LOW-TEMPERATURE HEAT PIPES WITH VAPOR INJECTION

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Two types of heat pipe with low-pressure injectors are considered. The pipes are intended for operation on any orientation in the gravitational field.

Heat transfer in the direction of the gravitational field or in the presence of g loads constitutes a special case in which the usual flow friction is supplemented by the losses involved in overcoming the hydrostatic pressure of the fluid column.

Existing heat-pipe designs [1-4] permit heat transfer at low temperatures and any orientation in the gravitational field over distances of 1 m or more. However, further improvements in the heat transfer characteristics and length of heat pipes for operation at any orientation in the gravitational field and in the presence of high loads depend upon new developments in design.

The heat pipes described below make possible a substantial reduction in the hydrostatic pressure of the fluid column without a corresponding decrease in heat transfer distance (Fig. 1a, b).

The first type of heat pipe (Fig. 1a) has a hermetically sealed body and consists of an evaporator 1 with a capillary-porous lining 2, and injector—condenser with total vapor condensation 3, and a heat exchanger 4, connected by condensate tubes 5, 6, 7.

The lining of the evaporator encloses a reservoir 8 communicating with an expansion chamber 9, together with vapor channels 10. Connection 11 serves for evacuating and filling the pipe.

The evaporator lining consists of finely porous nickel and fits tightly into the evaporator body.

TABLE 1. Design Characteristics of Heat Pipes (Body Material—Stainless Steel)

	Design (Fig. 1a)		Design (Fig. 1b)
Heat transfer	Water	Acetone	Acetone
Wick material	Nickel. Most pores on interval d_p = 6-10 μ	Nickel. Most pores on interval d_p =2-3 μ	Nickel. Most pores on interval d_p = 0.6-1 μ
l_1 mm	1500	1500	1200
l_2	100	100	150
l_3	150	150	150
l_4	70	67	75
l_5	60	60	60
l_6	10	10	10
l_7	2,0	5,0	0
D_1 mm	29	29	29
D_2	15	16	11
D_3	1,2	1,2	0,5
D_4	2,5	2,5	2.5
D_5	2,0	2,0	2.0
D_6	4,0	4,0	3,0
δ_1	1,0	1,0	1,0
δ_2	1,0	1,0	1,0

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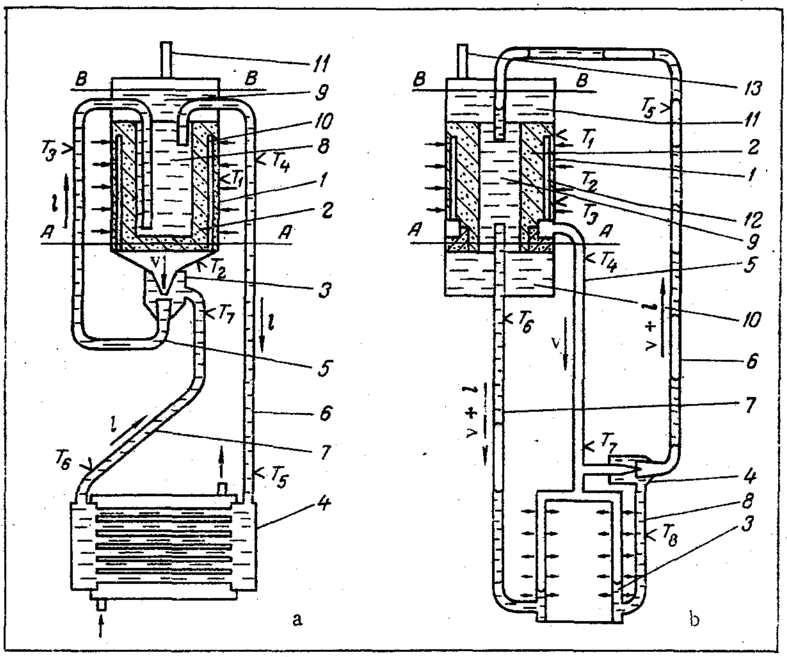

Fig. 1. Heat pipes with vapor injection: a) with injector—condenser; b) with discrete transportation of heat-transfer agent.

The principal feature of the design (Fig. 1a) is the fact that the entire vapor flow is directed through the nozzle into the injector, where the vapor jet condenses in a space filled with constantly renewed liquid [5]. As a result of the condensation of the vapor jet and momentum transfer between the condensed vapor flow and the injected liquid [6] the thermal and kinetic energy of the vapor jet is converted into static head. This head causes the heat-transfer agent to circulate through the closed loop at a corresponding ratio of the circulation and evaporator feed flows. Heat is transferred to the receiver through a heat exchanger inserted in the loop.

The heat pipe is filled with heat-transfer agent up to level A—A. Thus, in the nonoperating pipe the porous lining of the evaporator is always in contact with the heat transfer agent and saturated with liquid. When the evaporator is heated, the vapor pressure forces the liquid out of the vapor channels and the injector nozzle and raises it to level B—B in the expansion chamber, which corresponds to the rated load.

Obviously, the system can operate when

$$\Delta P_\sigma \geqslant \Delta P_{wv} + \Delta P_{wl} + \Delta P_n + \Delta P_{cl} + \rho_{cl} g h \sin\varphi,$$

where ΔP_σ is the capillary pressure difference determined by the capillary structure and the nature of the liquid; ΔP_{vs}, ΔP_{wl} are the flow friction of the wick with respect to vapor and liquid; ΔP_n, ΔP_{cl} are the flow friction of the nozzle and the circulation loop with respect to vapor and liquid, respectively; $\rho_l g h \sin\varphi$ is the hydrostatic pressure of the liquid column. A heat pipe of this design has certain advantages over ordinary heat pipes when, other things being equal ("height" of pipe, working fluid, capillary structure, etc.), the sum of the flow resistances external to the wick and the hydrostatic pressure of the liquid column is less than the corresponding sum for existing heat-pipe designs.

In the pipe described, this is achieved, despite the vapor flow resistance in the injector nozzle, by positioning the injector—condenser next to the evaporator, which substantially reduces the hydrostatic pressure of the liquid column.

Thus, minimizing the distance between evaporator and injector ensures vapor condensation near the heat source and largely relieves the evaporator lining from the return of condensate to the evaporation zone, leaving it basically with the function of pumping vapor and organizing the circulation of the liquid for heat transfer over the required distance.

We investigated two heat pipes of the same design as that illustrated in Fig. 1a. Their design and technological characteristics are given in Table 1. Heat was supplied by an

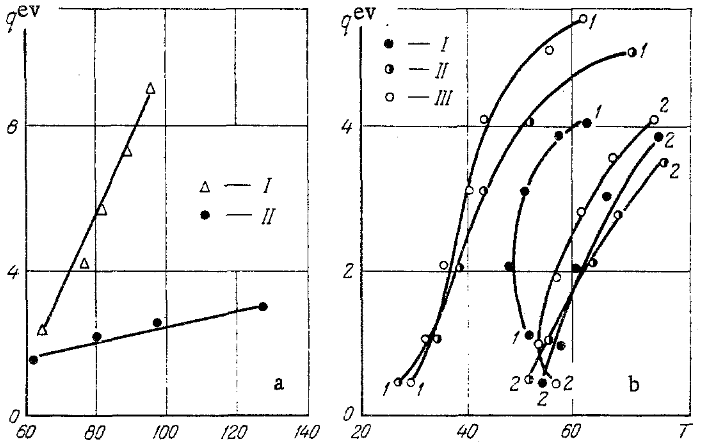

Fig. 2. Transferred heat flux density per unit heat supply surface q^{ev}, W/cm², as a function of the evaporator wall temperature T, °C: a) T_{cw} = 20°C; φ = 90°; I) water; II) acetone; b) acetone: 1) T_{cw} = 25°C; 2) 50°C; I) φ = 90°; II) 0; III) 90.

electric heater over a 45 mm length of the evaporating chamber, while the heat exchanger was cooled by water pumped from a TS-16 thermostat. The flow of cooling water was maintained at 65 ± 0.5 liter/h and monitored with an RÉ-02 rotameter. The temperature of the cooling water was maintained at 20 ± 0.1°C. The temperature was measured at the points indicated in Fig. 1a by means of copper—Constantan thermocouples and an R-37/1 potentiometer. The temperature difference between the inlet and the outlet of the heat-exchanger cooling jacket was also measured.

With the porous linings used in this system, operation on acetone proved possible at pressures in the pipe somewhat above atmospheric. This mode of operation was achieved by placing in the expansion chamber a sealed sylphon filled with air at 20°C.

The orientation of the pipes was varied from φ = 90° (evaporated above condenser) to φ = —90°. The results of the tests are presented in Fig. 2a, Fig. 3a, and Fig. 4a.

An analysis of the experimental data showed that in the case of heat transfer over distances on the order of several meters or in the presence of accelerations the pipes have certain advantages over existing types. Moreover, the heat-transfer agent circulating in the loop can be used for cooling additional heat sources. In this case heat sources and sinks, each with a power not exceeding that of the main source, alternate along the circulation loop.

Nonetheless, in order to function the heat-pipe design with total vapor condensation in the injector requires a considerable temperature difference between the heat source and the sink. This is attributable to the small condensation surface in the injector—condenser and the small pressure drop created by the capillary structure of the wick.

The heat-pipe design illustrated in Fig. 1b does not have this disadvantage to the same extent.

The main feature of this design is the fact that the heat-transfer agent is transported discretely through the condensate tubes — in the form of alternating columns of liquid and vapor. The absence of a continuous liquid column substantially reduces the hydrostatic pressure. The liquid column is broken up by injecting vapor into the condensate loop.

The heat pipe (Fig. 1b) consists of an evaporator 1 with a capillary-porous lining 2, a condenser 3, and an injector 4, connected by a vapor tube 5 and condensate tubes 6, 7, 8. In the evaporator lining there is a through-channel 9 communicating with the symmetrically disposed expansion chambers 10 and 11, together with vapor channels 12. Connection 13 serves for evacuating and filling the pipe.

The evaporator lining consists of finely porous nickel and fits tightly into the evaporator body. The capillary structure is relatively homogeneous with an open porosity of 60%. Most of the pores lie on the interval d_p = 0.6-1.0 μ.

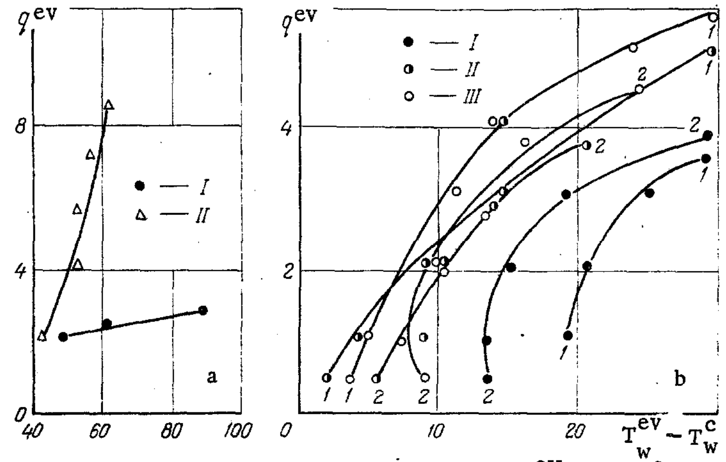

Fig. 3. Surface heat flux density q^{ev}, W/cm², as a function of the external temperature difference of the heat pipe $\Delta T = T_w^{ev} - T_w^c$, °C: a) $T_w^c \approx T_7$; $T_{cw} = 20$°C; $\phi = 90°$; I) acetone; II) water; b) acetone: $T_w^c \approx T_8$; 1) $T_{cw} = 25$°C; 2) 50°C. I) $\varphi = 90°$; II) 0; III) 90.

In the absence of a heat load the liquid level in the heat pipe corresponds to the section A—A. Accordingly, the lining is always saturated with heat transfer agent. When heat is supplied to the evaporation zone, the vapor pressure forces the heat-transfer agent out of the vapor channel 5, the injector nozzle 4, and part of the condenser 3, and into through-channel 9 and part of the expansion chamber 11, in which a certain level B—B is established.

In this system there are two condensation zones, remote from the heat source: a condenser with total, and an injector with partial vapor condensation. The main vapor flow enters the condenser through the low-resistance vapor tube and condenses on the developed walls. A smaller vapor fraction (determined by the nozzle cross section) enters the injector, where it partly condenses and partly enters the condensate tube, breaking up the liquid column.

An additional condensate tube 7 is provided to remove heat and vapor from the reservoir of the evaporator 9 and expansion chambers 10 and 11. It connects the evaporator 9 with the injector intake chamber via the condenser.

When the heat pipe is operating under the severest conditions, with the evaporator on top, the airlift effect is used; this is achieved by locating the injector at the bottom, together with the condenser.

Thus, in addition to its direct function — breaking up the liquid column in the condensate tube — the injector also serves to pump the vapor—liquid mixture through the closed circulation loop.

The incomplete vapor condensation in the injector makes it necessary to provide for condensation elsewhere. This condensation takes place, first, in expansion chamber 10 or 11, depending on the orientation of the pipe, and, second, as a result of part of the vapor being dumped discretely into the condenser through the additional condensate tube 7.

Moreover, as preliminary experiments have shown, discrete transportation of the heat transfer agent through the capillary structure is also possible. In this case part of the vapor from cavity 9 penetrates into the condenser through the vapor channel.

Tests which confirm this assumption were carried out on the heat-pipe evaporator. The supply of the evaporator chamber with vapor—liquid mixture was simulated using such inert gases as helium, argon, or neon. For this purpose a continuous flow of liquid was supplied to the evaporator chamber and the heat load was applied. Then, under steady-state thermal conditions, inert gas under slight excess pressure was admitted to the expansion chamber. An analysis of vapor samples in an MI-1305 mass spectrometer revealed the presence in the vapor of the corresponding inert gases in amounts considerably exceeding those corresponding merely to the solution of the gas in the liquid.

Thus, it may be assumed that small bubbles of both gas and vapor can penetrate to the evaporation surface through the wetted capillary-porous wall of the wick.

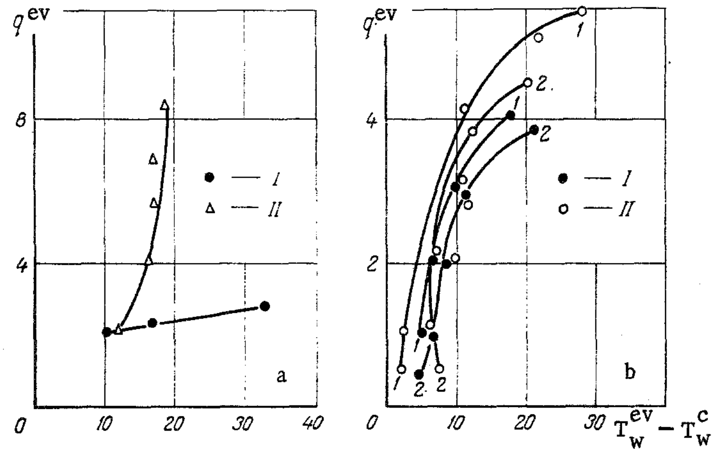

Fig. 4. Surface heat flux density q^{ev}, W/cm², as a function of the temperature difference $T_w^{ev} - T_{sat}$, °C: a) T_{cw} = 20°C; φ = 90°; I) acetone; II) water; b) acetone: 1) T_{cw} = 25°C; 2) 50°C; I) φ = 90°; II) 90°.

We investigated the 1200-mm acetone heat pipe whose design is illustrated in Fig. 1b. Heat was supplied from an electric heater over part of the evaporator chamber 50 mm in length and 30 mm in diameter, and the condenser was cooled by pumping water from a TS-16 thermostat. The flow of cooling water was maintained at 60 ± 0.5 liter/h and monitored with an RÉ-02 rotameter. The temperature of the cooling water was either 25, 40, or 50°C. The pipe was located in a room with the air temperature at 25°C. We measured the temperatures at the characteristic points indicated in Fig. 1b, together with the temperature difference between the inlet and the outlet of the condenser cooling jacket. For the temperature measurements we used copper–Constantan thermocouples and an R-37/1 potentiometer.

For visual observation purposes the condensate tubes 6 and 7 were made of glass; their inside diameter was 3 mm. The vapor tube was 5 mm in diameter.

In both condensate tubes we observed the discrete transportation of the heat-transfer agent in the form of alternating columns of liquid and vapor traveling at about 0.3 m/sec, there being less vapor in tube 7 than in tube 6. If in tube 6 the volume content of liquid and vapor is the same, then, as a simple calculation shows, the mass of the vapor

$$m_v = m_l \rho_v / \rho_l .$$

The heat released as a result of the condensation of this vapor mass

$$Q_l = m_v r$$

is much less than that transferred to the condenser by the heat pipe

$$Q_l = m_l r / u,$$

where u is the ratio of the circulation and feed flows. Then

$$Q_v / Q_l = u \rho_v / \rho_l .$$

For acetone at 56°C when u = 5 this quantity is equal to

$$Q_v / Q_l = 0.015.$$

Thus, if the heat pipe transfers 150 W, about 2.2 W is expended on breaking up the liquid column in the condensate tube, and the hydrostatic pressure in the heat pipe is reduced by a factor of two.

In Fig. 2b and Fig. 3b we have plotted the surface heat flux density (per unit heat-supply surface) against the mean temperature of the evaporator wall under the heater coil and against the external temperature difference of the heat pipe for various orientations and condenser cooling conditions. It is clear from the figures that the heat pipe yields optimal results when the evaporation is located below the condenser (φ = –90°). In Fig. 3b, it is clear that at φ = 90°, when the evaporator is above the condenser, the external temperature difference falls as the cooling water temperature increases. This indicates that in these circumstances the operating point of the heat pipe is shifted upward along the saturation curve of the heat-transfer agent, where the pressure difference required to force the heat transfer agent into the evaporation zone is provided by a smaller temperature difference. the pressure difference required to force the heat transfer agent into the evaporation zone is provided by a smaller temperature difference.

In Fig. 4b we have plotted the surface heat flux density in the evaporation zone (at the temperatures indicated in Fig. 2b) against the difference of the evaporator wall and saturated vapor temperatures.

NOTATION

Q, heat flux in pipe; q, heat flux density; ρ, density; m, mass; r, heat of vaporization; ϕ, angle of inclination of pipe with respect to horizontal; g, acceleration of gravity; h, distance between beginning of vapor channels in evaporator lining and uncompensated level of liquid in heat pipe; d_p, wick pore, diameter; Π, wick porosity; l_1, l_2, l_3, l_4, l_5, l_6, and l_7, lengths of heat pipe, evaporating chamber, heat exchanger (condenser), wick, vapor channels, injector mixing chamber, and distance between nozzle and mixing chamber, respectively; D_1, D_2, D_3, D_4, D_5, and D_6, inside diameters of evaporating chamber, evaporator reservoir, nozzle, mixing chamber, vapor channels, and circulation loop tubes, respectively; δ_1, distance of vapor channel from evaporating chamber wall; δ_2, distance between vapor channels; T_w^{ev}, evaporator wall temperature; T_w^c, condenser wall temperature; T_{cw}, temperature of cooling water; u, circulation and evaporator feed flow ratio.

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EXPERIMENTAL STUDY OF LOCAL HEAT EXCHANGE BETWEEN AN INCLINED PLATE AND AN IMMOBILE

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The local heat-liberation-coefficient distribution is described for the surface of a plate immersed in a dispersed layer, with the angle of incidence to the main gas flow varied over a wide range.

The recent wide use of boiling layers as heat-transfer media in the operation of immersed heat exchanges requires more detailed study of the intensity of heat liberation from bodies located in dispersed layers.

The literature has considered the questions of local heat exchange from vertical and horizontal tubes and tube clusters immersed in a layer, and from vertical and horizontal plates [1-3]. The effect of the orientation of the immersed body relative to the incident flow of draft medium has been considered in less detail [4-7]. The latter studies considered the cases of flow around a plate inclined at 45° to the vertical, with 40-50% of the cross section obstructed.

The goal of the present study is a clarification of the distribution of the heat-liberation coefficient over the surface of a small plate immersed in a dispersed layer over a wide range of attack angle in both an immobile and a boiling layer.

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