A Study of the Short Heat Pipes by the Monotonic Heating Method

AV Seryakov and A P Alekseev
LLC Rudetransservice, Sankt-Petersburg Street 64,
Veliky Novgorod, 173003, Russia
e-mail: seryakovav@vandex.ru

Abstract. The results of studies using the monotonous heating method in a vacuum adiabatic calorimeter the thermal resistance and heat capacity of short linear heat pipes (HP’s) designed for cooling spacecraft and satellites with strict take-off mass regulation are presented. Experimental results of thermal resistance, obtained at high thermal loads and starts boiling of the working fluid in the HP’s evaporator allow us to determine the evaporator heat capacity by solving the inverse coefficient problem of thermal conductivity equation. This allows us to estimate the thickness of the working fluid layer and the specific heat of evaporation, the heat capacity of the liquid and vapour phase of the boiling two-phase working fluid on the evaporator surface.

I. Introduction
Heat transfer coefficient increasing of short linear HP’s is an important technological task in the development of spacecraft with strict takeoff mass regulation. The use of short linear HP’s is justified in the case of constructive impossibility of contour HP’s placing, as well as to improve the long-term reliability and stability of passive cooling systems without no distributed supply and discharge lines of contour HP’s [1-4].

In the short linear HP’s with a Laval-liked vapour channel and with large amount of the working fluid in the insert (wick) pores and at high temperature head on the evaporator arise a high thermal conductivity and significantly reduces the response time of the unit. This allows us to apply the method of monotonic heating to study the HP’s internal characteristics. Studies using monotonic linear heating in time (dynamic thermal analysis) are widely used in the practice of thermophysical experiments [5-6], and short HP’s is no exception.

The response time is also reduced by a smaller thermal mass and length of the HP, this also allows to enter the monotonic heating mode very quickly.

The direct problem of thermal conductivity is to find the body temperature (surface temperature for thin HP in one-dimensional model of heat propagation in an adiabatic calorimeter) that satisfies the differential equation of thermal conductivity and with specified boundary and initial conditions of unambiguity [5].

The inverse thermal conductivity problem is to determine the parameters of internal heat transfer also in one-dimensional model of heat propagation in an adiabatic calorimeter, thermal resistance
(thermal conductivity) and heat capacity of the short linear HP at high thermal loads (high temperature head) and the boiling beginning in the evaporator. The formulation and solution of coefficient inverse problems of thermal conductivity are the theoretical basis of experimental methods for studying the heat capacity of the evaporator fragment of short linear HP using thermal resistance measurement and calculation of the transmitted heat power along the longitudinal axis, which is reduced due to internal friction losses [3-4].

Thus, we have a coefficient inverse thermal conductivity problem, when the known experimental data on the thermal resistance $R_{HP}$ and HP surface temperature, and the measured heat flow on the surface of the evaporator, it is possible to restore the values of the heat capacity $C_{ev}$ of the evaporative fragment as a function of the temperature head on the HP.

The finite dimensions of the HP lead to a distortion of its temperature field due to the heat exchange of the end surfaces with the external environment, although the equations imply a one-dimensional temperature field inside the HP [5].

2. Experimental setup

A schematic diagram of the experimental test setup is shown in Figure 1 and Figure 2. In this study was applied stainless steel HP’s with shaped vapour channel in the converging-diverging Laval nozzle -liked form with nonlinear varying diameter and with flat top and bottom covers.

![Figure 1. HP's diagram: 1 – top cover; 2 – cylinder body of the HP's; 3 – cone-shaped turbulator; 4 – capillary-porous insert; 5 – bottom cover; 6 – capillary injector channels; 7 – bottom flat capillary-porous insert-evaporator. There are capacitance sensors 8, 9, installed inside the top cover [3-4], one of which is intended for a condensate film thickness measurement, while another one has a sensing element of CT3-19 thermistor mounted on its electrodes to measure the film temperature.](image)

The diethyl ether $C_4H_{10}O$ is used as the working fluid, which has the boiling temperature under the atmospheric pressure of $T_B = 308.65K$ (35.5°C), freezing temperature $T_F = 156.95K$ (−116.2°C) and critical parameters $T_C = 466.55K$ (193.4°C), $P_C = 3.61MPa$. Filling HP’s is carried out with a limited amount of diethyl ether, and only equal to the volume of the longitudinal pores in the vertically oriented capillary-porous metal mesh insert. The restriction in the HP’s filling amount is made so that the evaporator is not flooded with ether at a high temperature head and the film boiling beginning on
the HP lower cover surface (the evaporator surface). The mass of diethyl ether with the density of 713.5 kg/m³ (20°C) in the insert pores is equal to 11.858·10⁻³ kg. The charging ratio of the HP with Laval-lipped vapour channel (ratio of the diethyl ether volume to the total volume of the HP) is equal to 16.62·10⁻⁶ m³ / 3.14·10⁻⁵ m³ = 0.529.

The developed sensors enables us to perform measurements of local characteristics of the film thickness and temperature, without making any major disturbances in the flow. The HP’s dimensions are chosen in such a way that it is possible to establish capacitive sensors [3–4] for film thickness and temperature measuring.

All heat transfer coefficient measurements in stationary mode were carried out using two HP’s, measuring one and reference one [2]. The main HP, called measuring, is filled with diethyl ether and the reference one, which is completely identical to the main HP, is filled with dehumidified air at a pressure of 1 bar with dew point temperature lower than 233.15K (–40°C). The heat transfer coefficient K_{HP} of the second HP does not exceed 0.15% from the first one (measuring one) K_{HP} and is not taken into account. The second HP, completely identical to the first one, performing the reference function in measurements with the help of high-frequency generators and capacitance sensors the condensate film thickness in the first HP.

3. Results and discussion

The diameter of the HP is small, \( D_{HP} / L_{HP} \leq 0.2 \), so the temperature field inside can be considered in a one-dimensional mode without heat loss through the side surface by using an adiabatic calorimeter. For the one-dimensional mode of heat conductivity along the \( z \) axis in the short linear HP with length \( L_{HP} \) and cross-section area \( F(\bar{z}) \), the equation of heat transfer in the cylindrical coordinate system looks like this:

\[
C_{HP}(t)\dot{t}(r, \tau_k) = \frac{1}{r} \frac{\partial}{\partial r} \left( r \lambda_{HP}(t) \frac{\partial t}{\partial r} \right) L_{HP} F(\bar{z}) .
\]

By replacing the \( r \) coordinate for the experimentally determined values of one-dimensional temperature field \( t(\bar{z}, \tau) \) along the HP’s longitudinal \( z \) axis with the dimensionless \( \bar{z} \) coordinate the thermal conductivity equation for the one dimensional case can be written as follows:

\[
C_{HP}(t)\dot{t}(\bar{z}, \tau_k) = \frac{1}{\bar{z}} \frac{\partial}{\partial \bar{z}} \left( \bar{z} \frac{\partial t(\bar{z}, \tau)}{\partial \bar{z}} \right); \quad \bar{z} = \frac{z}{L_{HP}} .
\]

\[
R_{HP}(t) = \frac{L_{HP}}{\lambda_{HP}(t)F(\bar{z})_{ev}}, \quad \frac{K}{W} ; \quad C_{HP}(t) = \rho_{HP} c_v(t)L_{HP}F(\bar{z}), \quad \frac{1}{K} .
\]

The initial conditions in the form of the experimental temperature values \( T \) for solving equation (1), taking into account all components of thermal resistance \( R_e, R_w, R_a \) [2], including contact resistance and the nonzero heat capacity of the resistive heater \( C_{H2} \), figure 2, are written in the standard way:

\[
t(0, \tau) = \tau_e = T_{ev}(\tau) - (R_c + R_w)Q_{ev}(\tau); \quad t(L_{HP}, \tau) = t_{cond} = T_{cond}(\tau) + (R_a + R_w)Q_{cond}(\tau) .
\]

\[
t(0, 0) = t_0; \quad t(0, \tau_k) = t_0 + \tau_k \cdot 3 \cdot 10^{-3} \frac{K}{s}; \quad q_{ev} = \lambda_{HP}(\frac{\partial t}{\partial \bar{z}})_{\bar{z}=0} = E_{H2} - C_{H2} \ddot{t}_{ev} ; \quad \dot{t}(\bar{z}, \tau_k) = \frac{q_{ev}}{C_{HP}} .
\]

The equation (2) of heat flow \( q_{ev}(\bar{z}, \tau_k) \) propagation along the HP \( z \)-axis is divided and presented as a system of two equations for calculating the heat capacity \( C_{HPk}(t) \) and heat flow, based on the thermal resistance \( R_{HP} \) and for the time moment \( \tau_k \):
\[  \bar{z} C_{HPk}(t) \dot{t} + \frac{\partial q(\bar{z}, \tau_k)}{\partial \bar{z}} = 0 . \]  \hspace{1cm} (6)

\[ q_{ev}(\bar{z}, \tau_k) = -\bar{z} \frac{L_{HP}}{R_{HP}(\bar{z})} \frac{\partial t(\bar{z})}{\partial \bar{z}} . \]  \hspace{1cm} (7)

The values of the heat power \( q_{ev}(\bar{z}, \tau_k) \) at the HP’s upper and lower flat boundaries are equal to:

\[ q_{ev}(0, \tau_k) = Q_{ev} ; q_{ev}(1, \tau_k) = Q_{cond} . \]  \hspace{1cm} (8)

Energy losses due to friction in the vapour channel and incomplete adiabaticity of the calorimeter lead to the fact that \( Q_{ev} \) and \( Q_{cond} \) are not equal to each other:

\[ \frac{(Q_{ev} - Q_{cond})}{Q_{ev}} \leq 2.6 \% . \]

The value of the thermal resistance \( R_{HP} \) [7] to solve equation (7) in the stationary state was determined by the formula (9), using a vacuum adiabatic calorimeter, in which all the HP’s parameters, including the temperature and the transmitted heat power, were measured using the experimental stand, shown in Figure 2, and the temperature head on the evaporator, \( \delta t_{ev} = T_{ev} - T_B \), relative to the boiling point \( T_B \) of diethyl ether at atmospheric pressure can be large and ensure boiling in the evaporator and pulsation mode of vapour generation and propagation in the HP’s vapour channel:

\[ R_{HP}(t) = \frac{T_{ev} - T_{cond}}{Q_{ev}} . \]  \hspace{1cm} (9)

Condensation zones of the HP’s are provided with insulated thermocouples and set to a depth of the \( 1 \) diameter into the vortex continuous-flow calorimeter, shown in Figure 2, with stabilized water flow. To ensure accurate measuring of thermal power and heat removal augmentation in the HPs, jet flow of input water is swirled, values of flow velocity and vorticity due to air bubbles are recorded. The Reynolds number \( Re_{cal} \) in calorimeter with water temperature \( T_{cal} = (293 \pm 0.03) \text{ K} \) and a density \( \rho_{H_2O} = 998.2 \text{ kg/m}^3 \) is equal \( Re_{cal} = 3.68 \cdot 10^3 \), the Nusselt number is \( Nu_{cal} = 77.3 \), and heat transfer coefficient \( \alpha_{cal} = 2.4 \cdot 10^3 \text{ W/m}^2\text{K} \).

Figure 3 shows the results of received in 0.5 K increments the experimental stationary \( R_{HP} \) values of the thermal resistance of short HP 100 mm long with a Laval – shaped vapour channel and a similar HP with a cylindrical vapour channel, with equal overall dimensions and the same mass of diethyl ether filling and with the \( R_{HP} \) values in the temperature head range on the evaporator \( \delta t_{ev} = T_{ev} - T_B = 0 \) to 20 K [2].
**Figure 2.** Scheme for the heat conductivity and thermal resistance measuring of the short HP’s in a vacuum adiabatic calorimeter, combined with a vortex flow calorimeter. 1-vortex flow calorimeter; 2-HP’s mounting flange; 3-glass cover; 4-cover mounting; 5-reference HP; 6-flat resistive heater; 7-calorimeter drain fitting; 8 –calorimeter swirl inlet fitting; 9-sealed input of measuring wires; 10-capacitive sensors for measuring the thickness of the liquid condensate film; 11-measuring and reference generators located on the HP’s upper end caps; 12-external digital generator; 13-power amplifier; 14-digital oscilloscope; 15-computer; 16-controlled switch; 17-digital voltmeter; 18-constant water pressure vessel; 19-air bubble generator; 20-flowmeter; 21-Dewar vessel; 22-vacuum chamber; 23-adiabatic shell; 24-adiabatic heaters; 25-differential thermocouples for measuring the HP surface temperature; 26-differential thermocouples of the HP’s adiabatic system.
Figure 3. HP’s thermal resistance values. 1– black dots, experimental stationary $R_{HP}$ values of a short HP with a Laval-shaped vapour channel; 2– white dots, experimental stationary $R_{HP}$ values of a short HP with a cylindrical vapour channel. Solid lines – calculated values of thermal resistance according to equation (1) and equation (7) with experimental values of surface temperature $t(z)$ along the longitudinal $z$ axis of the HP’s.

The polynomial equation (10) describing in dimensionless form the experimental values of the thermal resistance $R_{HP}$ of a short HP with a Laval nozzle-liked vapour channel, depending on the overheating value $\delta t$: $0 \leq \delta t \leq 20$ (and also in dimensionless form) looks like this:

$$R_{HP} = -6.98489 \cdot 10^{-6}(\delta t)^5 + 2.47563 \cdot 10^{-4}(\delta t)^4 - 3.3561 \cdot 10^{-3}(\delta t)^3 +$$
$$+2.21053 \cdot 10^{-2}(\delta t)^2 - 8.23987 \cdot 10^{-2}\delta t + 0.27799.$$  (10)

The standard variance value $\sigma=0.0077433$, the Fisher criterion $R^2=0.9993717$.

After integrating equation (2) by the coordinate $\bar{z}$ and taking into account experimentally set first kind boundary conditions (4) and (5), for the HP’s heat capacity we obtain the following integral equation:

$$\int_{\bar{z}}^{\bar{z}_{1}} \bar{z}C_{HPk}(t)\frac{\partial t}{\partial k} d\bar{z} - \bar{z}q_{ev}(\bar{z}, \tau_{k}) = 0.$$  (11)

In order to evaluate the evaporator operation, we will calculate the heat capacity of the lower HP’s fragment with a length of 0.1 $\bar{z}$ (10 mm long), Figure 2, including HP evaporator, with precisely measured temperature head (heat power) on the HP’s evaporator surface. We will limit the heat capacity calculation to height $\bar{z}_{1} = 0.1$ and write equation (11) in a refined form (12):

$$\int_{\bar{z}}^{\bar{z}_{1}} \bar{z}C_{HPk}(t)\frac{\partial t}{\partial k} d\bar{z} - \bar{z}_{1}q_{ev}(0, \tau_{k}) = 0.$$  (12)

Solution of the inverse problem of the HP’s thermal conductivity (6) and (7) and the solution of the integral equation (12) to determine the heat capacity of the lower (evaporative) HP’s fragment was
performed using a developed program in the Fortran language [3-4].

The entire numerical implementation of the presented algorithm was carried out by replacing all the mentioned equations with their difference analogues, the integrals were calculated using the Simpson method [6], and the well-known Crank-Nicholson numerical scheme was used in solving boundary value problems in combination with the run-through method [5-6]. The calculation of the equation (12) was carried out with constant control and minimization of the discrepancy in the calculated values of the heat capacity, an array of experimental data on the amount of evaporator overheating \( q(0, \tau_k) \) with a linear increase in time of the evaporator surface temperature from (5):

\[
\sum_k \delta C_{HPk}(t) = \int_{\tau_k}^{\tau_k+1} \left[ \int_0^{\bar{z}} \tilde{z} C_{HPk}(t) \, d\tilde{z} - \bar{z} \delta q_{ev}(\tau_k) \right] d\bar{z} - \bar{z} \delta q_{ev}(\tau_k) \right] \, d\tau .
\]

where the parameters of temperature \( t(\bar{z}, \tau_k) \) and heat power \( q(\bar{z}, \tau_k) \) are determined by the solution of a boundary problem with boundary conditions of the second kind (6), (7) and (8). The value of the time derivative of temperature \( t(\bar{z}, \tau_k) \) is determined by the solution of equation (6).

When we solve the inverse problem, the iterative process is stopped when a discrepancy \( \delta C \) is reached the random error value of the input data:

\[
\delta C_{ev} = \frac{\delta q_{ev}}{t} + \frac{q_{ev} \delta t}{(t)^2} \leq 0.05 \frac{1}{K} .
\]

The described numerical algorithm for solving the inverse problem of the short HP thermal conductivity and recovery the heat capacity of the evaporative HP’s fragment, using depending on the temperature head \( \delta t = T_{ev} - T_{B} \) experimentally determined thermal resistance \( R_{HP} \) allows us to accurately evaluate the parameters and characteristics of the boiling evaporator, see Figure 4. The verification of the obtained heat capacity results can be performed as follows. The enthalpy of vaporization of a working evaporator with diethyl ether is represented in the form of the following equation:

\[
H_{ev} = G_{vp} r(T_{B}) + G_{vp} C_{vp} (T_{ev} - T_{vp}) + G_{l} C_{l} (T_{ev} - \overline{T_{ev}}) , \ W
\]

We substitute the known table values for diethyl ether [8] and get the expression for the effective heat capacity of a linearly heated capillary-porous HP's evaporator at the diethyl ether boiling beginning by the formula (16):
Figure 4. Calculated value of the HP’s evaporator heat capacity $C_{ev} / C_{ev0}$: $C_{ev}$ - heat capacity of the saturated with diethyl ether evaporator, J/K; $C_{ev0}$ - heat capacity of the HP evaporator filled with dried air, J/K. 1-black dots, values of the relative heat capacity of the lower evaporator fragment of a short HP’s with a Laval nozzle-like vapour channel, obtained by solving the inverse problem using equation (13) in 0.5 K increments $\delta t = T_{ev} - T_B$ with diethyl ether starting to boil; 2- a polynomial of tenth degree for smoothes the heat capacity dots.

$$C_{ev} = \frac{H_{ev}}{t} = \frac{\left(345 \frac{kJ}{kg} + 1.75 \frac{kJ}{kg \cdot K} \cdot 11K + 2.34 \frac{kJ}{kg \cdot K} \cdot 16K\right) \cdot 1.57 \cdot 10^{-4} \frac{kg}{s}}{3 \cdot 10^{-3} K/s} = 21.02 \frac{kJ}{K} .$$  (16)

The ratio of heat capacities of the evaporator with diethyl ether to the dry evaporator $C_{ev} / C_{ev0}$ taking into account the heat capacity of the resistive heater $C_{H2} = 1.2$ K/J when the diethyl ether begins to boil is as follows:

$$\frac{C_{ev}}{C_{ev0}} = \frac{C_{ev0} + H_{ev}/t}{C_{ev0}} = 1 + \frac{H_{ev}/t}{C_{ev0}} = \frac{21.02 + 1.2}{1.2} = 18.5 .$$  (17)

which is very close to the actual maximum value of the heat capacities of the HP’s evaporator.

References

[1] Akachi H. 1990 Structure of Heat Pipe US patent 4921041
[2] Seryakov A 2018 Am. J. Modern Phys. Intensification of heat transfer processes in the low temperature short heat pipes with Laval nozzle formed vapour channel 7 48
[3] Seryakov A 2019 J high energy phys, gravitation Numerical modeling of the vapour vortex 5 218
[4] Seryakov A 2019 *Int. J. Heat Mass Tran* Computer modeling of the vapour vortex orientation changes in the short low temperature heat pipes 140. 243

[5] Platunov E 1973 *Thermophysical measurements in a monotonic mode*. (M: Energy) p 144

[6] Beck J, Blackwell B and Clair C jr Inverse heat conduction Ill posed problems NY 1985 p308

[7] Faghri A 1995 *Heat Pipe Science and Technology*. Washington USA, Taylor and Francis.

[8] *Tables of physical values* 1976 Guide under the editorship of Kikoin I (Moscow: Atomizdat) p1008