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Abstract. Results are presented of the development and tests of a 1 m long ammonia ramified loop heat pipe, with two cylindrical evaporators (24 mm in diameter with an active zone length of 150 mm) and two condensers (length 200 mm, diameter 24 mm), made as pipe-in-pipe heat exchangers. Tests of the device at different orientations in 1-g have shown that it can efficiently operate at symmetrical and non-symmetrical heat load distributions between the evaporators, and also at different temperatures of the condensers cooling. The maximum total transport capacity is 1100-1400 W. Shutting down the active cooling of one condenser results in an abrupt decrease in the maximum transport capability of the device.

INTRODUCTION

Ramified loop heat pipes, equipped with multiple evaporators and condensers (mecLHPs), are quite promising heat transfer devices for application in thermal control systems (TCS) of spacecraft. They make it possible to reduce the system mass, making systems more compact. Most typical cases for which mecLHPs can prove to be expedient are:

1. The heat-load source has a large thermal contact surface and/or capacity. In this case the mecLHP evaporators have, as a rule, the same dimensions, they are arranged in parallel at a relatively small distance from each other and are joined to a common "cold plate", on which the heat load source is located.
2. The heat load sources are located at a relatively large distance from each other and have the same or different capacity. In this case the number of evaporators should correspond to the number of heat load sources with which they are in thermal contact. The evaporators may have different dimensions and be located at a different distance both from each other and from the heat sink.
3. There are two or more heat sinks remote from each other. In this case the LHP is provided with the corresponding number of condensers, each one with its own heat sink.

Different combinations of the above variants are also probable. The most complicated of them shall envisage the possibility of a different heat load distribution between the evaporators and/or different operating temperatures of objects to be thermally controlled, and also different condensers cooling conditions (including heat sink temperature and cooling intensity).

The development and testing of an LHP with two evaporators were first performed in the mid-80's (Maydanik et al., 1988). The experimental device, with a length of 580 mm, was equipped with two identical titanium wick cylindrical evaporators and one "pipe-in-pipe" heat exchanger type condenser. The working fluid was acetone. The LHP was tested at different orientations in 1 g and at different heat load distributions between the evaporators. The maximum capacity achieved was 350 to 500 W, depending on the device orientation. It was shown that such an LHP was capable of operating efficiently at both symmetrical and non-symmetrical heat load distribution between the evaporators. The results of the development served as a basis for creating a number of experimental LHP's with three evaporators, including one for a flight experiment aboard the spacecraft "Gorizont".

Theoretical and applied problems arising in the creation of ramified LHPs are quite complicated. They require considerable and purposeful investigations and developments, which are being conducted in different directions. They brought to the appearance of various types and combinations of LHPs containing from one to three evaporators and one or two condensers.

At present we can differentiate three main types of mecLHPs, which differ predominantly in arrangement or means of joining compensation chambers, which play an important role in the operation of such devices.

First: Every LHP evaporator has its own compensation chamber, which has no direct connection with the compensation chambers of other evaporators (Maydanik et al, 1988, Bienert et al., 1997). Such a design seems to be the most flexible and universal one from the viewpoint of possible applications in TCS. Its peculiarity consists of the fact that the evaporators may be located fairly independent of each other and at different distances from the heat sink. However, it is the most complicated design for control and the least investigated one.

Second: Every LHP evaporator has its own compensation chamber, which is connected directly with the compensation chambers of other evaporators. At present such a design is the most typical one (Goncharov et al., 1998, Ku et al., 2001). Evaporators in such devices are rigidly connected with each other, are at a relatively close distance from each other and can be easily joined by a common “cold plate”.

Third: The evaporators have one common compensation chamber located separately, which may be symmetrical or non-symmetrical about the evaporators (Maydanik et al., 1996). Such an LHP design can be quite easily turned into a CPL with active temperature control in the compensation chamber.

There also exist different variants of mecLHP in which compensation chambers have a capillary link with the evaporator wick (Maydanik et al., 1990, Van Oost et al., 1996), which ensures a more reliable operation of the device in 0-g conditions.

Despite the fact that certain success has been achieved in this field, the information accumulated is not sufficient for predicting with confidence the results of using mecLHPs in the whole variety of actual conditions.

This paper is an attempt to make one more step on the way of investigation of ramified loop heat pipes. The paper contains the results of development and tests in normal conditions of an ammonia mecLHP with two parallel evaporators and a condenser. It also presents a hydrodynamic model, which makes it possible to evaluate the device maximum capacity with symmetrical and non-symmetrical heat-load distribution between the evaporators. The results of calculations have been compared with the results of experiments.

DESCRIPTION OF EXPERIMENTAL MULTIPLE-EVAPORATOR-CONDENSER LHP

Figure 1 presents the general view of a multiple-evaporator-condenser LHP.

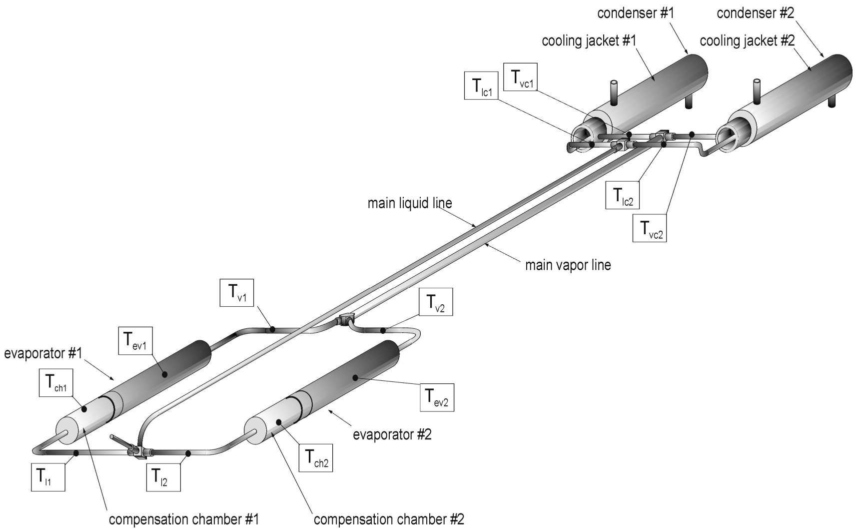

FIGURE 1. General View of the mecLHP.

The device is made of stainless steel, contains two cylindrical evaporators and two condensers made in the form of “pipe-in-pipe” heat exchangers. The evaporators are provided with nickel wicks with a breakdown pore radius of 1.1 micron and a porosity of 67%. The compensation chambers, having the same diameter, are located in the same body as the evaporators. The condensers are equipped with jackets for pumped liquid cooling. The evaporators and the condensers are positioned parallel to each other and are symmetrical with respect to the main vapour and liquid lines. The working fluid is ammonia. The main design characteristics of the mecLHP are given in Table 1.

TABLE 1. Main Design Characteristics.

Characteristics	Value
Total Length (mm)	1000
Evaporator Diameter (mm)	24
Evaporator Active Zone Length (mm)	150
Condenser Diameter (mm)	24
Condenser Length (mm)	200
Outside/Inside Diameter of the Main Vapour Line (mm)	6/4
Outside/Inside Diameter of the Main Liquid Line (mm)	4/2
Outside/Inside Diameter of Main Vapour and Liquid Collectors (mm)	4/3

OBJECTIVES AND TEST CONDITIONS

The main objective of testing consisted of obtaining information on the thermal characteristics of mecLHP's in different conditions. The test conditions included variations in the device orientation in the 1-g gravity field, different heat load distributions between the evaporators, and different cooling of the condensers. The measured thermal characteristics were the mecLHP temperature at 12 points, the heat load on each evaporator, and the temperature of the cooling liquid. The device orientation was characterised by the slope φ with respect to the horizontal plane. The heat-load distribution between the evaporators was symmetrical and non-symmetrical. The condensers were cooled by running water with a temperature 10±2°C. Special tests were done with non-symmetrical cooling of the condensers. In some cases only one of the condensers was actively cooled, in other cases the cooling temperature of one of the condensers varied between 20 and 70°C.

As a rule, the device start-up was realised with a heat load of 10 W on each evaporator. Besides, special tests were successfully conducted with 5 W during the start-up. The transition to a higher heat load was realised when a stationary vapour temperature was reached. The heat load was defined as maximum if with its further increase a stationary state was not achieved. The main test conditions of the mecLHP are presented in Table 2.

TABLE 2. Test Conditions of mecLHP.

Orientation (ű)	Heat load distribution (Q)	Cooling of the condensers
ű = 0° horizontal position, the evaporators and the condensers are at the same level	$Q_1 = Q_2$ - variable	$C_1 = C_2$ - constant
	Q_1 - no, Q_2 - variable	$C_1 = C_2$ - constant
	$Q_1 = Q_2$ - variable	C_1 - no, C_2 - constant
	$Q_1 = Q_2$ = constant	C_1 - constant, C_2 - variable
ű = 90° vertical position, the evaporators are above the condensers	$Q_1 = Q_2$ - variable	$C_1 = C_2$ - constant
	Q_1 - variable, Q_2 - no	$C_1 = C_2$ - constant
	Q_1 - no, Q_2 - variable	$C_1 = C_2$ - constant
	$Q_1 = Q_2$ = constant	C_1 - constant, C_2 - no
ű = 0°* horizontal position, evaporators and condensers are in the vertical plane	$Q_1 = Q_2$ – variable	$C_1 = C_2$ - constant

TEST RESULTS

Typical mecLHP test results are shown in the figures 2 to 5, depicting the heat load dependence of the temperatures at some characteristic points, for three different orientations and the same cooling of the condensers. The analysis of these results proves the following:

- The total value of the maximum capacity varies in the range from 1100 W to 1400 W for all device orientations. The maximum heat load on one evaporator, with zero power to the other evaporator, reaches values of 950 W to 1100 W. The weak dependence of the maximum capacity on the device orientation in 1 g is ensured by the high capillary pressure (due to the fine-pored wicks of evaporators) and the relatively small effective length of the device. At a heat load in the range up to 150-200 W the "vapour-liquid" boundary in the compensation chambers is retained. The device operates in the so-called auto-regulation regime, when its temperature with increasing heat load decreases or changes only slightly. In this case the difference between the temperatures of the compensation chambers T_{Ch1}, T_{Ch2} and the vapour temperature in the vapour line T_V is quite small. When the compensation chambers are filled, which happens with a further increase in the heat load, this temperature difference increases considerably, and the dependence T = T(Q) acquires the character of monotonic rise.

- The temperature difference of the compensation chambers (for a uniform heat load distribution between the evaporators) is between 0.1 and 0.5°C. At the same time the temperature difference of the wall of the evaporators at the maximum heat load reaches 12 to 15°C. This great difference is caused by the drawback of the design of the used electric heaters, and the poor control of the local conditions of the temperature-sensitive elements under them.

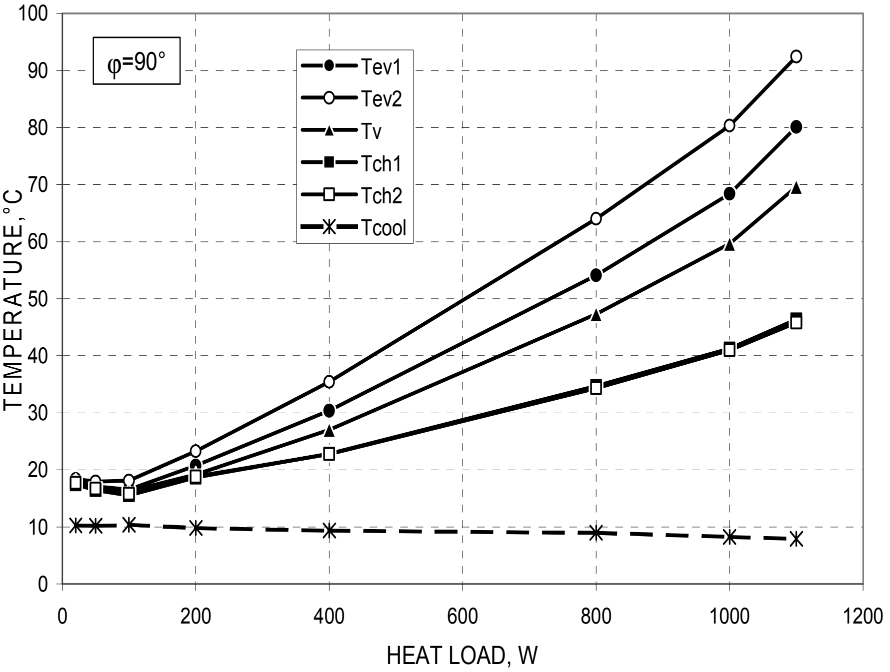

FIGURE 2. Tests with Heat Load Variations: $Q_1=Q_2$, $C_1=C_2$ – constant.

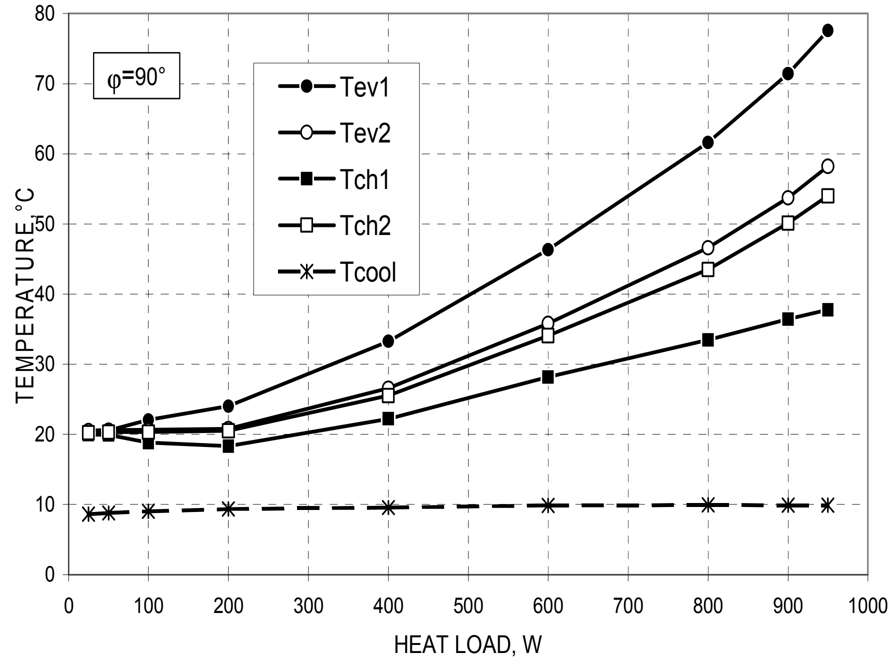

FIGURE 3. Tests with Heat Load Variations: Q_1-variable, $Q_2=0$, $C_1=C_2$ – constant.

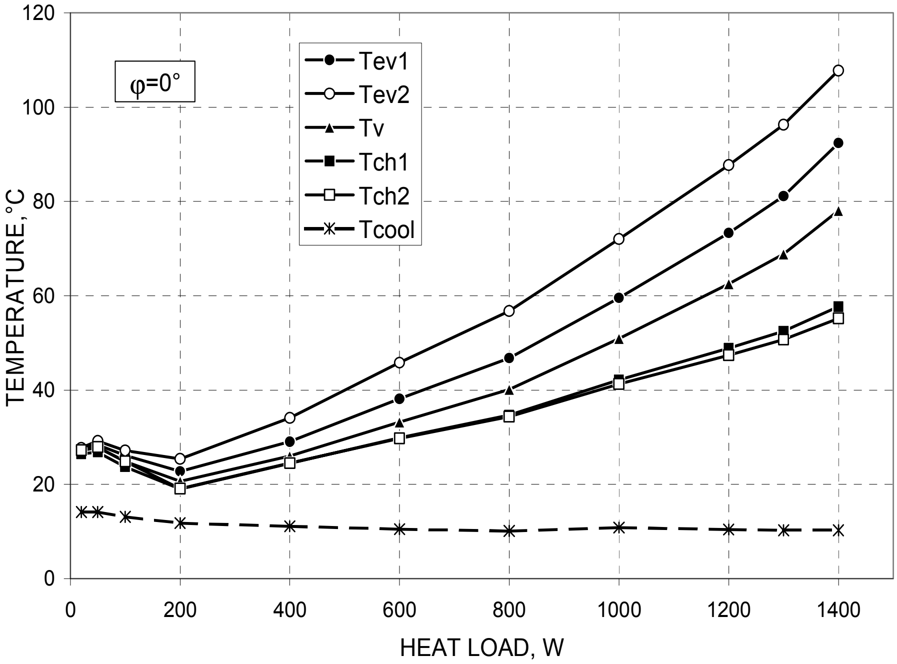

FIGURE 4. Test with Heat Load Variations: Q_1=Q_2, C_1=C_2 – constant.

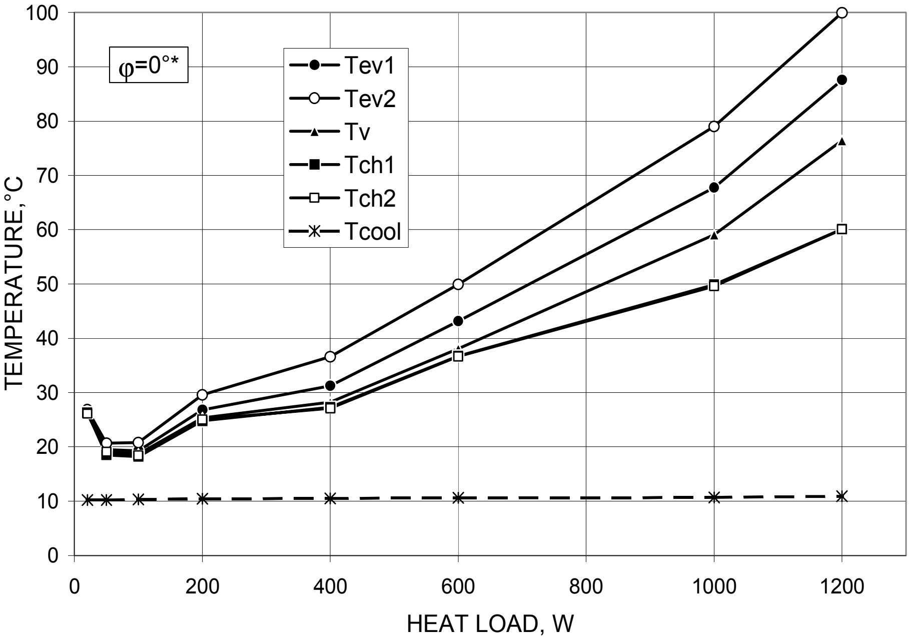

FIGURE 5. Test with Heat Load Variations: Q_1=Q_2, C_1=C_2 – constant.

Figure 6 presents the time diagram of the mecLHP operating temperature for a heat load of 300 W on each evaporator, when the cooling temperature of one of the condensers is kept constant at 20°C, and the cooling temperature of the other is stepwise increased to 70°C. These tests show that the device operates properly until the difference in the cooling temperatures of the condensers reaches 50°C. After that the temperature of the evaporators increases abruptly, which means initiation of a crisis due to vapour penetration into the liquid line. A similar picture is also observed in the absence of active cooling of one of the condensers (Fig. 7). A crisis, which has the same nature, begins here even at a heat load of 200 W on each evaporator. It is preceded by wide temperature variations arising at a heat load of 100 W. It should be noted that such phenomena are not fatal for mecLHPs. They can be prevented with sufficient facility by special design improvements of these devices.

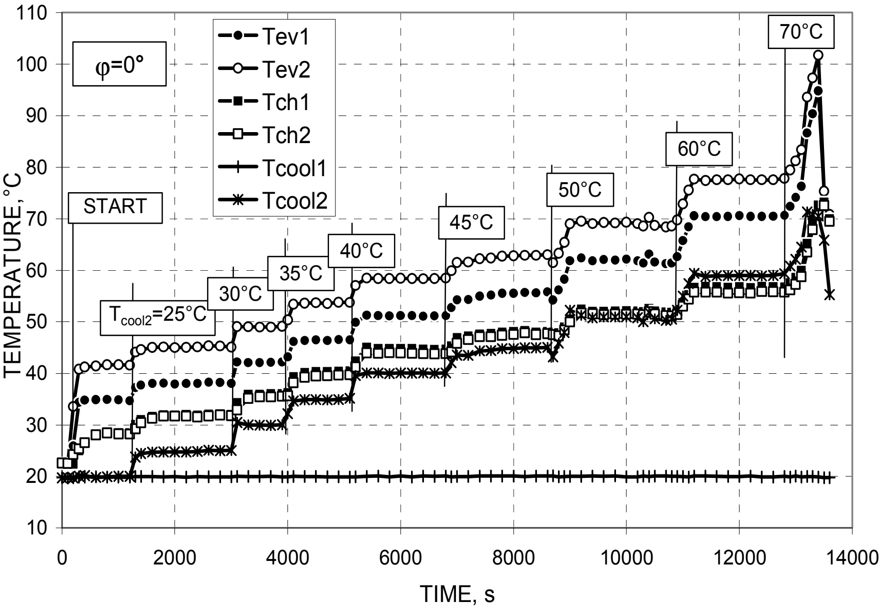

FIGURE 6. Time Diagram with Cooling Variations: $Q_1=Q_2=300W$, $Tcool_1=20°C$, $Tcool_2$-variable.

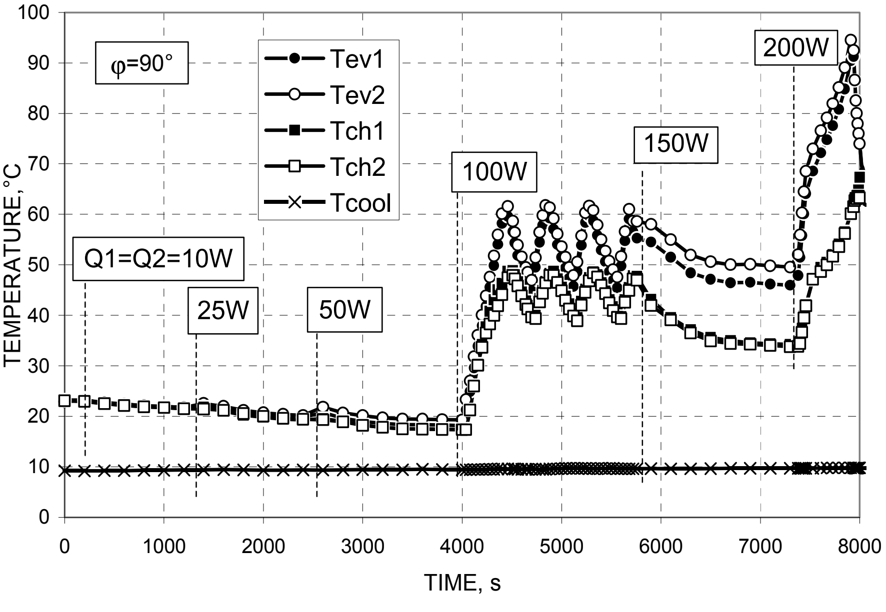

FIGURE 7. Time Diagram with Heat Load Variations: $Q_1=Q_2$, C_1– constant, C_2-no.

HYDRODYNAMIC MODEL

The creation of an adequate mathematical model for a ramified LHP, which would fully describe the device operating characteristics in different situations, is quite a complicated task, which remains to be solved. The hydrodynamic model given below makes it possible to calculate the maximum capacity of a mecLHP at different heat-load distribution between the evaporators and different device orientations in 1-g. The circulation of a working fluid in an LHP is by the capillary heads ΔP_{c1} and ΔP_{c2} created in the evaporators. The pressure balance in the general case is:

$$\Sigma\Delta \; P(Q) \leq \Delta P_c(r_c) \, , \tag{1}$$

where Ɇφ P(Q) is the sum of pressure losses in all the sections of the working fluid circulation, which is a function of the heat load Q. $\Delta P_c(r_c)$ is the capillary head created by a wick with an effective pore radius r_c.

Fig. 8 presents an analytical schematic of a mecLHP. The scheme is divided into elementary sections, which are hydrodynamic homogeneous. The boundaries of the sections are determined by the nodal points A, B, C, D. The hydraulic resistances of pipes and channels of the branches in section AB are defined as R_1 and R_2. The wick resistances as R_{w1} and R_{w2}. Arrows show fluid mass rate circulation direction G and places of heat load supply/removal.

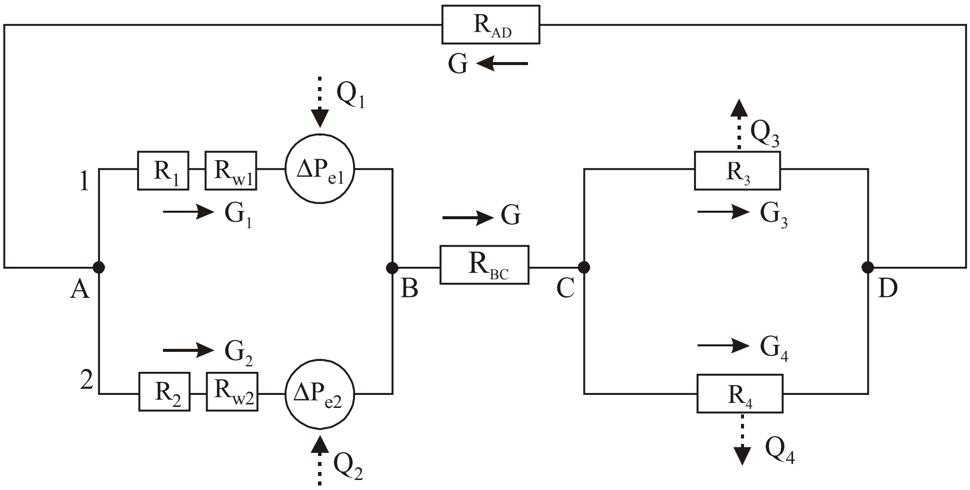

FIGURE 8. Analytical Schematic of mecLHP.

According to this scheme, relation (1) for each evaporator, can be written as:

$$\Delta P_{AB1} + \Delta P_{BC} + \Delta P_{CD} + \Delta P_{DA} + \Delta P_g \leq \Delta P_{c1}(r_{c1}) \tag{2}$$

$$\Delta P_{AB2} + \Delta P_{BC} + \Delta P_{CD} + \Delta P_{DA} + \Delta P_g \leq \Delta P_{c2}(r_{c2}) \ . \tag{3}$$

ΔP_{AB1} and ΔP_{AB2} are pressure losses in each of the branches of section AB. The heat load is formed by the heat loads applied to each evaporator:

$$Q = Q_1 + Q_2 \ . \tag{4}$$

According to the law of conservation of energy and mass, one can write:

$$Q_1 + Q_2 = Q_3 + Q_4 \tag{5}$$

$$G_1 + G_2 = G_3 + G_4 \, . \tag{6}$$

The pressure losses during the motion of a working fluid in pipes and channels (allowing the viscous and the inertial component in the general form) are determined by the equation:

$$\Delta P(G) = R \cdot G^2 \ . \tag{7}$$

For the liquid motion in a wick the pressure losses are determined by the Darcy equation:

$$\Delta P_w(G) = R_w \cdot G \ , \tag{8}$$

R_w is the hydraulic resistance of a wick.

The total hydraulic resistance of section CD can be expressed in terms of the resistances of its branches:

$$R_{CD} = \frac{R_{c1} \cdot R_{c2}}{R_{c1} + 2 \cdot \sqrt{R_{c1} \cdot R_{c2}} + R_{c2}} \ . \tag{9}$$

The conditions (2) and (3) can be written as follows:

$$(R_{BC} + R_{DA} + R_{CD}) \cdot (G_1 + G_2)^2 + R_1 \cdot G_1^2 + R_{w1} \cdot G_1 + \Delta P_g \le \Delta P_{c1}(r_{c1}) \qquad (10)$$

$$(R_{BC} + R_{DA} + R_{CD}) \cdot (G_1 + G_2)^2 + R_2 \cdot G_2^2 + R_{w2} \cdot G_2 + \Delta P_g \le \Delta P_{c2}(r_{c2}) \qquad (11)$$

Equations (10) and (11) make it possible to calculate the mecLHP maximum capacity at different heat load distributions between the evaporators ($Q_{max} = Q_1 + Q_2$) and at different vapour operating temperatures and device orientations in 1-. Figure 9 gives the results of such calculations and their comparison with experimental data Q_{exp}.

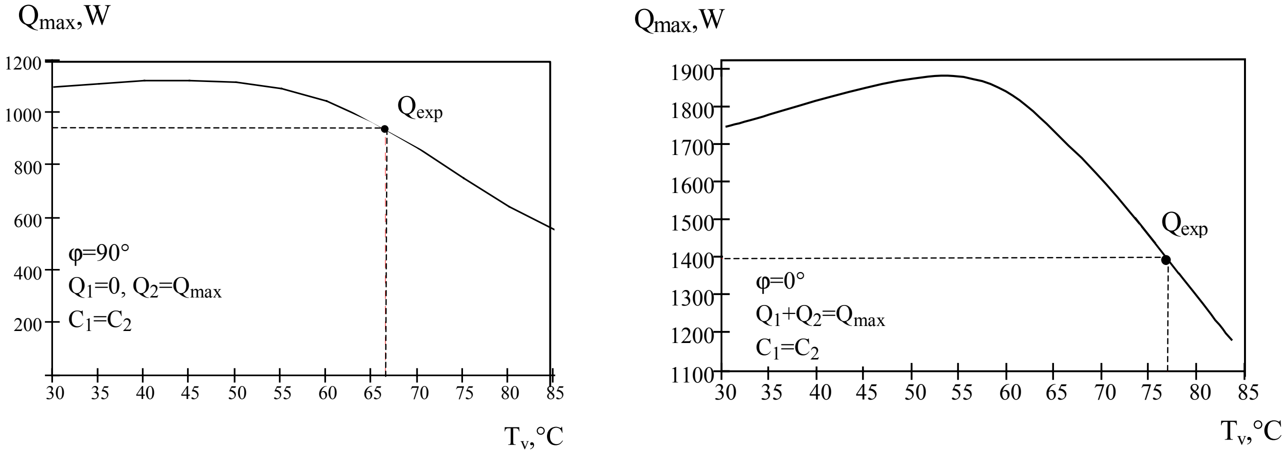

FIGURE 9. The mecLHP Maximum Capacity as a Function of the Vapour Temperature in Different Conditions: Left – The heat load is applied to one evaporator, Right – The heat load is distributed uniformly between the evaporators.

The curve $Q_{max}=f(Tv)$ presented in these plots limits from above the region of admissible heat load values, which can be distributed between the evaporators in different ways. The points lying on the curves correspond to the values of Q_{max} that may be realised at a given vapour temperature.

CONCLUSIONS

1. Tests of a mecLHP containing two evaporators and two condensers at different orientations in 1-g have shown that its maximum capacity varies moderately within the range 1100 to 1400 W.
2. A different heat load distribution between the evaporators does not disturb the device operation.
3. The “vapour-liquid” boundary in compensation chambers exists under given test conditions until the heat load on each evaporator exceeds say 150 to 200 W. The results of tests do not allow us to assert that the aforementioned boundary can exist only in one of the compensation chambers.
4. The device operation remains proper at different temperatures of condensers cooling until the temperature difference reaches a certain critical value, which depends on the heat load on the evaporators.
5. The maximum capacity of the device decreases abruptly if only one of the condensers is actively cooled. The reason for this phenomenon is the penetration of a large quantity of vapour into the liquid line.

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NOMENCLATURE

C	cooling of condenser		<u>subscripts</u>	
G	mass flow rate	(kg/s)	ad	adiabatic (section)
ISTC	International Science & Technology Center		c	condenser/capillary
LHP	Loop Heat Pipe		ch	compensation chamber
mecLHP	multiple-evaporator-condenser LHP		cool	coolant
ΔP	pressure head/drop	(Pa)	ev	evaporator
Q	heat load, power	(W)	l	liquid
R	resistance		max	maximal
r	radius	(m)	v	vapour
T	temperature	(°C, K)	w	wick
TCS	Thermal Control System		∅	orientation w.r.t. horizontal (°)