Part 3. Low temperature heat pipe systems for optical sensors cooling

Sources:

- V.Baturkin et al. Thermodiode system application for the achievement of low temperature for optic sensors at external disturbances. Proc. of 22nd ICES. Seattle, USA, 1992, report No 921209.
- V.Baturkin et al. Working out of Heat Pipes for Low Temperature Radiative Cooling Systems for Space Optic Sensors. Proc. of 26th ICES, Monterey, USA, 1996, report No. 961603

ABSTRACT

The substantiation of heat pipe usage in passive radiative cooling systems on temperature level (190...240) K for space optical sensors is presented. Heat pipes can be sound practice like heat conducting lines between sensor and radiator particularly at distances more 0.2 m and irreplaceable at distances (0.5...2) m. Embedding heat pipe with radiator allows to create the uniform temperature basis in case of several sensors connection to single radiator and to improve radiator efficiency.

It is analyzed approach to design of thermocontrol and cooling radiative systems with heat pipes to reduce sensitiveness to external light disturbances and to enlarge area of radiative system application. The results of design, thermovacuum test and flight operation of thermocontrol radiative system samples are under discussion as well.

INTRODUCTION

Questions of thermal regime providing on-board space housekeeping and scientific equipment always remain as actual. One of them it is ensuring of required temperature level of sensitive elements of optoelectronic light receivers which widely use in TV observation, IR technology. Special feature of these devices is strong dependence of output characteristics from temperature which lay in range (-90...-40) °C and less (depending on material type used and type of optical sensor).

Own heat released by sensitive elements is enough small (tens - hundreds milliwatts), but design of receiver has to foresee special measures for reducing of heat leakage from surrounding apparatus and mounting places which have much higher temperature level up to +50° C. Existing constructions of receiver heads allow to get level of additional heat inputs in range (0.3...2.5) W at vacuum surrounding. Discussed temperature level (-90 ... -40) °C and value of heat rejected assume to use for sensitive elements cooling the systems based on principle of thermoelectric batteries, heat emission to space and their combination. In every system there is additional element which fulfills the role of heat connecting line between hottest point and source of required temperature. Sometimes usage of heat pipe [1] in this role gives good thermal and design solution.

In presented report some design conceptions of thermal control and cooling systems based on application of heat pipes for space light receivers are considered. Systems were designed and manufactured in Kiev Polytechnic Institute (Ukraine) in cooperation with Institute of Space Research of Russian Academy of Sciences (Moscow, Russia) for benefit of series of International space projects.

THE BASIC PRINCIPLES OF THERMO-CONTROL SYSTEM CONCEPTIONS

There are two main widely used approach to building of thermocontrol systems (TCS) for space sensor cooling. First one deals with usage of thermoelectric battery Peletier. The matrix (sensitive element) of receiver has perfect thermal contact with "cold junction" of thermoelectric battery which ensure required matrix temperature level; "hot junction" has connection with heat sink. Thermal scheme of such thermocontrol system is presented on figure 1,a.

Heat leakage Q_{mp} and heat released in matrix Q_d are input to thermoelectric battery Peletier (its cooling productivity). Heat rejection from "hot junction" Q_{TB} which is essentially more than heat input ((4...15) times, against dependency from temperature drop) has to be rejected by additional cooling system like liquid, radiator or mounting place.

Usage of such type system is perspective when source of low temperature less than matrix

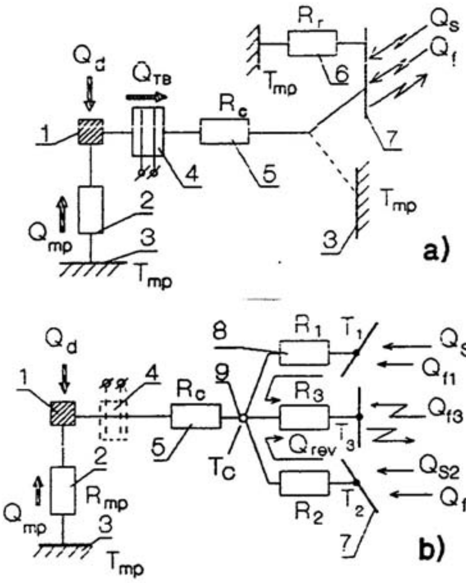

Figure 1: Thermal scheme of active TCS with thermoelectric battery Peletier (a) and passive radiative TCS (b): 1 - device cooled; 2 - thermal resistance "mounting place - device"; 3 - mounting places; 4 - thermoelectric battery Peletier; 5- heat conductive line; 6 - resistance "radiator - mounting place"; 7 - radiators; 8 - resistance between center point and radiator; 9 - united center point; Q_s, Q_f - external light disturbances from Sun and other sources

temperature is absent in exploitation conditions. Such system operates like active with expenditure of external electrical energy.

Passive radiative thermocontrol systems use low temperature of environment and do not require for its work supplementary energy. The principal scheme of such system is given on figure 1,b. In this case heat energy (Q_{mp}+Q_d) from light receiver is removed by heat conductor with the thermal resistance R_c of any type onward low temperature radiator(s) and is scattered to surrounding space.

In this scheme of the most difficulties could arise with the achievement of low temperature of radiator(s) (170...240) K taking into account the external light disturbances (solar flux, planet fluxes, reflection from surrounding equipment), heat leakage to it from through mechanical support and isolation.

The ambition to minimize system requires to reduce to minimum thermal resistance thermoconducting line by usage of heat pipe (R_c less (0.3...1)K/W) to develop all contact resistances, to improve efficiency of radiator by optimizing radiator thickness and material optical and thermophysical properties. Designing the cooling systems it is obligatory to take into account that connection of matrix (or receiver head) with heat transfer line has to have mounting flexibility to provide the adjustment of optic axis and to avoid the excessive mechanical loads.

It is possible to combine principle of operation of both above mentioned systems. In second type system the insertion of thermoelectric battery (with (1...3) stages) between matrix and heat conducting line can assist in matter of ground test (to get required matrix temperature without source of low temperature but with expenditure of electrical power). Thermoelectric battery may be designed like dismountable element or like indivisible part of receiver which will not operate in space exploitation (will be like additional thermal resistance).

COMPARISON OF SOLID DIATHERMAL FINGER AND HEAT PIPE (HP)

Simplest solution to transport the heat released to radiator is solid finger made from materials presented in table 1. Thermal resistance of finger was proposed

Comparison of heat-conducting fingers and heat pipes

Table 1

Finger materials				Distance "device - radiator" or HP adiabatic zone length							
Name of material	Properties			0.05 m		0.1 m		0.25 m		0.5 m	
	Density ρ kg/m³	Heat conductivity, λ W/(mK)	Heat capacity J/(kgK)	Diam. of finger, mm	Mass of finger kg	Diam. of finger, mm	Mass of finger kg	Diam. of finger, mm	Mass of finger kg	Diam. of finger, mm	Mass of finger kg
Cooper	8920	410	330	12.5	0.05	17.6	0.22	27.8	1.36	39.5	5.45
Aluminium	2700	220	500	17.0	0.03	24.1	0.12	38.0	0.77	53.8	3.10
Berilium	1840	194	850	18.1	0.02	25.6	0.10	40.5	0.59	57.3	2.40
Silver	10490	420	230	12.3	0.06	17.4	0.25	27.5	1.56	38.9	6.25
Heat pipe	SS + metal fibres CS wall thickness 0.5 mm			12.0	0.04	12.0	0.05	12.0	0.08	12.0	0.13
Heat pipe	Extrusion aluminium profile wall thickness 2.0 mm			12.0	0.04	12.0	0.05	12.0	0.08	12.0	0.14

to be same and to equal 1 K/W, temperature level -90° C. One can find that cross section area and mass of solid finger grow extremely with respect of finger length. At distance more (0.2...0.3) m mass of finger made best heat conductive material became more 0.6 kg. So the mass restriction has imposed the reasonable limits to distance between sensor and radiator. To solve the problem of radiator accommodation in spacecraft zone with lowest external light fluxes one can use heat pipe which has essentially smaller mass. This comparison is placed in table 1 as well. Here it was assumed that heat pipe has length of evaporator 0.02 m, condenser 0.1 m, heat transfer coefficients in zones are 2000 W/(m^{2}K).

The heat pipe usage as long connection line (1...3) m is attractive technical solution for better radiator arrangement. Nevertheless this one has rocks like additional heat leakage from low conductance support structures and multilayer isolation to heat pipe body. These factors can reduce or even bring to nothing the displacement of radiator to new zone of spacecraft. Simple balance equation which shows thermal acceptableness of long heat pipe line could be expressed like next:

$$(Q_s + Q_f)_B >> (Q_{mp} + Q_{mli})_{HP} + (Q_s + Q_f)_N, \quad (1)$$

where $(Q_s + Q_f)$ - heat absorbed by radiator from light disturbances for basic (index B) and new (index N) radiator positions; $(Q_{mp} + Q_{mli})_{HP}$ - heat inputs to HP. Here it was assumed that value Q_d, radiator temperature, leakage to radiator are the same in both radiator positions.

HEAT PIPE UTILIZATION FOR RADIATOR DESIGN IMPROVEMENT

Connection of device cooled or heat conducting line with low temperature radiator could be imagined like point with sizes essentially smaller than required radiator surface. So heat inputted have to be spread along radiator by conductance of material or other manner. Sample of radiator temperature distribution is presented on figure 2.

Heat (3 W) is inputted in center, contacting area 0.01 x 0.01 m. Radiator made aluminum alloy (λ ~120 W/(mK)) has thickness 0.0005 m. There is no external background fluxes. Temperature drop along axis X for case I is 15 K. Embedding of single heat pipe (case II) has given reducing of heated zone temperature on 12K, of two crosswise heat pipes-15K. Effect of heat pipe utilization will be more evident at background heat fluxes when geometric sizes of radiator will increase. If several heat transported lines "devices - radiator" (several devices work on single radiator) have contact with radiator in its different zones thermal coupling of these zones by HP will be helpful to ensure equal basis temperature of every heat canal.

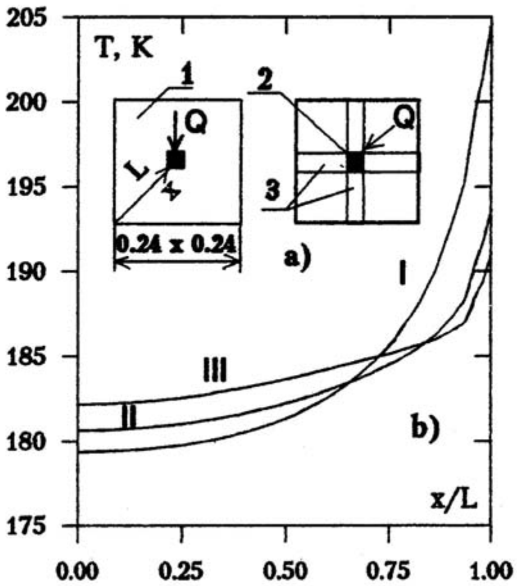

Figure 2: Radiator temperature distributions; a - locaion of heat input and heat pipes; b - temperature along axis X; 1 - radiator, 2 - heated zone, 3 - heat pipe(s). I - local heat input, no heat pipe, II - local heat input, one heat pipe embedded; III - local heat input, two heat pipes embedded

HEAT PIPES FOR TEMPERATURE LEVEL (190 ... 240) K

The survey of low temperature (cryogenic) heat pipes design and test results are presented in [1], recent data in [2] [3]. The most used constructions are aluminum extrusion profile (round, rectangular or special) with axial grooves, stainless steel shell with mesh capillary systems. Potential heat carriers and their properties are presented in table 2 (see appendix). Level of achieved by HPs heat productivity is from units (W·m) for mesh wicks (working fluid - C_2H_6, HP diameters 3...6 mm) up to 35 W·m for axial grooves (working fluid - C_2H_6, HP diameter 14 mm) at temperature level (180...200) K. HP length can reach (2...3) m. Information concerning to HP thermal characteristics like limitation of heat flux density in evaporator, heat transfer coefficient are in zones practically unnoticed.

Problems associated with HP on temperature level (180...240) K could be listed according [1]. As to practical HP utilization there are some important moment: (1) thermal test in ground condition; (2) startup behavior. First one requires of precise HP orientation in gravity forces, exact estimation of all leakages to heat pipe. Second one deals with strong function of parameter N (other name Figure of Merit) with temperature. Starting cooling from temperature level 290 K some of heat carriers have parameter N in (2...10) times less compare with operating temperature level, same are in supercritical state. Heat pipe connects two objects like device and radiator with different thermal capacities. Sometimes the nonidentical rates of radiator cooling and device from initial condition can generate essential temperature difference and correspondingly enough large heat transfer between these bodies. If heat pipe at initial temperature level has scarcity in heat productivity the limitation in functioning occurs. Radiator and device will be cooled like bodies coupling with great thermal resistance. Cooling of device can stretch. This effect has to be estimated at HPs utilization in real systems.

PASSIVE TCS DESIGN WITH HEAT PIPE UTILIZATION FOR HEAT TRANSFERRING

Such system was actualized for the cooling of CCD matrixes of receivers of two television cameras of interplanetary probes "VEGA-1" and "VEGA-2" of the International Project "Venus - Halley" for TV observation of comet Halley. General external cooling system diagram is presented on figure 3.

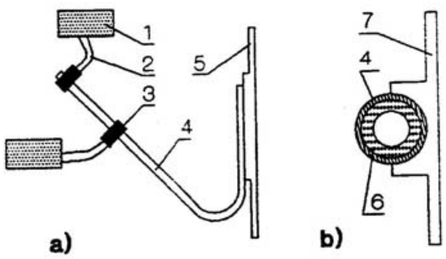

Figure 3: Scheme of TCS for "VEGA" TV cameras: a - principal thermal scheme, b - heat pipe cross section; 1 - receiver cooled, 2 - flexible heat-conducting line, 3 - detachable thermal interface, 4 - heat pipe shell, 5 - radiator, 6 - capillary structure made metal sintered fibers, 7 - flange of condenser zone

Every heat pipe serves two light receivers. Heat released by CCD matrix itself and accumulated by structural elements from the outside is removed with the help of a flexible copper heat-conducting line to heat pipe and transfer onward on the low temperature radiator (F_r = 0.0535 m^2; α_s=0.26, ε = 0.85). To compensate the direct solar rays influence occurring in some exploitation regimes the phase change accumulators were used, embedded to radiator body.

Basing on TCS arrangement specifications the heat pipes have cylindrical casing with outer diameter 8 mm. The pipe shell was bent by the radius 48 mm with the angle equals to 80 degrees. On one straight section of HP it is situated the heating zone (one or two detachable thermal interfaces, 16 mm in length), on another section the condensation zone is (flange, 125 mm in length). Selection of material casing, of capillary structure and heat carrier is attributed by the requirements of operating temperature level (-80...-40) °C, heat flow transferred over (up to 4 W) and long operation time. At such requirements the outgassing inside HP is inadmissible, that is why for flying specimen it was chosen such combination of materials: casing and metal-fibrous capillary structure (MFCS) made from stainless steel 12X18H9T, heat carrier - refrigerant R-22 (T_{mp} = -160 °C). Computation of capillary structure (CS) parameters were carried out with the help of methodology [4]. CS had following parameters: fibers diameter is 32 micrometers, length of a filament is 3 mm, porosity of porous media with annular configuration is 80 % and its thickness is 1 mm. Porous media has been made by sintering process in vacuum furnace.

In some ground thermovacuum tests of TCS in assembly with TV systems on the whole, when it was obligatory to provide HPs operating against gravity forces (heat has to be lifted on altitude up to 250 mm), additional heat pipes with ammonium were designed and used (ammonia owns by higher capillary - transport properties). On figure 4 the comparison of calculated heat - transferred capacities for HPs with ammonia and refrigerant R-22 depending on the required altitude of the heat-carrier uplift is presented.

Evidently HP with refrigerant R - 22 can run only in horizontal position or with insignificant deviation from it. Computation was conducted as to dependence [4]

$$Q_{max}=4 \cdot N \cdot (1 - \Delta P_h/\Delta P_{cs}) \cdot K_{cs} \cdot F_{cs}/D_{ef}/L_{ef}, \qquad (2)$$

where Q_{max} - heat flow transferred over by HP, W; $N = \sigma r/\nu_l$ - parameter of working fluid merit, W/m^2; $\Delta P_h/\Delta P_{cs}$ - ratio of a pressure loss with fluid uplift in capillary structure on required altitude to the stated capillary head, D_{ef} - effective diameter of CS pores, m; L_{ef} -effective length of the heat transport, m K_{cs}, F_{cs} - penetrability and area of CS cross section, m^2.

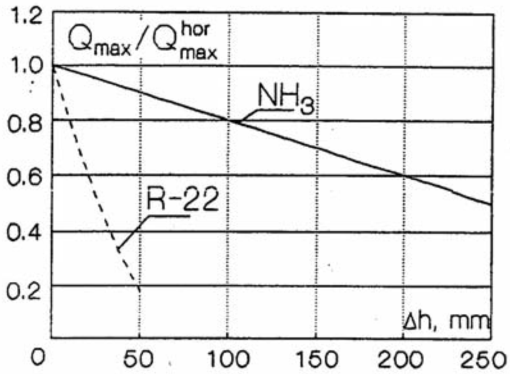

Figure 4: Comparison of HP heat productivity with R-22 and ammonia like working fluids at operation against gravity forces. Temperature level -40 °C

Except these, under development and optimization of TCS architecture the analysis of the influence on the system temperature regime of various exploitation factors was fulfilled, for example, temperature change of mounting places, thermal resistance of flexible line and radiator supports, heat-capacity of radiator (in unsteady regimes) etc. [5][6].

It was carried out the complete cycle of ground tests for particular heat pipes and TCS on the whole, including thermal-vacuum (with heat releasing on screen cooled by liquid nitrogen).

On figure 5 it is presented values of heat-transfer coefficients in evaporator zones inside HPs of various their modifications obtained in experiment (a) and results of HP life-testing (b).

Intensity of heat transfer in evaporator for NH_3 HPs is higher compare with R-22 HPs in whole range of heat flux density. Dependence of heat transfer coefficients on flux density is explained by evaporation regime existence at densities up to 2000 W/m^2 and boiling at larger density. Figure 5,b illustrates results of long life test for R-22 HPs in form of comparison current (measured in time) thermal resistance with initial. Deviation of this function in time deals with experimental errors (identity of test conditions, instrumental errors, etc.).

Except stationary characteristics it was investigated the transient temperature behavior of the system at the changing of outer conditions on some sections of flight program. The most heat load stressed the section on the rapprochement stage with the comet Halley is, when radiator is absorbed direct solar radiation causing growing of it temperature. Figure 6 shows temperatures of the light receiver thermal interfaces with flexible line T_{dev}, HP thermal interface "flexible line - heat pipe" T_{hp} and radiator

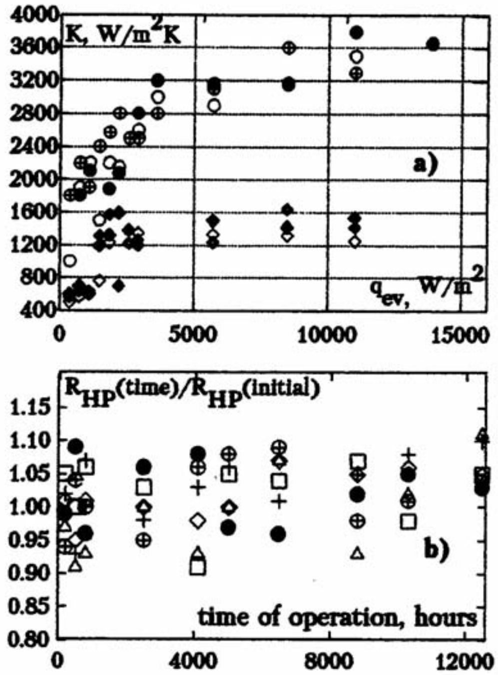

Figure 5: Heat transfer coefficients in evaporator zone of HPs with refrigerant R-22 (rectangular symbols) and ammonia (circle symbols) (a) and long life test results of refrigerant R-22 heat pipes (b). Symbols define design of heat pipe. R_{HP} -HP thermal resistance

T_{rad} have gotten in a thermal-vacuum chamber and results of the telemetry in-flying-obtained.

EMPLOYMENT OF HEAT PIPES FOR IMPROVEMENT OF THE RADIATOR EFFECTIVENESS

In development of TCS for the videospectrometric complex (VSC) of the project "Fobos" (Interplanetary mission to Mars and TV observation of Mars' satellite Fobos) the general object arrangement lets to situate a radiator in immediate vicinity from the light receivers and to remove the heat directly by flexible copper bus-bars [7].

However, mounting places of four bus-bars from four receivers are located non-uniformly as to the radiator rear face. These brings to temperature stratification between particular receivers that impedes its tuning and calibration, and also decreases the radiator effectiveness.

Rising of isothermality of radiator by means of the augmentation of its thickness brings to the significant

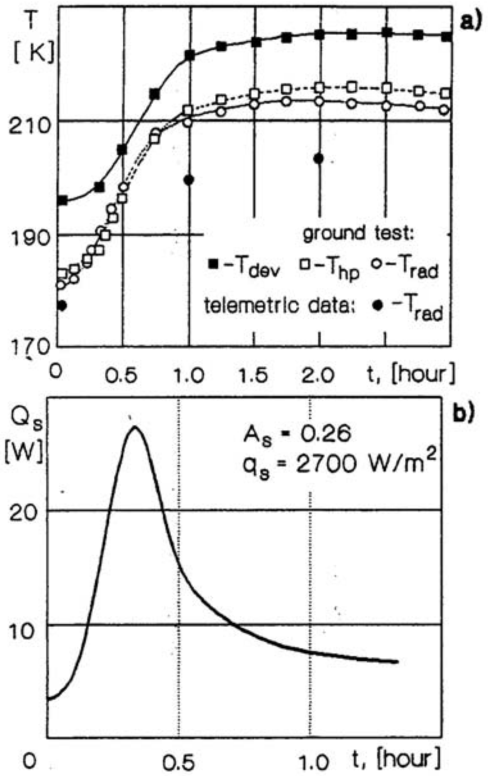

Figure 6: Experimental simulation of flying section near comet Halley: a - temperatures of TCS elements obtained in ground test and flying telemetry; b - assumed solar heat flux absorbed

mass of TCS. That is why in this TCS architecture for the equalization of the radiator temperatures the heat pipe was used, which transferred a heat with low thermal resistance from more hot radiator part (with three heat-sinks) to more cold one.

Comparison of temperature fields for TCS radiator with heat pipe and without it is presented on figure 7.

Computations of temperatures are accomplished by means of program package "HEAT-90" [8] actualizing the method of finite elements.

Employment of heat pipe allowed to lower the temperature stratification between detectors from 14 up to 1.5 degrees, in this case the mean temperature of radiator was reduced on 4 degree that was equivalent 8 % augmentation of the radiator area. In real design HPs made from stainless steel with metal fiber sintered capillary structure and refrigerant R-22 as heat-carrier were used. As to the information of flying telemetry the radiator temperature did not exceed -80 °C up to the moment of the communication termination with probes that well conforms with calculated data.

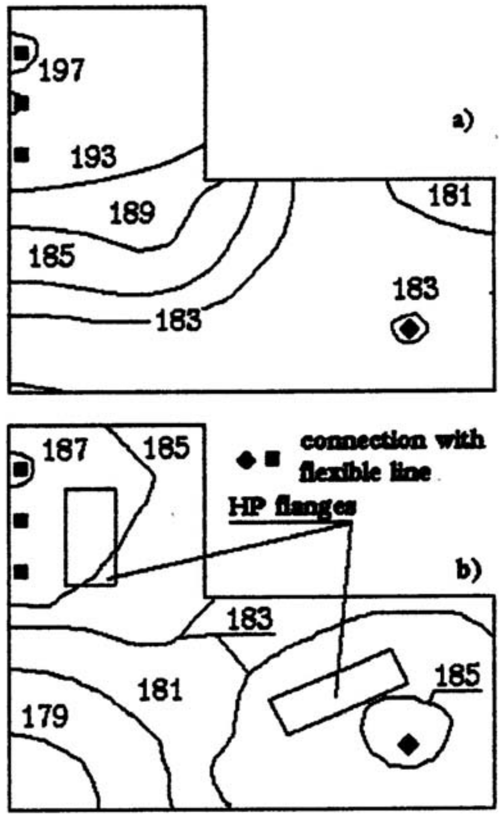

Figure 7: Temperature pattern in radiator: a - without heat pipe, b - with heat pipe embedded

UTILIZATION OF HP WITH VARIABLE THERMAL RESISTANCE

Major restriction of passive radiative systems on the way to achieve temperature level (190...200) K is the effect of background fluxes from ambient spacecraft elements, planets etc., and this limitation could be reflected in next balance equation:

$$Q_{dev} + Q_f < \varepsilon\sigma F_r \eta (T_{r,max}^4 - T_o^4), \quad (3)$$

where $T_{r,max}$ - highest feasible radiator temperature, η - efficiency of radiator; Q_f - sum of background fluxes absorbed by radiator, Q_{dev} - heat removed from devices, F_r - radiator surface area.

The TCS with several radiators variously oriented with reference to the Sun is promising. In such case one radiator will always reside in favorable conditions and provide heat removal on demanded temperature level.

Analysis of such TCS with 2-3 radiators was given in a work [9]. It was shown that such systems can be effective at utilization of variable thermal resistance heat pipes like thermodiodes (TD) which have significant thermal resistance in reversal regime (from 'hot' (exposed on Sun) radiator to a cold device). Simulation of thermal behavior has shown that this TCS principle can be useful at solar constant range up to 2700 W/m^2.

For OSS (Optical Star Sensor) Project it was developed TD system with two opposite oriented radiators (figure 8).

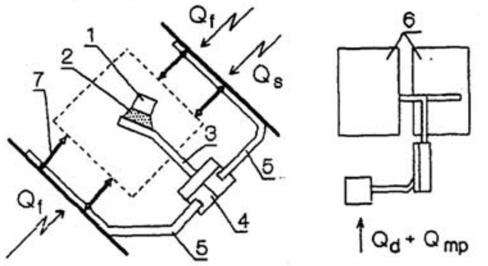

Figure 8: Scheme of TCS with two opposite oriented radiators: 1 - device cooled, 2 - flexible heat-conducting element, 3 - heat pipe, 4 - central united point, 5 - thermodiodes, 6 - radiators, 7 - thermoisolating supports

In this system heat from cooled element was removed by constant conductance HP to central junction and onward by TDs on radiators. Thermodiode construction used principle of liquid trap laid in the evaporative zone (placed in central united point). Material of HP casing - stainless steel 12X18H9T, capillary structure - same SS metal sintered fibers, heat carrier refrigerant R-22.

In the ground tests it was included the exploration of thermodiodes' operating in reversal regime. On figure 9 it is shown time transformation of TD thermal resistance and of value of heat flow backward transferred.

In initial moment the TD runs as conventional heat pipe with the alternate heat-carrier mass. After accumulation of major mass of working liquid in liquid trap the complete drying of the radiator zone and adiabatic zone occurs and heat transferred is decreased essentially. For chosen conditions computation displays that TD resistance grows up to

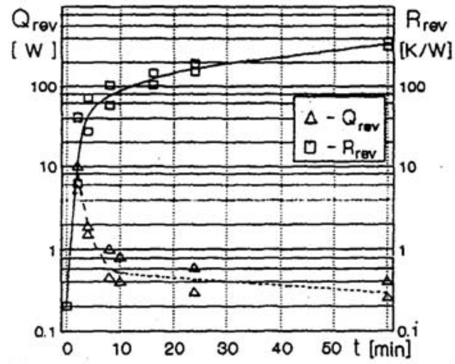

Figure 9: Thermal resistance and reverse heat flow for thermodiode in reverse regime

values 100 K/W in first 10 minutes that well agrees with experiment.

Analyzing the common operation of two thermodiodes as components of TCS, it should be noted that when system will change orientation of radiator respectively Sun there is time range, when one canal begins to close, and second one yet is closed, because this radiator has not cooled less temperature of central point.

In this case the heat coming from devices and radiators goes to the junction point of heat pipes. This has the important consequence: heat-capacity of central junction must be highest (as far as possible), and heat-capacities of radiators must be minimal for assurance of their fast cooling up.

On figure 10 results of thermalvacuum tests of TCS with two radiators at the imitation of the action of external radiated fluxes are presented. Specified values Q_s and Q_f correspond to values of heat absorbing by the radiator from solar and background fluxes (for orbits with q_s=1400 W/m^2).

Carried out ground experimental explorations justified computations and sustained the outlook of TD employment in TCS with several radiators. In addition, it was developed a variant of combined TCS for the case of near-Earth orbits. Temperature of the sensible element of optical-electronic device is maintained at the level -40 °C by the thermoelectric battery, and released heat from its 'hot junction' is removed by the heat pipe on the centraljunction and onward by TD on radiators. System retains its productivity at values of the total external flux absorbed by radiators up to 335 W/m^2 for open canal and up to 602 W/m^2 for enclosed canal. Capacity of thermoelectric battery is 3 W.

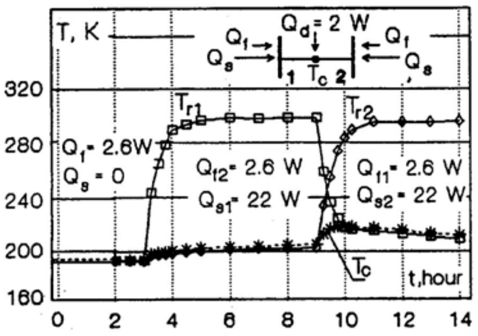

Figure 10: Transient TCS behavior at device heat dissipation Q_d = 2 W. Indexes 1, 2 define number of radiator, T_r , T_c - temperature of radiator and central point

SUMMARY

Principles of passive radiative TCS design for optical-electronic devices of the broad spectrum of space objects are presented. High-performance heat - transferred devices - the heat pipes with metal-fibrous capillary structure. are analyzed to improve TCS thermal characteristics and give them a new possibilities. These pipes steadily run in the range of temperatures (-80...+30) °C with usage refrigerant R-22 (and ammonia as well) as heat carrier. Capillary transport characteristics of pipes with MFCS with usage of ammonia as heat carrier have assisted at the arranging of ground tests of scientific apparatus in assembly when heat has to be lifted against gravity forces.

Heat transfer coefficients in the evaporative and condensation zones are enough high (degree (2000...4000) W/m^2).

Principal tasks have decided by the applying of heat pipes:

a) heat transfer with low thermal resistance from a device to the radiator;

b) equalization of the radiator temperature field for the rising of its effectiveness and providing of basic temperature level for several devices;

c) organization of directional heat fluxes in multi - radiators systems with TD employment and building of TCS with smaller sensitivity to external disturbances.

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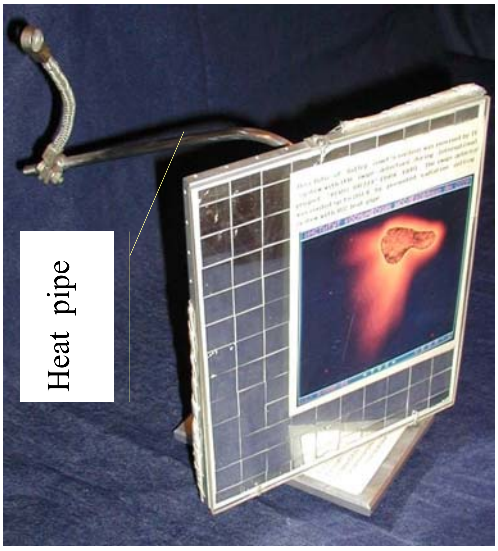
APPENDIX

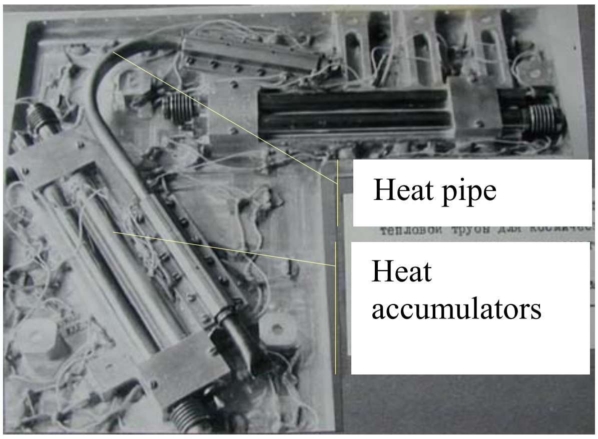
Potential working liquids for low temperature heat pipes

Table 2

Name of liquid	Temperature T, K	Vapour pressure P_s,MPa	Density liquid ρ_l, kg/m^3	Density vapour ρ_v, kg/m^3	Surface tension, σ $N/m \cdot 10^3$	Heat of evapor., r, KJ/kg	Parameter N, $W/m^2 \cdot 10^{-9}$
Propane	290	0.768	505	16.6	7.83	349.3	11.8
C_3H_8 T_{mp} = 85 K	240	0.147	571	3.4	11.9	415.9	17.9
T_{np} = 232 K	190	0.011	627	0.29	21.96	465.6	19.0
Propylene	290	0.926	507	19.1	7.52	355.0	14.1
C_3H_6 T_{mp} = 88 K	240	0.186	571	3.9	14.37	418.0	26.0
T_{np} = 225K	190	0.014	624	0.27	22.0	470.7	21.5
Ethane	290	3.51	365	77.1	1.16	229.8	2.1
C_2H_6 T_{mp} = 90 K	240	0.967	475	17.6	7.26	394.3	16.2
T_{np} = 184 K	190	0.13	541	2.64	14.9	481.5	29.6
Ethylen	290	—	—	—	—	—	—
C_2H_4 T_{mp} = 104 K	240	1.77	461	33.3	4.75	337.0	9.2
T_{np} = 170 K	190	0.29	555	6.02	12.72	449.6	24.7
Chlorodifluoromethane	290	0.84	1219	36.4	9.2	189.9	8.7
CHF_2Cl (R22)	240	0.14	1386	6.56	17.1	228.4	16.3
T_{mp} = 113 K T_{np} = 170 K	190	0.008	1525	0.45	25.5	259.2	13.2
Ammonia	290	0.78	615	6.1	21.8	1197.8	65.3
NH_3 T_{mp} = 195 K	240	0.1	682	0.9	34.1	1368.5	102.6
T_{np} = 240 K	213	0.03	714	0.03	40.6	1434.0	116.0

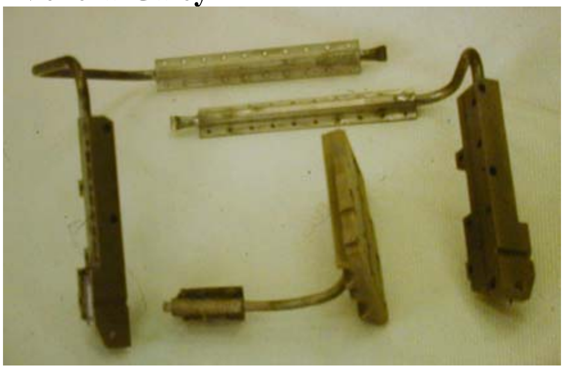
T_{mp} - temperature of melting point;
T_{np} - temperature of saturated vapour at standart pressure

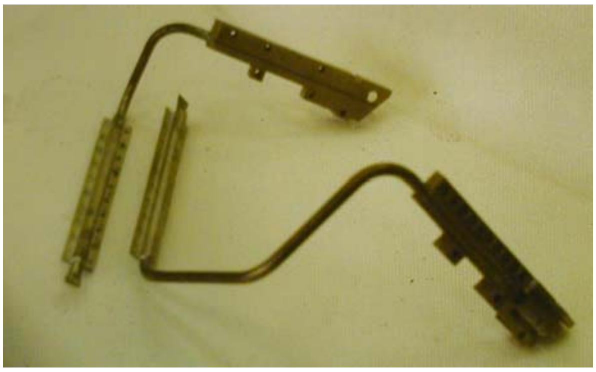




Radiative cooling system for mission "Venera-Galey"

Radiative cooling system for project Phobos





Different types of thermodiodes. Courtesy of NTUU "KPI", Ukraine