Morphology of temperature field of heat pipe in conditions of photon

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Abstract. Within the framework of computer experiment with the use of finite element method the technique of visualization of nonstationary temperature field morphology in the design of heat pipe having a symmetrical structure in the presence of localized heat flow source of a certain shape is developed. The features of the structure of isothermal surfaces and lines are visualized and investigated for the uniform starting mode model in the range of power values of the heat flux source induced by photon heating P = 0 ÷ 5W on both sides of the center of the pipe.

1. Introduction
Photon heating, as a physical phenomenon, represents certain scientific and practical interest among researchers engaged in the development of new diagnostic methods in various industries and medicine, as well as technologies that protect objects, for example, from exposure to laser radiation [1–4]. The known technological advantages of photon heating, including varying the size and shape of the heating zone, obtaining high energy density (power), the demanded monochromatic radiation and the simple implementation of the “flying spot” principle have provided it with wide application in the processes of modifying materials and non-destructive thermal control of layered, composite, cellular and other products [1, 5–9]. Highest rates of informativeness and reliability of the new method of quality control of containment with localized phase transitions, including heat pipes (HP), through the use of non-contact optical methods of heat supply and temperature measurement [10, 11].

Currently, HPs are products of mass production and they are widely used in special cooling systems for various electronic means, including computer technology [12]. In this regard, high requirements are imposed on their reliability, which stimulate the development of methods for studying thermal modes of HP. However, in the literature devoted to research in the field of design, testing and application of HPs, there is no information on the morphology of temperature fields during photon heating of HPs [13].

Therefore, this paper presents the results of a study aimed at creating a method of computerized visualization of the morphology of a nonstationary thermal field of TTs under the conditions of using photonic heating of HPs with a symmetric structure.

2. Research methods
When developing the technique, was chosen the flat aluminum HP with capillary-porous structure of which was implemented as a net fuze of aluminum wire as the basic design. Ammonia served as a coolant. The dimensions of the geometric model of HP were, respectively (figure 1): length L = 200mm; width B = 20 mm; thickness H = 4 mm, and the shell wall thickness was assumed to be 0.7 mm [12].
Figure 1b shows the structure of the thermal model of the HP. The central location of a flat circular heat source (HS) (2), induced by external luminous flux, allows using known heat transfer mechanisms to create reversible heat fluxes \( Q \), which move in the material of the body, fuze, steam channel and cause a change in time of the corresponding temperatures \( T_k, T_f, T_p \). Thus, a diagnostic signal is formed within the surface of the HP casing in the form of thermal contrasts, carrying information about morphological changes in temperature fields.

When stating the problem of analyzing unsteady heat transfer in the axial section of the “surface heat dissipation – heat pipe” system, the following basic assumptions were made:

1) The thermophysical characteristics of the medium in the area of the fuze HP are calculated as effective taking into account the volume fractions of each component. Such approach is widely used in solving problems in porous media and has performed well in calculating the thermal conductivity of fairly complex systems. At the same time, the effective thermal conductivity coefficient \( \lambda_E \) remains constant both for the condensation zone and for the TT evaporation zone [12, 14]:

\[
\lambda_E = \frac{\lambda_L + \lambda_S - (1-\varepsilon)(\lambda_L - \lambda_S)}{\lambda_L + \lambda_S + (1-\varepsilon)(\lambda_L - \lambda_S)}
\]  

(1)

where \( \lambda_L \) and \( \lambda_S \) - coefficient of thermal conductivity of the liquid and the fuze skeleton; \( \varepsilon \) - fuze porosity (for aluminum mesh is 0.6 [14]).

The thermophysical characteristics of the materials and elements of the heat pipe are given in table 1.

| System element | Material or substance | Density, \( \rho \), kg/m\(^3\) | Specific heat, \( C \), J/(kg·K) | Thermal conductivity, \( \lambda \), W/(m·K) |
|----------------|-----------------------|--------------------------|--------------------------|--------------------------|
| Case           | Aluminum              | 2689                     | 951                      | 237,5                    |
| Fuze Capillary structure | Aluminum            | 1691,5                  | 2484                     | 60,25                    |
| Fuze Refrigerant | Ammonia              | 0,8                      | 2160                     | 0,022                    |
| Refrigerant    | Ammonia              |                          |                          |                          |

2) We consider that the processes of heat release in the condensation zone do not directly affect the formation of the temperature field in the evaporation zone. In this case, the dynamics of heat transfer in the HP is determined by the process of thermal conductivity in the wall of the body and the fuze, as well as their heat capacity. Moreover, the coolant vapor (ammonia) is an ideal thermal bond, the heat capacity of which is close to zero.

3) HP is considered as a system of bodies interconnected by an ideal thermal bond, which is in heat exchange with the environment and instantaneous regularization of the thermal regime. Then, to calculate the non-stationary thermal mode of the HP, the known dependence of the rate of temperature change for the regular mode of the system of bodies can be used [15]:

\[
m = \psi_c S_C \alpha_C \frac{C_{HP}}{C_H}
\]  

(2)

where \( \psi_c \) - coefficient of uneven temperature of the heat removal zone; \( C_{HP} \) - heat capacity of the heat pipe, j / c; \( S_C \) - area of the condensation zone m\(^2\); \( \alpha_c \) - heat transfer coefficient of the condensation zone W / (m · K). The accepted assumptions do not impose fundamental restrictions on the generality of the problem statement and reflect a rather real operating mode of the heat pipe.
Figure 1. Geometric (a) and thermal (section along the axis) (b) HP type models with a round HS. 1 – Case; 2 – HS.

It should be noted that free (natural) convection at normal pressure and thermal radiation [14, 15] were chosen as the main mechanisms for dissipating thermal energy from the surface of a horizontally oriented HP body to an air (gas) medium with temperature $T_c$ in the thermal model. When constructing a mathematical model, the non-stationary heat conduction problem in a plane-parallel formulation was considered. For this case, using the original heat conduction equation (3) [15]:

$$C \rho \frac{\partial T}{\partial t} = \lambda x \frac{\partial^2 T}{\partial x^2} + \lambda y \frac{\partial^2 T}{\partial y^2} + \lambda z \frac{\partial^2 T}{\partial z^2} + q_0.$$  

(3)

The system of equations (4) was compiled describing the heat transfer in the fuze, the steam channel and the body of the HP [13,15]:

$$C_{1\rho_1} \frac{\partial T_1}{\partial t} = \lambda_1 \left( \frac{\partial^2 T_1}{\partial x^2} + \frac{\partial^2 T_1}{\partial y^2} \right); \quad 0 < x < (L - \delta_{ct} - \delta_f); \quad (\delta_{ct} + \delta_f) < y < (H - \delta_{ct} - \delta_f)$$

$$C_{2\rho_2} \frac{\partial T_2}{\partial t} = \lambda_2 \left( \frac{\partial^2 T_2}{\partial x^2} + \frac{\partial^2 T_2}{\partial y^2} \right); \quad 0 < x < (L - \delta_{ct}); \quad (\delta_{ct}) < y < (H - \delta_{ct})$$

$$C_{3\rho_3} \frac{\partial T_3}{\partial t} = \lambda_3 \left( \frac{\partial^2 T_3}{\partial x^2} + \frac{\partial^2 T_3}{\partial y^2} \right); \quad 0 < x < L; \quad 0 < y < \delta_{ct}$$

(4)

where $C$ – isobaric heat capacity, J/(kg · K); $T$ – temperature, K; $x, y$ – coordinates, m; $\lambda$ – coefficient of thermal conductivity W/(m · K); $\rho$ – density, kg/m$^3$; $t$ – time, s; $L$ – heat pipe length, m; $H$ – height of the heat pipe, m; $\delta_{wp}$ – wall thickness of the heat pipe, m; $\delta_f$ – fuze thickness, m. The indices in (4) correspond to the following bodies: 1 – steam channel; 2 – fuze; 3 – heat pipe body.
For the developed thermal model under conditions of uniform start-up of HP’s, the following initial and boundary conditions were used [14, 15]:

1) at the initial moment of time for the edges of all bodies included in the model, the temperature was fixed:
\[ T_{i,0} = T_c = \text{const}; \quad (5) \]

2) for all HS edges, taking into account the surface isothermality, a condition of the first kind was set:
\[ T = T_s; \quad (6) \]

3) for the inner ribs forming the vapor channel, the fuze and the body, the boundary conditions of the second and fourth kind were set:
\[ \frac{\partial T_i}{\partial n} = \psi, \quad (7) \]
\[ -\lambda_j \frac{\partial T_i}{\partial y} = -\lambda_2 \frac{\partial T_3}{\partial y}; \quad (8) \]

4) on the external fins (casing) of the HP model, a condition of the third kind was defined, which describes both convective and radiant heat exchange with the environment:
\[ \lambda_p \frac{dT}{dn} = -\alpha_c(T - T_c) - \beta(T^4 - T_c^4) \quad (9) \]

where \( \beta \) – a value equal to the product of the Stefan-Boltzmann constant \( (\sigma_0 = 5.7 \times 10^{-8} \text{ W} / \text{m}^2\text{K}^4) \) and the coefficient of radiation of the surface of the probe material; \( \alpha_c \) – convection heat transfer coefficient. The numerical values of the coefficient \( \alpha_c \) were obtained from preliminary calculations using the well-known similarity method [15].

The removal of heat flux through conductive connections of HP fasteners was not taken into account. The system of equations (4–9) was solved by a numerical method (finite element method) on a PC. Using the software "ANSYS 13.0" [15], a series of calculations was performed, which made it possible to estimate the degree of influence of the power of the induced HS of round shape on the field characteristics of the thermal model of the HP. The grid step was chosen optimal, based on the available resources of the PC, and was 1.4 mm for the whole model and 0.5 mm in the field of HS. Studies were performed at an ambient temperature of \( T_C = 300 K \), the duration of the HS was 120 s, and its power changed in the range of values \( P = 0 \div 5 \text{ W} \).

3. Research results and discussion

It is known that most HPs are designed for isothermal operation at a certain temperature and transmitted power [12–14]. At the same time, in practical use, virtually every HP goes through a starting mode and its duration, as well as the nature of the temperature field change depends on the design features of the pipe, on the methods of heat supply and cooling, and on the time variation of the input power.

The research results showed that when using the model of uniform starting mode HP, the nature of the change in the maximum surface temperature in the interval of time values \( 0 \div 120 \text{s} \) in the area of localization of HS is close to linear (figure 2). This character of dependences is preserved with an increase in the values of the power induced by radiation on the surface of the HS HP. This, in turn, made it possible, during a computer experiment, to study the conditions for removing the HP to the operating mode, the nature of the temperature distribution along the pipe, the factors that impede the launch, and others. Thus, using the method of field characteristics [16], the morphology features of the nonstationary temperature field of HPs with an HS of circular shape were visualized. Typical examples of simulation results are shown in figure 3.

Their analysis showed that in the thermogram mode (figure 3a) for a fixed point in time, the morphology of the HP temperature field on either side of the HS is represented by a system of symmetric isothermal surfaces (zones). At the same time, it should be noted that only near the HS, the shape of the zone quite well repeats the shape of the round HS.
Figure 2. The change in the maximum surface temperature of ammonia HP in time. Surface HS power: 1 – 1W; 2 – 2W; 3 – 5W.

Figure 3. Morphology of the temperature field in ammonia HP. Input data: HS power P = 5 W; a – thermogram mode; b, c – contour mode; g – mode selected circuit (along the axis of the HP).
As they move away from HS, the shapes of the zones acquire an almost equally curved convex profile, which is formed due to the external heat exchange of the HP surface [13, 14]. Additional calculations showed that with increasing HS power, the length of the zones only increased, and their morphological features did not undergo significant changes. It is quite clearly seen that in the isoline mode (figure 3b), the isotherms continuously repeat the boundaries of the corresponding zones. In the course of the research, the effect of the front lag of the corresponding isotherm was found on the opposite to the HS surface of the HP (figure 3c). This effect, in our opinion, is due to the peculiarities of the transport of heat flow from the HS in the transverse direction of the HP [12, 14].

The nature of the temperature change along the length of the HP (along the selected contour) reflects quite well the graph in figure 3d. In the absence of defects specially introduced into the HP model, it has strict symmetry about the center of the pipe.

4. Conclusion
According to the computer experiment, in the model of a uniform start-up of ammonia HP, the shape of isothermal lines near the induced HS radiation reproduces the shape of this source quite well, and the effect of the front of the corresponding isotherm on the opposite to HS surface of the HP occurs.

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