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INFLUENCE OF ADVERSE ACCELERATIONS ON THE OPERATION OF AN "ANTIGRAVITY" HEAT PIPE

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UDC 536.248.2

The authors present results of an experimental investigation of the influence of accelerations directed along the heat-transfer vector on the operation of heat pipes with separate channels for vapor and liquid.

There is a considerable gulf between the capability of elements of electronic equipment (EE) and heat pipes (HP) to function under the action of vibrations, accelerations, and other factors unfavorable for heat pipes. The literature has practically no information on the operation of heat pipes under the action of dynamic, arbitrarily directed accelerations of more than 1g.

The case of heat transfer in the direction of the vector of the accelerations acting on the heat pipe is more complex for the operation of a heat pipe, since it requires added expenditure of energy to move the liquid heat transfer agent into the heat supply zone against the action of mass forces.

The hydrostatic pressure can be compensated for by increasing the capillary potential due to a reduction of the size of pores of the heat pipe wick, which in turn leads to a

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TABLE 1. Some Parameters of AGHP Heat-Transfer Agents for $a = 10g$ and $\bar{d}_p = 2 \cdot 10^{-6}$ m

Heat-transfer agent	$\sigma \cdot 10^2$, N/m for $T = 50$ °C	ρ, kg/m³ at $T = 25$ °C	$\frac{dP}{dT}$, kPa/K, for $T=50$ °C	$\Delta P_{\sigma \max}$, kPa, for $T=50$ °C	ΔT_a, K	Bo
n-Pentane	1,3	621	3,8	26	4,80	0,72
Acetone	2,0	785	2,8	40	8,40	0,58
Freon-11	1,5	1475	6,6	30	—	1,42
Ammonia	1,6	562	72,0	32	0,23	0,44

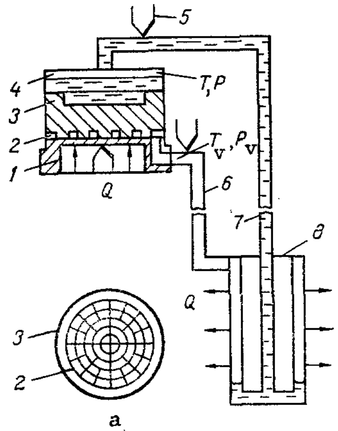

Fig. 1. Schematic diagram of a heat pipe: 1) evaporation chamber; 2) vapor discharge channels; 3) wick; 4) compensation cavity; 5) thermocouple; 6) vapor channel; 7) condensation channel; 8) condenser; a) vapor discharge channels in the heat supply zone.

sharp increase of the hydraulic resistance of the capillary pump and a considerable decrease of the heat flux transferred. Therefore, the use of heat pipes with a capillary-porous structure located along the entire length of the heat transfer is of low efficiency in a variable field of mass forces.

In heat pipes with separate channels for vapor and liquid, which have been called antigravity heat pipes (AGHP), the hydraulic resistance of the wick is substantially reduced by locating it only in the heat supply zone, so that the heat and mass fluxes in the wick of an AGHP are directed opposite to one another [1, 2]. Here the path length of the liquid heat-transfer agent along the capillary channels of the wick is considerably less than the heat-transfer length in a heat pipe, and is several millimeters. This circumstance leads to the fact that the necessary capillary potential to operate AGHP's is used not so much to drive the liquid heat-transfer agent along the fine pores of the wick as to overcome the hydrostatic pressure of the column of liquid in the heat pipe. Thus, the first condition for operability of an AGHP in the field of mass forces can be written in the form

$$\Delta P + \rho a L \sin\varphi \leqslant \frac{4\sigma \cos\Theta}{\bar{d}_p}. \tag{1}$$

The pressure balance condition in an AGHP is necessary but not sufficient to ensure its operability.

To move the heat-transfer agent from the heat supply zone (Fig. 1) to the compensation cavity requires a specific pressure difference. This required pressure difference is achieved by capillary forces, but is formed by the difference of the temperatures of saturated vapor in the heat supply zone and the compensation cavity:

$$P_v - P = (\Delta P - \Delta P_{1w}) + \rho a L \sin\varphi \simeq \left.\frac{dP}{dT}\right|_{\frac{T_v+T}{2}} (T_v - T). \tag{2}$$

The driving temperature head $\Delta T_\sigma = T_v - T$ forms when a heat flux is supplied and is determined by the conditions of heat and mass transfer through the wall of the wick of an AGHP, seen in the role of a hydraulic and thermal gap between the evaporation zone and the compensation cavity. The driving temperature head ΔT_σ is a necessary contribution to the thermal resistance of an AGHP.

When one analyzes the operation of a heat pipe in a mass force field with an adverse orientation ($\sin\varphi > 0$), the degree to which the capillary forces exceed the mass forces can be described by the Bond number (Bo):

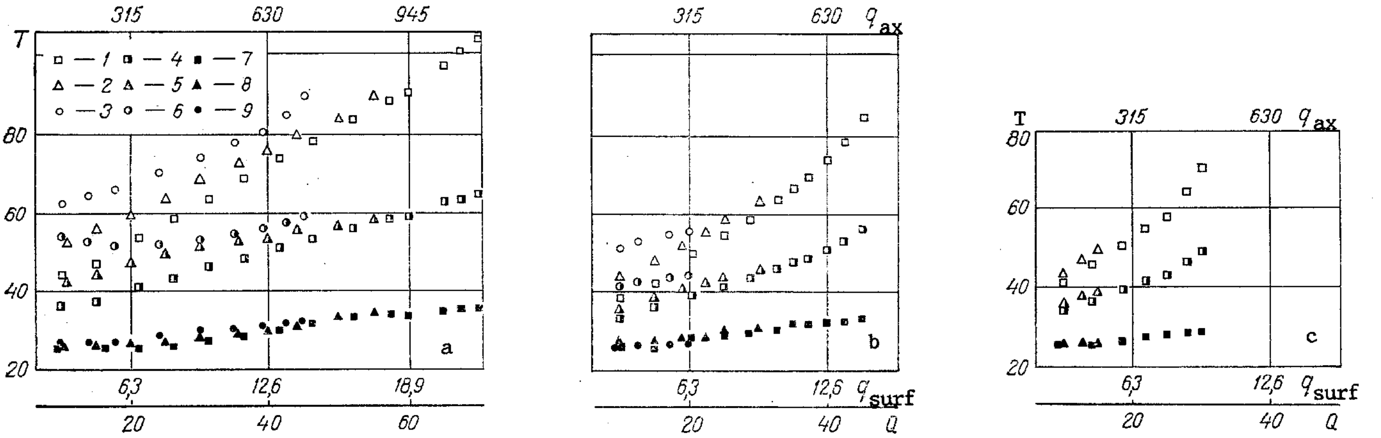

Fig. 2. Representative temperatures T, °C, of acetone (a), pentane (b), and Freon (c) in AGHP as a function of the heat flux Q, W, the surface q_{surf}, 10^4 W/m², and the axial q_{ax}, 10^4 W/m², heat flux density for an orientation of $\phi = +90°$, temperature and flow rate of the cooling water of 25°C and 0.01 kg/sec. 1, 2, and 3 indicate T_h at 1, 5, and 10 g; 4, 5, and 6 are T_v at 1, 5, and 10 g; 7, 8, and 9 are T_ℓ at 1, 5, and 10 g.

$$\mathrm{Bo} = \frac{\rho a L \sin\varphi d_p}{4\sigma\cos\Theta}. \tag{3}$$

To allow for the possible use of AGHP's in facilities of arbitrarily varying orientation relative to the acceleration vector, Bo is determined for an orientation for which the heat transfer occurs along the acceleration vector ($\sin\varphi = 1$) and at the maximum possible acceleration $\underline{a}$ for operation. Clearly the condition Bo < 1 must hold for operation of the AGHP. Below we shall consider only the most adverse orientation, for which $\sin\phi = 1$.

The driving temperature head ΔT_σ in the mass force field can conveniently be represented as $\Delta T_\sigma = \Delta T + \Delta T_a$. Then condition (2) can be written in the form

$$(\Delta P - \Delta P_{1w}) + \rho a L \simeq \left.\frac{dP}{dT}\right|_{\frac{T_v + T}{2}} (\Delta T + \Delta T_a), \tag{4}$$

where ΔT is the temperature head required to overcome the hydraulic resistance $(\Delta P - \Delta P_{\ell w})$, and ΔT_a is for hydrostatic resistance ρaL.

For limiting small power Q we can neglect the hydraulic resistance $(\Delta P - \Delta P_{\ell w})$ in comparison with the hydrostatic value ρaL and evalute ΔT_a:

$$\Delta T_a \simeq \rho a L \Big/ \left.\frac{dP}{dT}\right|_{\frac{T_v + T}{2}}. \tag{5}$$

Thus, an increase of acceleration $\underline{a}$ acting on an AGHP, other conditions being equal, leads to an increase of the hydrostatic pressure ρaL, and consequently to a decrease of the maximum heat flux Q in accordance with condition (1), and to an increase of the thermal resistance of the AGHP due to increase of the temperature head ΔT_a.

Some of the data used to evaluate AGHP heat-transfer agents are shown in Table 1.

We experimentally investigated the influence of the most adverse accelerations of magnitude 1, 5, and 10 g, directed along the heat-transfer vector ($\sin\varphi = 1$), on the operation of a heat pipe with separate vapor and liquid channels.

The AGHP shown in Fig. 1 was made of type 12Kh18N10T steel and had the following dimensions: heat pipe length L = 0.3 m; diameter of the planar circle in the heat supply zone $20 \cdot 10^{-3}$ m; condenser length $85 \cdot 10^{-3}$ m; diameter $9 \cdot 10^{-3}$ m; internal diameter of the vapor channel and condenser channel $2 \cdot 10^{-3}$ m. The wick, of thickness $8 \cdot 10^{-3}$ m and diameter $26 \cdot 10^{-3}$ m, was made of nickel metal-ceramic of 64% porosity with an average pore diameter of $\bar{d}_p = 2 \cdot 10^{-6}$ m. In the heat-supply zone the AGHP wick had a system of mutually intersecting vapor-removal channels communicating with the vapor channel.

The AGHP examined was designed for cooling semiconductor elements. During the experimental investigations we used an ohmic heater with a copper core as the heat supply element, to provide a uniform heat flux in the AGHP heat-supply zone. The heat flux transferred was controlled by the electrical power of the ohmic heater and was measured calorimetrically from the flow rate and temperature difference of cooling water in the cooling sleeve of the AGHP condenser. The difference between the heater electrical power and the calorimeter (transferred) power did not exceed 10% in all the tests. As representative temperatures, we measured the heater temperature at the center of the zone of contact with the AGHP (T_h) and the temperatures of the vapor channel (T_v) and condenser channel (T_ℓ). The AGHP was charged in a special rig, after pumping down to a pressure of about 5 N/m^2 and simultaneously heating the AGHP to 130°C, in sequence with identical volume amounts of n-pentane, acetone and Freon 11. The choice of heat-transfer agent was decided by the temperature range of AGHP operation (from −70 to +130°C) [3].

The operating characteristics of the AGHP were measured at an acceleration of 1 g in the field of gravity (evaporator upwards), and at accelerations of 5 and 10 g on a specially constructed centrifuge such that the evaporator was oriented toward the centrifuge axis, and the condenser faced the periphery. The radius of rotation of the centrifuge was 2.1 m, which achieved uniformity of acceleration along the AGHP to within ±10%.

The operating characteristics of the AGHP are presented in Fig. 2 for various heat-transfer agents in the form of dependence of the characteristic AGHP temperatures on the heat flux Q, on the heat-flux density in the heat supply zone q_{surf}, and on the axial heat-flux density in the transport zone q_{ax}.

It was impossible to operate the Freon AGHP under the action of adverse accelerations of 10 g. This accords with the fact that the number Bo > 1 for Freon 11 at a = 10 g.

During operation of the AGHP under conditions of an acceleration varying in magnitude and direction, we are interested, in addition to the steady operating characteristics, in the dynamics of change of the AGHP thermal parameters and also the behavior of the AGHP at limiting accelerations (Bo = 1) and at the maximum heat flux.

For a rapid change of acceleration or power or of acceleration and power simultaneously the heater temperature T_h is changed between the corresponding stationary characteristics in a law close to exponential whose time constant is mainly determined by the heat capacity of the object being cooled.

For accelerations which correspond to Bo $\geq$ 1 the AGHP heat transfer ceases, the heater temperature T_h increases, and if the acceleration is subsequently decreased (Bo < 1) the heat transfer resumes.

If the heat input of the heater exceeds the maximum heat flux transferred by the AGHP, it remains at the maximum level, and the excess energy causes a rapid increase of heater temperature T_h. When the heater power is reduced the AGHP thermal parameters return to the appropriate stationary operating characteristics. The heater temperature T_h in all the experiments did not exceed 140°C.

The difference between the temperatures of the heater and the vapor changes insignificantly with an increase of adverse acceleration, indicating a weak dependence of the vapor formation processes in fine-porous wicks on acceleration.

Thus, as a result of the above experimental data, we have shown that the AGHP can be used in semiconductor cooling systems operating under conditions of several g.

It should be noted that there are some special features of the operation of AGHP's under the action of adverse accelerations:

1) when operating with adverse accelerations, the AGHP has a large thermal resistance;

2) the maximum heat flux is reduced;

3) the temperature difference between the heater and the vapor depends little on the acceleration;

4) the increase of the AGHP thermal resistance under the action of accelerations is determined mainly by the increase of temperature drop between the vapor in the vapor channel and the liquid in the liquid channel.

NOTATION

g, free-fall acceleration; ΔP, hydraulic resistance of the circulating heat-transfer agent in the AGHP along a closed circuit; ρ, density of the liquid heat-transfer agent; $\underline{a}$, acceleration; L, length of heat-transfer section of the AGHP; ϕ, slope angle of the AGHP to the horizontal; σ, surface tension; Θ, wetting angle; $\bar{d}$, mean diameter of the AGHP wick pores; T_h, T_ℓ, temperature of the heater and condenser, respectively; T_v, P_v, temperature and pressure of the vapor in the heat supply zone; P, T, pressure and temperature of the saturated vapor of the heat-transfer agent at the phase boundary in the AGHP compensation cavity; $\Delta P_{\ell w}$, hydraulic resistance to the flow of liquid heat-transfer agent along the wick; ΔT_σ, driving temperature difference; Bo, Bond number; ΔT, ΔT_a, temperature heads required to overcome the hydraulic and hydrostatic resistances, respectively; Q, heat flux transferred by the AGHP; q_{surf}, q_{ax}, heat flux densities in the heat supply zone and the transport zone (ratio of the heat flux transferred Q to the area of the heat supply zone and to the total area of the internal sections of the vapor channel and condenser channel, respectively); $\Delta P_{\sigma max}$, maximum capillary head.

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VARIATION OF THE CONTACT ANGLE IN THE QUASISTATIC GROWTH OF A VAPOR BUBBLE ON A HORIZONTAL SURFACE IN A BOILING LIQUID

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The variation of the contact angle at the solid–liquid–vapor phase interface during the quasistatic growth of a bubble in the boiling of water is studied experimentally.

Scientists have been turning their attention in recent years to the variation of the contact angle at the phase interface in the process of growth of a vapor bubble. This interest is stimulated by the fact that the contact-angle dynamics determines to a large extent the stability of the vaporization centers as well as the bubble size and breakoff frequency [1-3]. In addition, the introduction of a bubble model with a "microlayer," i.e., a thin layer of liquid at its base, affords the most natural description of the dynamic angles during the motion of the bubble wall along the interface [3, 4]. The deviations of the contact angle from the equilibrium value are usually attributed to dynamic effects. The theory developed in [3] shows that the deviation of the contact angle is more pronounced, the higher the velocity of the bubble base. A comparison of the theoretical calculations with the experimental data [1, 2] shows that if only dynamic effects are considered, it is impossible to describe completely the behavior of the contact angle as the bubble grows. It is important in this connection to investigate experimentally the variation of the contact angle during the slow growth of a vapor bubble, when dynamic effects are negligible.

We have carried out a series of experiments, during which we determined the fundamental laws governing the variation of the base of a vapor bubble and the corresponding variations of the contact angle under various conditions of wetting of the heat-emitting surface by the boiling liquid. We investigated the boiling of distilled water on a horizontal surface. The heating surface was a nickel plate of length $5 \cdot 10^{-2}$ m, width $5 \cdot 10^{-3}$ m, and thickness

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