becomes this sagging on the operating curve $T_{ev} = f(Q_{load})$. But if these temperatures differ from each other only slightly, the U-shaped sagging on the temperature curve becomes quite insignificant. When the cooling temperature T_{sink} considerably exceeds the ambient temperature T_{amb}, the U-shaped sagging on the experimental curve $T_{ev} = f(Q_{load})$ is absent. In this case the LHP operating temperature always only increases with increasing heat load and the temperature curve has the form of a quasi-linear dependence characteristic of the regime of constant conductance.

The experimental data presented in Fig. 4, which were obtained in testing an ammonia LHP, show a typical form of the operating curve $T_{ev} = f(Q_{load})$ for the regime of variable conductance. The device operated at a vertical orientation with a slope $\varphi = +90°$, when the evaporator was located above the condenser. The heat-transfer length was equal to 700 mm, and the evaporator diameter was 6 mm. LHP investigations were conducted in conditions with $T_{sink} < T_{abm}$. The temperature of the condenser

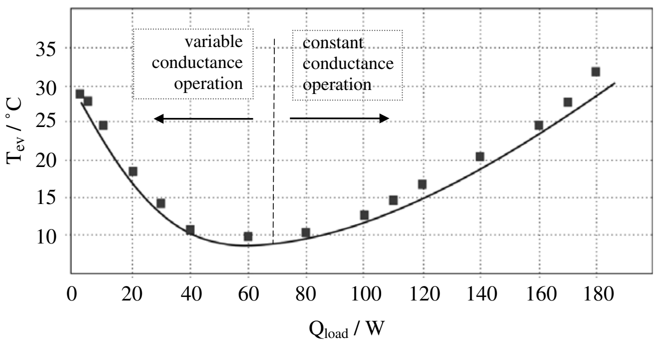

Fig. 4. Heat–load dependence of operating temperature. The working fluid is ammonia, $\varphi = +90°$, $T_{sink} = -20\,°C$. (■) Experiment; (—) calculation.

cooling T_{sink} was equal to −20 °C, and the temperature of the ambient medium T_{amb} was 19 °C. The LHP start-up was realized with a minimum heat load $Q_{min} = 3$ W . It is seen that with increasing heat load the operating temperature initially decreases. Then there appears a section of quasi-stabilization, after which the temperature begins to rise. Such a behavior of the LHP operating temperature is caused by the simultaneous action of two factors: the vacation of the condenser at the expense of the liquid displacement into the compensation chamber and the change of temperature in T_{cc} as a result of the complex heat and mass exchange processes occurred here. A cold liquid enters the CC from the condenser, and its mass flow rate changes with changing heat load Q_{load}. Simultaneously with this, part of the heat flow supplied to the evaporator penetrates into the CC and heats the working fluid there. It is the so-cooled parasitic heat flows, which in its turn, also depends on the value of the heat load Q_{load}. The total thermal action on the compensation chamber in the long run determines the thermal state of the working fluid in the CC and forms the temperature of saturated vapor over the vapor-liquid interface T_{cc}. According to Eq. (27), this temperature determines the operating temperature in the evaporation zone T_v, and according to Eq. (28), the operating temperature T_{ev}. From the graph it is seen that from a certain value of Q_{load} and on the operating temperature only increases. Such a behavior is characteristic of the LHP operating mode with a filled CC.

An analysis of the operating curve makes it possible to assume that at low heat loads the LHP functioned in the variable-conductance mode. All this time inside the LHP there proceeded a process of liquid redistribution between the CC and the condenser. It was terminated when during a regular heat-load increase a new portion of the liquid displaced from the condenser entered the CC filling the whole of it. It was the closing stage of the process of the working fluid redistribution between the condenser and the compensation chamber. This is followed by a change of operating mode, and at higher heat loads the LHP operated in the mode of constant thermal conductance.

The trustworthiness of the assumptions made on the mechanism of formation of the operating temperature depending on the degree of CC filling and the two modes of LHP operation has been confirmed by the calculations presented in Fig. 4. The data were obtained by the procedure of calculating the operating temperature of an LHP with mixed operating modes. At low heat loads the temperature T_{ev} was calculated for the LHP mode of variable conductance. In this case allowance was made for the liquid redistribution between the condenser and the CC. Simultaneously the heat load at which the liquid filled the whole compensation chamber was traced. Such a procedure made it possible to control "the point of mode changing". After its detection further calculation of the operating temperature for higher heat loads was made for the LHP mode of constant conductance. From the same graph it is seen that there is sufficiently good agreement between experimental and calculated data.

5. Comparative analysis of two operating modes of a loop heat pipe

If we compare Eqs. (13) and (28), we can note that at the same heat load Q_{load} and in similar external conditions an LHP may have different operating temperatures depending on the mode which it operates. The difference in operating temperatures ΔT_{ev} is determined with the help of the following expression:

$$\Delta T_{ev} = T_{cc} - T_c + (\Delta P_l + \Delta P_g)\cdot \left.\frac{dT}{dP}\right|_{\overline{T}}. \tag{29}$$

From analysis of Eq. (29) it follows that in the presence of the vapor phase of the working fluid in the CC a loop heat pipe has to operate at a higher temperature level. The increase of the operating temperature is caused by the contribution of two components, viz. the temperature component:

$$\Delta T_{cc\_c} = T_{cc} - T_c, \tag{30}$$

and the hydraulic component:

$$\Delta T_{hyd} = (\Delta P_l + \Delta P_g)\cdot \left.\frac{dT}{dP}\right|_{\overline{T}}. \tag{31}$$

The indicated temperature difference ΔT_{ev} was noted experimentally in testing different LHPs [11,12]. The incongruity of operating temperatures was observed during a cyclic change of the heat load. The incongruity manifested itself in the fact that in the region of low heat loads on the operating curve $T_v = f(Q_{load})$ there was a hysteresis loop. One of the examples of such a hysteresis behavior of operating temperatures may be the graph $T_v = f(Q_{load})$ presented in Fig. 5. The experimental data were obtained in testing an LHP with a heat-transfer length of 750 mm. The length of the vapor line was 500 mm, and its internal diameter was 4.5 mm. The length and the internal diameter of the liquid line equaled, respectively, 750 mm and 3 mm. The external diameter of the evaporator was 24 mm. The tests were performed at $T_{sink} = 25\,°C$. The LHP was located vertically, the evaporator was above the condenser, i.e.

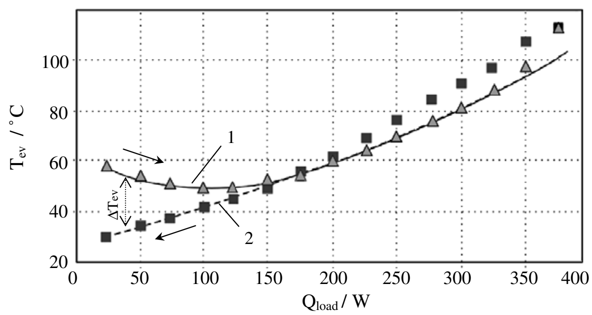

Fig. 5. Heat–load dependence of the operating temperature. The working fluid is pentane, $\varphi = +90°$, $T_{sink} = 25\,°C$. (—, - -) calculation; (■, △) experiment.

the LHP orientation was $\varphi = +90°$. Pentane was used as a working fluid. In the course of tests the operating temperature T_{ev} was registered at different values of heat load Q_{load}, which changed cyclically. A cycle included two stages of changes in Q_{load}. At the first stage the heat load increased (forward motion of changes in $Q_{load}\uparrow$), at the second stage it decreased (reverse motion of changes in $Q_{load}\downarrow$). In the graph the direction of changes in the heat load is shown as arrows. It is seen that the experimental curves behave in different ways. In the range of low heat loads the operating curve $T_{ev} = f(Q_{load}\uparrow)$ has a U-shaped sagging, and on the curve $T_{ev} = f(Q_{load}\downarrow)$ it is absent. Besides, at low heat loads the LHP operated at a higher temperature level during the forward motion of changes in $Q_{load}\uparrow$. Here the curve $T_{ev} = f(Q_{load}\uparrow)$ is situated above the curve $T_{ev} = f(Q_{load}\downarrow)$. The greatest difference in operating temperatures ΔT_{ev} is observed at minimum values of Q_{load}. Thus, for instance, at a heat load $Q_{load} = 25$ W the temperature difference ΔT_{ev} is almost 30 °C. With increasing Q_{load} the value of ΔT_{ev} gradually decreases and reaches the zero value at a heat load of 175 W. At higher values of Q_{load} both the curves behave in the same way and have a quasi-linear dependence. It should also be mentioned that there is an insignificant discrepancy between operating temperatures in the region of high heat loads. Here the experimental curve for $Q_{load}\uparrow$ is situated slightly below the curve obtained on the reverse motion of changes in $Q_{load}\downarrow$. The most probable reason for such a discrepancy of T_{ev} is capillary hysteresis. It manifests itself in the fact that on the attainment of sufficiently high heat loads, close to maximum ones, large pores in the wick surface layer are dewatered. As a result, the evaporating thermal layer shifts into the wick, which leads to an increase in its thermal resistance and a decrease in the intensity of heat-exchange processes in the evaporation zone. This tells on the LHP operating temperature, whose rate of rise increases. When the heat load changes in the opposite direction, the filling of previously-dewatered pores proceeds with a certain delay. The inertial character of restoration of the evaporating layer is the reason for elevated values of T_{ev} on the reverse motion of changes in the heat load $Q_{load}\downarrow$ in the region of its high values. Thus, the incongruity of operating temperatures at the low and high values of Q_{load} is caused by different reasons. The incongruity of T_{ev} at low heat loads is caused by different operating modes of the LHP, and at high heat loads – by capillary hysteresis.

Calculated curves 1 and 2 presented in Fig. 5 confirm the assumption that in the range of low heat loads at $Q_{load}\uparrow$ the device operates with an unfilled CC, and at $Q_{load}\downarrow$ with a filled one. Curve 1 corresponds to data calculated by the procedure described in Paragraph 4, which takes into account the redistribution of the working fluid between the CC and the condenser. In the region of heat loads from Q_{min} to Q_{ch} on Curve 1 there is a characteristic U-shaped sagging indicative of the LHP operation in the mode of variable conductance. Curve 2 corresponds to data obtained by a calculating programme for a one-phase liquid state in the CC. The dependence $T_{ev} = f(Q_{load})$ presented by Curve 2 simulates the LHP operation on the reverse motion of heat load $Q_{load}\downarrow$. Thus, the calculated dependence $T_{ev} = f(Q_{load})$ for a complete LHP operating cycle may be obtained by combining Curves 1 and 2. Sufficiently good agreement between calculated and experimental data in the range of heat loads from Q_{min} to Q_{ch} points to the correctness of model presentations of LHP functioning.

As for the results at heat loads from Q_{ch} and higher, in this interval experimental and calculated data for the forward motion of changes in Q_{load} also coincide, and on the reverse motion one can observe an insignificant discrepancy. Experiments have yielded higher values of the temperature T_{ev} than those predicted by calculation. The most probable reason for such a discrepancy is capillary hysteresis, which took place in the experiment but was not taken into account in the calculation. When the evaporating menisci are shifted deep into the capillary structure, an additional excess pressure ΔP^{v}_{wick} is required for vapor removal from the wick. And this should be taken into account in calculating the operating temperature T_{ev} on the reverse motion of changes in $Q_{load}\downarrow$. In calculating the vapor pressure losses instead of Eq. (11) one should use its modernized analog:

$$\Delta P_v = \Delta P_{vc} + \Delta P_v + \Delta P_{v\_cond} + \Delta P^{v}_{wick}. \tag{32}$$

At the same time the presence of capillary hysteresis complicates the problem considerably for, as is well known, hysteresis effects in capillary structures with all typical peculiarities are difficult for theoretical description [13].

Another demonstration of the effect of changing LHP operating modes on the operating temperatures T_{ev} may be the experimental data presented in Fig. 6, which have been obtained in testing the same LHP filled with acetone. The tests were conducted in similar conditions, including the cyclic change of the heat load. Worthy of attention here is the fact that on the reverse motion of changes in the heat load Q_{load} there occurs boiling-up of the working fluid in the CC with a subsequent change of operating modes. The transfer process noted in the experiment is designated in the graph as the zone of mode changing – “zone C”.

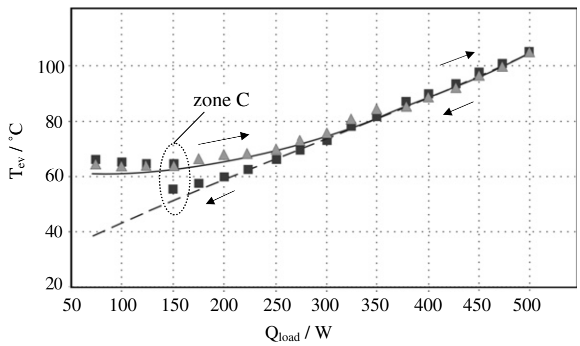

Fig. 6. Heat–load dependence of the operating temperature. The working fluid is acetone, $\varphi = +90°$, $T_{sink} = 25$ °C. (—, – –) Calculation; (■, △) experiment.

It is well known that a liquid in the CC may be in the superheated state. According to the theory of nucleation of the vapor phase in a superheated liquid, the boiling-up has a random nature [14]. This is how matters stand in the experiment with the LHP too. The process developed as follows. At a regular heat load decrease from 175 W to 150 W temperature-sensitive elements recorded stabilization of the operating temperature. It meant that the transient processes in the LHP had been successfully completed, and the heat-transfer device had passed into a new operating level. A temperature jump ΔT_{ev} was recorded after a stable LHP operation for about 15 min. The device began to operate at a higher temperature level. Such changes might be caused by a random formation of the vapor phase in the CC followed by a change of operating modes. Notable here is the fact that as a result of these internal changes the operating temperature T_{ev} increased to the value that the LHP had on the forward motion of changes in $Q_{load}\uparrow$ at the same heat load. From the graph in Fig. 6 it is seen that with a further decrease in the heat load to Q_{min} the mode of variable conductance was retained. The operating temperatures T_{ev} are situated on the high-temperature branch of the cyclogram, i.e. correspond to the temperatures characteristic of the operating mode with an unfilled compensation chamber.

The above example shows that the change of the LHP operating temperature under changes of operating modes may be realized in different ways. It proceeds either smoothly, as, for instance was the case on the direct motion of $Q_{load}\uparrow$, or stepwise, as was observed on the reverse motion of $Q_{load}\downarrow$. It is significant that the LHP, as a self-regulating heat-transfer system, independently realizes a transfer from one operating mode to the other. From the viewpoint of thermodynamics of heat and mass exchange processes inside the LHP, the transfer is realized from a less stable into a more stable state. The internal changes influence the thermal state of the heat-transfer device in the form of an ambiguous behavior of operating temperatures which take place at the same external conditions and the same heat load. Here it is important to understand that such a hysteresis behavior is not a manifestation of any troubles or malfunctions in the LHP operation. But at the same time, it is necessary to be aware of such peculiarities and take them into account in designing LHPs developed for thermal regulation of objects with variable heat generation.

A few words should be said about the behavior of the experimental curve $T_{ev} = f(Q_{load})$ at high heat loads (see Fig. 6).The values of temperatures T_{ev} on the forward motion of $Q_{load}\uparrow$ and the reverse motion of $Q_{load}\downarrow$ coincide and correspond to calculated values of operating temperatures T_{ev}. The absence of a hysteresis loop on the experimental curve means that limiting heat loads, i.e. loads close to maximum one, were not achieved in LHP tests. Thus, by setting the upper limit of the operating range of heat loads somewhat lower than their utmost values one can avoid the ambiguous behavior of operating temperatures caused by the effect of capillary hysteresis, which is what was experimentally demonstrated with the last example (see Fig. 6).

In comparing the two modes of LHP operation mention should be made of the advantages and the drawbacks of each of them. Thus, when operating with a filled compensation chamber an LHP has lower operating temperatures T_{ev} and a lower thermal resistance R_{LHP}. In the operating mode with a partially-filled CC parasitic heat flows have an appreciable effect on the LHP operation, increasing the operating temperature of the device T_{ev}. In the operating mode with a filled CC such an effect is absent, and therefore the temperature T_{ev} proves to be easier to predict. A considerable advantage of the operating mode with a partially-filled CC is the possibility of active regulation of the LHP operating temperature by means of an additional thermal action on the compensation chamber [6–8,15,16]. By regulating the temperature T_{cc}, one can realize temperature control over the object to be cooled. This procedure makes it possible to realize thermostabilization of a cooled object with a variable heat generation at a certain predetermined temperature level.

6. Conclusion

1. As a result of experimental and analytical investigations two main operating modes of LHP have been revealed:
   - An LHP operates in the presence of the working-fluid vapor phase in the compensation chamber.
   - An LHP operates in the absence of the working-fluid vapor phase in the compensation chamber.
2. It has been found that the LHP operating temperature depends on the operating mode of the heat-transfer device.
3. It is shown that at the same heat load and identical external conditions an LHP has a higher operating temperature when it functions with a partially-filled compensation chamber.
4. The reason of temperature hysteresis in the region of low heat loads is different LHP operating modes, and in the region of high heat loads – capillary hysteresis.

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