the next conception model of cooling system /6/ that gave possibility to realize the heat output from electronic components by means of heat conducting structure to two heat pipes (see figure 11).

The unit consists of 9 uniform module-card 5, a base plate 7, a cover of the unit 8.

Each electronic card has heat conducting sub layer 6 intended for heat collection from the electronic components and heat supply to two heat pipes 1, 2. These heat pipes are extended through printed card of all modules. Thermal connection between sub layer 6 and HPs 1, 2 is realized by construction 4.

The installation of an additional heat pipe 3, connecting condensation zones of both HPs 1 and 2, increases the temperature uniformity of electronic circuits and reliability of the TCS.

Heat removal is provided through the mounting places on the base plate. "Cold platform", emitting radiator, or VCHP may be used for heat rejection from the base plate.

Above mentioned unit had been designed with regard of the next technical data:

- total heat output is up to 80 W;
- temperature of mounting place is -20...+50°C;
- temperature level of any electronic components on card do not exceed 350 K;
- dimensions of individual card are 280*230*30 mm.

Heat pipes were manufactured from cooper tube (diameter 14 mm) with cooper metal fiber structure. Methanol used in HPs 1,2, water - in HP 3. Increasing of HPs' Q_{max} (up to W*m for methanol) was reached by using of longitude liquid artery placed in vapor channel of HP. HPs 1,2 have bending in two perpendicular planes, HP 3 - in one plane.

TEST RESULTS - Thermovacuum tests were conducted for the electronic unit in assembly with HPs. There were investigated two types of heat conducting structure: rectangular form and radial one.

The least thermal resistance between electronic component and place contacting with heat pipe was provided by radial heat structure. Heat pipe 1, 2 (figure 11) had provided the predicted thermal resistance and temperature of mounting places for every electronic board at level T_{mp}=325 K and heat rejection from every card 15 W.

Maximum temperature of the electronic block was less than 350 K.

Electronic block with mentioned conception cooling may be designed for space scientific apparatus working in open space (non-hermetic modules)

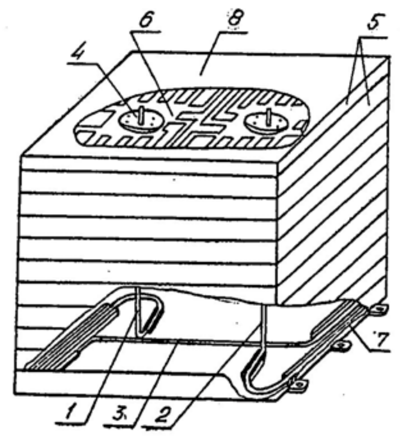

Figure 11. TCS of electronic block

SUMMARY

It was developed the ways of construction of electronic block for space devices. Applied autonomous systems on base of variable conductance heat pipes (VCHP) may function in passive or active control regimes (using an additional electronic controller or not) and are integrated into electronic blocks with different dimensions and purposes. Advanced values of control accuracy were 300±2 K at oscillation of mount places' temperature -20 ... 50 °C, oscillation of heat rejected 1.5 ... 13 W, and oscillation of external absorbed heat flux 50 ... 300 W/m^2. Electronic plate fixing on the flat heat pipes gives an opportunity to abolish of dimensional limitation and to increase density and power of heat rejected by electronic components. For the system *electronic plate + heat pipes + TCS* it was solved a task of heat transfer from the electronic elements and their thermal control at existence of non-uniform heat rejected by elements (1 ... 10 W).

Heat pipes can reject the heat release of electronic elements installed on printed cards. The combination of heat conducting structure, thermal conducting clamp and heat pipes are proposed to support temperatures of any elements below 353 K at 60 W total output of 9 printed cards.

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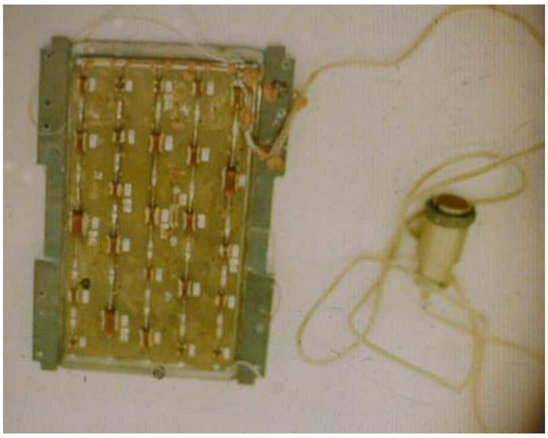
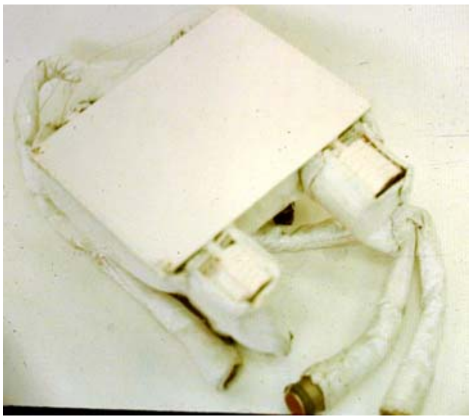
Thermostable electronic block with MLI



Printed card – flat heat pipe



Electronic block with isothermal electronic card

Base of electronic block with heat pipes

Part 1.1. Passive Thermal Control Systems with usage of Solar Energy for operation

References:

V. Baturkin et al. Passive Thermostatting System with Application of Gas-Filled Heat Pipes and Thermal Energy of Solar Radiation. Proc. of the 4th ESSECS, 1991, Florence, Italy; ESA, 1991, vol.2, p.769-774

ABSTRACT

An approach to the creating of passive or half-passive thermocontrol (TC) systems, using a heat energy of the solar radiation for its functioning and variable conductance heat pipes (VCHPs) as temperature control elements, is analysed with conditions of considerably altering (more 2 times) solar constant conditions. Results of the working out of control system specimens with two parallel metal-fibrous methanol HPs are presented for the typical instrument unsealed module with next conditions: range of solar constant values - 500..1400 W/m^2; variations of the temperature on mount places - 253...323 K, device temperature level 280...300 K.

Keywords: solar heat energy, variable , conductance heat pipe, device, thermostatting

1. INTRODUCTION

For objects, oriented to the Sun and worked in conditions of the negligible alteration of distances to the Sun, it is possible the application of passive TC systems based on an utilization of solar radiation heat energy. The high stability of solar constant qs has assure a constancy of the device temperature, that follows from the solution of heat balance equation:

$$Q_l + \varepsilon_r F_r \sigma (T^4_{dev} - T^4_0) - \alpha_s F_{ab} \cos\varphi\, q_s = Q_{dev} + (T_{mp} - T_{dev})/R_{mp} \pm Q_{bal} \quad (1)$$

where ε_r , F_r - emissivity factor and surface area of the emanating radiator; α_s, F_{ab} - solar radiation absorptivity and surface area of the solar absorber; σ = $5.67 \times 10^{-8} W/(m^2 K^4)$; φ - slope angle of Sun rays incident on the absorber; T_{dev}, T_{mp}, T_0 - temperatures of thermostatting device, mount places for object and environment, accordingly; Q_l, Q_{dev} - heat loss flows through the insulation and device heat realise.

An increase of the thermal resistance value between device and mount places, as well as an increase of F_{ab} allows to reduce the influence of T_{mp} and "to tie" the device temperature T_{dev} to predetermined value, and by the value q_s. At the variation of q_s, which is caused by the object movement relative to the Sun, the object temperature alterates very considerably by now. On fig.1 results of computation with Eq.1 are presented for two mount place temperatures (253 and 323 K) and typical values of R_{mp} = 0, 5, 15 and ∞ K/W. One may, theoretically, with T_{mp} to be constant, select such system parameters, which assured small deviation (± 10 K) of T_{dev}. However, it is impossible, with predetermined fluctuations of T_{mp} and with functional uncertainty between this values and q_s to obtain acceptable level of the thermostabilization T_{dev} = 280...300 K (T_{min} < T_{dev} < T_{max}) within given conception of TC system construction. First of all this associates with a fact, that the system heat balance includes greatly variated values $\alpha_s q_s F_{ab}$ and $(T_{mp}-T_{dev})/R_{mp}$.

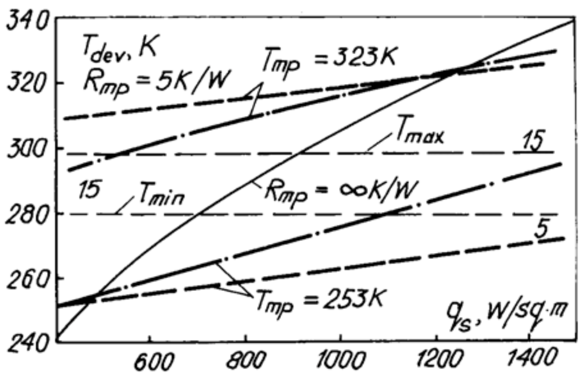


Fig.1. Influence of thermal resistance between device and mounting places "R_{mp}" and temperature of mounting places "T_{mp}"

For the heat balance ensuring at negligible alterating of T_{dev}, one may use systems with variable values F_{ab} and F_r, which realize owing to usage of bimetallic elements, liquid or electric power drives (Ref.1), as well as such systems, heat balance of which contains the controlled value Q_{bal}, compensating a deficiency or a surplus of the heat energy. If the system design performs in the way, that Q_{bal} < 0 for all conditions, than variable conductance heat pipe may be as a control instrument for Q_{bal}.

2. CONCEPTUAL PRINCIPLES

Showed on fig.2 possible design modification of such TC system includes: scientific equipment unit 1, mount place 8, radiator-flange 2 with light-absorbing coating 10, VCHP's 3 with reservoirs 5 for permanent gas, and shaded radiator-emitter 4. VCHPs 3 and both radiator-flange 2 and radiator-emitter 4 was joined by a brazing. Connection between unit 1 and radiator-emitter 4 was performed using elements 6, 7 with the low thermal conductivity (for decreasing of non-controled heat transfers), but between radiator-flange 2 and unit 1 the connection with high thermal conductivity was ensured. All surfaces of unit 1, non-working parts of radiators 2, 4 was covered by a heat insulation 9 marked "MLI". Number of HPs may be more than 1 for the raising of an operate reliability. TC system working conditions provides the temperature alteration of mount place 8 in the range $T_{mp,min} \ldots T_{mp,max}$, and stabilization diapason of the unit 1 temperature T_{dev} is within of this range $T_{mp,min} < T_{dev} < T_{mp,max}$.

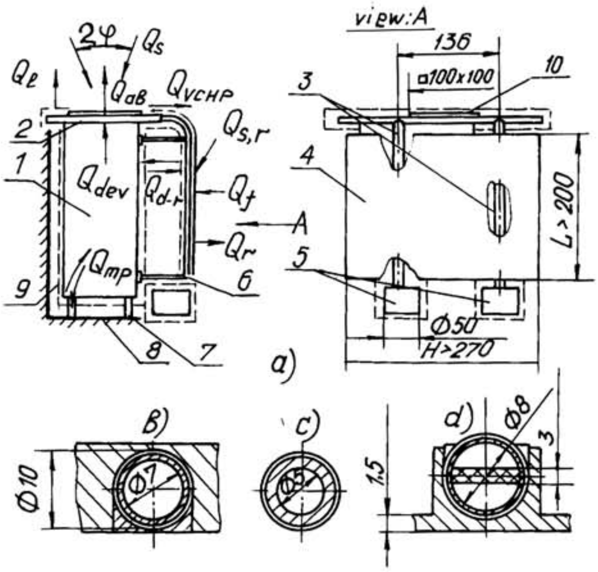


Fig.2. Scheme of thermocontrol system, b-d-cross-section of HP wall in zones

TC system works in the following way. The heat supplies to TC system by means of the heat realise Q_{dev} of unit 1 and the solar radiation energy α_s absorbed by the coating 10. One part of the heat is spended on a compensation of heat withdrawals to mounting places $(T_{mp} - T_{dev})/R_{mp}$ and another part radiates from the surface of the flange 2, opened to the space, Q_{ab} and Q_l. Remaining heat Q_{vchp} removes by VCHP to shaded radiator 4 and emits, with that the compensation of heat balance alterations and the correction of a value of heat flow removed Q_{vchp} are performed by the self-adjustment of HPs thermal resistances, and temperature level of the stabilization will be determine using the mass of non-condensable gas into a reservoir (Ref. 2). Heat balance of the device unit may be written as follows:

$$Q_{dev} + Q_s + Q_{mp} = Q_{ab} + Q_l + Q_{vchp} + Q_{d-r} \quad (2)$$

and for shaded radiator of the TC system:

$$Q_{vchp} + Q_{d-r} + Q_{s,r} + Q_f < Q_r \, , \quad (3)$$

$$Q_s = \alpha_s q_s F_{ab} \cos\varphi$$

$$Q_{ab} = \varepsilon\sigma F_{ab}(T_{ab}^4 - T_0^4)$$

$$Q_{mp} = (T_{mp} - T_{dev}) / R_{mp} \quad (4)$$

$$Q_{s,r} = \alpha_{s,r} q_s \sin\varphi F_r$$

$$Q_r = \varepsilon_r \sigma F_r \eta_r (T_r^4 - T_0^4)$$

$$Q_{d-r} = (T_{dev} - T_r)/R_{d-r}$$

In Eqs. 2-4 α_s, F, ε, η - solar radiation absorptivity, surface area, emissivity factor and efficiency of radiator; T_r - shaded radiator temperature, R_{d-r} - heat resistance between unit and shaded radiator, concerned with a heat transfer through "MLI" and fixing elements; Q_f - background flow absorbed by shaded radiator (reemission from devices, insulation, etc.). Eqs.2-4 are considered for the limit working conditions of the TC system, with an account of possible alterations among coating optical characteristics and the condition $Q_{min} < Q_{vchp} < Q_{max}$, where Q_{min} - minimum control heat flow, at which VCHPs begins to assure the control level desired, Q_{max} - maximum heat flow to be withdraw by the radiator-emitter for possible geometrical and optical characteristics and outer radiant perturbations on it ($Q_{s,r}$, Q_f). Proposed construction of the TC system with VCHPs was realized like the heat mock-up. In an analysis of heat perturbations next initial data was assumed:

- solar constant alteration, caused by the space vehicle movement in an outer environment and by seasonal fluctuations of the solar activity q_{Smax}/q_{Smin} = 2.8;
- deviations of the slope angle of Sun rays incident to the coating heat absorbed, which indices by the possible alteration of object orientation relative to the Sun direction $\varphi = \pm 20^o$;
- temperature deviations of mount places $T_{mp,min}$ = 253 K, $T_{mp,max}$ = 323 K; device temperature 280...300 K;

With solving Eq.2 in conditions R_{mp}=15 K/W, α_s = 0.90 and ε = 0.2, the value F_{ab}>0.01m^2 was determined, at which for all perturbations Q_{vchp}>0. Heat conditions of the shaded radiator depends on the heat-supply flow Q_{vchp}, the exposure by Sun rays $Q_{s,r}$ and the value of background radiation Q_f. The determination of a surface area of the shaded radiator at maximum values Q_{vchp}, $Q_{s,r}$ and Q_f leads to value F_r>0.054 m^2.

List of typical heat flux balance

φ		0.00	20.00	0.00	20.00
q_s	W/m^2	1400	1400	500	500
Q_s	W	12.60	11.84	4.50	4.23
Q_{ab}	W	0.92	0.92	0.70	0.70
T_{dev}	K	300	300	280	280
Q_l	W	1.00	1.00	1.00	1.00
Q_{mp}	W				
at T_{mp}=253 K		-3.13	-3.13	-1.80	-1.80
T_{mp}=323 K		1.53	1.53	2.87	2.87
$Q_{vchp,min}$	W	7.55	6.79	1.00	0.73
$Q_{vchp,max}$	W	12.21	11.45	5.67	5.40
Q_f	W	2.80	2.80	1.00	1.00
$Q_{s,r}$	W	0.00	7.83	0.00	2.80

The value Q_{vchp} equals to 0.7...12.2 W. Guaranteeing of the system serviceability at lower power level $Q_{vchp} > 0.7$ W is very problematic because of considerable influence of magnitudes Q_{mp}, Q_L on heat balance. For keeping the condition q_s->min it is needed to have the margine in the minimum magnitude of $Q_{vchp,min}$. Increase of Q_{vchp} owing to the correction of values α_s, ε and R_{mp} is not real in practice because of design and technological limitations obtained. With increase of F_{ab} the effect has been reached without delay, but it results in an augmentation of the radiator surface and hence the radiator mass at condition $q_{s\,max}$. Furthermore, subsequent development of this approach is restricted by a considerable rise of F_r. An expansion of the growing applicability of this regulation principle at optimal dimensions F_{ab} and F_r may be achieved by introducing of additional element – a cover once being opened (closed). Opening of the cover changes the value F_{ab} and, consequently, increases a quantity of heat Q_s being introduced into a system. The cover operation should be unitary in order to simplify the lock construction and the drived mechanism and should be performed with a help of a command from without. The range of values q_s, at which the operation is carried out, depends on real values Q_{mp}, Q_L and relationship of magnitudes F_{ab} before ($F_{ab'}$) and after ($F_{ab''}$) opening. On fig.3 data of the heat flows balance are shown for conditions R_{mp} = 10 and 15 K/W; $F_{ab'}$ = 0.01 m^2; $F_{ab''}$ = 0.02 m^2.

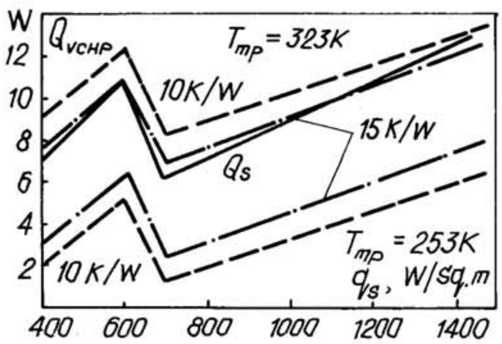


Fig.3. Influance of qs on heat flux through VCHP for advanced thermocontrol system

2.1 Modelling VCHP performance

The determination of VCHP heat characteristics was been carried out by models (Refs.3 – 6) without taking into account the influence of the heat-carrier diffusion on a temperature pattern of the condensator zone. It was been solved a differential equation in the form:

$$C(x)\frac{dT}{d\tau}=\frac{d}{dx}\left(A(x,T)\frac{dT}{dx}\right)+(T-T_v)/R_{in}(x,T_v,T)-$$

$$(T-T_0)/R_{ex}(x,T,T_0)+q_{ex}(\tau,x,T,T_0) \qquad (5)$$

which was been composed for all VCHP zones $0< x <L_{vchp}$ (evaporator, adiabatic zone, radiator-condenser and reservoir). In addition to Eq.5 there was composed balance integral equations for the heat transfer by vapor and the mass of a noncondensable gas M_g:

$$\int_0^{L_{vchp}} (T-T_v)/R_{in}(x,T_v,T)dx = 0 \qquad (6)$$

$$\int_0^{L_{vchp}} (P_v-P(T_v))/T_vF_v(x)dx = M_gR_\mu \qquad (7)$$

In Eqs.5-7 x – longitudinal coordinate; τ – time; T, T_v, T_0 – temperature of housing wall, vapor space and heat sink consequently; C, A – heat capacity and thermal conductivity of housing cross-section; R_{in}, R_{ex} – linear thermal resistances of internal and external heat transfer processes; $F_v(x)$ – function of vapor cross-section area, $q_{ex}(T, x)$ – outer heat flow, P_v, $P(T_v)$–vapor pressure and partial. The solution of a system Eqs. 5-7 was been performed numerically using the finite difference method. On the basis of temperature distributions obtained in a housing the boundary conditions of the VCHP functioning was been determined.

For refinement of heat transfer conditions in cross-sections at non-unifom heat supply (heat removal) along a section and for the determination of thermal resistances R_{in} the method of finite elements in two-dimensional formulation was been used. The temperature pattern in device housing was also been determined by a two-dimensional simulation, when one of VCHPs was got out of operation. Selection of the VCHP type and working fluid is determined by an accuracy of the devise temperature stabilization, the reservoir placement possibilities, it temperature condition and a power consumption for system functioning. For heat mock-up to be work out it was been supposed no usage of automatic devices for the reservoir temperature stabilization, at which a maximum accuracy of the temperature stabilization was assured (Ref.6). Possible potentialities consists in the use a system with reservoir being covered by a capillary structure or not, mounting in a zone witn constant temperature (e.g., on the device surface).

For a determination of the regulation accuracy, assured by one or another schemes, it is needed to know a range of temperature deviations of the reservoir or condensator zone end, being adjoined to reservoir. This values may be approximately accounted by the magnitude of radiant flows incident on radiator and reservoir on the basis of balance equations:

$$(T_{r'}-T_b)/R_{r,b}+Q_{s,b}+Q_{f,b}+(T_{mp}-T_b)/R_{mp,b}=$$
$$\varepsilon_bF_b\sigma(T_b^4-T_0^4) \qquad (8)$$

where $T_{r'}$ – temperature of the condensator end, $Q_{s,b}$ and $Q_{f,b}$ – solar and background radiant flows to be absorbed by the reservoir surface, $R_{r,b}$ and $R_{mp,b}$ – thermal resistivities between condensator and reservoir and between mount place and reservoir, accordingly. More accurate determination of change limits $T_{r''}$ and T_b is possible at a solution of the complex problem Eqs. 5-7.

For a computation of the regulation accuracy next values are adopted: minimum temperature in filled by non-condensable gases radiator and reservoir zones (with minimum power supplied Q_{vchp} and minimum condensator exposures) $T_{b,min}$ = 178 K, maximum temperature in filled by non-condensable gases radiator and reservoir zones (with maximum power supplied Q_{vchp} and maximum condensator exposures) T_{b}^{max} = 265 K. A length of the gas plug L_g, which determines the radiator surface being opened to radiation, is maximum in a first mode and equals to the radiator length L_r, but in a second mode it is minimum and equals to 0,2 L_r. The regulation accuracy $\Delta T/2$ assured by a TC system to be calculated in accordance with (Ref.6) as a function of the volume of reservoirs vapor space is shown on fig.4.

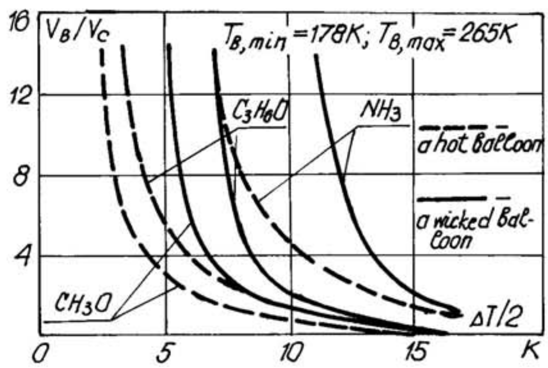


Fig.4. Ballon-condenser volume ratio as temperature accuracy

Analyzing correlations obtained we had concluded, that the construction with "hot" reservoir and without capillary structure assures the better regulation accuracy for all working liquids selected. Hovewer, this scheme have a shortcoming, concerned with a possibility of the working liquid ingress into reservoir,and dificulties of it return to the heat-up zone (it's possible in the course of groung trying out,during starting and manoeuvres of a spacecraft), that results in a considerable rise of the thermostatting level to (+50...+60) ^{0}C. In this connection the scheme with reservoir being covered with capillary structure is took for a heat mock-up. The methanol, having best regulation characteristics, low freezing point (- 98 ^{0}C) and acceptable thermal and mechanical properties, is used as a working fluid. Volumes ratio V_b/V_c for the working out scheme is adopted equal to 13, that principally defined by zone dimensions for the reservoir placement and by a nature of changing of the function $V_b/V_c = (\Delta T_v/2)$.

3. DESIGH OF THERMAL SAMPLE

On fig.2 a construction of the heat mock-up is shown. Heat absorbing plate 10 with coating being made with a technology (Ref.7) is glued by an epoxy adhesive to the flange 2. Cylindrical HP housings in the heat-up zone regions are soldered in flange channels (fig.2b). Zone of the radiator-emitter is formed of rectangle sections with wide 0,01 m (15 pieces) and 0.005 m (6 pieces) and with length 0.27 m. HP housings for an augmentation of the heat contact is soldered in slots of radiator sections by Sn - Pb solder. Both flange 2 and radiator 4 are made from the aluminium alloy, chemically coated with Ni for an ensuring of soldering conditions to the VCHP housing.
In TC system two identical VCHPs are used with reservoirs, inner surface of which is covered by a capillary-porous structure. Concerning with limited overall dimensions for the radiator-emitter placement reservoirs are situated in a direct contact with the end section of a radiator, heat condition of which determines the temperature deviations range of reservoirs. Capillary-porous structure of VCHP is made as a fibrous one (Ref.3). Fabrication technology applied permited to make the structure with different thickness and shape in heat-up, transport and radiator-emitter zones (fig. 2b,c,d). Structure porosity was (80 ± 2)% in all zones. In a radiator zone the artery is mounted, allowing to decrease an area of the vapor channel section at simultaneous increase of the transfered heat flow. An artery is gone out from condensation zone to the reservoir, having also a high efficient artery, and joined to it. The aim is in avoiding of the liquid accumulation in a reservoir for conditions of peak exposures of a radiator by the Sun, when vapor-gas front moves in a reservoir. VCHP housing and capillary structure are made from stainless steel, assuring a considerable resours with main potential heat carriers (aceton, metanol, ammonium).

4. INVESTIGATION OF MOCK-UP

The investigation of TC system on the basis of VCHPs was accomplished in normal conditions with a heat remove by convection into environment and in vacuum chamber with shields being cooled by a liquid nitrogen. Imitation of the TC system functioning was been implemented in next way.Heat supply to the absorber was been carried out with a help of either resistance heater,mounted on the non-working side, or solar radiation's simulator with a diameter of the working zone 0.15 m. Modelling of the solar and background radiation for both radiator and reservoir was been realized by three methods.On initial stages of investigations flows Q_s and Q_f was simulated by radiation from the painted aluminium plate to be mounted in parallel to radiator on it nonworking side. On subsequent stages flows Q_s and Q_f was simulated by heaters being mounted on the radiator and reservoir directly. At more detail investigation of the system to combine with a devise the flow Q_f was been simulated by radiation of specially oriented heated surfaces, to be characterizing the system environment, and the quantity Q_f was simulated with a help of electric heaters. Each of methods proposed has a different approximation to the real process. All ones has enough simple realizations in practice and the test execution does not required to use continually of solar radiation source with enough large light spot (more than 0.3 m).
Environment imitation was realized with use of shields having a temperature 79 -83 K in the vacuum chamber. Those formed the closed space.
Copper-constantan thermocouples mounted in heat supply zone (NN 1-6), adiabatic zone (NN 7,8) condensation zone (radiator) (NN 9-22) and on a reservoir (NN 23-26) was used as temperature detectors.

The measures for decreasing of not taking into account heat flows from measuring and power wires, through heat insulation from non-working zones of the heat-absorbed flange, imitator of perturbations of a shaded radiator and reservoirs were been took at TC system installation in a chamber. HPs housings in those bend plane (see fig.2) was leveled for avoiding of the gravitation forces influence on heat transfer characteristics of VCHPs.
TC system had been mounted in a chamber such way, that emitting surface of the radiator was parallel to the chamber axis. The execution procedure of investigation involved a study of the influence of the external radiant perturbation value $Q_{ex} = (Q_{s,r}+Q_r)$ and of the value Q_s on TC system regulation characteristics. Tests had been conducted without device unit with a purpose of diminishing of the time of transient processes, and in subsequent runs - jointly with device and simulators of mount places. Electric power supplied to the heater of solar radiant absorber Q_s, im was altered in a range (2...18) W, and by using a solar radiant simulator the flow density was 290, 550, 1100 and 2200 W/m^2.
The typical distribution of temperatures in a VCHP housing at the least value of heat perturbations on the radiator Q_{ex} = 1 W (density of the flow absorbed) is shown on fig.5. Temperature pattern on a heat-up zone is practically uniform: lengthwise temperature deviations on each of HPs was 0.4 ^oC and the temperature difference between HPs was no more than 0.6 ^oC, that may be explained by distinctions in mass of non-condensable gases, geometrical dimensions and heat and physical characteristics.

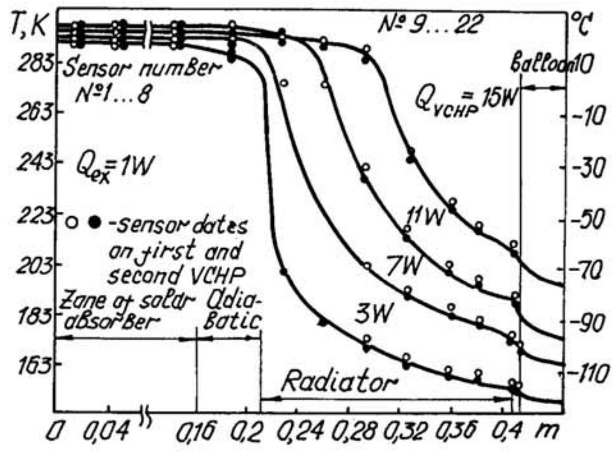


Fig.5. Temperature distributions along VCHP as function of transmitted heat flux

Temperatures in HPs transport zones differs each other by a value no more than 0.6 ^oC and at the mode with $L_g < L_r$ are correspondent to vapor temperatures.
Obtained with another values Q_{ex} lengthwise temperature profiles showed that increase of a level of the shaded radiator exposure has resulted in a temperature rise of all radiator and reservoir elements and in a displacement of the vapor-gas boundary toward a reservoir (at equal Q_s).
In steady conditions at value $(Q_{vchp} + Q_{ex})$ > 22 W the radiator will have a temperature more than + 30 ^oC, that higher than thermostatting point. Gas plug in this case is minimum and a major mass of the gas will transfer onto reservoir. Rising of the reservoir temperature T_r results in an increase of a temperature level in VCHP, since

$$(P_v-P(T_b))/T_b = \text{const}, \tag{9}$$

$P(T_b)$ - pressure of saturated vapor as function of temperature.
In modes with Q_{ex} = (1...2.8) W and Q_{vchp} < (4...6) W radiator element temperatures are less than a level of the heat-carrier freezing point. Serviceability worsening of the TC system in that modes and in start up of the VCHP with frozen heat-carrier do not detected.
Integral regulation characteristics of the TC system are shown on fig.6 in a form of the function of temperature on heat received flange T_{ab}, transport zone T_a, reservoir T_b at discretely alterated values Q_s (2...16W) and at three fixated values Q_{ex} (1; 2.8 and 13 W). Systems has two limits at little values of Q_s (Q_s < 4 W) and at maximum this values Q_s+Q_{ex} > 22 W. All curves has an inflection near Q_s = 4 W, which characterises the front placement in a transport zone.

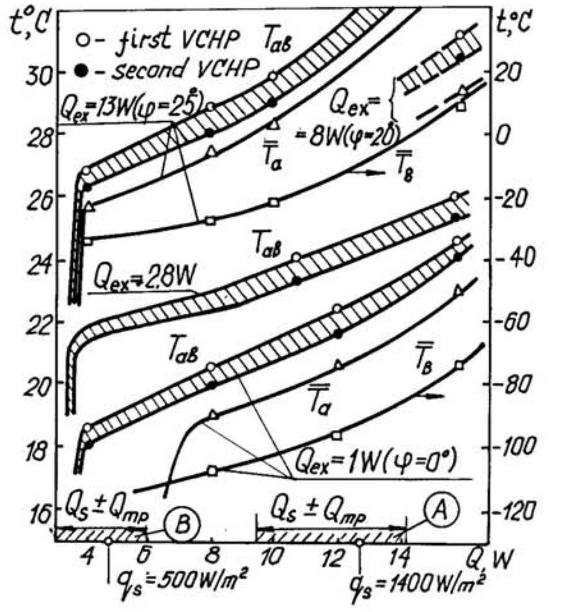


Fig.6. Control characteristics of VCHP system. Right axe is balloon temperature, "A" & "B" - regions of extrem regime

Correspondent value Q_{vchp} is less 2.5 W. With a decreasing of Q_s to 2 W the value T_{ab} falls up to 0...5 ^oC. With an increasing Q_{ex} the regulation level rises because of the reservoir heating. The magnitude $\partial T_{ab}/\partial Q_{ex}$ is 1 K/W.
Experimental data shows system abilities to control the temperature T_{dev} at boundary values q_s and T_{mp}. Zones "A" and "B"(fig.6) defines regions of the heat energy variation being entered into a system ($Q_s \pm Q_{mp}$) under q_s = 1400 and 500 W/m^2. Magnitude Q_{mp} is selected as a most considerable one and computed at R_{mp}=15 K/W.For conditions "A" (q_s =1400 W/m^2) at T_{mp} = 253...323 K and φ < 5^o the system has assure a device

stabilization in the range (23...26) ^oC. At increasing of the angle φ to 20^o a regulation level is raised to (28...33) ^oC because of an influense of solar rays incident on the radiator and reservoir (at $\alpha_{s,r}$ < 0.4), but for φ = 25^o- to 29...37 ^oC With the influence reduction φ may be diminished by a decreasing of value $\alpha_{s,r}$ to less than 0.4. Under conditions "B" (q_s= 500 W/m^2) system has maintain the temperature on a level (18...21) ^oC at φ = 0...20^o. However, at heat withdrawal values Q_{mp} > 1 W (it corresponds to T_{mp} < 0 ^oC) a temperature T_{ab} is lowered on the level 0...7 ^oC, that arousing by quite enough little value Q_{vchp} = (0.5...1)W, which determines the heat balance stability of a whole system. Therefore, TC construction proposed may satisfy working characteristics in the whole perturbation parameters range except a combination q_s= 500 W/m^2, T_{mp} < 0^oC. This defect may be removed in practice by two ways: rising of surface F_{ab} and hence F_r by 25 ... 60 %, and putting a supplementary element - the cover, which discloses in addition 30...100 % from initial value Fab and ensure the value ($Q_s \pm Q_{mp}$) > 4 W at any value T_{mp} and in a more broad range of R_{mp}. Cover construction mass is no more than 0.08 kg. Cover opening mechanism operates at 27 V of a power supply over 10 s. Value q_s, at which it is needed to open the cover, is defined after experimental account of limits R_{mp} and α_s. Correction of this value q_s occurs for data of telemetric temperature sensors.

CONCLUSIONS

Principles developed of the autonomous thermocontrol system construction may be used for devices with at least single solar orientation axis. Deficit of an auxiliary power is compensated by the solar heat energy. Variable conductance heat pipe supports heat balance in the system and is responsible for a temperature control.

Scope of disturbances covers required control temperature range (288...303)K, the variation of a solar constant 500...1400W/m^2 temperature of mount places (253...323) K. Main design difficulties is found to support the thermal balance at smallest solar constant and lowest mount place temperature. Achieved device temperature control accuracy is better than (18...31) ^oC for advanced design with opening cover. Total mass of thermocontrol system is less than 12% of device mass. The indicated principle may be applied for building of thermocontrol systems at the conditions of auxilliary electrical power deficit and wide range of oscillations q_s, for example, for Mars missions.

AKNOWLEGMENT

Authors thanks Doodejev A.A., Dr.Savina V.N and Dr.Zhuk S. K. for consultations and advices during the work performing.

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1.+2.+4.: books
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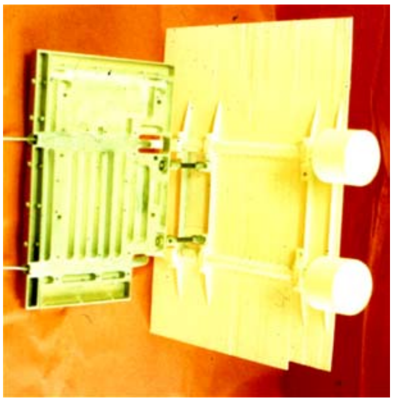
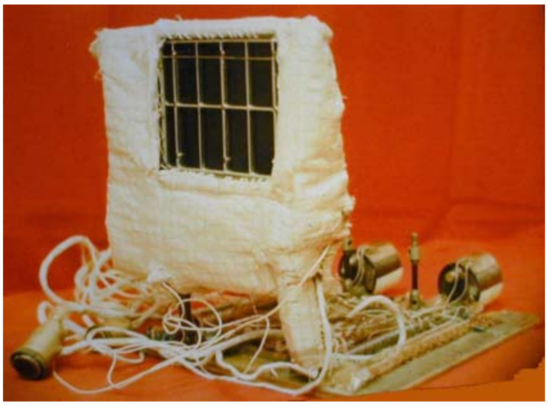
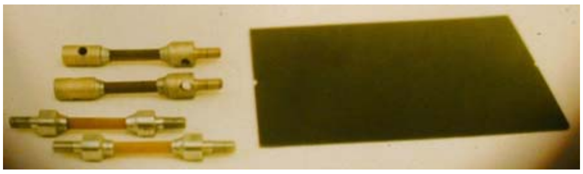


VCHP outlook



VCHP assembly



Components of thermal control system: solar flux absorber, low conductance supports