OPERATION OF CERTAIN CONSTRUCTIONS OF A POROUS EVAPORATOR OF A COAXIAL HEAT-TRANSFER DEVICE

G. P. Nikolaev, V. A. Shamanaev,
V. E. Prokudenko, L. N. Negodyaev,
and É. V. Mazurov

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Presented here are the results of an investigation of the operation of a metallic-ceramic evaporator working with water in a variant of a coaxial heat-transfer device.

The use of numerous heat-stressed elements in modern devices and equipment and the need for removal of heat from these or its deconcentration in near-isothermal conditions have stimulated the development and use of efficient heat-transfer devices called heat pipes and vapor chambers [1-3].

Devices with a cylindrical evaporator and condenser inserted one into another are usually called coaxial. They are of interest as deconcentrators of heat flux [3] for cooling cylindrical heat-releasing objects with heat transfer into the intermediate heat-transfer agent or to the surrounding medium.

The processes of evaporation of the working liquid [4] play a significant role in the operation of the heat pipe; these processes have not been adequately studied. In the present work we give the results of tests of individual constructions of a metallic-ceramic porous evaporator used in a variant of the coaxial heat-transfer device (vapor chamber) for removal of heat flux Q not less than 400 W from the inner cylindrical heater of diameter 24 mm and length 80 mm with three-dimensional heat release. The use of a wick-type material is necessary for continuous maintenance of the level of the heat-transfer agent in the evaporating segment, since the heating element can have an arbitrary position with respect to the gravitational field.

The investigated evaporator (evaporation head) is made by the classical method of powder metallurgy from nickel with particles of dispersity of 1-20 μ in the form of cylinder ($d=50$ mm, $l=70$ mm). The porosity of the sample, determined by the weight–volume method, comprised 69%. The predominant diameter of the pores was ~100 μ and the presence of "puddles" of heat-transfer agent (distilled water) permitted to completely soak the evaporation head by the working liquid for all orientations.

The tests of different variants of the evaporation head were conducted in a cylindrical chamber (Fig. 1) of diameter 60 mm and length 80 mm cooled by a water-jacket thermostat. The chamber was made demountable to make possible a change in the evaporation head. The inner frame 14 on which the metallic-ceramic head 15 is tightly fitted is intended for introducing the heating element 13. The outer frame 12 has a cooling jacket 11.

The heat-transfer agent was poured in after pumping the working volume to a pressure of 10^{-1} mm Hg. The amount of the liquid was chosen depending on the constructional pecularities of the evaporator (70-120 cm^3). For all variants the excess heat-transfer agent (puddle) in the chamber remains identical ~40 cm^3 corresponding to an imbedding of the wick of the evaporator at ~5 mm for horizontal and vertical positions of the chamber. For soaking the first solid sample 83 cm^3 of heat-transfer agent was additionally required (total filling 123 cm^3). The presence of vapor channels in the subsequent constructions reduced the soaking volume.

The temperature was measured by 10 copper–Constantan thermocouples with electrodes of 1.2 mm diameter. Thermocouples 1-4, 9 (corresponding to temperatures T_1-T_4, T_9) were placed inside the porous material of the evaporator. Thermocouples 5 and 6 control the temperature T_5 and T_6 of the vapor space inside the chamber. Thermocouple 8 measured the temperature T_8 inside the heater (according to the technological conditions it should not exceed 300°C, which also limited the output power). The readings of the differential thermocouples 7 (T_7) and the flow rate of the liquid through the cooling jacket offer the possibility of controlling the removed heat flux. The amount of heat supplied to the device was determined from the readings of a VK7-10A/1 voltmeter and a M-1108 ammeter.

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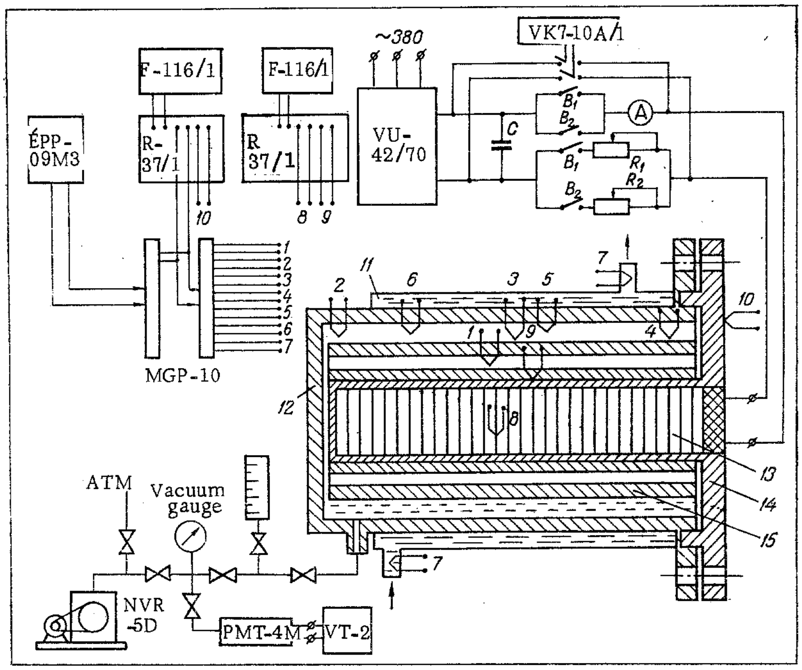

Fig. 1. A schematic diagram of the experimental setup: 1–10) copper–Constantan thermocouples; 11) cooling jacket; 12) external frame of the chamber; 13) cooled heater; 14) internal body of the chamber; 15) evaporation head; MGP-10) small switch; PMT-4M) manometric thermocouple transducer; VT-2) vacuum-gauge thermocouple; F-116/1) photocompensated micromultimeter; R-37/1) potentiometer; NVR-5D) forevacuum pump; VU-42/70) rectifier; VK7-10A/1) numerical voltmeter; A, M-1108-type ammeter; R_1 and R_2) resistance variables, B_1 and B_2) switches; C) capacitor bank; ÉPP-09M3) electronic automatic potentiometer.

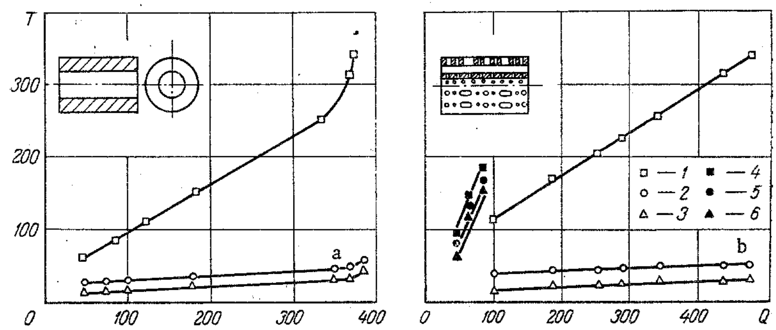

Fig. 2. Dependence of the evaporator temperature increase on the supplied heat flux: a) variant 1, cooling-water temperature 20°C, horizontal orientation, filling volume 123 cm^2; b) variant 5, cooling water temperature 20°C, horizontal orientation; filling volume 75 cm^3; 1, 2, 3) temperature values inside the heater T_8, on the heater surface T_9 and on the evaporation-head surface T_3, respectively; 4, 5, 6) the same temperatures in the experiments without heat-transfer agent; T, °C; Q, W.

TABLE 1. Results of an Investigation of the Operation of Various Constructions of Evaporation Heads

Evaporation variant	$Q_{300°}$, W	T_8, °C	T_9, °C	ΔT, °C	$q_{300°}\times10^{-4}$, W/m²	$\alpha_{300°}\times10^{-4}$, W/m²·°K
1	365	275	67,0	6,1	6,91	0,79
2	360	290	48,5	4,5	6,82	0,90
3	400	280	51,0	4,8	7,58	1,26
4	390	280	44,0	3,0	7,39	2,12
5	395	280	46,0	3,5	7,48	

Two P-37/1 potentiometers were used for recording the emf of the thermocouples. Photocompensated microvolt–microammeters of type F-116/1 served as reference instruments. For controlling the transitional processes an ÉPP-09M3 automatic potentiometer was used. The accuracy of the temperature measurements was ±0.1°C and of the heat flux ±5 W.

In the experiments we determined the dependence of the local temperature of the evaporation head and the heater on the supplied heat flux $T = f(Q)$.

The first variant of the evaporator is a porous cylinder [$d = 50$ and $l = 70$ mm (Fig. 2a)]. During the investigation of the operation of the first variant (temperature of the cooling water 20°C, horizontal orientation, filling volume 123 cm³) the temperature at all the control points in the chamber increased linearly with the increase of the supplied power. At the heat-flux intensity of ~350 W the temperature inside the heater T_8 was equal to 275°C; at the surface of the heater the temperature was $T_9 = 67$°C and the radial drop along the wick ΔT was 6.1°C. The heat-transfer coefficient at a temperature of the heater equal to 300°C was $\alpha_{300°} = 0.79 \cdot 10^4$ W/m²·°K (Table 1). A further increase of the power supply led to a sharp increase of the temperature inside the heater (curve 1, Fig. 2a).

The visual observations in the experiments with air showed that the evaporation of the heat-transfer agent at small powers occurs mainly from the surface of the material of the wick and for heat flux of ~350 W a vapor sublayer is formed inside the evaporator near the heating surface. The vapor emerges from the position of the seam of the wick and the heater, which indicates the need for organizing vapor removal from the heating surface.

In the second, third, and fourth constructions of the evaporation head the dimensions, number, and arrangement of the vapor channels were changed. The second variant of the evaporator differs by the presence of 17 open axial vapor channels of 2.5 mm diameter, welded at a distance of 1.5 mm from the inner surface of the head. The axial channels direct the vapor flow to the end faces of the chamber. The number and diameter of the channel insure a sufficiently small resistance to the flow of the forming vapor.

In the third and the fourth variants 300 radial channels of 1.5 mm diameter and additionally 9 axial channels of 7 mm diameter were made.

The fifth variant of the evaporative head and radial windows of 7 mm diameter, directly connect the axial channels to the condensation surface (Fig. 2b). The transport area of the wick of the evaporator was sufficient for stable operation up to a power of 600 W, as shown by the experiments.

For a power supply of ~350 W the temperature at the surface of the heater T_9 (curve 2) changed to 46°C, the temperature drop over the wick decreased to $\Delta T = 3.5$°C. The supplied power at a temperature $T_8 = 300$°C was $Q_{300°} = 395$ W (Table 1). A sharp increase of the temperature at all measurement points was not observed up to a power of 600 W compared with the first variant of the evaporative head. The organization of the vapor flow in the sample and improvement of the vapor removal from the evaporation zone led to an increase of the heat-transfer coefficient changing from the first variant to the last. The maximum value of $\alpha_{300°} = 2.12 \cdot 10^4$ W/m²·°K is attained for the fifth evaporation head.

All the changes in the construction of the evaporator were made successively on a single porous sample. Each of the subsequent variants differ from the proceeding by a weight reduction. The weight of the last variant decreased by about 35% compared to the first.

Estimates of the heat removal due to thermal conductivity of the material of the chamber, obtained by dry (without heat-transfer agent) tests (curves 4, 5, 6) showed that these losses do not exceed 10% of the supplied power. Similar results are obtained from an estimate of the removed power from the readings of the differential thermocouples 7 (Fig. 1) and the flow rate of the liquid through the cooling jacket.

The change in the orientation of the evaporation head (horizontal and vertical positions) and also the increase in the temperature of the cooling water to 60°C had an insignificant effect on the amount of the removed power and the temperature within the heater.

NOTATION

d, *l*, diameter and length of the porous sample, respectively; T_1-T_{10}, temperatures, measured by the thermocouples 1-10 (Fig. 1); $\Delta T = T_9 - T_3$, radial temperature drop along the wick; Q, supplied power; $Q_{300°}$, $q_{300°}$, α_{300}, supplied power, supplied heat-flux density, heat-transfer coefficient at the temperature $T_8 = 300°C$, respectively.

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THREE-DIMENSIONAL RADIATIVE HEAT-TRANSFER PROBLEM WITH SHADING

V. F. Kravchenko and V. M. Yudin

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The problem of radiative heat transfer between diffuse gray surfaces bounding a closed volume of arbitrary configuration is discussed.

Often in calculations of the heating of airframe structures it is necessary to solve problems of radiative heat transfer between the surfaces of various structural elements forming the interior compartments of an aircraft. In many cases the entire bounding surface is nonconvex and has such a complex configuration as to present serious difficulties in applying the zonal method.

For situations in which one of the dimensions of such a bounded volume is much greater than all the rest, we have proposed [1] a method for solving the planar radiative heat-transfer problem with allowance for shading and have demonstrated the substantial influence of this factor on the distribution of q_{inc} over the surface of compartments of real structures. In the present study we elaborate the method of analysis of radiative heat transfer with allowance for shading in the three-dimensional case.

We consider the problem of radiative heat transfer between diffuse gray surfaces bounding a closed volume of arbitrary configuration. An open volume can be closed by the addition of a fictitious surface with $\varepsilon = 1$ and $T = (q_\infty/\sigma)^{1/4}$, where q_∞ is the dissipated heat flux from the surrounding medium.

We assume that the bounding surface comprises N plane faces having the shape of a convex rectangle. These are actually the kind of surfaces that occur in the majority of real problems, and any continuous surface can always be approximated with sufficient accuracy by a system of plane faces. The temperature and emissivity distributions over each face are variable.

The radiative heat transfer in such a region is described by a system of Fredholm integral equations of the second kind in the incident flux density:

$$q_{\text{inc}}(p_i) = \sum_{j=1}^{N} \int_{F_j} \{\sigma\varepsilon(p_j)\,T^4(p_j) + [1-\varepsilon(p_j)]\,q_{\text{inc}}(p_j)\}\,K(p_i,\ p_j)\,dF_j, \quad i = 1, 2, \ldots, N, \tag{1}$$

where $K(p_i, p_j)$ is a function of the angular coefficients:

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