

SORPTION HEAT PIPE – A NEW THERMAL CONTROL DEVICE

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Abstract

Sorption heat pipe is a novelty and combines the enhanced heat and mass transfer typical for conventional heat pipe with sorption phenomena in a sorbent structure. Sorption heat pipe can be used as a heat source / sink and be applied as a heat pipe. Sorption heat pipe is insensitive to some "g" acceleration and it is suggested for space and ground application.

KEYWORDS

Heat transfer, evaporation, sorption, loop heat pipe.

INTRODUCTION

Sorption heat pipe is a combination of a heat pipe and solid sorption cooler with some specific interaction between these elements. This device combines the enhanced heat and mass transfer in conventional heat pipes with sorption phenomena of sorbent bed inside it [1]. It means, that this device could be used as a sorption heat transfer element for cooling/heating, and be cooled and heated as a heat pipe. Sorption heat pipe is beneficial for the power electronic components cooling (IGBT, thyristors, et.) and for the space cryogenic two-phase thermal control systems. Now the electronic component cooling is a key problem in the industry and space application [2]. The major problems associated with cryo-coolers are usually 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 of adsorption systems applications 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 coupled use of heat pipe and solid sorption cryo-coolers, realized in sorption heat pipe. 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. Sorption heat pipe have a sorbent bed (adsorber/desorber and evaporator) at one end and a condenser + evaporator at the other end (Figs. 1-2). In this paper we consider a sorption heat pipe as a combination of loop heat pipe (LHP) and ammonia/ (active carbon fiber + chemicals) solid sorption cooler. Such system extends the limits of the two-phase thermal control and ensures successful mode of electronic components cooling even in very harsh environmental conditions (ambient temperature 40 °C, or more). Sorption heat pipe could be also considered as a cryo-cooler applied for deep cooling of sensors in space down to the triple point of the hydrogen. Traditional two-phase thermal control system for space application is sensitive to the vehicle acceleration and vibration (spacecraft launching time change of the orbit, et.). To avoid this inconvenience a sorption heat pipe as a coupling pair of CPL (LHP) and solid sorption cooler (SSC) is a good opportunity [5] Fig. 2. The active mode of cooling/heating of the electronic components, or the other equipment could be realised using SSC integrated directly into the two-phase heat transport system. The solid sorption cooler begins to function (to be switched on), when the cooling possibilities of the conventional 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 on 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 [6], for space application in a large temperature range from 60 K up to 400 K. Heat releasing

compounds may be disposed inside the evaporator of such device [7], or to be in good thermal contact with the external surface of the heat pipe evaporator.

SORPTION HEAT PIPE

Sorption heat pipe include the advantages of conventional heat pipes and sorption machines in one unit. The major it advantage to compare with conventional sorption machines is convective (two-phase flow) mode of the heating/cooling of sorbent structure inside the heat pipe. The same working fluid is used as a sorbate and heat transfer media.

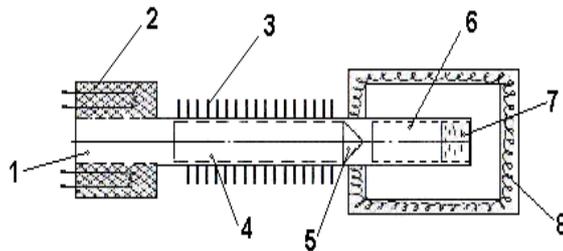


Figure 1. Heat Pipe Longitudinal Cross, Vasiliev, 1992.

- 1 – vapour 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 with thermal insulation.

Phase 1. At the beginning it is necessary to desorb a sorption structure (2), Fig. 1 of heat pipe due to absorption of the heat of a heat source. During desorption of a sorbent bed the vapor (1) of a working fluid need to go outside of a porous structure (2) and be condensed in the heat pipe evaporator/condenser (3). The vapor is generating inside the porous structure of a sorbent bed, the vapor pressure is increasing, and the vapor flow enters the condenser and is condensing releasing the heat to surroundings. The part of the cold working fluid is filtrated through the porous valve (5) and enters the evaporator due to the pressure drop between the hot part of a heat pipe and the evaporator. The other part of the working fluid is returning to the sorbent bed due to capillary forces of the wick (4) and increases the procedure of sorbent bed heating by the heater, following the micro-heat pipes phenomena inside the sorbent bed. When desorption of the sorbent structure is accomplished, the source of energy is switched of, the pressure in the sorbent bed is decreasing and the working fluid is accumulated inside the evaporator.

Phase 2. After the Phase 1 accomplishment the porous valve (5) is opened and the vapor pressure inside the heat pipe is equalizing following the procedure of the liquid evaporation inside the porous structure of the evaporator (6). During the liquid (7) evaporation in the evaporator (6), the cold generation is available inside the cold box (8). When the liquid evaporation is accomplished and the sorbent bed is saturated with the vapor, a porous valve is closed and the sorbent bed is beginning to be cooled with the help of the heat pipe condenser (3). The phase 2 is closed.

In our recent experiments a modified sorption heat pipes were used, Fig. 2. These sorption heat pipes are similar to CPL and LHP, but have a sorbent structure, integrated into the system. Loop heat pipes (LHP) seems to be very promising heat transfer devices [8-10] for space applications in a large temperature range from 60 K up to 400 K.

The sorption heat pipe system includes some basic phenomena interacting with each other: 1. in the sorbent bed there is a vapor flow (two phase flow) with kinetic reaction rate and pressure, vapor pressure, geometry, conductive and convective heat transport with radial heat transfer; 2. In the 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. Cryogenic sorption heat pipe (hydrogen, oxygen, and nitrogen) has no needs to be protected against super pressure influence at room temperatures,

because the pressure inside is regulated by the sorption structure and basically is low. For the conventional CPL and LHP there are some problems with start at cryogenic temperatures from supercritical state. New sorption heat pipe, which include a cryogenic (oxygen, propane, ammonia) loop heat pipe + solid sorption gas storage system easily can be used to solve this problem. Solid sorption gas storage device facilitate the gas adsorption at the room temperature, being adsorbed by the sorbent bed. When the LHP condenser is cooling and the solid sorption gas storage system is heating the working fluid (for example, hydrogen), being desorbed from the sorbent bed, is condensing in the condenser and is sucking by the porous wick of the LHP evaporator. Sorption heat pipe is ready to cool the electronic components (infrared sensor). It is interesting to compare possibilities of the conventional LHP and sorption heat pipe to pump the working fluid in different gravity field values.

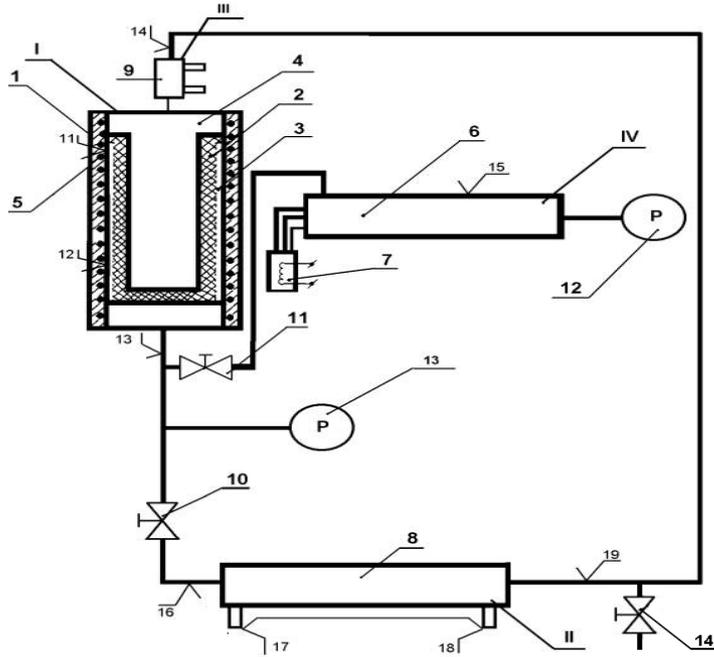


Figure 2. Schematic of a Sorption Heat Pipe Experimental Set-Up.

I – heat pipe evaporator, II – heat pipe condenser, III – liquid cooler, 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 – electric heater, 6 – sorption canister, 7 – sorption structure thermal control system, 8 – heat pipe condenser, 9 – liquid cooler, 10 – valve, 11 – regulated valve, 12 – pressure sensor, 13 – vacuum sensor, 14 – valve for the fluid charging, 11-19 – thermocouples

For LHP the maximum pressure rise due to the surface tension effects in the wick can be evaluated by La-Place equation:

$$(p_c)_{\max} = 2\sigma / r_c \quad (1)$$

where σ is a surface tension of working fluid and r_c - is the effective capillary radius of the wick.

In the real LHP design capillary pressure drop ΔP_c depends on some LHP parameters and need to be:

$$\Delta P_c \geq \Delta P_v + \Delta P_l + \Delta P_w + \Delta P_g \quad (2)$$

where ΔP_v and ΔP_l are the pressure drop in the vapor and liquid lines, ΔP_w is the pressure drop in the wick pores and $\Delta P_g = \rho_l g L_{\text{eff}} \sin \theta$ – pressure drop due to the gravity field action. In real devices this pressure head is less 1 bar.

In sorption heat pipe action the maximum pressure rise is determined by the vapor pressure difference in the evaporator and adsorber due to Clausius-Clapeyron equation:

$$d \ln P / d(1/T) = -L/R, \text{ or } -\Delta H/R \quad (3)$$

For such fluids as ammonia the pressure drop for sorption heat pipe 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 the sorption heat pipe the one external source of energy is also the heat input to the evaporator, the heat sink is the adsorber. The second external

source of energy needs to be used periodically for the adsorber regeneration. LHP can be used constantly, sorption heat pipe need to be used periodically (or we need to have at least two adsorbers, working out of phase).

The experimental set-up of a sorption heat pipe, Fig. 2, is composed with the loop, a compact solid sorption cooler and valves. The core of this set-up and a key element of a sorption heat pipe is a capillary pumped evaporator with the inverted meniscus of the evaporation. The evaporator design (wick structure, geometry, internal volumetric surface of pores, surface of the liquid meniscus and surface of liquid entrance, thermal conductivity, permeability, et.) is the dominating factor in the sorption heat pipe. This evaporator can be used alternatively as a heat pipe evaporator, or the evaporator of a solid sorption cooler.

Naturally, such evaporator needs to be compatible with these two systems of cooling. Its performance depends on the transport properties of the wick, i.e. permeability, thermal conductivity, as well as structural characteristics. 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 wick outer surface contacting with SS tube (heating element). The Ti wick was used with the ammonia as working fluid and was force-fitted within the stainless steel tube. Working liquid (Fig. 2) is entering from the heat pipe condenser to the liquid accumulator. Radial liquid flow occurs through the wick to the heated by the SS heat loaded wall and the vapor is moving along the axial and circumferential grooves to the vapor outlet. The Ti wick structure has porosity up to 55%, the length – 280 mm, the outer diameter – 38 mm, maximum pore diameter – 10 microns, medium pore diameter – 3-5 microns, wick thickness – 4 mm. The heat pipe condenser is made as a tube in tube heat exchanger. The heat output of the system is near 800 W, the thermal resistance of the evaporator – 0.03 K/W, pressure drop $\Delta P_{cap} = 200$ mbar. Sorption heat pipe experimental set-up is composed from the evaporator, condenser and adsorber (sorbent bed - active carbon fiber "Busofit" saturated with salts). The adsorber was filled with "Busofit" and saturated with $CaCl_2$ salt. The cold output is near 100 W. Solid sorption cooler (adsorber) being switched on by the valve in a certain time increase the heat transfer intensity inside the porous evaporator due to the volumetric evaporation of the working fluid and liquid accumulator under the pressure drop influence (some bars). By choosing a working fluid having a large heat of the evaporation (water, ammonia) one may expect to dissipate significant heat flux in the evaporator. In this two-phase thermal control system the mass flow needs to have any type of pumping device with energy consumption from an external source. For LHP this source of energy is a capillary pumped evaporator and the thermal sink is condenser. Sorption heat pipe evaporator is the same as LHP evaporator, but pressure head is generated due to superheated vapor in the sorbent bed and an adsorber forms the thermal sink of SSC. The experimental data of the sorption ammonia capacity for such a combination "Busofit" + $CaCl_2$ as function of pressure and temperature is shown on Fig. 3 [11].

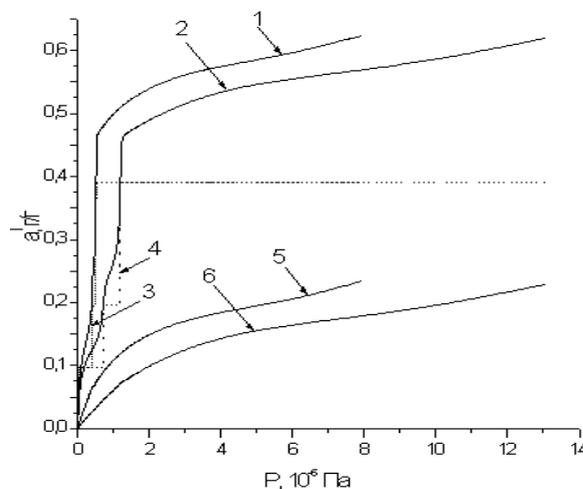


Figure 3. Experimental and Ammonia Isotherms of "Busofit+ $CaCl_2$ " at different temperature of equilibrium (1 – $T_{eqil}=20$ °C, 2 – $T_{eqil}=40$ °C) [11]

In the case with a heat loaded evaporator we realize the phenomena of two-phase flow forced convection through the porous structure with power generation inside a wick due to a high thermal conductivity of the porous metal layer sintered to the evaporator wall. There is a possibility to evaluate the heat transfer intensity inside the porous wick, considering the forced two-phase fluid filtration through the heat loaded high thermal conductivity metal porous layer and a volumetric coefficient of the heat transfer in the pores h_v (W/m³K) [12]. In this case we consider the heat transfer between the cold fluid, filtrating through the porous layer, and a porous structure with volumetric heat generation. The energy absorbed by the cold fluid can be evaluated by the number $\alpha_v (T - t)$ (W/m³), which determines the heat energy dissipation in the unit of the volume of porous structure at the 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:

$$[(1 - \Pi)c_s \rho_s + \Pi c \rho] \frac{\partial T}{\partial t} - Gc \frac{\partial T}{\partial x} = \frac{\partial^2 T}{\partial x^2} \quad (4)$$

where Π is porosity, c – specific heat, ρ - density, G – flow rate.

When the heat flow generation through 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:

$$(1 - \Pi)\rho_s c_s \frac{\partial T_s}{\partial t} = (1 - \Pi)\lambda_s \frac{\partial^2 T_s}{\partial x^2} - \alpha_v (T_s - T) \quad (5)$$

$$\Pi \rho c \frac{\partial T}{\partial t} + Gc \frac{\partial T}{\partial x} = \alpha_v (T_s - T) \quad (6)$$

where λ is thermal conductivity, α_v – volumetric heat transfer coefficient.

The equation (4) now can be presented as:

$$\frac{d^2 T}{dz^2} - B \frac{dT}{dz} = 0 \quad (7)$$

Here some non-dimensional values are applied $B = G\delta c/\lambda$. Non-dimensional complex B characterizes the ratio between the heat absorbed by the cold fluid and the heat transferred through the porous skeleton due to its thermal conductivity. δ – is the wall thickness.

The boundary condition for the case of the fluid flow with initial temperature t_0 through the porous wall is:

$$\begin{aligned} X = \delta \quad \lambda_{\text{eff}} dT/dX &= Gc (T(X) - T_1) \\ X = 0 \quad \lambda_{\text{eff}} dT/dX &= Gc T_1 \end{aligned} \quad (8)$$

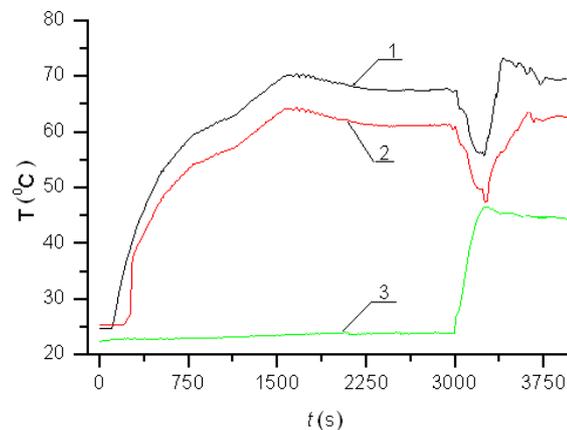


Figure 4. Temperature evolution in different parts of the sorption heat pipe:
1 – temperature evolution on the surface of the evaporator; 2 – temperature evolution of vapor in the transport zone; 3 – temperature evolution of the sorption canister

On the other hand the porous coating on the inner surface of the evaporator can be considered as mini-fins set and we can determine the heat transfer efficiency between the fluid and porous structure as:

$$Q = \alpha S (T_w - t_v)$$

(9)

where α - is the effective heat transfer coefficient between the fluid and the wall of the evaporator; S - the heat loaded wall surface; Q - heat flow; T_w - the temperature of the wall; t_v - the temperature of the vapor on the evaporator exit.

On the Fig. 4 the temperature evolution of the heat loaded wall of the evaporator (1), vapor on the evaporator exit (2), and the temperature of the sorbent bed canister surface (3) are demonstrated as a function of the time. When the sorption heat pipe switched on, during the time of it's preheating (0 – 450 s) the temperature of the evaporator surface and the temperature of the vapor is steadily increasing till the stationary state heat pipe functioning is fixed (time interval 1500 – 3000 s). After the time interval 3000 s, a canister with sorbent bed is connected with the heat pipe evaporator through the valve 11, Fig. 2. A sharp temperature decrease of the evaporator wall and vapor are immediately checked on during the time interval 3000 – 3500 s. Then sorbent bed saturation is finished (adsorption of the ammonia vapor) and the temperature of the sorbent canister begins to increase (curve 3) together with the increase of the temperature of the evaporator surface and the vapor output (time interval 3500 – 3750 s). This mode of evaporator cooling can be stationary, if we use at least two solid sorbent bed switched on and off alternatively. The time interval 0-3000 s. (Fig. 4) is typical for the conventional LHP, the time interval 3000 – 3500 sec is typical for the sorption heat pipe, switched on during 500 s. The coefficient of the heat transfer between the heat loaded wall of the evaporator and the cooling fluid is shown on Fig.5. When the sorbent bed structure is connected with the heat pipe evaporator through the valve a sharp increasing of the heat transfer intensity, (three times more to compare with conventional heat pipe function) is checked due to the forced convection heat transfer of the two-phase flow inside the pores.

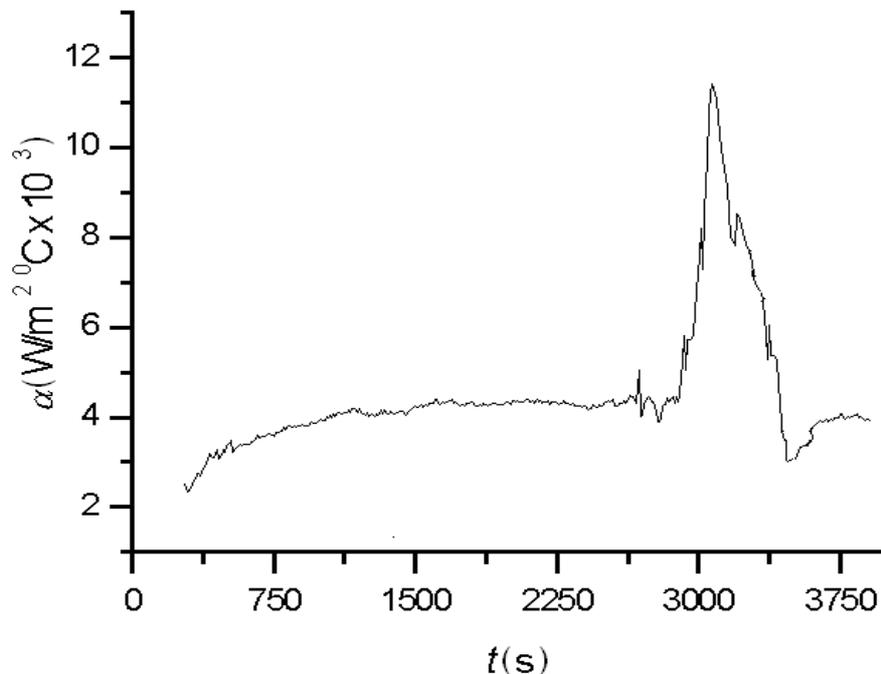


Figure 5. Heat transfer coefficient as a function of time for the sorption heat pipe evaporator

The main disadvantage of such cooling devices is its periodical functioning. To get a cold constantly we need to use at least two sorbent bed, switched on and off alternatively or we could use absorption based evaporator. Fig. 6 hows such heat transfer device.

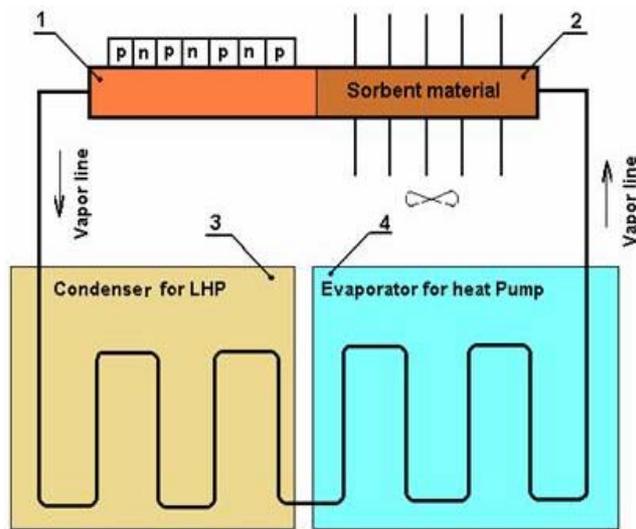


Figure 6. Thermally driven capillary/sorption compressor:
 1 – desorber, 2 – absorber, 3 – condenser, 4 – evaporator

This device consists on absorber, desorber (high temperature evaporator), condenser and the low temperature evaporator. In the high temperature evaporator, which is thermally linked with the heat releasing unit (electronic component) the vapor of the low temperature working fluid (for example ammonia) is released during its evaporation from the low concentrated solution (for example water+ ammonia) inside the porous structure (like in conventional LHP evaporation zone), and the ammonia vapor flow is going to the condenser, were it is condensing with the heat dissipation to the ambient. A pressure swing between the high temperature evaporator and the condenser (due to the temperature difference) stimulates the ammonia vapor flow and the vapor condensation in the condenser. The low temperature liquid (ammonia) enters the evaporator, where it evaporates due to the pressure difference between the low concentration solution in the absorber and the low temperature evaporator. The ammonia evaporation in the evaporator is accompanied with the heat flow absorption from the ambient through the wall of the unit. During the ammonia evaporation in the evaporation zone the temperature is decreasing below ambient. After the procedure of the ammonia vapor absorption by the low concentration solution of the high concentration fluid (water+ammonia), the high concentration solution is sucking to the evaporation zone of the high temperature LHP evaporator by capillary forces thus close to working cycle. After the ammonia release in the high temperature evaporator, the low concentration solution (mostly water) is going back to the absorber unit by the bubble pump action and there it is sucking by porous structure of the absorber.

This design of the new cooling device promises to be low cost, lightweight, reasonably efficient, and motion insensitive. First of all the unit is compact, the vapor/bubble pump for water with the low concentration ammonia solutions is disposed inside the porous heat and mass exchanger. The low concentration solution (low concentration in refrigerant) is pumped with the help of two-phase bubble type of siphon from desorber to the absorber. The surface of mass transfer for the ammonia and water is very large, because it is situated in the compact volume of porous structure. The intense heat and mass transfer inside the high thermal conductivity porous structure and the unit envelope (heat input in the desorption part and heat output in adsorption part of the unit) is ensured due to a large pores surface of the water and ammonia (or other pair) mass interaction. Some improvements over the above mentioned conventional absorption compressor include significant miniaturization and cost reduction in the absorber, generator, and heat and mass transfer area, because all these components are disposed inside the same cylindrical envelope (Fig. 6) .

CONCLUSIONS

1. The 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. Sorption heat pipe can be used as a heat source / sink and be applied as a heat pipe. The sorption heat pipe parameters are insensitive to some “g” acceleration and such heat pipe can be suggested for space and ground application.
2. New type sorption heat pipe family was developed and tested, oriented on the application in highly efficient sorption machines and electronic components cooling.
3. Sorption heat pipe extends the limits of the two-phase thermal control and ensures successful mode of electronic components cooling even in very harsh environmental conditions (ambient temperature 40 °C, or more).
4. Sorption heat pipe can be considered as a cryocooler, applied for deep cooling of sensors in space down to the triple point of the hydrogen.
5. Significant heat transfer enhancement (up to 3 times) was obtained in the sorption heat pipe to compare with a conventional loop heat pipe for the same fluid (ammonia) and evaporator dimensions.
6. A new absorption cooler is suggested for the sorption heat pipe to ensure a constant rate of its cold generation.

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