ANALYSIS OF THE THERMAL BEHAVIOR OF LOOP HEAT PIPE

Nadjara dos Santos, nadjara@dem.inpe.br

Roger R. Riehl, rriehl@dem.inpe.br
National Institute for Space Research – Space Mechanics and Control Division – INPE/DMC
Av dos Astronautas 1758, 12227-010 – São Jose dos Campos, SP Brazil
Phone ++ 55 12 3945-6205, Fax ++ 55 12 3945-6226

Abstract. *In order to analyze the thermal performance of a LHP, this paper presents an experimental and mathematical investigation of a LHP. The experimental performance of the LHP is analyzed, which was designed and built to promote the thermal management on power cycles of up to 80 W, operating with acetone as the working fluid. Results have shown that the proposed experiment presents reliable thermal control within the designed range of heat loads and startups. The experimental results were then compared to a mathematical model, which was able to predict the LHP operation temperature. The mathematical model was based on the steady-state energy conservation equations calculated for each component of the LHP. The comparison of the calculated and experimental results showed good agreement, with little deviation when the operation temperatures are compared. As the mathematical model presented good correlation with the experimental results, its validation is important to be used as a tool for future thermal control in space applications.*

Keywords: *loop heat pipe, thermal control, capillary evaporator, mathematical model.*

1. INTRODUCTION

Artificial satellites are constituted of several electronic and mechanical equipments, which usually dissipate heat and thermal conditions of operation require quite different approaches. Systems consisting of loop heat pipes (LHPs) are of great reliability in thermal control of electronic equipment, structures and satellites, by keeping their temperatures within very restricted operation range and using no moving parts. Those systems operate passively by effects of capillary forces generated in the capillary evaporator, which acquires heat from a source being transferred to a working fluid that operates in its pure state. Several applications of LHPs have showed reliable operation (Ku, 1999; Ku and Birur 2001a; Dutra and Riehl, 2003; Maydanik, 2005; Riehl, 2006c; Singh et al., 2007, Riehl and Santos, 2007). Investigations relating to the variables involved in the project type LHP systems and CPL have been made (Kaya and Hoang, 1999), and that they have been applied successfully in the design, construction and testing of a LHP (Dutra and Riehl, 2003; Launay at al, 2007; Riehl and Santos, 2008). The components of LHP are: capillary evaporator, condenser (or two-phase reservoir), liquid and vapor lines and compensation chamber. It operates by means of capillary forces generated in the evaporator, which is responsible for evaporating the fluid and generating the capillary forces that will drive the working fluid. Then the vapor flows in the vapor line towards the condenser, where it is condensed and flows back to the evaporator by the liquid line. The compensation chamber is responsible for controlling the internal pressure and by adjusting the temperature of the loop in which it operates, in addition to the liquid inventory required for operation of the LHP in the conditions imposed by a given power rate introduced into the capillary evaporator. Thus, the two-phase reservoir accomplishes the control of operation temperature and pressure in order to allow an efficient thermal control. Depending on the power applied to the capillary evaporator, the working fluid flow rate may increase or decrease, which will determine the amount of working fluid that must be supplied or removed by the compensation chamber. Investigations have been realized by Ku (1999) and Ku and Birur (2001b) to identify the parameters that affect the LHP, considering its geometry, the working fluid, the temperature of the heat power source and cold source and the influence of thermal conductance of the fluid on the compensation chamber. However, with respect to build and test an extensive study was conducted by Riehl and Vlassov (2003).

There is a search for alternative working fluid for ammonia, with the objective to replace it with an alternative fluid that represents less hazard of use (Riehl and Dutra, 2005). Several other studies have been made by Ku et al. (2002) Riehl and Siqueira (2005 and 2006) and Riehl (2006c) to improve the design, operation and implementation of the LHP and some mathematical models have been presented, in order to predict the LHPs operation (Kaya and Hoang, 1999; Watson et al, 2000; Vlassov and Riehl, 2006; Kaya and Goldak, 2006; Launay et al.,2008).

The objective of this paper is present an experimental investigation of a LHP using acetone as the working fluid, designed for the thermal management 80W. The experimental results, presented to the LHP operation temperature, were then compared to a mathematical model presented by Santos (2009), used to predict the steady-state operation of LHPs.

2. EXPERIMENTAL APPARATUS

Nowadays, one of the most important objectives of the space activities is the development of two-phase systems of capillary pumping for heat dissipation. Problems of thermal dissipation many times restrict certain applications, due to severe limitations regarding the use of certain components for a given mission.

For the purpose of this research, a LHP was built and tested in laboratory conditions at horizontal orientation. Tests using acetone as working fluid are necessary to better relate to the use of this fluid, especially related to the noncondensable gases (NCGs) influence. In this LHP, the liquid inventory of 25 grams was used, with 50% of the compensation chamber filled with liquid in the cold mode. Following the project requirements, the entire experimental setup was built in 316L ASTM stainless steel tubing and aluminum (alloy 6061) on the capillary evaporator saddle (70 x 45 mm) and condensation plate (300mm x 300mm x 4mm thick) with the geometric characteristics described in Table 1.

Table 1. Geometric characteristics of the LHP

Capillary Evaporator		**Liquid of Line**	
Total Length (mm)	85	Diameter ID (mm)	2.85
Active Length (mm)	70	Length (mm)	8.50
Diameter OD/ID (mm)	19.0/16.5	Material	Stainless steel grade 316L(ASTM)
Material	Stainless steel grade 316L(ASTM)		
UHMW Polyethylene Wick		**Condenser**	
Pore Radius (μm)	4	Diameter ID (mm)	2.85
Permeability (m^2)	10^{-13}	Length (mm)	1200
Porosity (%)	50	Material	Stainless steel grade 316L(ASTM)
Number of Circumferential Grooves	23+1 Axial (TCD-LHP3)		
Compensation Chamber		**Vapor of Line**	
Volume (cm^3)	20	Diameter ID (mm)	2.85
Diameter OD/ID(mm)	19.0/16.5	Length (mm)	5.50
Length (mm)	95	Material	Stainless steel grade 316L(ASTM)
Material	Stainless steel grade 316L(ASTM)		

Twenty type-T thermocouples (deviation of ± 0.3 °C at 100 °C) were installed throughout the loop as presented by Fig.1a. Special techniques for machining the porous wick have been developed to be able to produce the grooves (Fig. 1b), while keeping the proper porosity. Arteries can be obtained by special folding techniques of the secondary structure made of stainless steel screen mesh #200 (Fig. 1c). A data acquisition system was responsible for reading and recording the temperatures, which was used to monitor the loop's behavior during the tests. The condensation plate was the cover plate of a heat exchanger with embedded channels, circulating a mixture of 50% water and 50% ethylene glycol at a rate of 9 l/min. The condensation temperatures used to test the LHP varied from –20 °C to + 5 °C for all tests, which were intended to verify the LHP behavior in different condenser operation as found in many satellites applications. In order to improve the capillary evaporator thermal behavior, its housing was also machined with micro threads on its inner diameter, presenting between 2,500 and 3,000 micro threads per meter.

Heat was applied to the capillary evaporator through an aluminum saddle where a kapton skin heater was attached (15mm x 25 mm, 14.5 Ohms). The configuration for the LHP design presented above was established to perform the thermal control of up to 100 W, even though this specific application would carry up to 80 W. The fittings of the LHP capillary evaporator and compensation chamber were welded, as the rest of the loop, using an orbital automatic system in order to be in agreement with the requirements for space qualification procedures.

All tests were performed without pre-conditioning procedures prior the startups and without temperature control of the compensation chamber. The LHP was tested with a room temperature controlled between 18 and 20 °C, on power cycle test profiles as presented by Table 2. Tests have been carried out to check the LHP performance along time, in order to certify them as thermal control devices for space applications for regular satellites (low orbit) as well as geo-stationeries (high orbit), with attention to the possible effect of NCG influence on their operation along time.

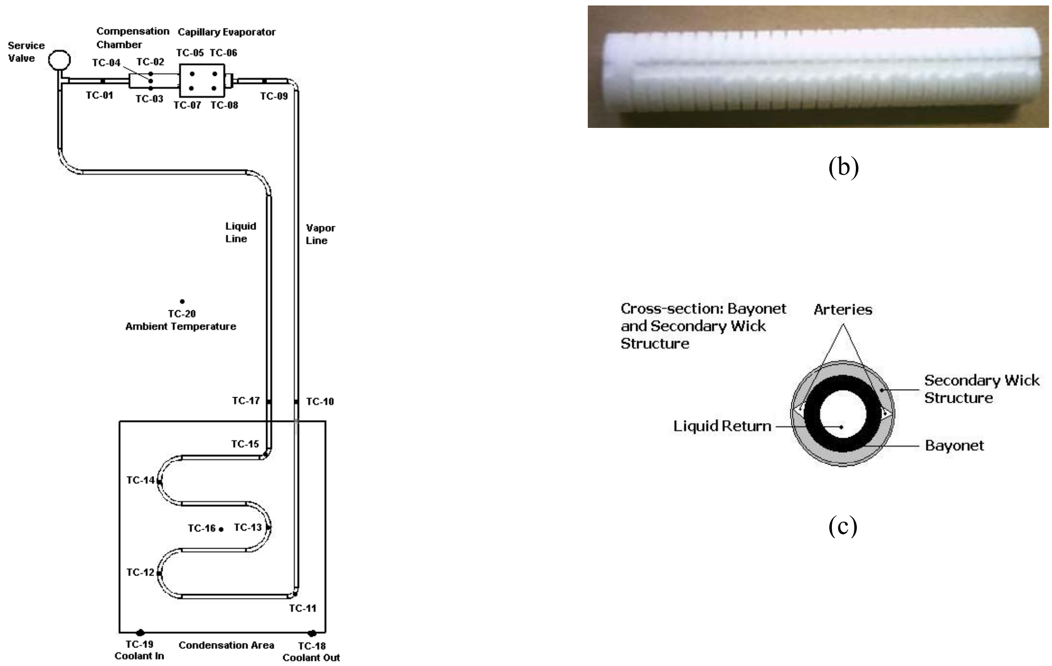


Figure 1. Loop heat pipe setup: (a) instruments locations; (b) circumferential grooves on primary wick; (c) arteries on secondary wick.

Table 2. Power cycles applied to test the LHP.

Profile	Power Cycles (W)	Startup Power (W)
1	20-2-30-2-40-60	20
2	40-10-60-5-20-80	40
3	2-5-1-2-1-5	2

Following the above-mentioned parameters, the LHP could operate upon applying the heat loads to the evaporator, which was important for the proper data analysis. For the startup tests, the objective was to investigate the LHP capability on initiating its operation, where the required superheating and time to start and reach the steady state regime were analyzed. It was also intended to investigate how the LHP operation conditions would interact in order to reach the operation temperature for a given heat load and condensation temperature. The power cycle tests were selected according to potential LHP operation modes. In this test sequence, an important attention is given to profile 3 were reduced heat loads are applied to the capillary evaporator of the LHP in order to evaluate their capability in promoting the heat transport in the sleeping mode. For this entire test sequence, it is important to analyze the devices operation along the life tests, in order to promote potential design improvements and better evaluate the use of the selected working fluid.

3. MATHEMATICAL MODEL

In this model, thermal interactions were considered between the system and the heat load, the sink and the ambient. In addition, to calculate the operation temperature of the system, heat load was applied in the evaporator considering some geometry parameters, temperatures of the sink and ambient as well as the working fluid.

The mathematical model was based on equations of conservation of energy on a steady-state, where for a given application of a heat load, knowing the temperature of condensation and environment, the temperature of operation of the LHP could be calculated. The computational model was developed to operate in steady-state; the distribution of temperature along the LHP does not change with respect to time. In the operation of the LHP, the wick in the evaporator must develop a capillary pressure enough to overcome the total pressure drop in the loop. The advantage of a capillary loop is that the meniscus in the evaporator wick will automatically adjust its radius of curvature in such a way that the resulting capillary pressure is equal to the total system pressure drop (Ku, 1999). The total system pressure drop, for flow in steady state, is the sum of the pressure drops of each component, as presented in Eq. (1) (Faghri, 1995):

$$\Delta P_{total} = \Delta P_{evap} + \Delta P_{wick} + \Delta P_{liq} + \Delta P_{vap} + \Delta P_{condenser} \tag{1}$$

The total pressure drops consists of steady-state pressure drops in the evaporator, evaporator wick, vapor and liquid lines and the condenser.

In the operation of the loop, with the increased power rate, the following condition must be satisfied, where Eq. (2) (Faghri, 1995):

$$\Delta P_{total} \leq \Delta P_{cap} \tag{2}$$

Thus, in steady state, the capillary pressure generated in the evaporator must be balanced with the total pressure drop. For the operation of the model, the list of parameters had to be properly supplied, such as details of the dimensions of each component, heat source, primary wick structure, as well as its permeability and the pore radius, mass flow rate, temperature of sink, external thermal conductance of the condenser. Typical values of external thermal conductance of the condenser per unit length depend on the flow and the thermal resistance of contact between the tubings and plate of the condenser (Kaya and Hoang, 1999).

The compensation chamber is responsible for balancing the temperature and pressure of the system and this is the result of heat transfer under three conditions (Kaya and Hoang, 1999), which were:

1. heat exchange between the evaporator and the compensation chamber (heat leak);
2. heat exchange between the compensation chamber and the environment, and
3. heat exchange between the compensation chamber and the liquid returns to the liquid line.

As presented on Eq. (3), the conservation of energy in steady state to the above conditions are as follows:

$$\dot{Q}_{HL} = \dot{Q}_{SC} + \dot{Q}_{CC-A} \tag{3}$$

The heat leak $\dot{Q}_{HL}$ from the high to the low pressure side of the primary wick is caused by conduction across the wick (Kaya and Hoang, 1999):

$$\dot{Q}_{HL} = \frac{2\,\pi\,k_{eff}\,L_w}{\ell n(D_o / D_i)} \Delta T_{ac.w} \tag{4}$$

The temperature difference $\Delta T_{ac.w}$ across the wick is the difference between the local saturation temperatures caused by system pressure drop:

$$\Delta T_{ac.w} = \left(\frac{dT}{dP}\right)_{sat} \left(\Delta P_{total} - \Delta P_{wick}\right) \tag{5}$$

where $\left(\frac{dT}{dP}\right)_{sat}$ is calculated by the equation of Clausius-Clapeyron (Moran and Shapiro, 2006).

In LHP, capillary forces are generated in the evaporator capillary through a porous structure, which acquires heat and transfers it to the working fluid that is in its pure and saturated state. The vapor generated is responsible for the displacement of the liquid to the condenser during the startup of the device. Once condensed, by the effect of capillary forces, the liquid with a certain degree of subcooling flows back to the capillary evaporator where the cycle is completed.

Following the cycle described above and performing the energy balance, the rejected heat on the two-phase portion of the condenser can be written as follows (Kaya and Hoang, 1999):

$$\dot{Q}_C = \dot{Q}_{APP} - \dot{Q}_{HL} - \dot{Q}_{VL-A} \tag{6}$$

The heat exchange between the vapor line and the environment can be positive or negative depending on the temperature of the sink. For cases of heat transfer with phase change inside the tube, where there is a constant heat flux at the wall, the model for determining the length at which this process occurs can be described by Eq. (7) through the energy balance (Incropera, 1996):

$$\dot{q}''\pi D dz = \dot{m} h_{lv} dx \tag{7}$$

Integrating the equation above, between the inlet and outlet quality, yields:

$$z = L_{TP} = \int_{xin}^{xout} \frac{\dot{m} h_{lv}}{\dot{q}'' \pi D} dx \tag{8}$$

Therefore, the heat rejected in the two-phase portion of the condenser tube occurs in two ways, to the environment and to the sink as shown in the Eq (9):

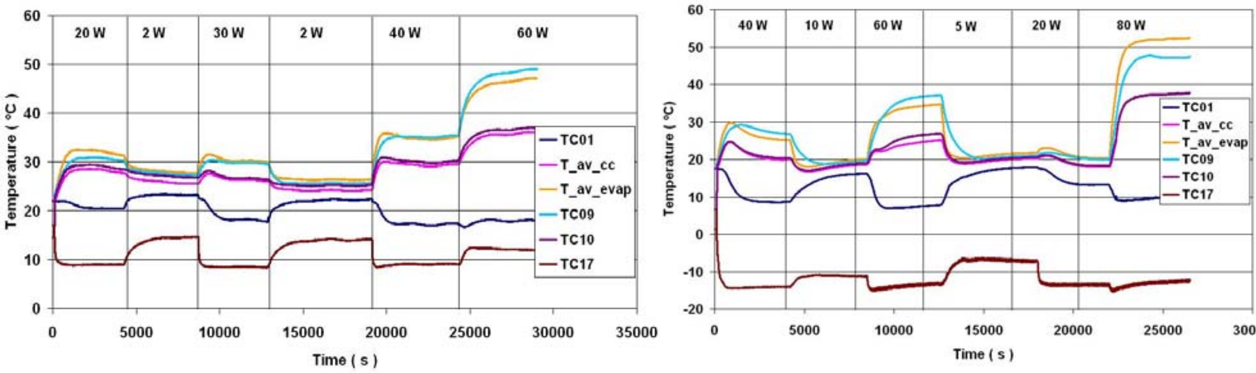
$$L_{TP} = \dot{Q}_C \times \int_{xin}^{xout} \frac{dx}{\left[(UA/L)_{c-s}^{2\phi}(T_{SAT} - T_{SINK}) + (UA/L)_{c-A}^{2\phi}(T_{SAT} - T_{AMB})\right]} \tag{9}$$

where the condenser outlet quality (Moran, Shapiro, 2006) in the equation above, is zero by considering a complete condensation, resulting in only liquid exiting the heat exchanger.

After obtaining the two-phase length in condenser, the liquid temperature at the condenser exit can be calculated, as well as the liquid temperature at liquid line exit.

4. RESULTS AND DISCUSSIONS

Experimental tests were carried out without temperature control of the compensation chamber and pre-conditioning procedures (common in capillary pumped loops-CPLs) for both startups and heat load profile tests. Improvements on the capillary evaporator thermal performance could be achieved when using circumferential grooves machined on the primary wick outer diameter (Riehl and Santos, 2006), which resulted in evaporator temperatures up to 50% lower when compared to a previous design with axial grooves. Figure 3 presents some performance test results for certain heat loads profiles using different condensation temperatures.

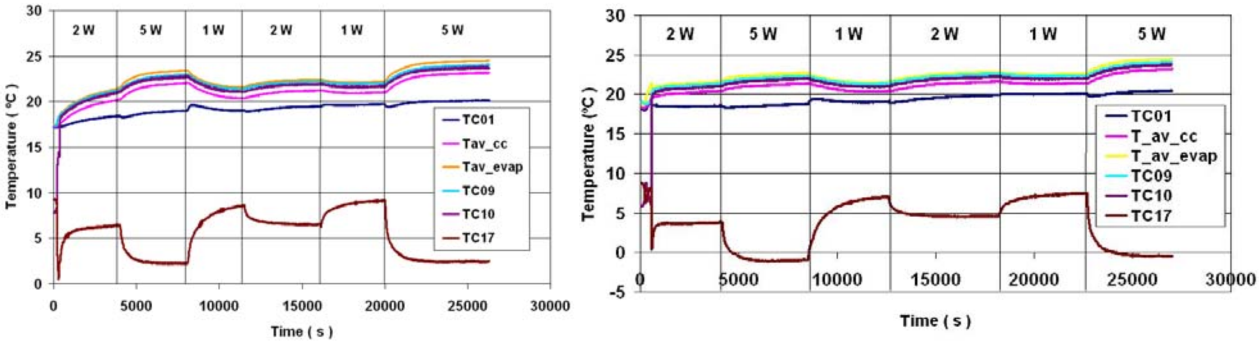


(a) Profile 1- Condensation at 5°C. **(b)** Profile 2 - Condensation at –20°C.

Figure 3. Performance Tests of the LHP.

With the design improvement obtained used the TCD-LHP3 conception it has been possible to perform all the tests required for certifying the LHP while observing reduced heat source temperatures. This is a direct result of the increase on the contact area between the evaporator inner surface and the wick structure, which resulted in an increase of 20% on the contact area while keeping the same active length. This modification on the project of the capillary evaporator resulted in a remarkable operation that could observe through the experimental results.

The LHP also showed reliable operation at both low and high heat loads, especially when its operation at the sleeping mode (profile 3) is evaluated. Continuous heat transport was observed when the LHP was operating at profile 3 for both temperatures of the sink, as presented by Fig. 4. It is possible to verify that the LHP design presented to be robust and efficient when operating in low heat loads.



(a) Condensation at 5°C. **(b)** Condensation at -10°C.

Figure 4. Profile 3-Performance Tests of the LHP: sleeping mode operation.

Evaluating the calculated results using the model presented in session 3 with the experimental data, Fig. 5 shows the comparison of the evaporator temperatures obtained in experiments (Texp) with the calculated (Ttheo).

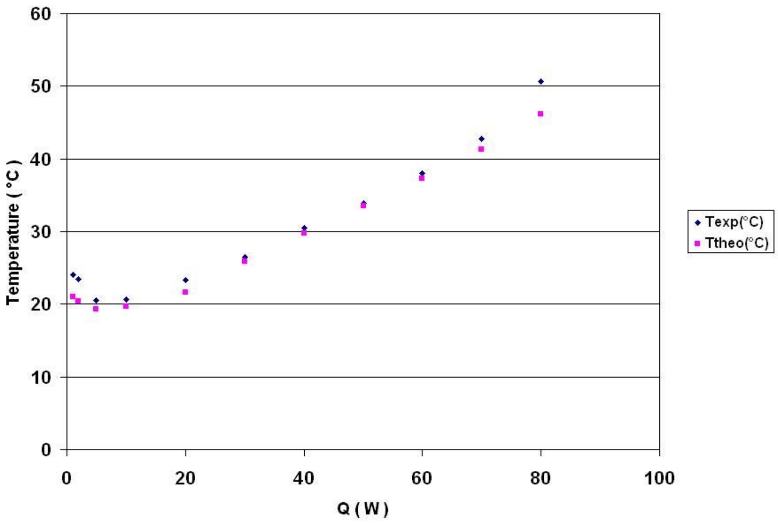
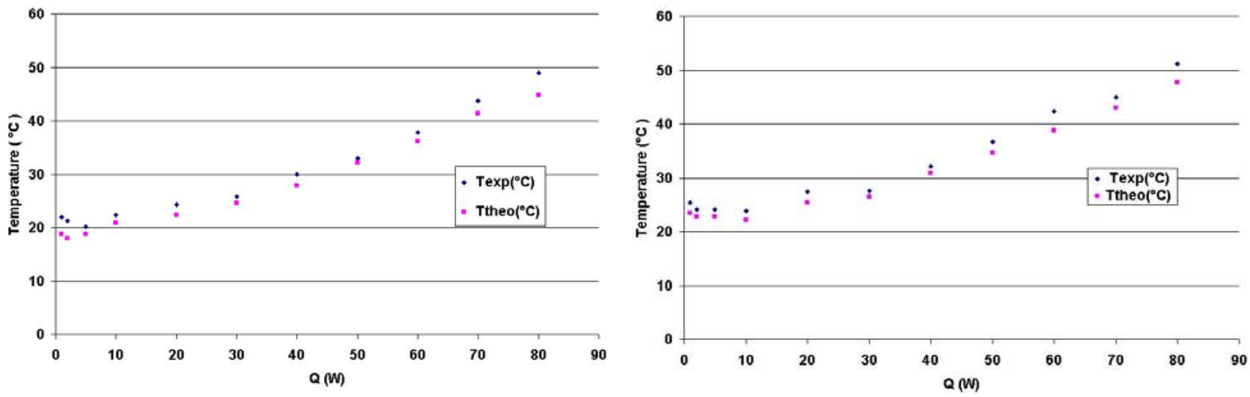


Figure 5. Comparison of experimental and theoretical results of LHP.

In the graph of Figure 5, it can be seen that initially the temperature of operation decreases as the heat load increases to 5 W, which is due to the cold liquid that returns the condenser. This case shows that until reaching this heat load, the LHP operates at a condition of variable conductance, where the sub-cooling is maximum and the liquid enters the evaporator in such a condition that its temperature could be lower than the ambient. After this heat load, the LHP starts operating at constant conductance as the compensation chamber presents a reduced available space where vapor is present, due to the displacement of the liquid in the lines, as the flow rate increase due to the increase of the heat loads. Under those circumstances, the condenser starts operating close to its maximum capacity, reaching the designed capacity at the highest heat load applied to the evaporator. From the comparison presented by Fig. 5, it is clear that the mathematical model is able to predict the experimental results with good accuracy.

Figure 6 (a) shows the comparison of theoretical and experimental results for the sink temperature of 0 °C and Fig. 6 (b) for the sink at 10 °C, assuming an ambient temperature of 19 °C. On both graphs it can be observed the first movement of the trend curve in the region where the variable conductance is occurring at low heat loads. That is when the sink temperature is close to ambient temperature, and when there is a slight subcooling, the temperature of the liquid at the condenser outlet is near the sink temperature.



(a) (b)

Figure 6. Comparison of Experimental and Theoretical Results for the temperature of LHP3 sinks at (a) 0°C and (b) 10 ° C.

Figure 7 compares the temperatures of the thermal resistance between the capillary evaporator and compensation chamber of the experimental data obtained (Texp) with theoretical calculations (Ttheo):

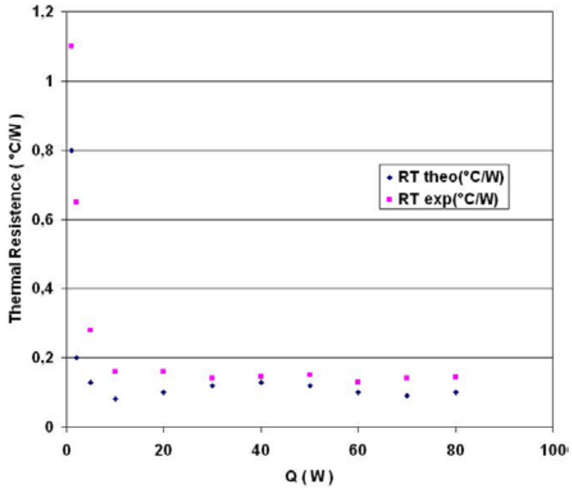


Figure 7. Comparison of thermal resistance: theoretical and the experimental LHP3.

The error between the calculated results with the model proposed above and the experimental results is considered small. In assessing whether the error exists in the readings of thermocouples, one can say that the discrepancy between the calculated results with respect to experimental is within a range of 8%, which is considered acceptable.

5. CONCLUSION

This paper presented the experimental results of an investigation in the LHP, designed to accomplish several project requirements towards its use in space missions as a passive thermal control device. The LHP presented reliable operation during all tests, with acceptable temperature control of the heat source for the highest heat load of 80 W as required by the project. Better thermal performances were achieved using the LHP that presents the capillary evaporator with circumferential grooves as lower heat sources temperatures were observed, which has been already discussed and analyzed in previous publications.

A mathematical model was also presented, where the energy balance was applied for all LHP's components, also considering the influence of the sink and ambient temperatures, heat leak from the evaporator to the compensation chamber and heat loss to the environment. The introduction of the validation of the mathematical model based on experimental results was important because the project could be evaluated in more details, as the model was used to evaluate the LHP as a system and also of each component separately. Moreover, it can be seen that the error between the calculated results with the model and the experimental results was considered small, as it can be observed that the discrepancy between the calculated results with respect to experimental is within a range of 8%.

The investigation presented in this paper has shown the great potential in using alternative working fluids such as acetone for a certain operation temperature, as well as the configurations of the primary and secondary wick structures with reliable results. Upon validating the proposed model with the experimental results, the computer code generated can be used as an important tool for future designs of LHP, since it has shown to predict the operation conditions of this thermal control device with good accuracy.

6. ACKNOWLEDGMENTS

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7. NOMENCLATURE

h_{lv}	= latent heat of vaporization, J/kg
k_{eff}	= effective thermal conductivity of wick, (W/m °C)
L_{TP}	= length of the two-phase flow portion in the condenser, m
L_w	= length of wick, m
$\dot{m}$	= mass flow rate, kg/s
$\dot{Q}_{APP}$	= total heat load applied to evaporator, W
$\dot{Q}_C$	= heat rejected by two-phase portion of condenser, W
$\dot{Q}_{CC-A}$	= heat loss/gain between compensation chamber and ambient, W
$\dot{Q}_{SC}$	= return liquid subcooling, W
$\dot{Q}_{VL-A}$	= heat loss/gain between vapor line and ambient, W
$(UA/L)_{c-s}^{2\phi}$	= thermal conductance per unit length from inner surface of condenser tube to outer surface of condenser plate, W/ (m.°C)
$(UA/L)_{C-A}^{2\phi}$	= thermal conductance per unit length from surface of compensation chamber to ambient, W/ (m.°C)
x	= thermodynamic quality
ΔP_{cap}	= capillary pressure, Pa
T_av_cc	= compensation chamber average temperature in the, °C
T_av_evap	= evaporator average temperature, °C

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