Journal of Engineering Physics and Thermophysics, Vol. 67, No. 3-4, 1994

ANALYSIS OF A RADIATIVE HEAT EXCHANGER FOR SYSTEMS FOR THERMAL CONTROL OF SPACE VEHICLES

L. L. Vasil'ev, L. E. Kanonchik, and V. A. Babenko

UDC 536.423

Starting from the solution of a two-dimensional heat conduction problem, a mathematical model of a heat pipe-based radiative heat exchanger is developed. Good agreement between the predicted and experimental results is obtained. The effect of operational and structural parameters on the characteristics of the radiative heat exchanger is analyzed.

Systems for thermal regulation of space vehicles involve radiative heat exchangers (RHE) to which heat is supplied by a single- or two-phase heat transfer agent and from which it is carried away to space by radiation. It is important for practical applications of such devices to be able to predict the basic characteristics of heat extraction and to select an optimum design for various situations. In [1] an experimental investigation undertaken to determine the optimum geometry of a radiator-emitter is described. In [2] the main problems and stages of mathematical simulation of an RHE and also of the sets of heat exchangers involved in thermal regulation systems are formulated. In [3, 4] the problems of the optimization of thermophysical parameters are solved. In [5] a numerical simulation of the relative characteristics of an RHE is carried out in order to optimize a device made in the form of a set of longitudinal fins of variable cross section. The calculations are based on a one-dimensional analysis of the heat transfer of an individual fin with allowance for radiative heat exchange.

In the present work, we developed a two-dimensional mathematical model of an RHE whose radiation fins are duplicated by heat pipes (HP). In the simulation we consider a computational element comprising one-eighth of one of the sections (Fig. 1). It is bounded by symmetry planes passing through the channel axis ($x = 0$), the HP axis ($z = 0$), and the middle of the distance between two adjacent HPs ($z = L$). The heat exchanger consists of a number of double elements or sections connected in series. A hot heat transfer agent moves through the channel in the direction of the z axis. A portion of this heat is dissipated by radiation from the outer surface of the channel $r = b$ into the surrounding space, and the remaining portion enters a cooling fin of half-thickness δ_j, which is duplicated by a HP of diameter D_p. Thereafter, the heat is released by radiation from both surfaces of the fin and from the HP surface. Between the channel wall and the HP a portion of the fin x_2 may be located whose dimensions are dictated by the thermal resistance of the contact; this portion exerts an effect on the working temperature.

In going from the real three-dimensional picture to the two-dimensional finite-difference model of an RHE in a steady state the following assumptions are made: 1) the mean temperature over the channel wall and fin cross sections coincides with the surface temperature, because the thermal conductivity of the metal is high; 2) in calculations the HP can be replaced by a fin band with equivalent heat transfer into the surrounding medium. The thermal conductivity of this band is

$$\lambda_p = \pi (L_e + L_c)/\left\{ 2\delta_f \, [1/(L_e\alpha_e) + 1/(L_c\alpha_c)] \right\}, \tag{1}$$

where the heat transfer coefficients α_e and α_c are calculated by the technique described in [6], and the lengths of the evaporator L_e and the condenser L_c are refined in the course of the calculations, since they are not predetermined in the present problem.

Academic Scientific Complex "The A. V. Luikov Heat and Mass Transfer Institute of the Academy of Sciences of Belarus," Minsk. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 67, Nos. 3-4, pp. 261-265, September-October, 1994. Original article submitted June 22, 1993.

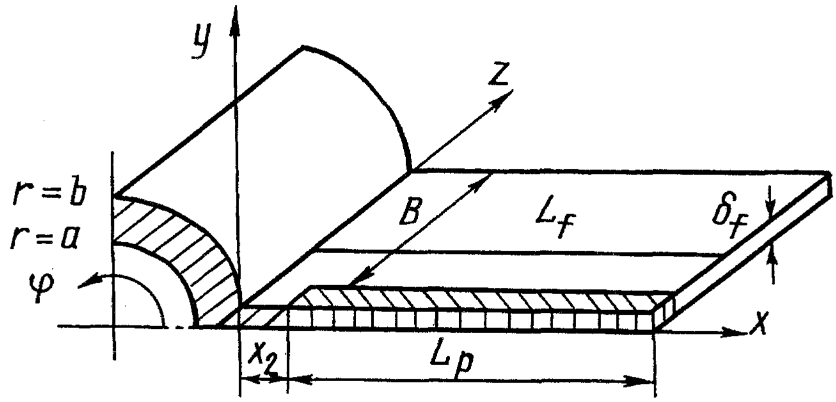


Fig. 1. Schematic of a computational element of a heat pipe-based radiative heat exchanger.

The core of the mathematical model is a two-dimensional heat conduction problem written down for the channel wall in a cylindrical system of coordinates:

$$\lambda_t \left(\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial t}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 t}{\partial \varphi^2} + \frac{\partial^2 t}{\partial z^2} \right) = 0 \tag{2}$$

with the boundary conditions

$$\lambda_t \frac{\partial t}{\partial r} \Big|_{r=b} = - \sigma \varepsilon \left(t_b^4 - t_0^4 \right), \tag{3}$$

$$\lambda_t \frac{\partial t}{\partial r} \Big|_{r=a} = \alpha_a \left(t_a - \bar{t} \right). \tag{4}$$

For the cooling fin the same equation in a Cartesian coordinate system has the form

$$\lambda_f \left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} \right) = 0, \tag{5}$$

where the boundary conditions are

$$\lambda_f \frac{\partial t}{\partial y} \Big|_{y=\delta} = - \sigma \varepsilon \left(t_\delta^4 - t_0^4 \right), \tag{6}$$

$$\lambda_f \frac{\partial t}{\partial y} \Big|_{y=0} = 0. \tag{7}$$

We linearize the nonlinear terms under conditions of radiative heat transfer on the outer boundaries $r = \delta$ and $y = b$ under the assumption that the density of the heat flux from the surface $q = \sigma\varepsilon(t^4 - t_0^4)$ is proportional to the first power of the local temperature difference, $q = \alpha(t - t_0)$, where $\alpha = \sigma\varepsilon(t^{*2} + t_0^2))(t^* + t_0)$ and the values of t^* are taken from the previous iteration.

Next, we integrate Eqs. (2) and (5) with allowance for the condition of conjugation with respect to the temperature and the heat flux at a point on the boundary of the fin and the channel:

$$\lambda_t \ln \frac{b}{a} \frac{\partial T}{\partial \varphi} \Big|_{\varphi=0} = - \lambda_f \delta \frac{\partial T}{\partial x} \Big|_{x=0}; \quad T|_{\varphi=0} = T|_{x=0}, \tag{8}$$

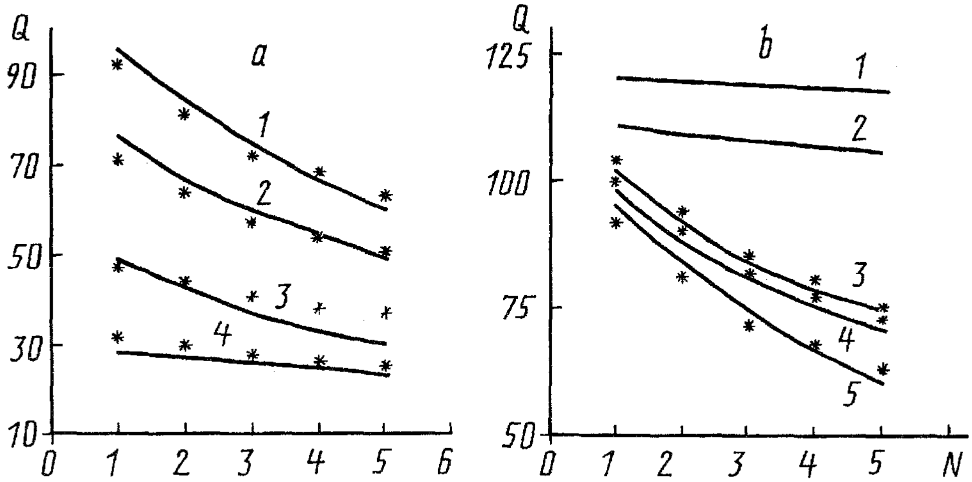


Fig. 2. Profiles of heat release along the length of a radiative heat exchanger in a steady-state regime (points, experiment; lines, prediction): a) for different gas temperatures: 1) 489; 2) 445; 3) 364; 4) 322 K and $G = 0.004$ kg/sec; b) for different gas flow rates: 1) 0.04; 2) 0.02; 3) 0.006; 4) 0.005; 5) 0.004 kg/sec and $T_g = 489$ K. Q, W.

and also introduce the new coordinate x' from the relation $dx' = -\delta d\varphi$, $dx' = dx$. As a result, we rewrite these equations for the average temperature T over the cross section in a unified form, as for a two-layer plate with discontinuous heat conduction coefficients and source terms:

$$\frac{\partial}{\partial x'}\left(\lambda_t \ln\frac{b}{a}\right)\frac{\partial T}{\partial x'} + \frac{\partial}{\partial z}\left(\lambda_t \frac{b^2 - a^2}{2\delta^2}\frac{\partial T}{\partial z}\right) = \frac{\alpha_b b + \alpha_a a}{\delta^2} T - \frac{\alpha_a a \bar{t}}{\delta^2} - \frac{\alpha_b b t_0}{\delta^2}, \tag{9}$$

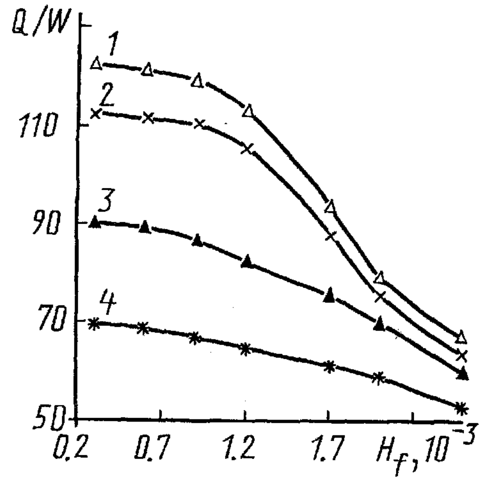
$$\frac{\partial}{\partial x'}\left(\lambda_f \frac{\partial T}{\partial x'}\right) + \frac{\partial}{\partial z}\left(\lambda_f \frac{\partial T}{\partial z}\right) = \frac{\alpha_\delta T}{\delta} - \frac{\alpha_\delta t_0}{\delta}. \tag{10}$$

The problem was solved by a modified Patankar method [7] in which the difference equations for the temperature at the nodes of a 21 × 21 grid were solved by the Buleev incomplete factorization method [8].

To confirm the validity of the mathematical model of an RHE, the results of a calculation were compared with experimental data for the steady state regime. Two series of experiments were carried out on a low-temperature vacuum setup [9] under conditions close to those in an orbit ($T_0 = 103$ K, $P_0 = 0.013$ Pa). In one of them we varied the temperature (322, 364, 384, 445, and 489 K) and in the other the flow rate (0.004, 0.005, 0.006 kg/sec) of the gas entering the channel.

In the calculations we specified the basic initial data corresponding to those used in the experiment of [9]. The radius of the channel was 0.011 m, and the channel wall thickness was 0.002 m. The HP had the following parameters: length 0.3 m, diameter 0.016 m, number of grooves 28, half-width of a groove 0.0025 m, fin thickness 0.0017 m, and half-width of a fin 0.052 m. To calculate a heat exchanger having N sections, it is necessary to use the developed procedure successively N times each time changing the gas temperature at the inlet and assuming it equal to the gas temperature at the exit from the previous section. We considered an HP-based heat exchanger (aluminum and Re-113) consisting of five successive sections of mass 0.445 kg, i.e., five combinations of "left-side" element and "right-side" element.

Some results of calculations and experiments corresponding to the steady-state operating regime of the RHE are presented in Fig. 2. It is seen that as the gas temperature increases, the absolute value and the nonuniformity of heat release along the heat exchanger length increase (Fig. 2a). An increase in the gas temperature from 322 to 489 K ensures an increase in the heat release from a five-sectioned RHE from 150 to 500 W. The effect of the gas flow rate amounts to an improvement in the heat exchange with the wall and an increase in the degree



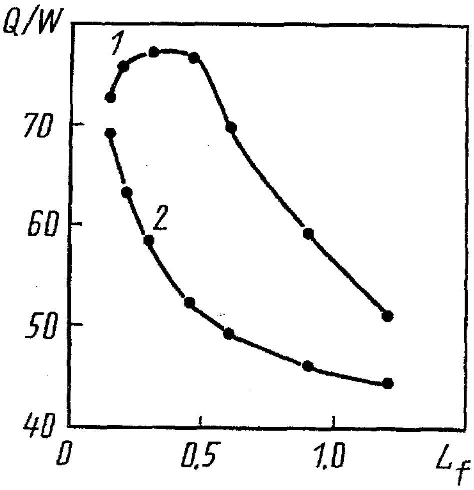


Fig. 3. Effect of the fin thickness on the mass compactness of a single section of an RHE for different widths of the section: 1) B = 0.092; 2) 0.072; 3) 0.052; 4) 0.032 m; L_f = 0.3 m, R = 0.009 m, H_t = 0.002 m. Q/W, W/kg; H_f, m.

Fig. 4. Effect of the height of a fin on the mass compactness of a single section of an RHE: 1) a fin duplicated by an HP; 2) a solid aluminum fin; H_f = 0.0017 m, R = 0.009 m, H_t = 0.002 m, B = 0.052 m. L_f, m.

of isothermicity of the channel along its length (Fig. 2b). An increase in the mass velocity of the gas leads to an increase in the heat release. For example, at a flow rate of the gas of 0.004 kg/sec its temperature in the channel decreased from 489 to 410 K. Furthermore, the first section released a quantity of heat almost 1.5 times larger than the last one. At a gas flow rate of 0.04 kg/sec the drop in temperature was 14 K and the amount of heat released changed from 120 to 118 W. The characteristics of an RHE computed at a flow rate of 0.04 kg/sec are essentially limiting for the given design since the temperature of the inner wall of the channel differs but little from the temperature of the gas itself.

From a comparison of the calculated data and the experimental results in Fig. 2 it follows that they agree rather well (the maximum deviation is 18%). On the whole, it can be said that the mathematical model of an RHE gives an adequate description of the steady-state mode of operation.

To elucidate the influence of the structural parameters on the characteristics of the RHE, we performed numerical experiments by the finite-difference model developed. In the calculations a number of quantities were assumed to be fixed, namely, the refrigerator temperature 103 K, the inlet gas temperature 320 K, the gas flow rate 0.04 kg/sec, and the HP diameter 0.016 m. The results for the dependence of the heat-transmitting characteristics of a single section on the wall thickness and the diameter of the central channel are given in [10].

Figure 3 illustrates the effect of the fin thickness $H_f = 2\delta_f$ and its half-width B on the mass compactness (the ratio of heat flux to mass). For values of H_f exceeding 0.0012 m, the mass compactness begins to decrease sharply because of the appreciable increase in the mass. The calculations performed as well as technological considerations permit one to recommend a fin thickness from 0.001 to 0.0017 m.

The mass compactness also decreases with a decrease in the fin width. The decrease in the released heat flux because of the decrease in the heat evolving surface turned out to be more important than the decrease in the mass. We note that there are limitations on the HP fin area that follow from the balance of heat fluxes: that supplied from the side of the inner surface of the pipe and that removed from the fin surface by radiation. Thus, for a gas temperature equal to 490 K the heat transfer limit of a channel HP amounts to 130 W and the half-width of a fin should not exceed 0.052 m.

The use of a heat pipe in a fin permits one to improve markedly the characteristics of an RHE (Fig. 4). The heat exchanger fin duplicated by an HP emits more heat for the same mass than a solid aluminum fin. The proposed design of the heat exchanger is more efficient (up to 1.5) compared to plane emitting fins. Moreover, the mass compactness of a fin with an HP has a maximum located near an HP length of 0.3 m. The presence of the extremum in the function of two variables that change in one direction is explained by the difference in the rates of increase of the heat flux and the mass when short and long tubes are used.

In conclusion it should be noted that the proposed mathematical model, which is implemented in the form of a block of programs for a personal computer, permits one to determine the optimum geometry of the design with an accuracy sufficient for practical purposes and to evaluate the influence of various factors on the efficiency of operation of an RHE. The results of experimental and numerical investigations have confirmed the potentialities of using a heat pipe-based RHE in systems for thermal regulation of space vehicles.

NOTATION

B, half-width of a fin; D, diameter; E, efficiency; G, mass flow rate; $H_f = 2\delta_f$; $H_t = b - a$; L, length; M, mass; N, number of a section; Q, heat flux; R, radius; t, local temperature; T, average temperature over a cross section; α, heat transfer coefficient; ε, emissivity; δ_f, half-thickness of a fin; λ, thermal conductivity; σ, Stefan-Boltzmann constant; τ, time. Subscripts: e, evaporator; c, condenser; 0, heat sink; p, heat pipe; f, fin; t, channel.

REFERENCES

1. I. Coggi, Heat Pipe Thermal Control of Spacecraft Batteries, California (1979).
2. A. A. Nikonov, G. A. Gorbenko, and V. N. Blinkov, Heat-Exchanging Loops with a Two-Phase Heat Carrier for Systems for Thermal Control of Space Vehicles [in Russian], Moscow (1991).
3. V. V. Malozemov and N. S. Kudryavtsev, Optimization of Systems for Thermal Control of Space Vehicles [in Russian], Moscow (1988).
4. V. V. Malozemov, V. F. Rozhnov, and V. N. Pravetskii, Life-Support Systems for the Crews of Aircraft [in Russian], Moscow (1986).
5. H. V. Chang, AIAA Paper No. 1788 (1984).
6. L. L. Vasil'ev, L. P. Grakovich, and D. K. Khrustalev, in: Proc. 4th Int. Heat Pipe Conference, London (1981).
7. S. Patankar, Numerical Methods for Solving Heat Transfer Problems and Problems of Fluid Dynamics [Russian translation], Moscow (1984).
8. G. I. Marchuk, Methods of Computational Mathematics [in Russian], Moscow (1977).
9. L. L. Vasil'ev and L. E. Kanonchik, Inzh.-Fiz. Zh., 67, Nos. 1-2, 93-97 (1994).
10. L. L. Vasil'ev, L. E. Kanonchik, D. K. Khrustalev, and V. A. Babenko, in: Proc. 4th European Symposium on Space Environmental Control Systems (Florence, Italy, 1991), pp. 1-6 (1991).