INVESTIGATION OF AN ACTIVE REGULATION HEAT PIPE

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Results are presented of an experimental investigation of an active regulation heat pipe with two heat sources.

At this time, gas-regulated heat pipes are widely used as heat regulating apparatus, thermostats, radiation panels, etc. [1,2]. However, the limits of temperature regulation of the evaporator of passive gas-regulated heat pipes are constrained to a comparatively narrow temperature band.

To extend the heat regulation limits, gas-filled active-regulation heat pipes [3] are used in which the gas reservoir temperature varies automatically as the heat flux in the evaporator changes.

Results are presented in this paper of an investigation of the parameters of an active-regulation gas-filled heat pipe in which the condensation zone and the zone occupied by the gas are in its central section, and the heat supply is by means of two energy sources located at the edges of the pipe. Such a heat pipe has no fixed operating temperature, therefore, the heat regulation process can be accomplished in a broad temperature band according to the desire of the experimenter.

Moreover, this heat pipe can be used as a thermal bridge and heat flux meter; its analogous construction is described in [4].

The purpose of the research is an experimental investigation of a controllable heat pipe. The temperature on the outer surface of the active zone of the condenser was used as controllable parameter, while the controlling action is additional heat flux (control heater). The process at which the controlling action is directed is an additional evaporation—condensation cycle.

Analysis of the Regulation Process. We make the following assumptions in examining the regulation process:

1. We use the flat wall model $r_v/r_{hp} \approx 1$ in describing the heat transmission.
2. The pressure and temperature drops in the vapor are taken to be almost zero.
3. We use boundary conditions of the second kind on the outer surface of the evaporator (controlled and controlling).
4. We take boundary conditions of the third kind (heat transfer) on the outer surface of the condenser.
5. The heat pipe parameters are δ, λ, α, S_e^1, S_e^2, R_{e_1} and R_{e_2} = const.
6. The reduced thermal resistivity of the condenser and evaporator are equal: $R'_e = R'_c$.
7. The heat pipe position is horizontal.
8. The heat pipe operating mode is evaporative.

The total temperature drop over the heat pipe has the form

$$\Delta T_{Q_1} = T_e - T_c = \Delta T^{e}_{wa,\,w} + \Delta T^{c}_{wa,\,w} + \Delta T^{c}_{\alpha}, \tag{1}$$

where

$$\Delta T^{e}_{wa,\,w} = \frac{Q_1 R'_{e_1}}{S_{e_1}}, \tag{2}$$

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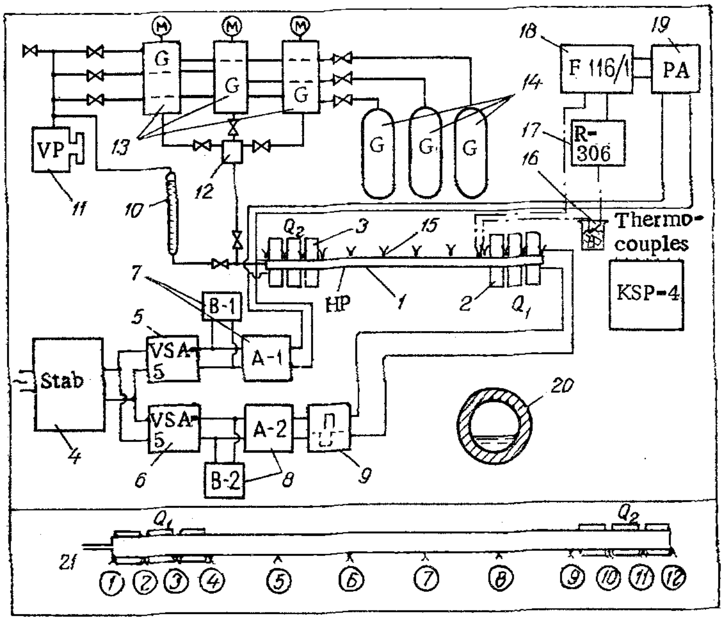


Fig. 1. Diagram of the experimental setup: 1) heat pipe; 2) heater Q_1; 3) heater Q_2; 4) power stabilizer; 5) dc source; 6) VSA dc source; 7) power measuring system of the heater Q_1; 8) power measuring system of the heater Q_2; 9) unit to produce rectangular pulses at the heater Q_1; 10) measuring vessel to measure the quantity of filled heat carrier; 11) vacuum pump; 12) valves; 13) measuring tanks with the noncondensing gas; 14) tanks with different gases; 15) sensors (thermocouples) of the regulation system; 16) zero junction of the sensors; 17) thermocouple emf compensation system; 18) photoamplifier of the regulation system; 19) regulation power amplifier; 20) heat pipe cross section; 21) thermocouple arrangement diagram; numbers in circles are the thermocouple number; G, gas.

$$\Delta T^{c}_{\mathrm{wa,w}} = \frac{Q_1 R'_{\mathrm{c}}}{S_{\mathrm{c}}}, \tag{3}$$

$$\Delta T^{\mathrm{c}}_{\alpha} = \frac{Q_1}{\alpha S_{\mathrm{c}}}. \tag{4}$$

The evaporator outer surface temperature of the controlled heater is expressed as

$$T^{Q_1}_{\mathrm{e}} = Q_1\left[\frac{R'_{\mathrm{e}_1}}{S_{\mathrm{e}}} + \frac{R'_{\mathrm{c}}}{S_{\mathrm{c}}} + \frac{1}{\alpha S_{\mathrm{c}}}\right] + T_{\mathrm{s}}, \tag{5}$$

where

$$\frac{R'_{\mathrm{e}}}{S_{\mathrm{e}}} + \frac{R'_{\mathrm{c}}}{S_{\mathrm{c}}} + \frac{1}{\alpha S_{\mathrm{c}}} = K. \tag{6}$$

We write the expression for the vapor temperature during operation of the second heater as

$$T^{Q_1+Q_2}_{\mathrm{v}} = (Q_1 + Q_2)\left[\frac{R'_{\mathrm{e}_2}}{S_{\mathrm{e}_2}} + \frac{1}{\alpha S_{\mathrm{c}}}\right] + T_{\mathrm{s}}. \tag{7}$$

The temperature checking factor has the form

$$\sigma = \frac{\Delta T^{Q_1}_{\mathrm{df}}}{\Delta Q_{1,\mathrm{df}}} = \frac{T^{Q''_1}_{\mathrm{v}} - T^{Q'_1}_{\mathrm{v}}}{Q''_1 - Q'_1}. \tag{8}$$

By using (5) and (8) the following expression can be written for the thermal checking factor

$$\sigma = \frac{\Delta T_{\mathrm{df}}}{\Delta Q_{\mathrm{df}}} = \frac{Q''_1 K + T_{\mathrm{s}} - Q'_1 K - T_{\mathrm{s}}}{Q''_1 - Q'_1}. \tag{9}$$

The purpose of active regulation is to maintain the thermal checking factor near zero

$$\sigma \to 0. \tag{10}$$

Condition (10) can be realized if the numerator in (9) is close to zero

$$Q''K + T_s - Q'K - T_s = 0. \tag{11}$$

For a two-heater heat pipe, when one of the heaters is the regulating unit, there follows from (7)

$$Q'' = Q_1'' + Q_2'' \quad \text{and} \quad Q' = Q_1' + Q_2'. \tag{12}$$

For the control process the expression (11) has the form

$$(Q_1'' + Q_2'')K + T_s - (Q_1' - Q_2')K - T_s = 0. \tag{13}$$

It follows from (13) that to maintain a constant temperature at the outer surface of the controlled heater, it is necessary to maintain the conditions

$$\sigma \to 0;\ Q_1'' + Q_2'' - Q_1' - Q_2' = 0 \tag{14}$$

or

$$\Delta Q_1 = -\Delta Q_2. \tag{15}$$

On the basis of the analysis performed, the deduction can be made that an active-regulation heat pipe will possess a minimum thermal checking factor under the assumptions made if conditions (12)-(15) are satisfied.

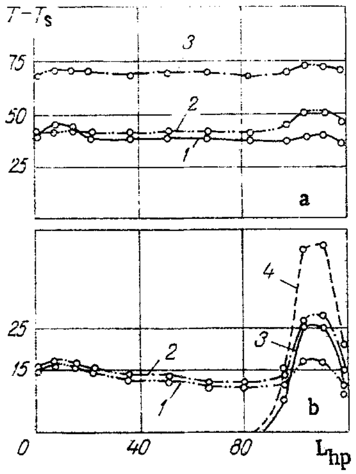
Technique of the Experiment. A horizontal wickless heat pipe made of stainless steel, 1200 mm long with 10-mm outer diameter and 0.3-mm wall thickness, and nitrogen ($P_g = 0.5 \cdot 10^5$ N/m²) as noncondensing gas, was used as the specimen to be investigated. The controlled and controlling heat fluxes were produced by three-section 200-mm-long ohmic heaters located at opposite ends of the pipe (Fig. 1).

A special unit was used to change the controlled heat flux, which permitted the production of rectangular current pulses 0.5 sec to 12 h in duration. Active regulation was accomplished by means of an electronic circuit which connects the temperature sensor (copper-Constantan thermocouple), the photoamplifier F116/1, and a specially developed power amplifier whose load was the controlling ohmic heater. To determine the value of the temperature in the active zone of the condenser a low-ohm dc potentiometer R-306 was connected into the F116/1 input loop. The heat scan was accomplished by free convection of air from the outer surface of the condenser in all the experiments. The temperature fields along the outer surface of the heat pipe were measured by using twelve Chromel-Copel thermocouples. Recording the readings of the thermocouples was done automatically by a KSP-4 electronic potentiometer.

The heat pipe parameters were measured in the stationary state with heat supplied both by one heater and by two simultaneously. The horizontal position of the heat pipe was determined to 0.2° accuracy by using a level gauge. Then both heaters were connected. Additional correction of the horizontal position of the heat pipe was made according to the thermocouple readings, and the heat pipe was considered installed horizontally when the temperature fields had identical form for identical heat fluxes.

The purpose of the first series of experiments was to investigate the influence of the heat supply from two opposite ends of the heat pipe on its characteristics. The results of this investigation are presented in Figs. 2a,b. Opposing vapor jets actively interacting in the condensation zone occur within the heat pipe. The result is a reduction in the thermal resistance of the pipe as compared with the usual heat pipe.

Operation of the two-heater heat pipe was investigated in the situation when the second heater was located above the first at a slope of 2°. The purpose of the experiment was to study the influence of the vapor flux and the presence of the condensate film on the heat elimination in the evaporation zone of the second heater. The results are presented in Fig. 2b. The substantial mutual influence of the heaters is seen from the figure. A deduction about the possibility of two-heater wickless heat pipe operation when one heater is located above the other can be made from an analysis of the data obtained. The operating capacity of two-heater heat pipes at slopes greater than 2° was noted in performing this series of experiments. But at larger angles, the "equipment" boiling mode started, which hindered analysis of heat pipe operation. The presence of the "equipment" boiling mode was determined



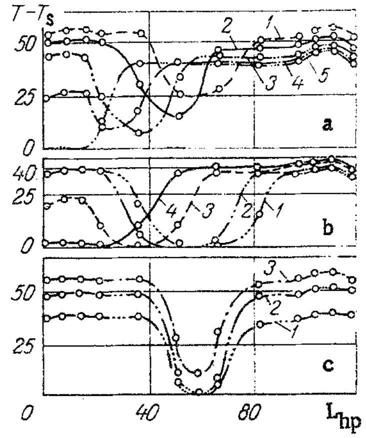


Fig. 2 Fig. 3

Fig. 2. Results of an experimental investigation of a heat pipe without gas under a bilateral heat supply $T - T_s$, °K; L_{hp}, cm: a) temperature distribution along the heat pipe length, located horizontally; $\theta = 0°$: 1) $Q_1 = 24.4$ W, $Q_2 = 0$ W; correspondingly 2) 0 and 24.4; 3) 24.4 and 24.4 (acetone 30 ml); b) temperature distribution along the heat pipe length at a 2° slope (second heater higher than the first): 1) $Q_1 = 3.5$ W, $Q_2 = 3.5$ W; 2) 3.5 and 7.0; 3) 0 and 3.5; 4) 0 and 7.0.

Fig. 3. Results of an experimental investigation of a heat pipe with a noncondensing gas under bilateral heat supply ($T - T_s$, °K; L_{hp}, cm): a) temperature distribution along the heat pipe length Q_2 = const: 1) $Q_1 = 17.0$ W, $Q_2 = 17.6$ W; 2) 11.9 and 17.6; 3) 7.0 and 17.6; 4) 3.5 and 17.6; 5) 0 and 17.6; b) temperature distribution along the heat pipe length $Q_1 + Q_2$ = const: 1) $Q_1 = Q_2 = 7.9$ W; 2) $Q_1 = 5.4$ W and $Q_2 = 10.5$ W; 3) 2.9 and 13.0; 4) 0 and 15.9; c) temperature distribution along the heat pipe length $Q_1 = Q_2$: 1) $Q_1 = Q_2 = 9.1$ W; 2) correspondingly 14.1; 3) 17.0 (acetone 30 ml; N_2, $P_g = 0.5$ atm).

by periodic temperature jumps over the whole length of the pipe with a ten-second repetition rate.

The second series of experiments investigated a two-heater heat pipe with a noncondensing gas. Primarily the possibility of blocking the gas in the central part of the condenser was investigated. Results of the experiment are represented in Fig. 3a. Analysis of the data obtained showed that the process of gas blocking in a two-heater heat pipe can be analyzed by using the mathematical apparatus used for analysis of ordinary (single-heater) gas-regulated heat pipes. The fact is noted that the active-zone stabilization temperature can be changed in two-heater gas-regulated heat pipes by changing the thermal flux supplied to the opposite part by the second heater. It is shown that the thermal resistivity of one heater is not dependent on the mode of the second.

An investigation was made of the influence of the location of the gas plug in the condensation zone as a function of the heat flux; the sum of the heat fluxes in the two heaters was hence kept constant in all cases. The results of the experiment are represented in Fig. 3b. It is shown that the location of the blocked zone depends strongly on the magnitude of the heat flux.

The length of the blocked zone was investigated experimentally as a function of the heat flux. The total heat flux was increased during the experiment with the heat flux from each separate heater being equal. The results are presented in Fig. 3c. It is shown that for large heat fluxes the length of the blocked zone does not depend as strongly on the total heat flux as in the case of low heat fluxes.

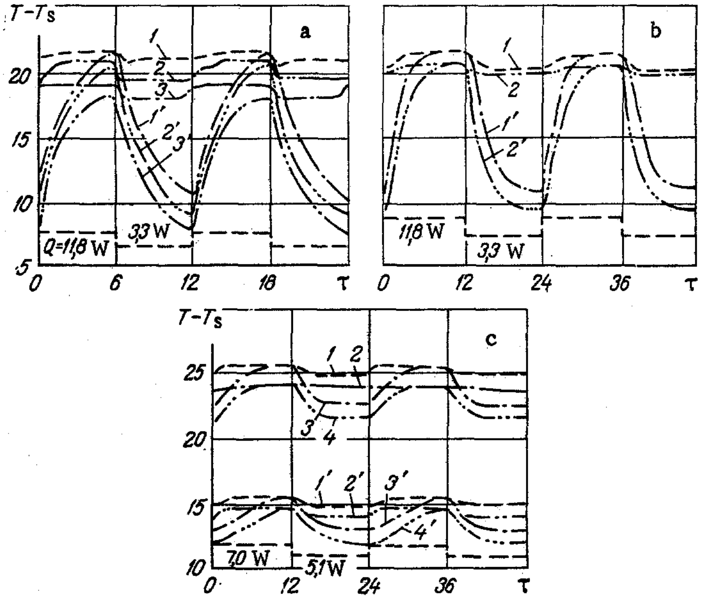


Fig. 4. Results of an experimental investigation of the active-regulation process ($T - T_s$; τ, min): a) temperature change with time for a heat pipe without a noncondensing gas (τ = 6 min), in the active-regulation mode: 1) temperature at the point T_1; 2) at T_2; 3) at T_7; in the mode without active regulation: 1') temperature at the point T_1; 2') at T_4; 3') at T_7; b) change in temperature with time for a heat pipe without a noncondensing gas (τ = 12 min) in the active-regulation mode: 1) temperature at the point T_1; 2) at T_4; in the mode without regulation: 1') at T_1; 2') at T_4; c) change in temperature with time for a heat pipe with noncondensing gas: 1,2) at points T_1 and T_4 with regulation; 3,4) temperature at the points T_1 and T_4 without regulation; 1',2',3',4') analogous characteristics for a heat pipe without a gas.

It should be noted that for a given heat pipe construction (boundary conditions of the second kind in the evaporator) the length of the blocked zone is a function of the temperature, which depends, in turn, on the heat flux. This result is in conformity with an analysis of gas-regulated heat pipe operation presented in [2].

It is noted that for large heat fluxes the temperature of the blocked zone is elevated somewhat, which is explained by heat transmission by both vapor diffusion through the blocking zone and by heat conduction along the wall and the fluid layer, where the second component of the heat flux predominates over the first. Moreover, the appearance of the Marangoni effect [5] probably affects the heat pipe operation under such conditions.

The third series of experiments included an investigation of heat pipe operation with active regulation. A rectangular heat flux signal was selected in the experiments as yielding poorly to regulation.

The active regulation of a two-heater gasless pipe was investigated primarily when one heater was the regulated element and the second heater was the regulating element. The possibility of regulation for a different heat pulse duration was studied. It is noted that for the heat pipe construction under investigation and the electronic circuitry used for regulation, control can be accomplished by starting with a $\tau > 3$ min heat pulse duration; this is determined by the inertia of the heat and mass transfer processes within the heat pipe and by the inertia of the electronic control system.

Results of an investigation of an actively regulated heat pipe without a gas but with a high degree of regulated heat flux are represented in Fig. 4a (for a pulse duration $\tau \approx 6$ min of the heat flux). Analysis of the data obtained shows that the change in heater temperature was 0.6°K. More exact regulation can be realized by using a more perfect electronic automation system. It is shown as a result of the investigation that the evaporator temperature was minimal during passage of the trailing edge of the regulated heat pulse, i.e., upon disconnection of the heater Q_1.

NOTATION

Q, heat flux; T, temperature; s, area; λ, coefficient of heat conduction; α, heat transfer coefficient; δ, thickness; r, radius; R', thermal resistivity; σ, temperature checking factor; P, pressure; θ, heat pipe slope; τ, time. Subscripts: hp, heat pipe; v, vapor; e, evaporator; c, condenser; s, heat sink; wa, wall; w, wick; df, destabilizing factor; g, gas.

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HEAT TRANSFER IN THE EVAPORATOR SECTION OF A ROTATING HEAT PIPE AT LOW ROTATIONAL SPEEDS

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The heat transfer in the evaporator section of a rotating heat pipe, with and without a wick, is investigated experimentally at low rotational speeds and low heat-flux densities.

Rotating heat pipes are used at the present time for the cooling of electrical machines subjected to heavy rotational-speed control demands (high-torque dc motors, frequency-regulated asynchronous motors, etc.). At high speeds of rotation, the condensate is uniformly distributed by centrifugal forces around the periphery of the heat pipe, and steady-state cooling prevails [1-4]. Periodically, however, the rotational speed of regulated motors approaches zero, whereupon the inertial overloads prove inadequate for uniform distribution of the condensate around the periphery. In this case, the condensate collects in the lower part of the pipe as a pool, and the wall-cooling regime becomes nonsteady. Accordingly, the following are the most timely problems in connection with the development of heat pipes for regulated electrical machines: determination of the rotational speed at which transition takes place from steady to nonsteady cooling of the pipe walls; assessment of the nature of the heat transfer in this transition regime and of the practicality of installing a wick in the pipe to improve the heat transfer at low rotational speeds.

We have investigated these problems experimentally on a special calorimetric arrangement with radiative heat input to the heat pipe and convective heat withdrawal from its condenser section by placement of the pipe in a flow-through water heat exchanger. The construction of the heat pipe and experimental apparatus was designed to create heat-transfer conditions such as occur in heat pipes used in regulated electrical machines. The vapor temperature in the pipe could be varied between 60 and 120°C, the heat-flux density in the evaporator section from $0.4 \cdot 10^4$ to $2 \cdot 10^4$ W/m², and the angular speed from 0.05 to 6 rad/sec. The evaporator section of the pipe had an inside diameter of 100 mm, length of 150 mm, and wall thickness of 10 mm; the condenser section had a diameter of 30 mm and length of 100 mm. The readings of

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