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# Sorption heat pipe—a new thermal control device for space and ground application

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**Abstract**

Sorption heat pipe (SHP) combines the enhanced heat and mass transfer in conventional heat pipes with sorption phenomena in the sorbent bed. SHP consists of a sorbent system (adsorber/desorber and evaporator) at one end and a condenser + evaporator at the other end. It can be used as a cooler/heater and be cooled and heated as a heat pipe. SHP is suggested for space and ground application, because it is insensitive to some "*g*" acceleration. This device can be composed of a loop heat pipe (LHP), or capillary pumped loop (CPL) and a solid sorption cooler. The most essential feature is that LHP and SHP have the same evaporator, but are working alternatively out of phase. SHP can be applied as a cryogenic cooler, or as a fluid storage canister. When it is used for cryogenic thermal control of a spacecraft on the orbit (cold plate for infrared observation of the earth, or space), or efficient electronic components cooling device (lased diode), it is considered as a cooler. When it is applied as a cryogenic storage system, it insures the low pressure of cryogenic fluid inside the sorbent material at room temperature.

*Keywords:* Heat transfer; Evaporation; Sorption; Loop heat pipe; Sorption heat pipe

## 1. Introduction

Sorption heat pipe (SHP) is beneficial for the power electronic components cooling (IGBT, thyristors, etc.) especially for transport application, high power electronic component cooling (laser diodes) and for the space two-phase thermal control systems. SHP is a combination of a heat pipe and solid sorption cooler with some specific interaction between these elements. This device is based on the enhanced heat and mass transfer in conventional heat pipes with sorption phenomena of sorbent bed inside it [1]. Now the electronic component cooling is a key problem in the industry and space application [2]. The major problems associated with cryocoolers are reliability, efficiency, vibration, size and mass, electromagnetic interference, heat rejection, and cost. Besides classical techniques of cryogenic refrigeration by dumping in cryogenic fluid (nitrogen, hydrogen, or liquid helium), magnetic refrigeration, Vuilleumier machine, Stirling and pulsed tubes actually many attempts to apply sorption coolers have been carried out to reduce the constraints of less vibration, low power consumption, reliability and long-term life [3,4]. This problem can be solved successfully with a coupled use

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## Nomenclature

| | |
|---|---|
| $a$ | sorption capacity (g/g) |
| $c$ | specific heat (J/kg K) |
| g | gravitational acceleration ($m^2/s$) |
| $G$ | flow rate per unit area or mass flux ($kg\ m^{-2}\ s^{-1}$) |
| $h_v$ | volumetric heat transfer coefficient ($W/m^3\ K$) |
| $H$ | enthalpy (J/kg) |
| $L$ | latent heat ($J\ mol^{-1}$) |
| $P$ | pressure (MPa) |
| $r$ | capillary radius of the wick ($m^{-6}$) |
| $R$ | thermal resistance (K/W) |
| $S$ | surface ($m^2$) |
| $T$ | temperature (K) |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | heat transfer coefficient ($W/m^2K$) |
| $\delta$ | thickness ($m^{-6}$) |
| $\theta$ | angel of wetting (°) |
| $\lambda$ | thermal conductivity (W/mK) |
| $\rho$ | density of working fluid ($kg/m^3$) |
| $\sigma$ | coefficient of surface tension (H/m) |
| $\tau$ | time (s) |

*Subscripts*

| | |
|---|---|
| c | capillary |
| eff | effective |
| max | maximal |
| l | liquid |
| v | vapor |
| w | wick |
| wl | wall |

of heat pipe and solid sorption cryo-coolers, based on sorption heat pipe phenomena. Such cryo-cooler ensures a deep cooling of space sensors down to the triple point of the hydrogen. A good example of its application is for lunar missions where during the day, conventional unshaded space radiators would have to look at either the sun, or the hot lunar surface, making difficult heat rejection directly to the environment. Another example is for cooling of electronic components during the lunar or Martian day, when the temperature of the surrounding could be higher than 50 °C or even more. SHP is very attractive for cooling IGBT modules widely used for railway train application. For such cases it is necessary to dissipate high heat flux, releasing from small heat loaded surface and to use ammonia or water. Ammonia is not comfortable as a working fluid for transport applications and has low latent heat of evaporation and liquid tension to compare with water. Water is not convenient also, if the system is stored at the temperature below 0 °C. By using SHP we can keep the fluid (adsorbed water) at the temperature below 0 °C. SHP includes the advantages of conventional heat pipes and sorption machines in one unit. Its major advantage to compare with conventional sorption machines is convective (two-phase flow) mode of the heating/cooling of sorbent structure inside SHP. The same working fluid is used as a sorbate and heat transfer media. SHP has a sorbent bed (adsorber/desorber and evaporator) at one end and a condenser + evaporator at the other end (Figs. 1 and 2) [5]. This device is working in such a way: phase 1, at the beginning it is necessary to desorb a sorption structure (2), Fig. 1, applying a heater. During desorption the vapor (1) of a working fluid is releasing from a porous media (2) and condensing in the evaporator/condenser (3). The part of the liquid due to the pressure drop between the hot part of a heat pipe and the evaporator is filtrated through the porous valve (5) and enters the evaporator (6). The other part of the liquid is going back to the sorbent bed due to capillary forces of the wick (4) and is used to enhance the heat and mass transfer inside the sorbent material (micro-heat pipe phenomena). When the desorption procedure of the sorbent structure is stopped, the heater needs to be switched off, the working fluid is accumulated inside the evaporator and the pressure in the sorbent bed is decreasing due to its cooling.

Phase 2. After the phase 1 accomplishment the porous valve (5) is opened and the vapor pressure inside the heat pipe is equalising following the procedure of the liquid evaporation inside the porous structure of the evaporator (6). The schematic of sorption heat pipe evaporator is shown in Figs. 1 and 3. During the procedure of the liquid (7) evaporation, the air inside the cold box, (8) is cooling. When the liquid evaporation inside the evaporator is finished and the sorbent bed is saturated with the vapor, a porous valve is closed and then the sorbent bed starts to cool down to the ambient temperature with the help of the heat pipe condenser (3). The phase 2 is finished.

In this paper we consider SHP as a combination of a loop heat pipe (LHP) and ammonia/(active carbon fiber + chemicals) solid sorption cooler. Such system extends the limits of two-phase thermal control and ensures successful mode of electronic components cooling even in very harsh environmental conditions (ambient temperature 40 °C, or more). The conventional two-phase thermal control system for space application is sensitive to the vehicle acceleration and vibration (spacecraft

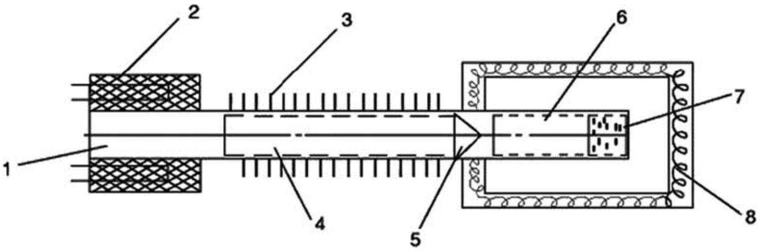

Fig. 1. Sorption heat pipe. (1) vapor channel; (2) porous sorption structure; (3) finned surface of heat pipe evaporator/condenser; (4) porous wick inside heat pipe; (5) porous valve; (6) heat pipe low temperature evaporator with porous wick; (7) working fluid accumulated inside the evaporator; (8) cold box.

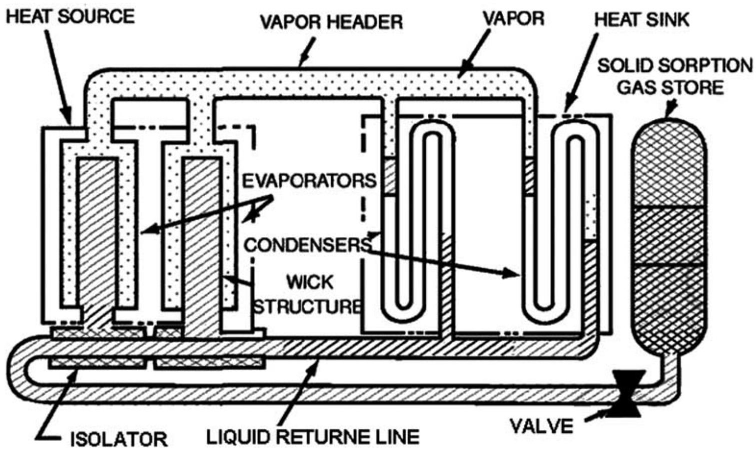

Fig. 2. The schematic of sorption heat pipe.

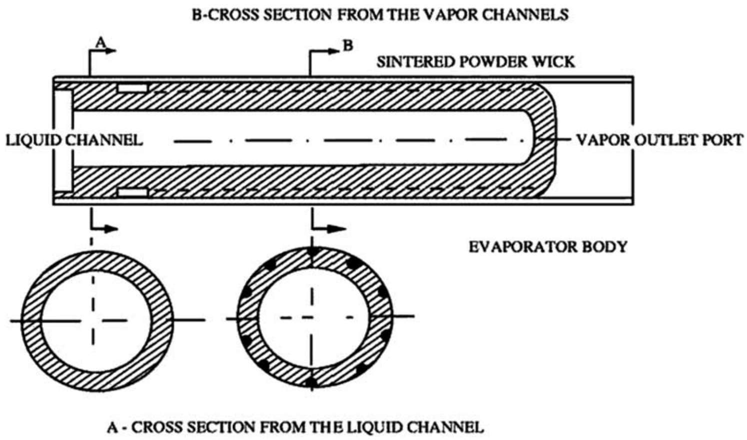

Fig. 3. Porous evaporator, used for SHP and LHP.

launching time, change the altitude of the orbit, etc.). SHP avoids this inconvenience. The active mode of cooling/heating of the electronic components can be realised using SHP integrated directly to the two-phase heat transport system. The solid sorption cooler of SHP begins to function (to be switched on), when the cooling possibilities of an ordinary heat pipe are exhausted (for example, the condenser is damaged, or covered by some insulation media, the special situation with LHP during the space vehicle launch to the orbit). Actually a lot of low temperature heat pipes are used for space applications (as a cold plate for infrared observation of the earth or space), or effective electronic components cooling. It is known, that conventional LHP seems to be very promising heat transfer devices [6–8] for space applications in a large temperature range from 60 K up to 400 K. Cryogenic LHP (hydrogen, oxygen, hydrocarbons and nitrogen) needs to be protected against super pressure influence at room temperatures [9–13]. The combination LHP with solid sorption gas storage canister (SHP) can be useful to solve this problem. There are some problems with the start of LHP at cryogenic temperatures from supercritical state. Solid sorption gas storage device facilitates adsorption of cryogenic fluid at room temperature. When the LHP condenser is cooling and the solid sorption gas storage system is heating the working fluid, being desorbed from the sorbent bed, is condensing in condenser and is sucking by the porous wick of the LHP evaporator. Now SHP is ready to cool the electronic components (infrared sensor).

In SHP there are some basic phenomena interacting with each other:

1. A vapor flow (two-phase flow) with kinetic reaction rate and pressure, vapor pressure, geometry, conductive and convective heat transport with radial heat transfer inside the sorbent material.
2. In the SHP condenser and evaporator there is a vapor flow, liquid flow, interface position, radial heat transfer with kinetic reaction pressure, liquid pressure, vapor pressure, condensation and evaporation, shear stress, geometry, adhesion pressure, convective heat transport, radial heat transfer under the influence of the gravity field.

## 2. The experimental set-up

The experimental set-up is composed of two-phase loop, valves and compact solid sorption cooler with its system of thermal control, (Fig. 4). The core of this set-up is a capillary pumped evaporator with the inverted meniscus of the evaporation [6]. The evaporator design, Fig. 3 (wick structure, geometry, and the internal volumetric surface of pores, the surface of the liquid meniscus and the surface of liquid entrance, thermal conductivity, permeability, and etc.) is the dominating factor in SHP operation. In our experiments the heat pipe evaporator was made from Ti sintered powder as a compact cylindrical heat transfer device. This evaporator has a liquid accumulator inside it and a set of longitudinal and circumferential vapor channels (grooves) on the outer surface of the wick contacting with SS tube (heating element). The Ti wick was force-fitted within the stainless steel tube. Working liquid (Fig. 4) is entering from the heat pipe condenser to the liquid accumulator through the liquid subcooler. The liquid sub-cooler heat exchanger ensures the reliable operation of the LHP evaporator at the time of its transient heating. The radial liquid flow is going from the liquid accumulator through the wick to the hot wall, cools it and the vapor is moving along the grooves to the vapor outlet. The Ti wick structure has porosity 45%, the length −280 mm, the outer diameter −38 mm, maximum pore diameter −10 μm, medium pore diameter −3 to 5 μm, wick thickness −4 mm. The heat pipe condenser is made as a tube in tube heat exchanger. The heat output of the system is up to 900 W, the mean thermal resistance of the evaporator $R = 0.03$ K/W, pressure drop $\Delta P_c = 400$ mbar.

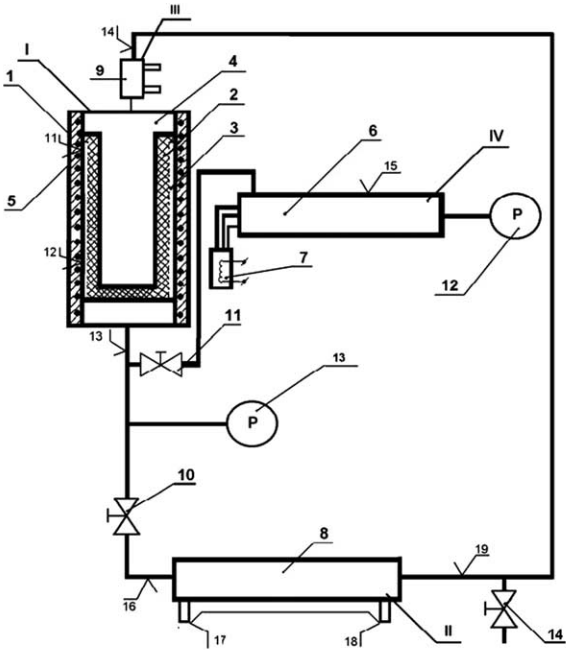

Fig. 4. Schematic of the experimental set-up. (I) heat pipe evaporator, (II) heat pipe condenser; (III) liquid subcooler; (IV) sorption canister: (1) heat pipe envelope; (2) capillary-porous wick; (3) vapor channels disposed along the heat pipe envelope; (4) liquid compensation chamber inside the porous wick; (5) electrical heater; (6) sorption canister; (7) sorption block thermal control system; (8) heat pipe condenser; (9) liquid subcooler heat exchanger; (10) valve; (11) regulated valve; (12) pressure sensor; (13) vacuum sensor; (14) valve for the fluid charging; (11–19) thermocouples.

This set-up is convenient to determine LHP and SHP parameters, due to its ability to join the evaporator alternatively to SHP, or LHP loop, (Fig. 4) during the experiment. When the valve 10 is opened and the valve 11 is closed this set-up is functioning as LHP. When the valve 10 is closed and the valve 11 is opened a set-up is functioning as SHP.

## 3. Analysis

The considered experimented set-up can be applied in two different ways. The first one is to use is it as a semiconductor sensor cooler with low heat dissipation to cool the sensor down to the ambient temperature. It is interesting to be applied in cryogenic range of temperatures. The second option is related with the cooler application for high energy dissipation devices (for example laser diode cooler). The first set of experiments were performed with SHP to demonstrate the basic possibility to decrease the temperature of the heat loaded wall to compare with the temperature of the same wall in the phase of LHP cooling mode.

In Figs. 5 and 6 the LHP evaporator heat transfer as a function of the heat load is shown. The most effective heat transfer coefficient (4500 W/m$^2$ K) is fixed for the heat flow $Q = 700$–$900$ W, when the maximal surface of evaporation inside the pores is activated. For LHP the maximum capillary pressure in the wick can be evaluated by the Laplace equation:

$$(P_c)_{\max} = 2\sigma/r_c, \tag{1}$$

where $\sigma$ is the surface tension and $r_c$ is the effective capillary radius of the wick.

In the real LHP design capillary pressure drop $\Delta P_c$ depends on some LHP parameters:

$$\Delta P_c \geqslant \Delta P_v + \Delta P_l + \Delta P_w + \Delta P_g, \tag{2}$$

where $\Delta P_v$ and $\Delta P_l$ are the pressure drop in the vapor and liquid lines, $\Delta P_w$ is the pressure drop in the wick pores and $\Delta P_g$ is the pressure drop due to the gravity field.

In SHP the pressure drop is equal to the vapor pressure difference between the evaporator and adsorber (Clausius–Clapeyron equation), Fig. 7:

$$\mathrm{d}\,\mathrm{Ln}P/\mathrm{d}(1/T) = -L/R, \text{ or } -\Delta H/R. \tag{3}$$

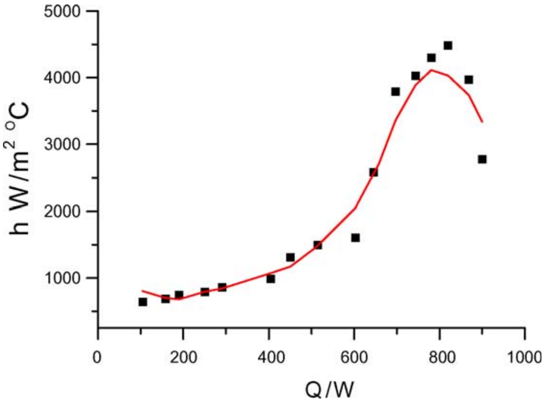

Fig. 5. Heat transfer between the hot wall of the LHP and the vapor at the evaporator exit versus a heat flow $Q$.

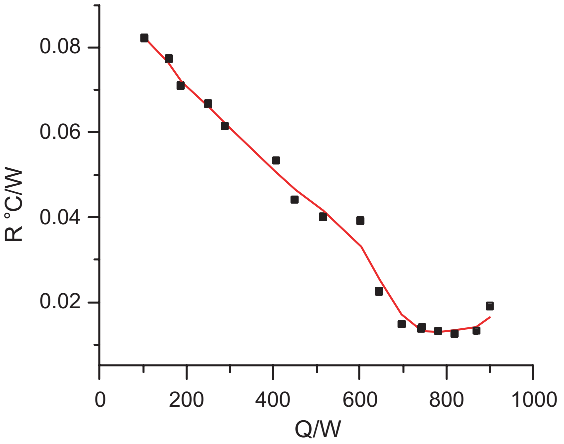

Fig. 6. LHP thermal resistance as a function of the heat flow $Q$.

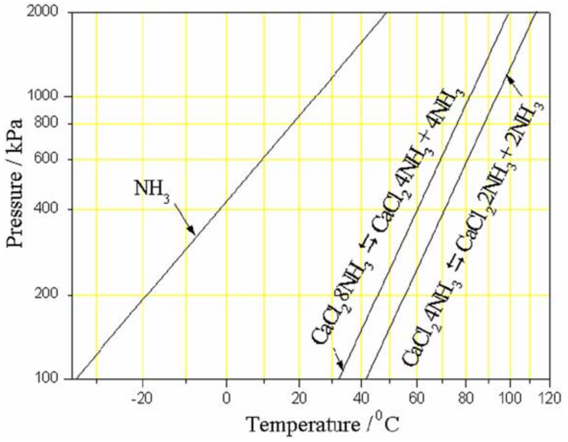

Fig. 7. Pressure as a function of the temperature for the ammonia and CaCl$_2$ saturated with ammonia.

For ammonia the pressure drop in SHP could be near 10 bars, it is 10 times more to compare with LHP. For LHP the external source of energy is the heat input to the evaporator and the heat sink is the condenser. For SHP the external source of energy is also the heat input to the evaporator, but the heat sink is the adsorber. The second external source of energy for SHP needs to be used periodically for the sorbent material regeneration. LHP can be used constantly, SHP needs to be used periodically (or at least two adsorbers, working out of phase). Naturally, such evaporator ought to be compatible with these two cooling systems.

SHP experimental set-up is composed of the evaporator, condenser and solid sorption cooler. By choosing a working fluid with a large heat of evaporation (water, ammonia) one may expect to absorb significant heat flux in the evaporator. The adsorber is filled with an active carbon fiber “Busofit”, saturated with CaCl$_2$ (in the mass ratio 70% of Busofit, 30% salt). Adsorbent increases the heat transfer intensity inside the evaporator due to the volumetric evaporation of the liquid in the pores under the heat load and pressure drop influence (some bars). In order to find the total sorption capacity of the sorbent material the Dubinin–Radushkevich equation is used:

$$a = (W_0/v)\exp\left(-BT^2[\lg(P_s/P)]^2\right) \tag{4}$$

where $a$ is the total sorption capacity, $W_0$ is the total volume of micro pores, $B$ is a structural constant, $P_s$, $P$ are the pressures of equilibrium and saturation. The experimental and calculated data of the sorption ammonia capacity for "Busofit + $CaCl_2$" are found as a function of pressure and temperature, Fig. 8. There is two-phase flow forced convection heat transfer of ammonia through the porous structure accompanied with an intense evaporator wall cooling.

To evaluate the heat transfer intensity inside the porous wick with two-phase fluid filtration through the porous media a volumetric coefficient of the heat transfer $h_v$(W/m$^3$ K) need to be determined. The energy absorbed by the cold fluid can be estimated by the number $h_v(T-t)$ (W/m$^3$), which represents the heat energy dissipation in the unit of the volume of porous structure per unit of time. The temperature of the porous structure in this elementary volume is "$T$", and the temperature of the fluid is "$t$". The temperature field inside the porous wall is determined as [11]

$$\lambda\frac{d^2T}{dZ^2} = Gc\frac{dt}{dZ} \tag{5}$$

When the heat flow dissipation in the porous layer is increasing, the temperature difference between the pore wall and fluid in the pore is also increasing. The temperature field inside the porous wall needs to be determined as

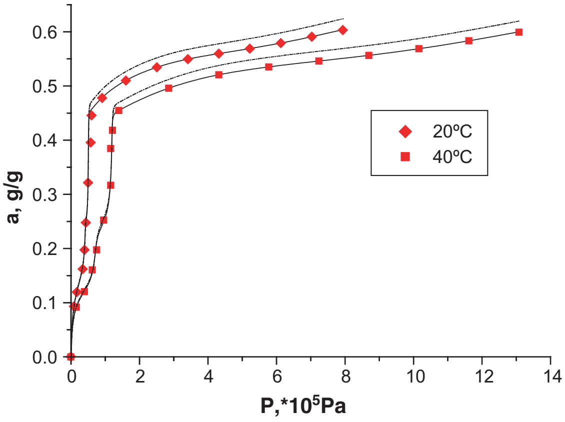

Fig. 8. Experimental and calculated ammonia isotherms of "Busofit + $CaCl_2$". Dashed lines are the calculated data at different temperature of equilibrium ($1 - T_{eqi} = 20$ °C, $2 - T_{eqi} = 40$ °C).

$$\lambda\frac{d^2T}{dZ^2} = h_v(T-t) \tag{6}$$

$$Gc\frac{dt}{dZ} = h_v(T-t). \tag{7}$$

The set of equations can be transformed into

$$\frac{d^2T}{dz^3} + A\frac{d^2T}{dz^2} - AB\frac{dT}{dz} = 0 \tag{8}$$

$$T = t + \frac{1}{A}\frac{dt}{dz} \tag{9}$$

Here some non-dimensional terms are applied:

$$Z = Z/\delta; A = h_v\delta/Gc; B = G\delta c/\lambda;$$

$$D_{1,2} = \frac{A}{z}[-1 \pm (1+4B/A)^{1/2}]. \tag{10}$$

Non-dimensional number B characterises the ratio of the heat absorbed by the cold fluid and the heat transferred through the porous media due to its thermal conductivity.

$\delta$ is the wall thickness. Eq. (5) now can be presented as

$$\frac{d^2T}{dz^2} - B\frac{dt}{dz} = 0. \tag{11}$$

The solution of set of Eqs. (6), (7)

$$t = C_1 + C_2\exp D_1z + C_3\exp D_2z \tag{12}$$

$$T = C_1 + C_2(1 - D_1/A)\exp D_1z + C_3(1 + D_2/A)\exp D_2z. \tag{13}$$

The boundary condition for the case when the fluid flow with initial temperature $t_0$is filtered through the porous media is

$$\text{At } Z = 0\ \lambda dT/dZ = Gc(t_1 - t_0) \tag{14}$$

On the other hand the porous media (coating) on the inner surface of the evaporator can be considered as mini-fins and there is a possibility to determine the heat transfer efficiency between the fluid and porous structure as

$$Q = \alpha S(T_{wl} - t_v), \tag{15}$$

where $\alpha$ is the effective heat transfer coefficient between the cooling fluid and the heat loaded wall S of the evaporator. $Q$ is the heat flow; $T_{wl}$ is the temperature of the wall; $t_v$ is the temperature of the vapor on the evaporator exit.

In Fig. 8 the temperature evolution of the evaporator (1), vapor on the evaporator exit (2), and the temperature of the surface of the sorbent canister (3) are shown as a function of time. When LHP is switched on, during the time of its preheating (0–1500 s) the temperature of the evaporator surface and the temperature of the vapor are steadily increasing till the steady state of heat pipe operation (time interval 1500–3000 s) is fixed. After the time interval of 3000 s, when the steady state condition of heat pipe operation was fixed with $Q$ max, the next possibility to dissipate the increasing heat load is by switching on the sorbent canister and connecting it with the evaporator through the valve 11 Fig. 3. At the same time the valve 10 (Fig. 3) ought to be closed, other ways the liquid will go directly to the sorbent bed. A sharp temperature decrease of the evaporator and vapor are immediately checked during the time interval 3000–3500 s. When the sorbent bed saturation by vapor is finished and the temperature of the sorbent material is increasing (curve 3) together with the increase of the temperature of the evaporator, the temperature of the vapor at the output also starts to increase (time interval 3500–3750 s). By opening the valve 10 (Fig. 4) during the desorption of the sorbent material we let go the vapor flow in two directions: one from the evaporator to condenser, another from the sorbent canister to condenser. Then we close valve 10 and start the system operation as loop heat pipe. This enhanced mode of the heat pipe action can be quasi-stationary, if we use the system with at least two sorbent beds, switching on and off alternatively. The time interval 0–3000 s. (Figs. 9 and 10) is typical for LHP, the time interval 3000–3500 s is typical for the SHP, operating during 500 s. The heat transfer coefficient between the evaporator and the cooling fluid is shown in Fig. 10. When the sorbent bed structure (SSC) is joined to the evaporator, a sharp increase of the heat transfer coefficient (three times more to compare with conventional heat pipe function) is fixed with the two-phase flow forced convection inside the pores.

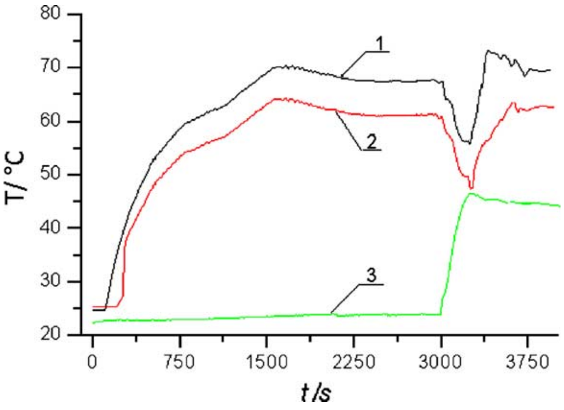

Fig. 9. Temperature field evolution in SHP: (1) surface of the evaporator; (2) vapor in the transport zone; (3) surface of the SSC (adsorber).

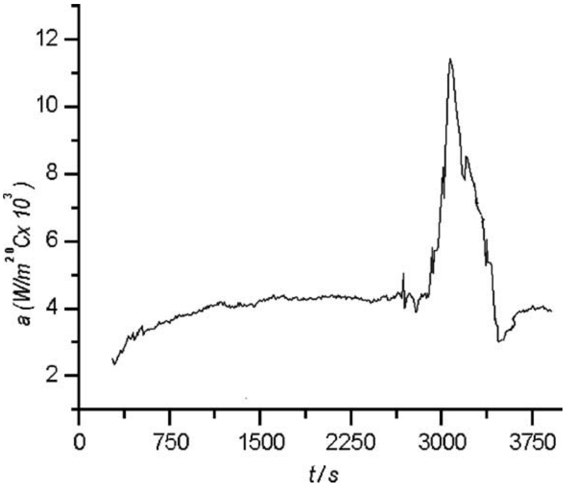

Fig. 10. Heat transfer coefficient as a function of time for SHP evaporator.

The sorption bed efficiency as a cooler depends on several parameters such as: total mass of the working fluid in HP, sorption capacity of sorbent material, flow rate of ammonia from evaporator to a sorption material and heat transfer rate inside of the evaporator.

There is another possibility to apply this experimental set-up as a high heat releasing component cooling system (Laser diodes stack, IGBT transistors). The goal of the second set of experiments is to estimate the SHP capability to remove high heat fluxes >200 W/cm$^2$ with low surface superheat of the evaporator (ammonia as a working fluid). This new cooling device can be applied

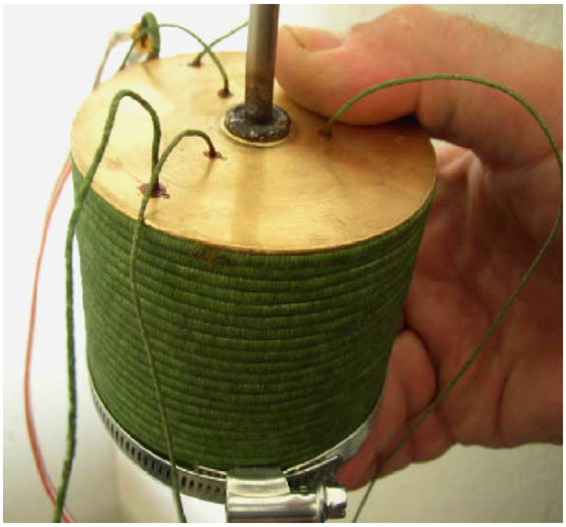

Fig. 11. Photo of SHP evaporator with electrical heater and thermocouples.

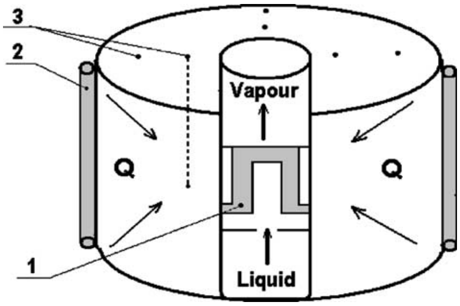

Fig. 12. Schema of SHP evaporator with electrical heater and thermocouples: (1) porous media body; (2) electrical heater; (3) thermocouples.

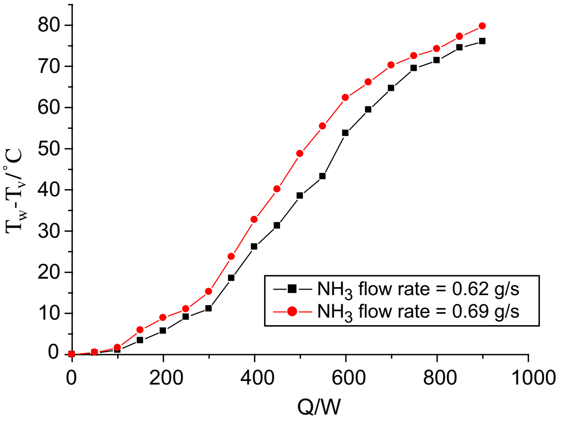

Fig. 13. Temperature drop between the heat loaded wall and the vapor (evaporator outlet) ($\Delta T = T_w - T_v$) as a function of heat load $Q$.

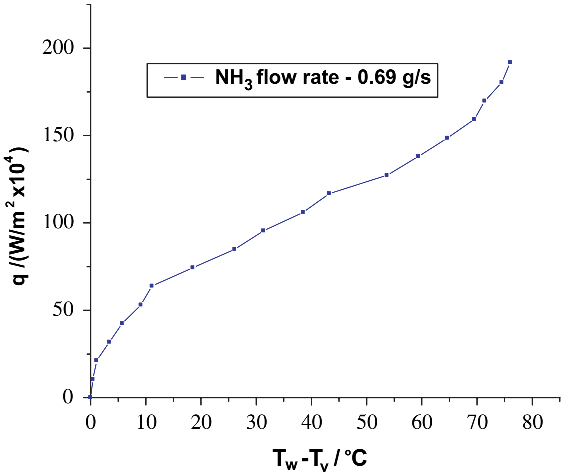

Fig. 14. Input heat flux as a function of temperature difference $T_w - T_v$.

for large area cooling at high heat fluxes, operating in microgravity space environment. Its relatively low pressure drop across the porous structure is an important advantage to compare with conventional systems of semiconductor components cooling. SHP as a cooler has a compact evaporator/micro multi nozzle spray chamber to subcool the heat loaded device and enhance heat transfer coefficient in the evaporator.

The experimental evaporator (micro two-phase jets generator) is shown in Figs. 11 and 12. The heater block of the evaporator includes a cavity machined directly in the centre of the copper cylinder with thick walls and is used to put in a nickel sintered powder evaporator. Some thermocouples are disposed inside the copper block to

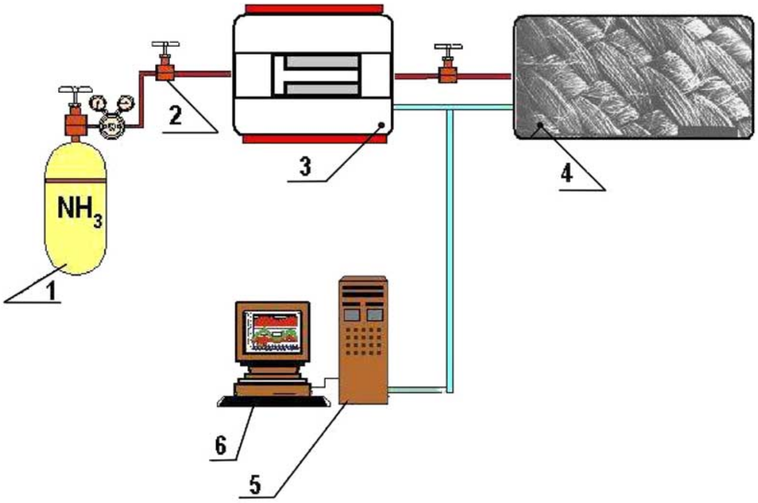

Fig. 15. Experimental set-up: (1) ammonia tank; (2) valve; (3) SHP evaporator; (4) sorbent canister; (5) data acquisition system with thermocouples set; (6) personal computer.

control the heat flow to the evaporator from the electric heater disposed on its outer surface. Heat input to the evaporator was calculated by conduction analysis using thermocouples that were placed along a known distance apart in the copper heater block (Fig. 12).

The SHP evaporator, Figs. 13 and 14 is considered as a main thermal resistance of the circuit. This evaporator thermal resistance includes the temperature drop due to the thermal conduction through the heat pipe wall and the porous structure. For the heat flux $Q = 100\text{–}200\ \mathrm{W/cm^2}$ and more the average evaporator resistance $R = 0.07\text{–}0.08\ \mathrm{K/W}$. So there is a possibility to cool successfully the electronic components, using a new generation of SHP. The limited possibility to increase the heat flux value more than $200\ \mathrm{W/cm^2}$ is the electric heater, which did not allow the heat input increasing to the evaporator.

The dominant mode of heat transfer in SHP for this application is a volumetric evaporation of the liquid inside the porous structure, micro-jet spray cooling of the heat loaded wall and thin film evaporation of micro droplets on its surface. The efficiency of the cooling process is the best for the heat loads $200\ \mathrm{W/cm^2}$ and more. Such mode of cooling allows reaching a large variation of the evaporator temperature. The volumetric phase change in pores and the micro droplets in the jets guarantee the additional thin film evaporation on the hot wall whereas the surface temperature is quite uniform (Fig. 15).

## 4. Conclusions

1. Sorption heat pipe is a novelty and combines the enhanced heat and mass transfer typical for conventional heat pipe with sorption phenomena in the sorbent structure. The sorption heat pipe parameters are insensitive to some "*g*" acceleration and such heat pipe can be suggested for space and ground application.
2. In the sorption heat pipe significant heat transfer enhancement (more than 3 times) is obtained to compare with a conventional loop heat pipe for the same fluid (ammonia) and evaporator dimensions.
3. Sorption heat pipe can be considered as a cryocooler, applied for deep cooling of sensors down to the triple point of the hydrogen (nitrogen) with cyclic or periodic operation in space. The working fluid can be stored in sorption material of SHP at low pressure.
4. Sorption heat pipe can be used as an efficient multi micro-jet cooler of the laser diode with good thermal contact of the SPH porous wick and heat loaded wall, insensitive to some "*g*" acceleration and capable to dissipate high heat fluxes $q > 200\ \mathrm{W/cm^2}$ for ammonia with low surface superheat.

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