PROBLEMS IN BUILDING VERTICAL HIGH-TEMPERATURE HEAT PIPES

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This paper examines problems arising in building vertical high-temperature heat pipes with simple and composite wicks. The author's test results with gas-free and gas-controlled sodium heat pipes are presented.

Vertically located high-temperature heat pipes can be used in nuclear engineering, electronic technology, chemical technology, and other technology areas. In building them one must frequently: 1) supply heat to a considerable height of pipe, sometimes up to 0.5 m; 2) insure high heat transfer; 3) provide thermal stabilization. It is not easy to fulfill these conditions, even individually. And in combination they present conflicting requirements as to choice of capillary structure.

To avoid overburning of the pipe under boundary conditions of the second kind, the capillary structure must be wetted with the liquid over the entire height of the heating zone, even in the start-up process. For example, this type of structure may be a mesh directly at the wall, very fine channels, or an annular gap of very small size under a finely porous screen. However, a uniform capillary structure satisfying this condition is incapable of high heat transfer for reasonable wick thickness.

High specific heat transfer (more than 0.5 kW/cm^2) can be obtained by using the so-called composite wicks (of ring or arterial type). In composite wicks the high capillary head is created by the liquid in very fine pores of the wick screens which take the form of fine sintered particles or a fine mesh, usually of serge-type weave. To reduce the hydraulic drag in composite wicks one must have capillary channels (of ring type or in arteries) of large enough hydraulic diameter. In contrast with plain wicks, the composite type are capricious in operation. They provide high heat transfer only with a full charge of liquid. The presence of unwetted areas, or vapor or vapor-gas bubbles in the heater zone leads to boiling of liquid inside the wick, the formation of "hot spots," to drying out of the wick, and even to overburning of the heat pipe [1]. It is difficult to build gas-controlled heat pipes with composite wicks because of the possible ingress of gas into the liquid filling the capillary structure. Combination simple wicks, which have channels of different sizes, are more reliable and capable of operating with noncondensible gases present. But they provide only moderate heat transfer.

The application and construction of vertical high-temperature heat pipes of moderate power has been discussed in [2-4]. An attempt to use heat pipes with composite arterial wicks in [3] did not provide positive results. Heat pipes (HP) with screw-type open grooves have been built, giving conditions of moderate heat transfer. However, the question of achieving operation of HP with composite wicks continues to remain an urgent one.

To satisfy conditions 1) and 2) one needs a capillary structure with at least two types of channels: fine channels to ensure wetting of the heater zone in the start-up period, and coarse channels, closed on the vapor channel side by a finely porous screen, and designed for high heat transfer.

With vertical heat pipes it is frequently difficult to count on guaranteed initial filling of a composite wick with liquid. One must therefore carry out the filling during operation.

We now consider conditions for start-up of an HP with a composite wick with channels of two sizes (Fig. 1). In the analysis we shall assume that the HP composite wick operates in evaporation conditions. At the

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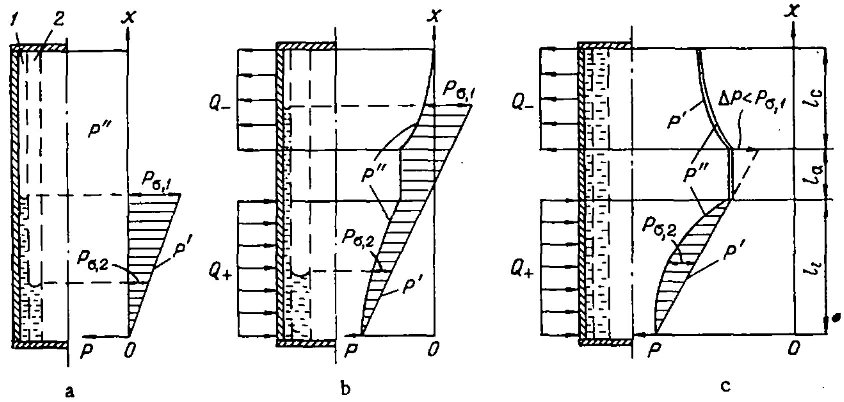

Fig. 1. Filling of channels with liquid in a vertical heat pipe with a two-channel composite wick.

initial time the wick is not completely filled with liquid. During the start-up a pressure drop will arise in the vapor region, and this can promote filling with liquid of the channel of larger diameter. We shall determine conditions for which there can be self-filling of the larger wick channel, if the condition for filling of the first (smaller) channel along the heater zone is satisfied:

$$P_{\sigma,1} \equiv \frac{2\sigma\cos\theta}{r_1} \geqslant \rho g l_l. \tag{1}$$

Figure 1 shows the location of liquid in the HP and the pressure variation at three characteristic times of the start-up: a) initially, when there is as yet no heat supply; b) when there is heat supply and the HP operating conditions are sonic with a vapor pressure drop in the condenser section; and c) at the onset of subsonic conditions, when both channels (1 and 2) can be filled over the entire height of the pipe, and ready to conduct liquid. At time c ("tinning") there is an excess of liquid below the pipe, and it is transported to the zone section by film flow. At time c, if conditions are favorable for vapor bubble condensation [1], the liquid fills the entire length of the larger channel. The least vapor pressure (or temperature) for which there can be filling of the larger channel of a composite wick in the evaporation zone (Fig. 1c) corresponds to the condition

$$|P''(x) - P'(x)|_{\max} = P_{\sigma,2}. \tag{2}$$

For simplicity we assume that the specific heat fluxes are constant along the evaporation zone and that the tinning height is small and can be neglected. At the low vapor pressures typical for the case considered, the inertia effect dominates over friction, and the pressure distribution along the heater zone is roughly parabolic:

$$P''(x) - P'(0) \simeq \frac{Q^2 x^2}{8\rho'' L^2 R^4 l_l^2}. \tag{3}$$

The liquid pressure variation in this zone is due mainly to the hydrostatic effect, and follows an almost linear law along the zone

$$P'(x) - P'(0) \simeq \rho' g x. \tag{4}$$

The maximum pressure drop for the case corresponding to time c (as can be shown by finding the extremum) lies in the middle of the evaporation zone, and is one-quarter of the total vapor pressure drop over the evaporation zone. Consequently, condition (2) will then take the form

$$\frac{1}{4}\,\frac{Q^2 x^2}{8\rho'' L^2 R^4 l_l^2} = P_{\sigma,2} \equiv \frac{2\sigma\cos\theta}{r_2}. \tag{5}$$

It is necessary that the large channel also be filled in the adiabatic zone. If we neglect the vapor pressure variation in this zone, then the condition for filling of the large gap there can be written as:

$$\rho' g l_a \leqslant P_{\sigma,2}. \tag{6}$$

To simplify the calculations it is convenient to construct and use a nomogram (Fig. 2). For a given length of evaporation zone we can use the nomogram to find the power and the temperature, and also the maximum sizes of the effective radius of the large channel for which self-filling of a composite wick can occur during start-up of the heat pipe. It follows from Fig. 2 that for constant specific heat flux along the evaporation zone, the quantity r_1 must be less than r_2 by a factor of no less than 4. The height of the adiabatic zone must be no more than one-quarter of the length of the heater zone.

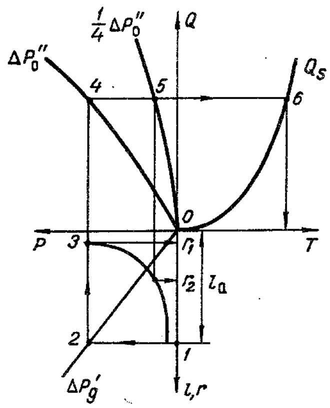

Fig. 2. Nomogram for checking the filling capacity of channels of a composite wick in a vertical heat pipe.

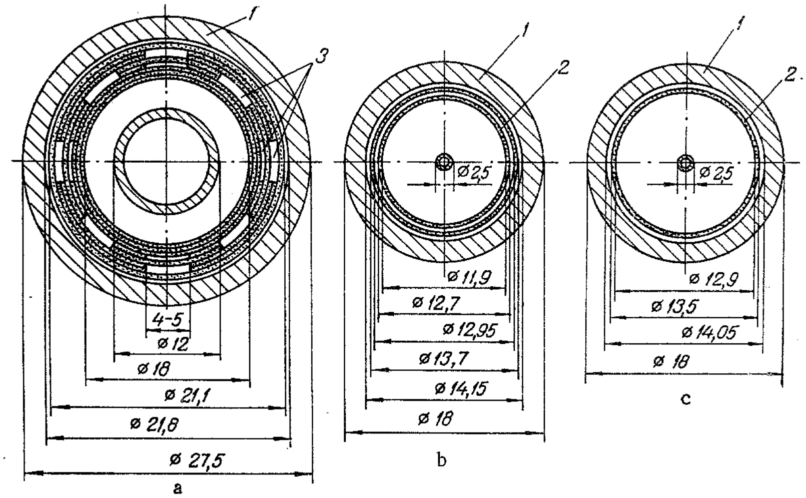

Fig. 3. Sections of heat pipes [a) No. 1; b) No. 2; c) No. 3]; 1) pipe body; 2) staggered fine-mesh screens; 3) combined wick of fine mesh with internal channels.

It is clear that conditions for filling of a composite wick with liquid can be improved by shaping the specific heat flux along the evaporation zone, or by shaping the vapor channel cross section. Estimates show that the conditions for self-filling of the arteries of the type GK-1 and GK-3 pipes in [3] are not fulfilled and therefore high heat transfer cannot be achieved.

The authors studied the heat transfer experimentally in three sodium heat pipes with simple and composite wicks (see Table 1). Heat pipe No. 1 had a combined simple wick, made of several layers of one-way serge weave mesh No. 450 (diameter of base wires 0.09 mm, and of the woof 0.055 mm). Along the wick, between the mesh layers, narrow strips of the same mesh were spot-welded, forming 12 channels, 8 of width two mesh layers, and 4 of width one layer (Fig. 3).

The heat pipe was tested with the lower part heated by noncondensible gas (to a height of 50 mm), i.e., the heat-transfer conditions were of the first kind. Heat was removed by means of a water calorimeter via the gas-controlled gap. In the first series of tests the condenser length was 280 mm, and in the second series it was 120 mm. The heat pipe was equipped with a system for supplying its vapor space with a quantity of helium, metered by means of a piston. Tests were conducted both with no noncondensible gas in the pipe vapor space, and also with helium there. The repeatability of the results was satisfactory, up to the limits of power transferred, fixed by the beginning of thermal resistance of the pipe. The test results are shown in Fig. 4. The limits of power transfer, as shown by the first series of tests, were independent of the presence of helium in the pipe, and varied very little as a function of temperature. Some difference in the power limits was ob-

TABLE 1. Parameters of the Sodium Heat Pipes

Heat pipe No.	1	2	3
Gas presence	He supplied	No gas	He supplied
Length, mm			
total	1000	960	680
heater zone	400	400	94
adiabatic 120*	280	20	40
condensation 280*	120	487	340
Diam., mm			
body, outer	27,5	18,0‡	16,9
body, inner	22,1	14,15	14,05
vapor channel	18,0†	11,9	12,9
Wick type	Simple	Composite	Composite
Effective radius of screen pores, μm	—	17	22
Heater	Condensation of sodium	Electron bombardment	Condensation of sodium
Cooling	By water through the gap	By radiation to vacuum	By water through the gap
Power reached, W	5800	1520	6900
Specific heat flux in the evaporation zone, W/cm²	25,0	10,0	166,5
Corresponding temp., °C	760—810	860	760

*In the first series of tests.

†At the center of the pipe there was a tube of diameter 12 mm for locating the auxiliary heater.

‡A copper layer of diameter 32 mm was soldered to the pipe body in the heater zone and in the cooling zone a copper layer of diameter 24 mm was soldered to the body and held by a screw.

served when a change was made to a larger length of adiabatic zone, and simultaneously a reduced condensation zone length. The characteristics of temperature variation at different powers in a gas-controlled heat pipe, obtained in the second series of tests, depend on the heat-transfer level. Within the thermal control region the least temperature variation for a given power variation was observed when the thermal conductivity of the gas gap was greatest. On the whole, it was established that by using a combined wick one can build gas-controlled heat pipes with comparatively high heat transfer (up to 5 kW/cm^2).

Heat pipe No. 2 had a composite wick (Fig. 3), in the form of two screens, joined "hermetically" to the body, and made of fine one-way No. 685 serge-weave mesh (diameter of base wires 0.064 mm, and of the woof 0.032 mm). The screens were made of three layers of mesh, tightly rolled and welded along the sides. The nominal gap between the body and the first screen was 0.225 mm, and between the screens it was 0.125 mm. However, since centering was done only at the ends of the pipe, the actual gaps differed from the above values.

The lower part of the heat pipe was heated by electron bombardment, i.e., the heat supply conditions were of the second kind. In the upper part of the pipe the heat removal to the water was by radiation in vacuum. Start-up was accomplished without preliminary heating by supplying a power step of up to 1520 W. Initial filling of the composite wick with heat-transfer agent while still in the horizontal position, combined with the massive construction to give the necessary thermal inertia, promoted absence of drying out of the heat pipe when subject to the stepwise power profile. Freezing of the heat-transfer agent in the cooling zone was prevented by the low-level heat removal method used: with radiation in vacuum the starting warm-up and the cooling after switch-off occurred in low heat-transfer rate conditions. Figure 4 shows the experimentally obtained relationship between the power supplied and the pipe temperature. The tests confirmed the possibility of an operable composite wick, using radiation in vacuum as a crude thermal stabilization method. The main advantage of the system used is the possibility of start-up by applying a power step without preliminary heating.

To prove the possibility of operating gas-filled heat pipes with composite wicks, we conducted tests on a horizontal heat pipe, No. 3. In this pipe the wick had a single screen, made in the same way as for the previous pipe, No. 2 (Fig. 3). The screen stood off from the housing body with a gap of 0.275 mm. The pipe was heated by condensed vapor, and cooled by a water calorimeter via the gas-controlled gap. There was a system for supplying measured portions of helium to the vapor space of the operating pipe. Tests conducted with no

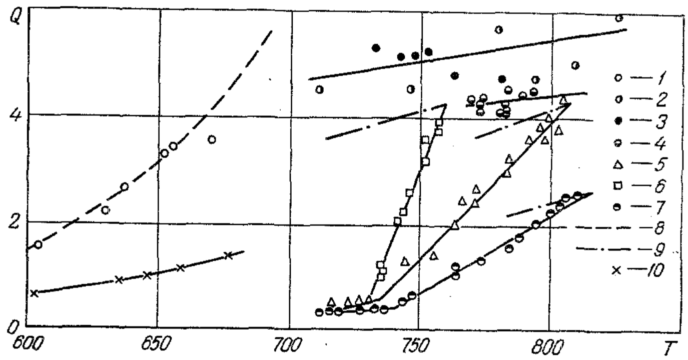

Fig. 4. Relation between the power supplied and the temperature for heat pipes Nos. 1 and 2 (Q in kW; T in °C). The tests on pipe No. 1 had a condenser length of 280 mm: 1) without gas filling, sonic limit; 2) without gas filling, power limits; 3) with gas filling, power limit. Tests on heat pipe No. 1 with a condenser length of 120 mm and gas filling: 4) the helium pressure in the gap varied, power limit; 5) argon in the gap, no limit; 6) helium in the gap, no limit; 7) vacuum in the gap, no limit; 8) sonic limit, theory; 9) pipe with no gas filling, theory. Tests on heat pipe No. 2: 10) no gas filling, condenser cooling by radiation to vacuum.

gas showed that the composite wick was operable. The maximum specific power at temperature 708°C was 4.16 kW/cm^2. Supplying measured amounts of helium to the vapor channel of the operating pipe did not lead to loss of operation. For example, at a temperature of 759°C the measured maximum heat transfer for a gas-filled pipe of this type was 5.25 kW/cm^2. Thermal drying out did not affect the operability of the heat pipe. On repeated supply of power, following preliminary chilling [5], the same high heat transfer was again obtained in the heat pipe.

NOTATION

g, acceleration due to gravity; l_l, l_a, l_c, length of the evaporation, the adiabatic, and the condensation zones; L, latent heat of vaporization; P', P'', pressure of the vapor and of the liquid; $P_{\sigma,1}$, $P_{\sigma,2}$, capillary pressure for the small and large channels; $\Delta P''_0$, vapor pressure drop in the evaporation zone; $\Delta P'_g$, liquid pressure drop in the evaporation zone due to the hydrostatic effect; r_1, r_2, effective radii of the small and large channels; x, longitudinal coordinate; R, radius of the vapor channel; T_0, temperature at the beginning of the evaporation zone; Q, thermal power transferred by the pipe; Q_S, sonic power limit; θ, wetting angle ρ', ρ'', density of liquid and vapor; σ, coefficient of surface tension; T, temperature.

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