FULL-SCALE TESTING OF HEAT PIPES FOR SPACE-VEHICLE THERMAL REGULATION SYSTEMS

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Results are presented of tests of heat pipes for space-vehicle thermal regulation systems with different capillary structures and heat carriers.

A full-scale experiment to check out the operability of an experimental heat pipe (HP) specimen to equilibrate the space-vehicle housing temperature, as conducted on the AES Interkosmos 11, is described in [1]. Tests of a HP with a capillary structure in the form of a set of cylindrical corrugated inserts laid along the inner surface of the housing (the corrugations were parallel to the housing axis) and with Freon-11 as heat carrier yielded affirmative results.

In order to obtain test data for the selection of an optimal HP construction (of the capillary structure, heat-carrier type) for space-vehicle thermal stabilization and for the practical confirmation of the HP fabrication and testing technology developed, a full-scale experiment was conducted on the AES Interkosmos 15 launched on 19 June 1976.

The method of performing the experiment was the following. The HP specimens to be tested were placed in a sealed gas-filled compartment of the satellite. Thermal fluxes of fixed intensity were delivered to their evaporator sections by using surface electric heaters, for a sufficient time to put the HP into the stationary operating mode. The heat fluxes supplied are transmitted to the condensation sections connected to a special radiator over transfer sections. The heat is taken off from the radiator by convective motion of the gas induced by a fan. During the experiment the HP housing temperature is controlled on the heat delivery and removal sections. A comparative estimate of the HP heat conductivity characteristics is made for different structures by the temperature drops between the evaporator and condenser sections for identical heat fluxes being transmitted.

There were three cylindrical HP 412 mm long with outer diameter 14 mm for each in the heat tube module fabricated for the full-scale tests (Fig. 1). The material of HP1 was AMg3, the heat carrier was Freon-11, and the capillary structure was in the form of rectangular axial channels on the housing inner surface. The channels were 0.6 mm deep, 0.5 mm wide, the width of the connectors (on the circumference of the projections) was 0.6 mm, and the diameter of the vapor channel was 11.3 mm. The housing material and the capillary structure of the HP3 was the same, the heat carrier was synthetic liquid ammonia according to GOST 6221-70.

The HP1 and HP3 housings with the capillary structures were fabricated by the core method. The HP2 housing was fabricated from a 14 × 1.5 AMg3 tube on whose inner surface were M12 × 0.5 cl. 3 threads. The HP2 is proved with a central artery (5-mm outer and 3-mm diameter) fabricated from a fabric mesh No. 1 Kh18N10T ChMTU-4-7-68. The heat carrier is ammonia.

The endface panels and loading units were welded to the housing according to GOST 14806-69-VI-Rn-3.

After filling the heat pipe with the heat carrier, the hole of the loading unit was covered with a fluoroplast packing and a special cover and was filled with epoxy resin ED-6 and hardener GOST 10587-72 in a 10 : 1 weight ratio.

The condenser sections of the HP were installed in the 62-mm-long holes 14A, of the

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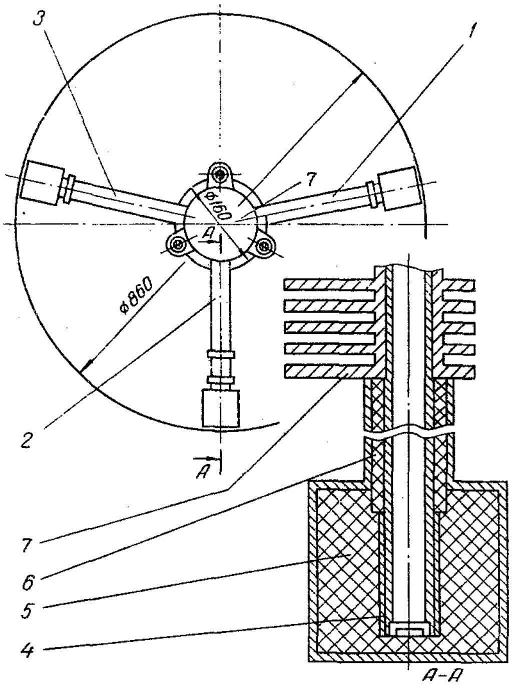

Fig. 1

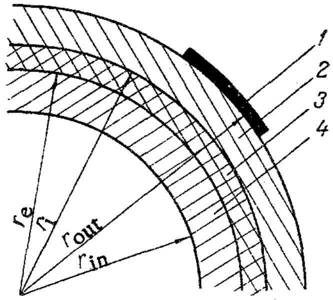

Fig. 2

Fig. 1. Construction of the heat-pipe module mounted on the Interkosmos 15 satellite: 1, 2, 3) heat pipes; 4) evaporation zone with electric heaters and temperature sensors; 5) thermal insulation of the evaporation zone; 6) thermal insulation of the transfer zone; 7) radiator.

Fig. 2. Disposition of the electrical heaters and temperature sensor in the evaporation zone with the notation for computing the temperature drop in the wall: 1) resistance thermometer; 2) heaters; 3) insulation layer; 4) pipe housing.

radiator by a mixture of the silicate glue MRTU6-15-U33-70 with aluminum oxide GOST 8136-56 in a 1 : 1 ratio by weight.

The cylindrical radiator (130-mm diameter, 170-mm height) milled from the material AD-1T, designed for reduction of the heat flux to 45 W in the rated operating temperature range (0-70°C) when air cooled by a fan (see below for characteristics), has three flanges around the circumference for fastening to the satellite frame.

Electrical equipment, including a sectional electrical heater from a bifilar winding of the wire PÉK on around a 60-mm-long section of the housing, and a temperature sensor (resistance thermometer) were placed in the evaporator section of each HP. A resistance thermometer was also installed in the condenser section of each HP. The electrical insulation was condenser paper SILKON with BT-95 lacquer, and the temperature sensors were glued to the condenser paper with BF-2 glue.

The arrangement of the electrical heaters and temperature sensors in the evaporation zone is shown in Fig. 2. Such a sensor disposition was determined by the technology of HP fabrication. Hence, to determine the actual temperature in the evaporation zone, a correction must be introduced for the temperature drop between the sensor and the inner wall of the heat pipe. Two separate domains (Fig. 2) must be examined to compute the temperature drop in the wall of the heating zone.

Heat liberation and heat condition affect the temperature drop in the domain $r_i < r < r_{out}$ in which the electrical heaters from copper wire in lacquered insulation were disposed, while there is no heat liberation in the domains $r_e < r < r_i$ and $r_{in} < r < r_e$ and the temperature drop is determined by pure heat conduction. Under these conditions the desired temperature drop can be written in the form

TABLE 1. Functions of the Heat Flux and Temperature

r	ϑ	W	r	ϑ	W
0	0	0	5,5	3,5178 1	2,4058 2
0,5	9,7317 —4	3,9069 —3	6	6,5634 1	4,9289 2
1	1,5644 —2	6,2662 —2	6,5	1,2302 2	1,0084 3
1,5	7,9621 —2	3,2059 —1	7	2,3165 2	2,0602 3
2	2,5522 —1	1,0418	7,5	4,3825 2	4,2051 3
2,5	6,4170 —1	2,6932	8	8,3255 2	8,5741 3
3	1,4016	6,1505	8,5	1,5877 3	1,7471 4
3,5	2,8188	1,3238 1	9	3,0381 3	3,5579 4
4	5,4137	2,7638 1	9,5	5,8319 3	7,2420 4
4,5	1,0162 1	5,7046 1	10	1,1227 4	1,4736 5
5	1,8917 1	1,1726 2			

$$\Delta T_w = T_{out} - T_{in} = (T_{out} - T_i) + (T_i - T_e) + (T_e - T_{in}). \quad (1)$$

As has been shown in [2], the temperature drop $T_{out} - T_i$ is determined by

$$T_{out} - T_i = -\frac{Q_{rem}}{2\pi\lambda_m} \frac{\vartheta_i - \vartheta_{out} - W_{out} \ln\left(\frac{r_i}{r_{out}}\right)}{W_i - W_{out}} \quad (2)$$

and heat is removed from the inner surface of radius r_i.

Integrals of the heat flux and temperature are presented in [3]. Values obtained for the functions of the temperature ϑ and the heat flux W are represented in Table 1 for certain radii. The numbers in the right side of the column display the order by which the number in the same row must be multiplied.

The heat-conduction equation

$$T_i - T_e = \frac{Q_{rem}}{2\pi\lambda_{i,a}} \ln\left(\frac{r_{i,e}}{r_{e,in}}\right) \quad (3)$$

is used to compute the temperature drop in the domains $r_e < r < r_i$ and $r_{in} < r < r_e$. Since the quantity of heat removed inside is $Q_{rem} = 2\pi q_{in} \cdot r_{in}$, where q_{in} is the heat flux density, then taking account of (1)-(3) we have

$$\Delta T_w = q_{in} r_{in} \left(\frac{1}{\lambda_i} \ln\left(\frac{r_i}{r_e}\right) + \frac{1}{\lambda_a} \ln\left(\frac{r_e}{r_{in}}\right) - \frac{1}{\lambda_m} \frac{\vartheta_i - \vartheta_{out} + W_{out} \ln\left(\frac{r_{out}}{r_i}\right)}{W_i - W_{out}}\right). \quad (4)$$

For the heat pipe module developed r_{out} = 8.5 mm, r_i = 7.5 mm; r_e = 7 mm; r_{in} = 5.5 mm. The computed temperature drop corrections ΔT_w are, respectively, 0.63°C for 5 W, 1.26°C for 10 W, and 1.89°C for 15 W.

To minimize dissipation of the heat flux delivered to the HP from the outer surface of the electrical heaters and the housing into surrounding space, heat insulation from PSV-N was provided. A cotton textile tape was wound above heat insulation in a layer of BT-95 lacquer. In such a construction of the evaporation zone of the heat pipe it can be considered that the heat flux from the electrical heater goes totally to the evaporation zone, i.e., the product $q_{in} r_{in}$ = const is determined by the power of the electrical heater section enclosed.

The heat pipes were fastened to the frame by 10-mm-wide straps of 20-1-1 tin plate through a heat-insulating 10M-A-S slab-rubber padding.

The heat-pipe module underwent ground checkout in the volume provided (including climate tests and verification of the stability to the effect of mechanical overloads and vibrations) with affirmative results. The HP heat-conduction characteristic in the design range of the working temperatures was taken off on a special test stand [4] to study the operability of the HP under almost exploitational conditions. Some of the data obtained here (steady-

TABLE 2. Results of Heat Pipe Tests

P, W		Ground 5	Ground 10	Full scale 5	Full scale 10
HP1	T_1	27,1	31,9	13,0	21,4
	T_2	25,4	26,6	11,9	17,3
	ΔT	1,7	5,3	1,1	4,1
HP2	T_1	35,8	39,8	26,4	—
	T_2	34,3	37,3	17,3	—
	ΔT	1,5	2,5	9,1	—
HP3	T_1	25,7	30,0	22,0	16,0
	T_2	24,3	25,8	19,1	12,8
	ΔT	1,4	4,2	2,9	3,8

state HP operating mode) are presented in Table 2 with the correction ΔT_w taken into account.

To support the experiment on the satellite, a fan of 28 liters/sec productivity for a 4.5 kgf/m² head, to blow around the heat pipe radiator module, and an electrical heater control module to accomplish conversion of the control command into a performance command in connecting (or disconnecting) the necessary electrical heater sections and the simultaneous connection (or disconnection, respectively) of the fan, as well as fuse-containing safety devices in the supply circuits of each electrical heater section were also mounted.

The temperature sensors were set up in the standard temperature measurement system and the storage of the satellite, hence measurement in the 0-70°C range was assured.

Characteristic data of a telemetry check of heat-pipe module operation during the full-scale experiment with the correction ΔT_w taken into account are presented in Table 2 (HP operating mode almost steady-state).

A comparative analysis of the ground and full-scale test results permits making the following deductions:

Heat pipes of the construction tested are capable of operating under weightless conditions, and after the actions of mechanical overloads and vibrations on the powered section of the injection trajectory.

The heat-conduction characteristics of the tested HP with capillary structure in the form of axial rectangular channels on the inner surface of the housing, under weightless conditions are similar in the operating modes realized to those obtained experimentally in the ground tests.

The probable cause of degradation in the heat-conduction characteristics of an arterial HP during operation under weightlessness is the formation of vapor bubbles degrading the transport properties of the arteries [2].

NOTATION

λ, heat-conduction coefficient, W/m·deg; Q, heat flux from the heater W; q, heat flux density, W/m²; T, temperature, °C; r_{out}, r_i, r_e, r_{in}, radius of the appropriate layer of the heat pipe evaporation zone, m; ϑ, W, heatflux and temperature functions, respectively; P, flux transmitted by the heat pipe, W; T_1, temperature in the evaporation zone, °C; T_2, temperature in the condensation zone, °C; ΔT, temperature drop between the evaporation and condensation zones, °C; ΔT_w, temperature drop in the multilayer wall of the evaporation zone, °C.

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INVESTIGATION OF A GAS REGULATABLE HEAT PIPE WITH A WICKED RESERVOIR

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Results are presented of an analytical and experimental investigation of the characteristics of a heat pipe with a wicked reservoir as the temperature of the cooling medium, and the heat supply intensity change.

Gas regulatable heat pipes (GRHP) with a wicked reservoir are used successfully to stabilize the temperature of heat-liberating objects operating under substantial changes in the thermal load (10-15-fold) and the cooling temperature, at different temperature levels [1-5]. The extensive application of GRHP of this kind in real apparatus is due to the high stability of the working characteristics, the absence of extremal situations during start-up (e.g., a substantial rise in the vapor pressure and temperature above their working level, as can be observed in structures with a wick-free reservoir when the working fluid is incident) as well as the comparative simplicity of fabrication.

Gas regulatable heat pipes with a wicked reservoir are also the basis for constructing thermostat systems with variable thermal resistance heat pipes which have a higher accuracy of temperature stabilization (e.g., GRHP with electrical feedback, thermodiodes, etc. [2-4, 6]).

The strict dependence of the partial vapor pressure of the working fluid on the reservoir temperature, which governs the circuit operation in many cases, is characteristic for a construction with a wicked reservoir. Using the model of a flat interface between the vapor and the uncondensed gas (UCG) as basis, we obtain a dependence to determine the accuracy of vapor temperature stabilization t_{v_1}, t_{v_2} on the primary parameters t_{c_1}, t_{c_2}, t_{r_1}, t_{r_2}, $V_r/\Delta V_g$:

$$\left(\frac{p_{v2}}{t_{r_2}+273}-\frac{p_{v1}}{t_{r1}+273}\right)-\left(\frac{p_{r2}}{t_{r2}+273}-\frac{p_{r1}}{t_{r1}+273}\right)=\frac{p_{v1}-p_{c1}}{t_{c1}+273}\frac{\Delta V_g}{V_r}, \tag{1}$$

where the partial vapor pressure of the working fluid p is related to the corresponding temperature of the vapor or the vapor–gas mixture t on the saturated curve. Equation (1) is based on conservation of the balance of the mass of gas for two positions of the vapor–gas interface. Its solution for the magnitudes of the pressures $p_{v2}(T_{v2})$ and $p_{v1}(T_{v1})$ permits determination of the accuracy of vapor temperature stabilization $\Delta t_{reg} = t_{v2} - t_{v1}$, and therefore, of the heat liberation source for given ratios $V_r/\Delta V_g$ and temperature fluctuations of the cooling medium ($t_{c1} - t_{c2}$) and of the storage reservoir ($t_{r1} - t_{r2}$). Balance equations of the form

$$Q = k_l\,(L_к - L_g)\,(t_v - \bar{t}_c) \tag{2}$$

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