SOME RESULTS OF INVESTIGATIONS OF LOW-TEMPERATURE HEAT PIPES, WORKING AGAINST GRAVITY

Yu. F. Gerasimov, Yu. F. Maidanik, Yu. E. Dolgirev, V. M. Kiseev, G. A. Filippov, and L. G. Starikov

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A number of constructional details and results are given for investigations of water and acetone heat pipes.

The use of heat pipes to cool objects which are accelerating or changing their attitude in a gravitational field is limited primarily by the fact that in a specific case one of the parameters such as the length or the transmitted power sharply opposes the action. This occurs because of the action of gravity on the return flow of heat-transfer agent and the need to use a wick with capillary channels of small diameter. Similar conditions arise also in transferring heat to a distance on the order of several meters by a heat pipe located horizontally or with an arbitrary attitude under weightless conditions. This applies primarily to low-temperature and cryogenic heat pipes for which the heat-transfer agents have low surface-tension coefficients.

Information relating to heat pipes capable of operating against gravity and transmitting a heat flux of several hundred watts in the temperature range 50-80°C is extremely limited [1-5]. The available information relates mostly to multisectional heat pipes [6-9], in which the height of the liquid capillary column within each section is part of the total length of the tube, or to pipes operating horizontally with the evaporator elevated slightly above the condenser [10-14]. When we consider that this temperature range includes the thermal conditions for most elements of radio and electronic equipment, it becomes clear that we must develop and investigate heat pipes of this type.

Works [1, 2] describe the operating principles and discuss certain features of a heat pipe capable of fulfilling the above conditions to a considerable extent.

In this paper we describe a somewhat different form of heat pipe (Fig. 1). The main difference is that the wick is located only in the evaporator chamber, while the compensation cavity, which is filled with liquid in nonconnected capillaries during the whole operation of the pipe, is buried to practically its whole extent in the evaporator zone. This construction ensures a minimum length for the liquid path in the porous medium, i.e., the least hydraulic loss in the liquid phase of the heat-transfer agent. The condenser chamber has the form of 16 parallel tubes of internal diameter 2 mm, joined to the vapor and liquid collectors. With a well-developed condensation surface such a chamber has a small volume capable of stable condensation of heat-transfer agent and able to return it to the evaporator zone, for any attitude of the tube in a gravity field.

The main element which predominantly determines the operation of the heat pipe is the evaporator chamber. Figure 2 shows the main types. The motion of the liquid in the wick of the evaporator chamber is preferentially along the heat-receptive wall (Fig. 2a,b) or normal to it (Fig. 2c,d). Chambers a, b, and d are of a capillary-porous construction, consisting of a finely porous layer close to the heat section of the chamber wall and a coarsely porous layer away from the wall and forming a compensation cavity. In the evaporator chamber c the compensation cavity is embedded in a finely porous structure for almost the whole section of the evaporator zone. The more homogeneous the finely porous structure, the smaller

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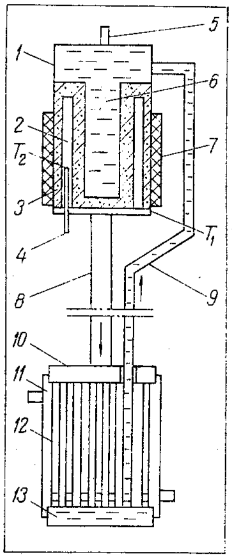

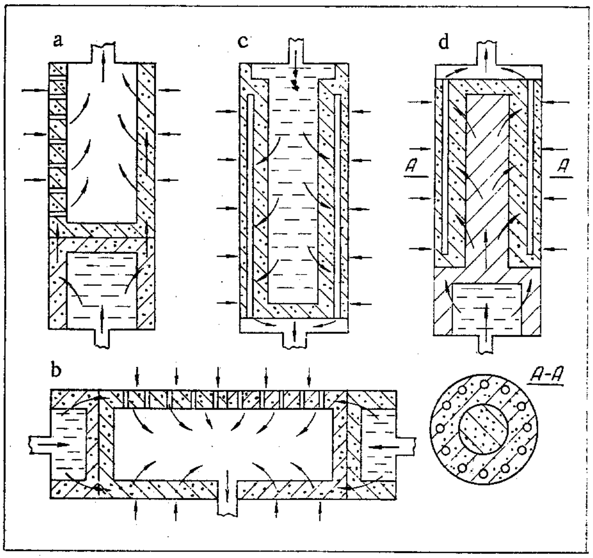

Fig. 1

Fig. 2

Fig. 1. Heat pipe: 1) evaporation chamber; 2) vapor-removal channel; 3) wick of evaporator chamber; 4) thermocouple sheath; 5) filling tube; 6) compensation chamber; 7) heater; 8) vapor channel; 9) liquid channel; 10) vapor collector; 11) cooling sleeve; 12) condenser tube; 13) liquid collector.

Fig. 2. Main types of evaporator chambers.

the probability of a dry open pore being formed and the smaller is the thickness of the wick. Chamber a can be used most efficiently when the heat supply section is short, while variants b, c, and d are more suitable when this is not so. The pressure loss in the vapor phase can be reduced by using a large number of radial vapor-removal channels (Fig. 2a,b). The diameter of these channels is 0.5-0.6 mm, and their number (15-20 per cm^2 of the outer surface of the wick) is determined by purely technical factors. These values can be obtained more accurately from the condition that the total pressure

$$\Sigma\Delta P = \Delta P_{lw} + \Delta P_{vw},$$

should be a minimum, where ΔP_{lw} and ΔP_{vw} are the hydraulic resistances of the wick with respect to liquid and vapor. The presence of vapor-removal channels also allows the process of formation of a continuous vapor layer at the chamber wall to be delayed and increases the radial heat flux density, e.g., by 20-30% for water [15]. Since the construction of a large number of microchannels is a very laborious process, it becomes very promising to use the so-called biporous capillary structure, which has balanced vapor and liquid hydraulic resistance. It is most efficient to use this kind of structure for the evaporator chambers of types b and c in the evaporator zone.

To ensure reliable start-up of a heat pipe of this type there must be constant contact of the working liquid with the evaporator chamber wick. This requirement can be met, if the total volume of the condenser and the vapor channel is approximately equal to, or somewhat less than, the volume of the compensation cavity. The amount of liquid in the heat pipe must ensure impregnation of the wick and must fill the compensation cavity and the liquid channel. When it absorbs heat the liquid flows from the compensation cavity towards the condenser, if it lies below the evaporator, and rises to some level in the vapor and liquid channels and in the interconnecting vessels, thus coming in contact with the evaporator chamber wick. To start this type of heat pipe with the evaporator upward it is necessary that the wick be wetted and the heat input be above a certain value. Otherwise, the liquid does not flow into the compensation cavity, i.e., there is no "Clapeyron" filling [16]. The minimum heat flux to ensure a stable start-up of the heat pipe depends on its length (or height), the dP/dT of the working liquid, the effective heat conduction and moisture content of the wick, and a number of other factors. For example, for an acetone pipe of height 900 mm the minimum ra-

TABLE 1. Constructional Features of the Heat Pipes

Heat-transfer agent	H_2O	Acetone
Wick material	Baked carbonyl nickel powder, $d_{av}=1\cdot10^{-2}$ mm P = 68%	Baked electrolytic nickel powder, $d_{av}=9\cdot10^{-4}$ mm P = 72%
Material of pipe body, mm	Stainless steel	Stainless steel
l_1	900	900
l_2	100	100
l_3	105	105
l_4	75	75
l_5	68	68
D_1	29,5	29,5
D_2	30	30
D_3	5	5
D_4	3	3
D_5	14,5	13,7
D_6	2	2
δ_1	1,5	1,5
δ_2	1,5	1,5

dial heat flux density was 0.3-0.5 W/cm², and for a water pipe of the same length it was 0.7-0.9 W/cm².

The present authors have investigated two heat pipes whose construction is illustrated in Fig. 1. The structural and technical characteristics of the pipes are given in the table. Measurements were made of the temperature T_1 of the evaporator chamber wall at a distance 3 mm above the heater, the temperature T_2 of the vapor in the vapor-removal channel (a thermocouple was mounted in a metal sheath and introduced into the channel to a depth of 15 mm), and also the temperature difference between the inlet and exit of the condenser cooling sleeve. Heating was supplied by an electrical heater of length 50 mm in the evaporator chamber section, and cooling was supplied by water from a type TS-24 thermostat. The mass flow of cooling water was maintained at 63 ± 0.5 liters/h and controlled by a type RE-02 rotameter. The cooling water temperature was maintained at 25 ± 0.1°C for the water heat pipe and at 30 ± 0.1°C for the acetone pipe. Copper—Constantan thermocouples and type R-304 and R-37/1 potentiometers were used to measure the temperatures. The transmitted heat flux power was calculated from the relation

$$Q = c\Delta tG.$$

The attitude of the pipes was varied from $\theta = -90°$ to $+90°$ in steps of 45°. The results are shown in Fig. 3a and b. The vapor temperature T_2 was qualitatively similar to the curves $T_1 = f(Q)$, but its value was about 1-2°C lower for acetone and 2-3°C lower for water.

Typical features of heat pipes of this construction at low heat load, ∿1-2 W/cm², are the presence of a section with an inverse dependence $T_1 = f(Q)$ for $\theta < 0°$ and also oscillations in temperature and in transmitted heat power which are synchronous in time but opposite in amplitude. Figure 3 shows the maximum heat flux and minimum temperature corresponding to the oscillations. These oscillations occur even in normal conditions and are associated with oscillations of the liquid level in the heat pipe. A similar picture is observed when there is insufficient heat-transfer agent, but in that case the oscillations are not cut short over the whole range of heat input.

Another feature is the presence of a hollow section for $\theta \leqslant 0°$, particularly for the water heat pipe. One can speculate that this feature may be due to at least two interrelated processes, occurring simultaneously and causing a weak dependence of temperature of heat flux. First, there is an overdrying of the largest pores in the wall of the wick, increasing progressively with increase of heat flux and beginning to play the role of vapor microchannels; secondly, there is an increase in the speed of circulation of the heat-transfer agent.

For heat inputs exceeding ∿2 W/cm² for acetone and ∿10 W/cm² for water the curves for ($\theta > 0°$) lie above the curves for ($\theta \leqslant 0°$), although the conditions for wetting the wick become more favorable. In this case there is partial flooding of the wick by the heat-transfer agent and also some reduction in the transmission cross section of the vapor channel due to the flowing film of condensate, which leads to a sharp increase in the hydraulic resistance of the vapor and, consequently, to a rapid temperature rise.

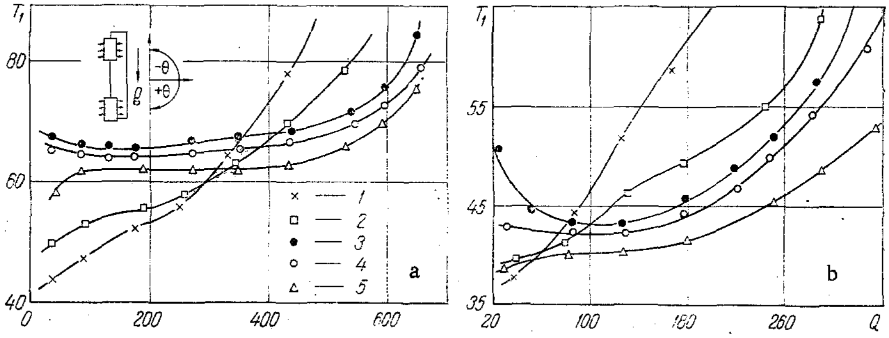

Fig. 3. The evaporator chamber wall temperature T_1, °C as a function of the transmitted heat flux power Q, W, for water (a) and acetone (b): 1) $\theta = +90°$; 2) $+45°$; 3) $-90°$; 4) $-45°$; 5) $0°$.

The following conclusions can be drawn.

1. Heat pipes of the construction considered, in addition to an upper limit of heat input associated with vaporization in the capillary structure, have a lower limit for $\theta < 0°$, due to the presence of oscillatory processes and the associated unstable operation of the heat pipes.

2. For $\theta > 0°$ the operating region shifts in the direction of lower heat inputs and lower temperatures.

3. The largest width of the operating region is achieved for attitudes close to $\theta = 0°$.

4. Heat pipes of this construction are designed primarily for the attitude $\theta = -90°$. For other attitudes the operating conditions depart from optimum.

NOTATION

Q, heat flux power in the pipe; T, temperature; ΔP_{lw}, hydraulic resistance of the liquid in the wick; ΔP_{vw}, hydraulic resistance of the vapor in the wick; g, acceleration of free fall; θ, angle of inclination of the pipe; c, specific heat; Δt, temperature difference; G, mass flow rate of liquid; d_{av}, average diameter of wick particles; P, porosity; l_1, l_2, l_3, l_4, l_5, lengths of the heat pipe, evaporator chamber, wick, and vapor-removal channels, respectively; D_1, D_2, D_3, D_4, internal diameter of the evaporator chamber, condenser collectors, and vapor and liquid channels, respectively; D_5, diameter of the compensation cavity; D_6, diameter of the vapor-removal channels; δ_1, distance of the vapor-removal channel from the evaporator chamber wall; δ_2, distance between the vapor-removal channels.

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A CRYOGENIC HEAT PIPE WITH A RESERVOIR IN THE TRANSPORT ZONE

V. V. Gil' and A. D. Shnyrev

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The article discusses an explosion-proof design for a cryogenic heat pipe.

Cryogenic heat pipes have come into increasingly wide use in recent years; they are particularly promising for thermally stressed devices of modern design whose optimum operating conditions require intensive cooling. The basic requirements for the design of such devices include a combination of compactness, independence, and high reliability with minimum weight.

However, cryogenic heat pipes of traditional construction have a number of specific characteristics that substantially reduce the reliability of the design; the most important such characteristic is the sharp increase of pressure in the heat-pipe cavity when the temperature of the surrounding medium is increased. An absolute majority of cryogenic devices are in an inoperable condition an average of 50% of their total service time, i.e., at an ambient temperature of $\sim 20°C$; in addition, the possibility of breakdown situations that will bring increased temperatures is not excluded.

If in such a device cryogenic heat pipes are used for cooling some elements, the operating liquid may be in the supercritical state. Let us consider as an example a cryogenic heat pipe operating with liquid nitrogen and having the following geometric dimensions: length $l = 0.5$ m, inner diameter of jacket $d_{in} = 0.01$ m, thickness of cylindrical porous bronze wick baked onto the inner wall of the jacket $\delta = 0.002$ m, and porosity $\sim 40\%$.

The maximum heat flux transmitted by such a heat pipe at a working temperature of 70-80°K can be calculated by the formula [1]

$$Q = \frac{i\rho kA}{\eta l}\left(\frac{2\sigma\cos\theta}{r_e} + \rho g l \cos\varphi\right) \quad (1)$$

and amounts to approximately 4 W if the pipe is charged with liquid nitrogen in an amount of

$$m = \rho_l Alk + \rho_v \pi\left(\frac{d_{in} - 2\delta}{2}\right)^2 l, \quad (2)$$

which, with the heat-pipe parameters assumed above, amounts to $5.81 \cdot 10^{-3}$ kg, corresponding to a pressure of $p > 1{,}000$ abs. atm on the base at 300°K [2]. It is clear that the reliability of a system having such a stressed construction is sharply reduced, not to mention the fact that the increase in the thickness of the jacket walls leads to an increase in the temperature drop in the evaporation and condensation zones, which makes the main contribution to the total temperature drop along the pipe [3], and this must be kept down to a minimum.

In order to reduce the pressure in the operating cavity of a cryogenic heat pipe, we can make use of additional gas reservoirs connected to the jacket of the tube by a capillary or placed on the jacket of the heat pipe beyond the condensation zone. In some cases such con-

A. V. Lykov Institute of Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR, Minsk. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 30, No. 4, pp. 587-589, April, 1976. Original article submitted June 19, 1975.