STEADY-STATE AND STARTING CHARACTERISTICS OF A HEAT PIPE WITH A WICKLESS TANK

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This paper presents results of an investigation of the characteristics of a gas-controlled heat pipe. The experimental results are compared with theory of a model with a planar front between the vapor and gas.

Gas-controlled heat pipes (GCHP) are being applied with increasing frequency in various fields of science and technology, where they are solving the problem of temperature stabilization of heat-emitting objects with variable heat flux Q and coolant temperature t_o. GCHP's with wickless tanks, located both inside the vapor volume and alongside it, provide high accuracy of temperature control, react less to a variation of the cooling conditions than other types of GCHP constructions, do not require auto-control devices for their operation, and are simple to manufacture [1-5].

To construct GCHP's with wickless tanks one must take into account the special features of unsteady processes, and take steps to avoid the vapor of the working liquid passing into the tank of noncondensable gas (NCG).

This paper describes an experimental study of the steady-state and transient characteristics of a heat pipe with an external wickless tank, and the development of methods of calculating the configuration of GCHP's of this type. The basic parameters of the test heat pipe were as follows: the heat-pipe length L_{hp} = 500 mm; the heating-zone length L_{evap} = 150 mm; the heat-removal length L_c = 290 mm; the external diameter × the wall thickness $D_{ex} \times \delta_{wall} = 28 \times 2$ mm; the material of the shell and wick is copper; the type of wick is an individual baked fiber with d_f = 40 μ; the wick thickness is δ_w = 0.5 mm; the porosity Π = 83%; the effective diameter of the wick pores is D_{av} = 0.0916 mm; the effective thermal conductivity of the wick is λ_{eff} = 4.7 W/m·deg; and the ratio of the tank and condenser volumes is V_t/V_c = 2.1; 4.2.

The tank of NCG communicates with the condenser volume by means of a connecting pipe with d_{in} = 6 mm, made of type 1Kh18N10T stainless steel. The diffusion vapor flow time constant of this system τ_d, calculated using the method of [2], is 20 hours. A system of heating and cooling the tank is provided to maintain the required temperature.

The control system characteristics were investigated on equipment shown schematically in Fig. 1. We determined the temperature field in the body wall $t_{wall} = f(L)$ and in the vapor stream of the heat pipe $t_v = f(L)$ in all the zones, while varying the power supplied Q, the coolant temperature t_o, and the mass flow rate G of coolant liquid.

The heat flux supplied was generated with a resistance heater wound on a copper block, simulating the heat-emitting object, and was varied from 30 to 1200 W. The required temperature of the coolant water was maintained using heat exchangers and varied from +10 to +65°C. The temperature field in the vapor volume was varied by means of 15 Chromel-Copel thermocouples, of which 1 was located in the heat supply zone, 10 in the condensation zone, and 4 in the tank. Thin-walled stainless-steel capillaries were used to insert the thermocouples into the internal volume of the heat pipe.

The temperature distribution in the heat-pipe body wall was determined by means of 5 thermocouples in the evaporator zone, 2 in the transport zone, and 18 in the condensation zone. All the thermocouples were inset in channels milled in the body wall. A type R37/1 potentiometer was used as a recording device for steady-state measurements, and type KSP-4 automatic potentiometers were used in the transient conditions.

Figure 2 shows the steady-state characteristic of a GCHP $t_v = f(Q, t_o)$ for various temperature levels, determined by the mass of NCG contained in the system. The nature of the

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Polytechnic Institute, Kiev. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 33, No. 3, pp. 381-387, September, 1977. Original article submitted September 29, 1976.

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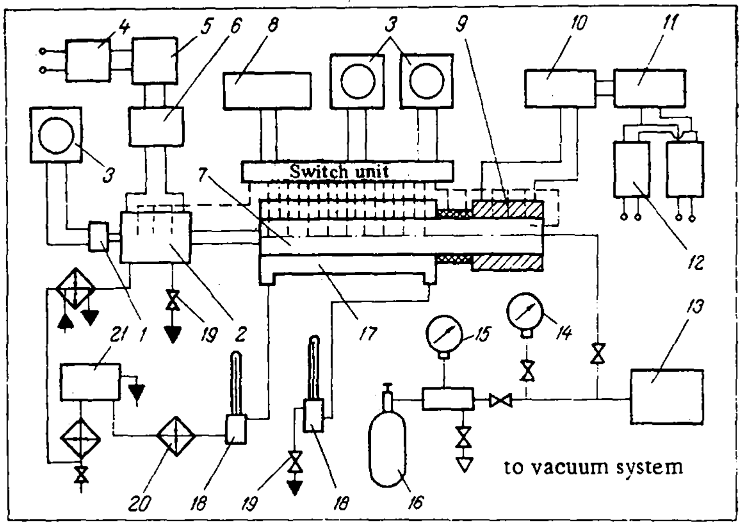

Fig. 1. Diagram of the experimental equipment: 1) pressure sensor; 2) tank of NCG; 3) type KSP-4 potentiometer; 4) type S-0.5 voltage stabilizer; 5) voltage regulator; 6) watt meter; 7) heat pipe; 8) type R37/1 potentiometer; 9) heater; 10) type K-50 measuring unit; 11) voltage regulator; 12) type S-0.9 stabilizer; 13) system for charging the working liquid; 14) standard vacuum gauge SVG; 15) vacuum manometer; 16) tank charged with NCG; 17) condenser; 18) thermometer; 19) fan; 20) heat exchanger; 21) constant level tank.

function $t_v = f(Q, t_o)$ varies when a minimum value of heat flux Q_{min} is reached (see points A_1, A_2, and A_3). This value depends on the form of the transport zone, the level of the control temperature, and the cooling conditions, and it marks the beginning of the control zone. In our experiments the quantity Q_{min} fell in the range 30-120 W. Because of the finite lenght of the vapor—NCG transition zone (l_{v-g} = 10-15 cm) part of the condenser is not used for active vapor condensation, and, therefore, the maximum transmitted power is limited (curves II). Curves I show the maximum transmitted power in the case where all of the condenser is actively used. In conditions where the temperature of the coolant liquid (t_o= 50, 65°C) is high, the slope of the curve $t_v = f(Q)$ is increased somewhat, due to the nonzero partial vapor pressure in the condenser.

For higher temperature levels and increased NCG mass m_g, the heat-pipe sensitivity $\sigma = dQ/dt_v$ increases. For example, at t_o = 23°C the sensitivity for all temperature levels, determined by the masses m_g = 0.049, 0.3, and 0.52 g, was 115, 170, and 280 W/°C, respectively. The range of variation of the vapor temperature increases somewhat with increase of NCG mass, because of the increased pressure in the heat pipe. An increase in the tank volume leads to a reduction in the slope of the curves $t_v = f(Q, to_o)$, i.e., the system has greater potential for thermal control of objects.

The basic technique used to compute the heat pipe is a planar vapor—gas front model [5]. A comparison of the calculated results with experimental data obtained on the test heat pipe without a tank showed that the vapor—gas transition zone has an appreciable influence on the accuracy of the computation. Because the mass of gas concentrated in the transition zone is a small fraction of the total amount of NCG in the system, for pipes with a reservoir, one expects good agreement between the calculated and experimental data. The working characteristics of the heat pipe can be calculated using the following system of equations, based on the conservation equations for energy and NCG mass:

$$Q = \frac{k_{l,\mathrm{ex}} k_{l,\mathrm{in}}}{k_{l,\mathrm{ex}} + k_{l,\mathrm{in}}} L_{\mathrm{c}} (t_{\mathrm{v}} - \bar{t}_0) \left(1 - \frac{L_{\mathrm{g}}}{L_{\mathrm{c}}}\right); \tag{1}$$

$$Q = \pi L_{\mathrm{evap}} D_{\mathrm{v}} k_{\mathrm{evap}} (t_{\mathrm{evap}} - t_{\mathrm{v}}); \tag{2}$$

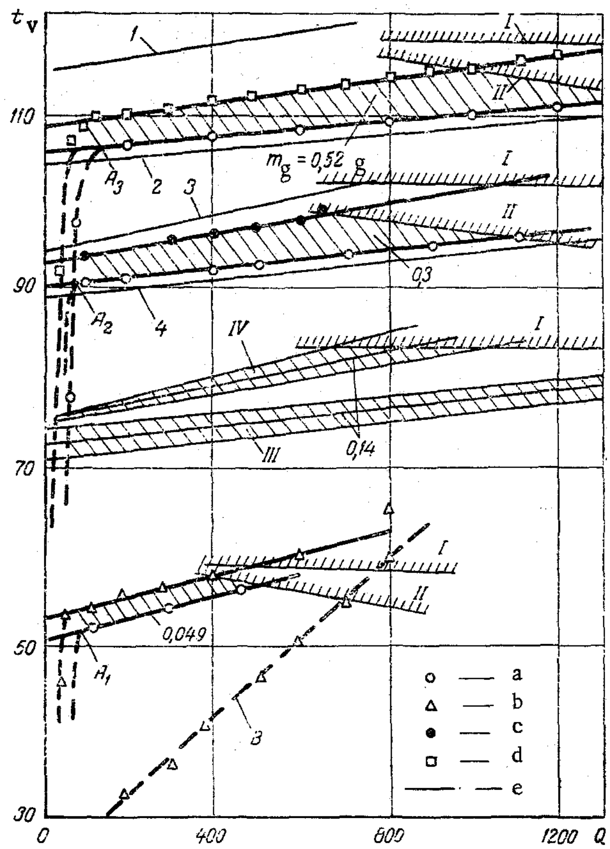

Fig. 2. Steady-state characteristics $t_v = f(Q, t_0)$: I) maximum transmitted power with all of the condenser surface operating; II) limit of transmitted power, obtained when the vapor—NCG front lies in the condenser zone; III) region where the NCG mass deviates from the theoretical value by ±5%; IV) region where the coefficient k_l deviates by ±20% from the theoretical value; the variation in vapor temperature at $m_g = 0.52$ g for $t_0 = 65°C$ (1) and 10°C (2), the variation in vapor temperature at $m_g = 0.3$ g for $t_0 = 50°C$ (3) and 10°C (4). The experimental data: a) t_0 = 10°C; b) 23; c) 50; d) 65; e) theory; B is the heat pipe without NCG; t_v and t_0 are in °C; and Q is in W.

$$Q = k_{l,\mathrm{in}} L_c (t_v - t_c)\left(1 - \frac{L_g}{L_c}\right); \tag{3}$$

$$k_{\mathrm{evap}} = f(p,\ \delta_w\ \lambda_{\mathrm{eff}},\ D_{\mathrm{av}}\ \delta_{\mathrm{wall}},\ \lambda_{\mathrm{wall}}); \tag{4}$$

$$\frac{L_g}{L_c} = \frac{m_g R_\mu (t_0 + 273)}{M V_c (p - p_0)} - \frac{V_t}{V_c} \cdot \frac{t_0 + 273}{t_t + 273} \cdot \frac{p - p_t}{p - p}; \tag{5}$$

$$p = f(t_v). \tag{6}$$

Figure 2 shows the results of calculations using Eqs. (1), (5), and (6). The experimental data agree well with theory. There is some disagreement for low heat flux $Q \leq Q_{min}$, arising from the fact that the model does not account for heat transmission through the heat-pipe body wall. For $Q < Q_{min}$ the front is located in the transport zone. If one assumes that a large part of the heat flux is transmitted by conduction through the body walls to the condensation zone and is then removed by the coolant liquid, Eq. (1) then takes the form

$$Q = \frac{\sqrt{\alpha_{\mathrm{ex}} \Pi_c \lambda_c F_c}}{1 + L_c \dfrac{\sqrt{\alpha_{\mathrm{ex}} \Pi_c \lambda_c F_c}}{\lambda_{\mathrm{tr}} F_{\mathrm{tr}}} \left(\dfrac{L_g}{L_c} - 1\right)} (t_v - t_0). \tag{7}$$

Equation (7) shows that the minimum heat flux depends appreciably on the shape of the transport zone (λ_{tr}, F_{tr}) and the levels t_v and t_o, and this is confirmed experimentally.

The system of equations (1)-(6) enables us to study the influence of the individual parameters on t_v. Region III on Fig. 2 shows a mass divergence of ±5% relative to the calculated value $m_g = 0.14$ g. Region IV shows the influence of the coefficient of heat transfer from the vapor to the coolant liquid k_l on t_v. The boundaries are formed by the curves for which k_l is 20% greater and less than the theoretical value. Variation in the amount of NCG in the system leads to displacement of the $t_v = f(Q, t_o)$ curves, and a variation in k_l changes the slope of the curves. It is clear that the influence of m_g on t_v is very large, and, therefore, charging of the heat pipe is a very important operation. In the experiments the filling mass of gas was equal to the amount of NCG in the system, determined from results of measuring the temperature field in the vapor stream:

$$m_g = \frac{M(p-p_t)V_t}{R_\mu(t_t+273)} + \int_0^{L_c} \frac{F_v M[p-p_0(x)]}{R_\mu[t_v(x)+273]}\,dx,$$

where $t_v(x)$, $p_o(x)$ are the temperature and partial pressure of the vapor in the vapor—gas mixture along the condenser.

The experimental data as shown were obtained at constant tank temperature t_t = 20-25°C. Curves 1-4 on Fig. 2 show the variation of t_v in the case when the tank temperature is equal to that of the coolant medium.

A study of the temperature profiles in the vapor and the wall has shown that there is an appreciable drop between the vapor and the wall, both in the active condensation zone and in the vapor—NCG transition zone. This is evidence that the heat-transfer coefficients for the inner and outer surfaces ($k_{l,in}$ and $k_{l,ex}$) of the condenser are of the same order. It was noted also that the vapor—gas front oscillates about some position. The amplitude of the oscillations increases with increase of the heat flux supplied. The cause of this phenomenon probably lies in the special features of boiling in the wick of the heat-pipe heat-supply zone.

In studying the starting characteristics we determined the time for the test system to reach a steady-state from a state where the whole structure has the temperature equal to that of the coolant liquid. Figure 3a shows the starting characteristics of the system $t_v = f(\tau)$ (curves 1), and also the variation of the transport-zone temperature with time $t_{tr} = f(\tau)$ (curves 2). A characteristic feature of the function t_{tr} is the presence of the section AB, where there is an appreciable variation in the quantity $dt_{tr}/d\tau$. An increase in the curvature of the curve in this section is apparently due to intensified heating of the transport zone, associated with movement of the vapor plug. Figure 3b shows a generalized dependence of the time for the heat pipe to reach conditions τ_v, as a function of the heat flux supplied, the coolant temperature, and the nominal evaporator temperature $t_{evap,n}$. The time τ_v depends appreciably on t_o, $t_{evap,n}$ in the low flux, 30-250-W region. This influence diminishes with increase of the initial value of Q.

Thus, the following conclusions may be drawn: The heat pipe has quite a good control capability at various temperature levels, determined by the NCG gas, over a wide variation of the power supplied. For example, for a nominal evaporator wall temperature of $t_{evap,n}$ = 112°C, with simultaneous variation of the power supplied from 120 to 900 W and coolant temperature t_o from 10 to 65°C, a given wall temperature can be stabilized in the range $0.5\Delta t_{evap}$ = ±6°C.

Going to a lower level of temperature $t_{evap,n}$ reduces the maximum transmitted power. The range of variation of vapor temperature Δt_v when the front moved from the beginning of the condenser to the end is somewhat reduced. At a level $t_{evap,n}$ = 56.5°C a variation of evaporator wall temperature of ±4°C was obtained with variations of Q = 70-400 W and t_o = 10-23°C.

The vapor—NCG transition zone has a substantial length, because the heat-pipe body is made of material with good heat insulation [6], and, therefore, part of the condenser is not used for vapor condensation.

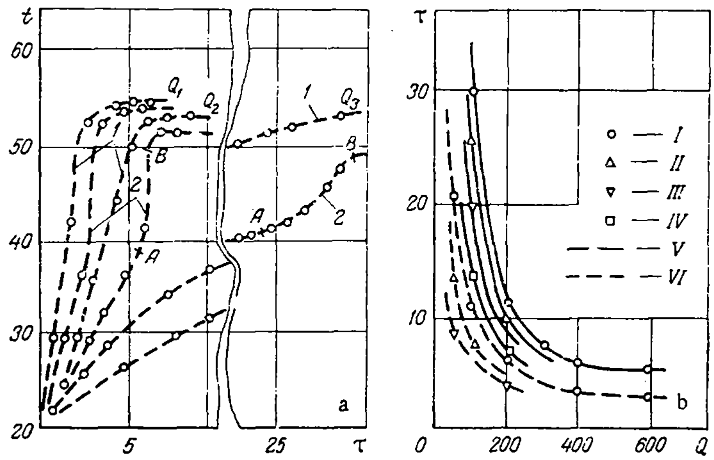

Fig. 3. Transient characteristics of the heat pipe: a) vapor temperature variation in the transport zone (Q_1 = 200 W, Q_2 = 100 W, Q_3 = 30 W, $t_{evap,n}$ = 56°C, t_0 = 23°C); 1) vapor temperature; 2) transport-zone temperature; b) starting characteristics $\tau_v = f(Q, t_0, t_{evap,n})$ [I) t_0 = 10°C; II) 23; III) 40; IV) 65; V) $t_{evap,n}$ = 112°C; VI) 56°C]; τ_v is in min; $t_{evap,n}$ and t_0 are in °C; and Q is in W.

In the low heat-flux region Q = 30-250 W, the time for the heat pipe to reach steady conditions depends appreciably on the coolant temperature and on the nominal temperature.

The method of computation, based on the planar front model, gives good agreement with the experimental data.

NOTATION

Q, transmitted power; p, t_v, p_0, t_0, p_t, t_t, partial pressure and temperature in the evaporator zone, the nonactive part of the condenser, and the tank, respectively; t_{evap}, t_c, temperature of the body wall in the evaporator zone and in the vapor condensation zone; t_0, average temperature of the coolant liquid; $t_{evap,n}$, nominal evaporator temperature; Δt_v, Δt_{evap}, variation in the vapor and evaporator temperatures; σ, sensitivity of the heat pipe to change in heat flux; m_g, M, the mass and molecular weight of the noncondensable gas; $k_{l,ex}$, $k_{l,in}$, linear heat transmission coefficients for the outer and inner surfaces of the condensation zone; k_{evap}, heat-transfer coefficient in the evaporator zone; k_l, linear coefficient of heat transfer from the vapor to the coolant liquid; λ_{well}, λ_{tr}, λ_c, thermal conductivity of the body wall material in the evaporator zone, the transport zone, and the condenser; F_{tr}, F_c, F_v, cross-sectional area of the body in the transport zone, condensation zone, and vapor channel; Π, α_{ex}, perimeter and coefficient of heat transfer from the outer surface in the condensation zone; L_c, L_g, length of the condenser and the vapor plug; δ_{well}, D_{av}, λ_{eff}, thickness and average diameter of the pores, and effective thermal conductivity of the wick; V_t, V_c, volumes of the tank and condenser; R_μ, universal gas constant.

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