LOOP HEAT PIPES EXPERTIMENTAL INVESTIGATIONS, MATHEMATICAL MODELING AND DEVELOPMENT

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Abstract

This paper presents the development of the loop heat pipe (LHP) technology that has been undergoing in this institute towards its application in future space missions. An experimental program has been developed to execute performance tests in LHPs to check their operationability over the time. An alternative working fluid has been used during the development of this program, which is the high grade acetone, as this fluid represents less hazard and reduced costs of purification and transfer when compared to the ammonia. The performance test results have been used to validate a transient type mathematical model that was conceived to evaluate LHPs performance in space conditions, focusing in future applications of LHPs in satellites payloads, where the condensate film thickness in the radiator condenser tube is determined by the solution of the conjugate equations of energy, momentum and mass balance in the control volume. Evaporator and compensation chamber are both described by a few transient nodes with generalized key parameters. In the transport lines the fluid temperature, pressure, enthalpy and vapor quality are tracked, making the model tolerant to eventual condensation on inner wall surface in the vapor line or vapor migration into the liquid line. The model is closed by the loop overall mass balance. The results of this investigation are of great importance for future LHPs design.

KEYWORDS: loop heat pipes, capillary evaporator, thermal control, mathematical model.

INTRODUCTION

As the interest in developing LHP technology for application in satellites and payloads, efforts have been made towards the development of such a technology that could deal with the thermal management requirements of current satellites and payloads. The LHP has been extensively developed during the last years as this passive two-phase thermal control device has presented to be an important device to proper execute the thermal management of current needs. During the last decade, LHPs have been extensively investigated in order to be applied as passive thermal control devices in spacecrafts and satellites [1-3]. Several applications for LHPs are possible not only for space but also for ground. In space applications, LHPs are mainly developed as thermal control devices of electronics, batteries, structures and sensors and an extensive program for qualifying these devices for flight is highly required. Such a qualification procedure involves launching forces of up to 12-g, thermal cycling and proper thermal management during the device's designed life. For ground applications, LHPs can be applied in several areas such as: refrigeration and air conditioning systems, avionics thermal control, anti-icing systems of aircraft turbines and wings, computer cooling, water heating systems, etc [4]. However, each application must have its own LHP development according to its requirements for proper operation. An important parameter that must be carefully considered is the presence of people where a LHP must operate as usually has ammonia as the working fluid. In this case, an alternative working fluid must be applied but before this can be done, extensive tests must be performed to proper consider a given substance as a potential working fluid. This is necessary because few working fluids have been applied to LHPs so far thus informations regarding long term operation of these devices are rare. Also, when using LHPs as passive thermal control devices, several considerations regarding their long term operation and reliability must be evaluated as their failure could cause serious damage to electronics components. Such considerations are related to materials used, potential chemical reaction between all parts, range of heat loads applied to the capillary evaporator, etc.

The development of such a broad technology requires efforts towards the design, construction, test and qualification of LHPs. In this case, one of the most important aspects related to this development is in regard to the working fluid selected. Instead of using ammonia, which is potential hazard and requires special equipments and training to manipulate it, a choice was done for a low pressure fluid such as acetone. Upon choosing this fluid, all the technological efforts required are focused on the infra-structure generation to use this fluid, which represents less hazard and reduced costs for purification and out-gassing. However, in order to qualify this fluid for future space applications, extensive performance life tests are necessary.

On the same way, as the application of LHPs are specially related to space components a mathematical model should be developed and validated with experimental data. After being validated over certain operation conditions observed in laboratory tests, the mathematical model could then be used to simulate LHPs under space operation conditions which results could be used to better evaluate their performances.

Focusing on the above mentioned aspects related to the current LHP technology development, this work presents a review of recent results obtained at INPE in the area of experimental investigation, mathematical modeling and development of LHPs. The experimental setups are presented along with some results regarding the past 24 months of performance tests using acetone as the working fluid, which are compared to a mathematical model conceived to evaluate LHPs operation in space conditions.

DEVELOPMENT OF THE LOOP HEAT PIPE PROGRAM

The LHP development program that has been undergoing deals with the design, tests and future qualification for space applications related to the current needs to heat management (up to 150 W). The most important variable that needs to be properly considered is related to the selection of the working fluid used. As there is a wish to substitute the so-used anhydrous ammonia, which represent hazard and expensive equipments for purification and fluid transfer, a low pressure working fluid has been applied. The choice is related to the use of high grade acetone that represents less hazard during manipulation and charging, reduced costs of purification and fluid transfer. On the same way, for the levels of operating temperatures (up to 80 °C), ammonia can present a working pressure of 39 bar while acetone presents only 1.2 bar.

As the working fluid must be chemically compatible to the materials applied on the LHP, the development has been focused on the use of UHMW polyethylene wick structures with pore sizes ranging from 2.5 to 6 microns and porosity of 50%, along with 316L stainless steel tubing. Two LHPs with one evaporator and one condenser have been built during the development of this program, being one with a compensation chamber (CC) detached from the capillary evaporator (TCD-LHP) while the other was an integral part of it (TCD-LHP2). During the performance tests, the second configuration was chosen for extensive tests towards space qualification as integrating it in the satellite's payload presents to be more adequate. Figure 1 presents an sketch of the LHP developed along with the instrumentation position. Table 1 presents the geometric characteristics of the LHPs.

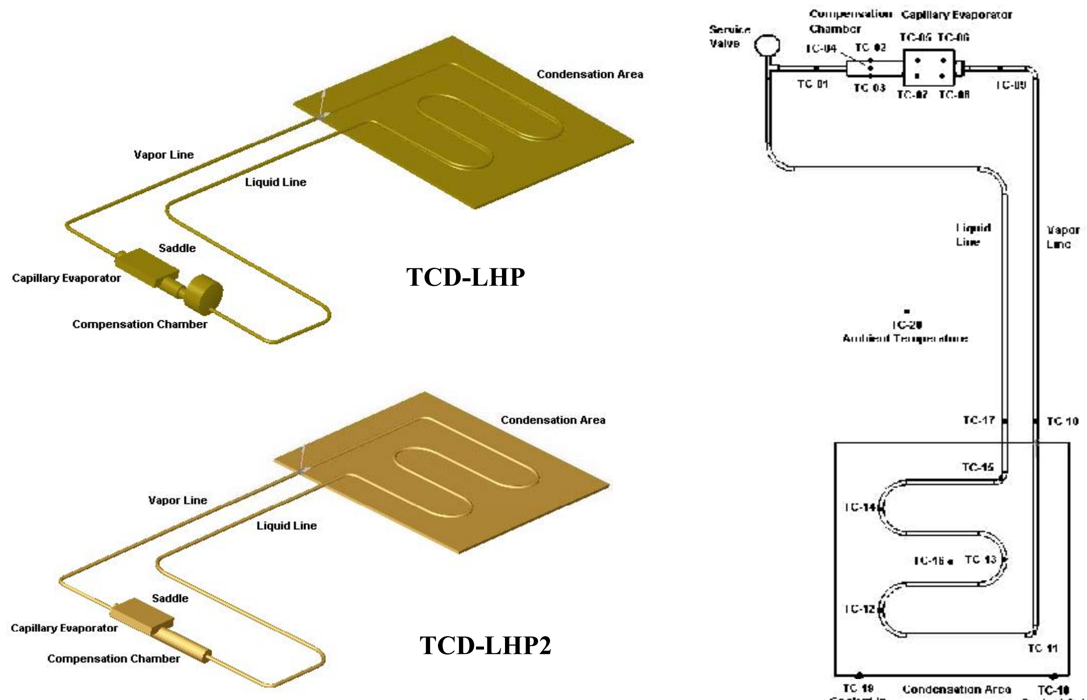


Figure 1. LHP sketch and instrumentation on the TCD-LHP2.

The LHPs are instrumented with 20 type T thermocouples that are read by a data acquisition system, which save the temperature in a spreadsheet file on a sampling frequency of 0.5 Hz. To administrate the heat loads, a kapton skin heater (70 x 25 mm, 14 Ohms) is used which is connected to a DC power supply, The condensation area is composed of a 300 x 300 mm x 4 mm thick aluminum alloy 6063 plate where the condensing tubing is in thermal contact. This plate is the cover of a heat exchanger with embedded channels where circulates a coolant fluid (50% water and 50% ethylene glycol) at a flow rate of 9 L/min. The coolant temperature is set to range from 5 to –20 °C and the LHP is tested over a power cycle routine where the heat load that the LHP must manage varies from 1 to 80 W. High-grade acetone is used as the working fluid, which is twice distilled and out-gassed prior to be transferred to the LHP, on an inventory of 35 ml, keeping 50% of void fraction in the CC in the cold mode. The entire performance test program has been executed in a room with controlled humidity and temperature (ranging from 18 to 20°C).

Table 1. Geometric characteristics of the LHPs.

Capillary Evaporator		*Liquid Line*	
Total Length (mm)	100	Outer Diameter (mm)	4.85
Active Length (mm)	67	Inner Diameter (mm)	2.85
Outer/Inner Diameter (mm)	19.0 / 16.5	Length (mm)	850
Material	Stainless steel grade 316L (ASTM)	Material	Stainless steel grade 316L (ASTM)
UHMW Polyethylene Wick		*Condenser*	
Pore Radius (μm)	6	Outer Diameter (mm)	4.85
Permeability (m^2)	10^{-13}	Inner Diameter (mm)	2.85
Porosity (%)	50	Length (mm)	1000
Diameter (OD/ID) mm	16.5 / 7.0	Material	Stainless steel grade 316L (ASTM)
Grooves height, width, angle	2.0 mm/2.5 mm/26°		
Number of Grooves	15		
Compensation Chamber		*Vapor Line*	
Volume (cm^3)	20	Outer Diameter (mm)	4.85
Screen mesh	# 200 Stainless steel grade 304L (ASTM)	Inner Diameter (mm)	2.85
TCD-LHP OD/length (mm)	45/25	Length (mm)	550
TCD-LHP2 OD/length (mm)	19/95	Material	Stainless steel grade 316L (ASTM)
Material	Stainless steel grade 316L (ASTM)		

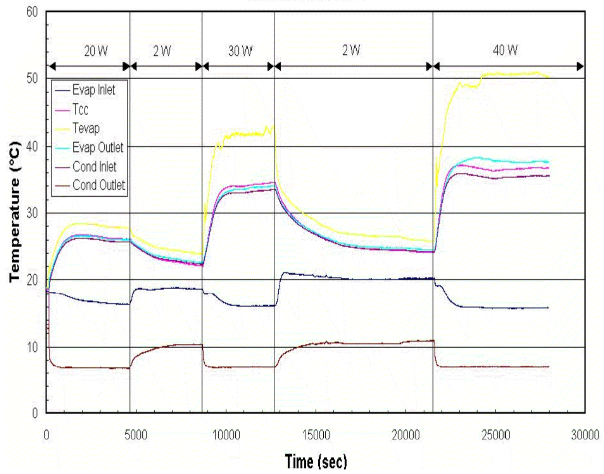
The LHP has been tested on a power cycle routine, using the heat loads presents on Table 2. These power levels are related to the potential heat loads that the LHP might face while operating in space conditions, however they can be changed according to new requirements. The main objective of testing the LHP over these power levels is related to evaluate its capability in managing sudden changes on the heat applied without presenting temperature overshooting or evaporator dryout. More details on the development of the program are given by Ref. [5].

Table 2. Heat load profiles applied to the LHP operation.

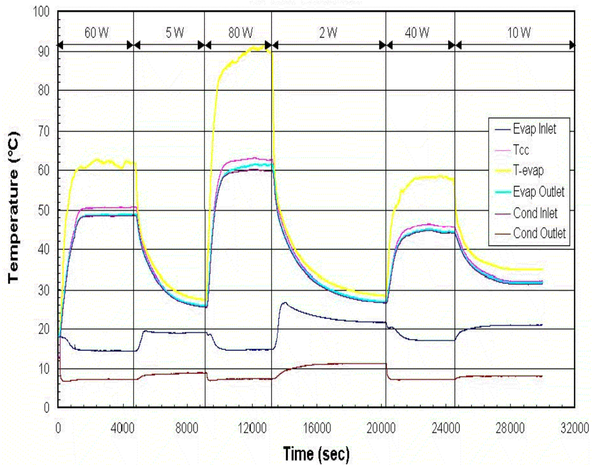
Profile	Power Levels (W)	Startup Power (W)
1	20-2-30-2-40	20
2	40-10-60-5-20-80	40
3	2-10-2-30-50-2	2
4	60-5-80-2-40-10	60
5	2-5-1-2-1-5	2

Experimental Results and Discussion

The TCD-LHP2 has been extensively tested with the objective of evaluating its performance over the time so the analysis related to the materials used and working fluid could be properly verified and potential changes on its design could be done. Some results from this investigation are presented by Figs. 2 and 3 where the power cycle tests were performed showing the reliable operation of the LHP when using the acetone for sink temperature of 5 °C. Temperatures were managed within the limits imposed by the design parameters and the continuous operation of the LHP shows that acetone can be largely applied as a working fluid for the range of power management of this level. During the entire test program that has been undergoing in laboratory conditions, the LHP has successfully performed the thermal management without indications of temperature overshooting or system failure. Actually, the results have been repeated along the time the LHP has been under test and the comparison with all temperatures throughout the loop has been within the thermocouples deviation, which shows negligible influence of non-condensable gases. Figure 3a shows the heat source temperature slope for the tests performed and Fig. 3b shows the thermal resistances obtained during the tests where the LHP was operating with sink temperatures ranging from 5 to –20 °C.

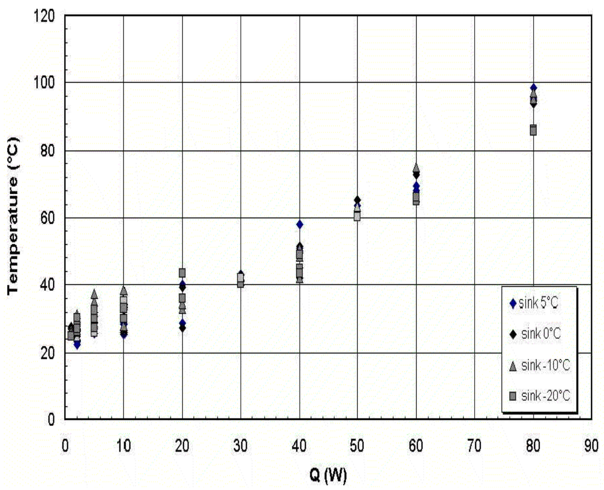


(a) Profile 2

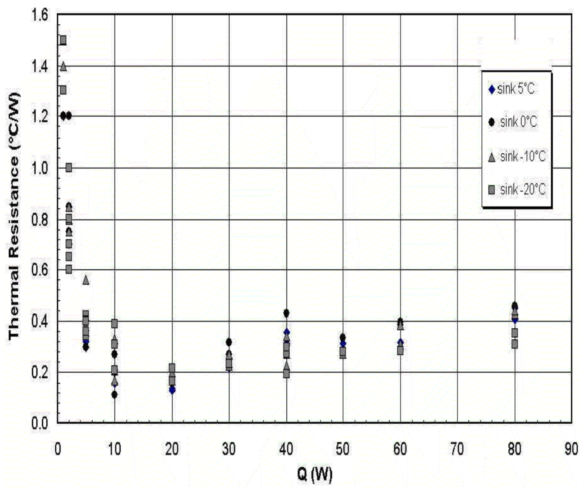


(b) Profile 4

Figure 2. Power cycle tests with the LHP.



(a) Heat source temperatures



(b) Thermal resistances

Figure 3. Heat source temperatures and thermal resistances.

The constant development of the LHP that has been considered for future space applications will result in more reliable configuration and thus operation. Applying acetone as the working fluid has presented to be an interesting option for alternative working fluid and the constant use of it can certainly certify this substance as a promised candidate for substituting the ammonia in LHPs in the near future.

Other LHP configuration has been under development, which is the reversible LHP. This is an interesting configuration as it can operate either way totally passive promoting the proper thermal control of a source and a sink that are constantly reversing their operation mode. A reversible LHP was designed to promote up to 150 W of power management and has been tested also using acetone as the working fluid. Table 3 presents the geometric characteristics of the reversible LHP used during the tests and Fig. 4 presents its schematics.

Table 3. Geometric characteristics of the reversible LHP.

Evaporator/Condenser	**Liquid/Vapor Lines**
Total Length: 185 mm	Length: 600 mm
Active length 85 mm	Outer/Inner Diameters (mm): 6.35/4.85
Inner/Outer Diameters (mm): 16.5/19	Material: 316L stainless steel
Material: 316L stainless steel	
Wick: polyethylene	**Compensation Chambers**
Mean pore radius: 4 μm	Length: 100 mm
Porosity: 55%	Inner/Outer Diameters (mm): 19/16.5
Number of grooves: 15	Material: 316L stainless steel

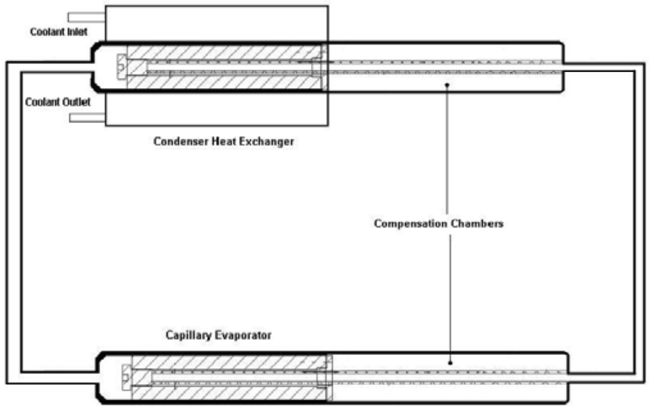


Figure 4. Reversible LHP schematics.

The performance tests for this reversible LHP have been performed as the tests done with the LHP, where power step and power cycles were applied to the capillary evaporator to check its performance. Then, the evaporator was switched to become a condenser and then the condenser became the evaporator and the tests were resumed. Either way, the reversible LHP presented acceptable operation being able to manage heat loads up to 130 W with sink temperature of –10 °C, as presented by Fig. 5. Some flow oscillations are observed during the tests, which are basically due to the draining effect of the porous wick present in the condenser. As the vapor that is condensed back to the liquid phase needs to be drained by the wick structure, some flow oscillations are expected specially when considering wick structures with fine pore radii. However, this behavior does not represent an issue for the r-LHP overall performance.

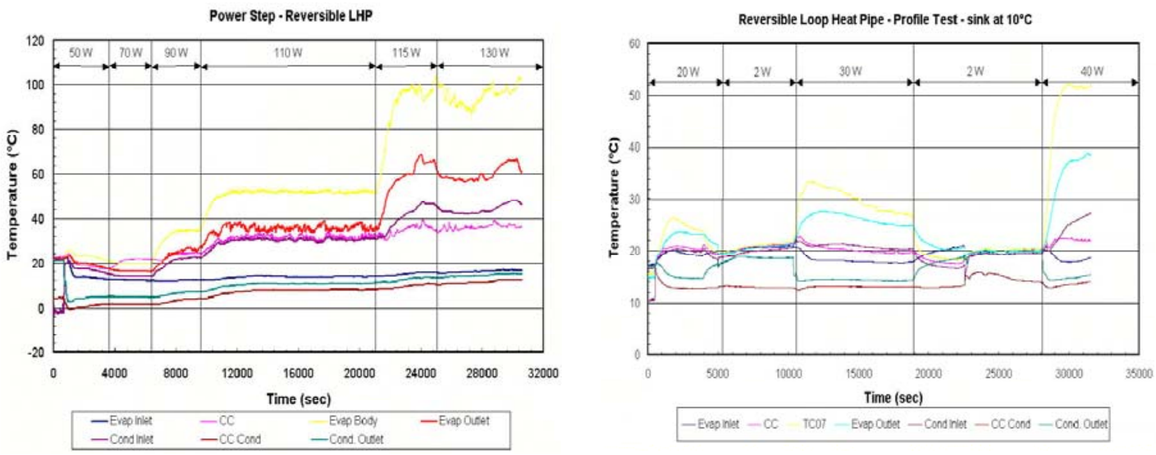


Figure 5. Reversible LHP performance test.

The reversible LHP has also undergoing continuous development as this important configuration finds several applications in both space and ground. Using acetone as the working fluid has also shown to be an interesting choice for this device and it can operate continuously with a reliable thermal management.

Mathematical modeling

A mathematical model has been developed in order to predict LHPs behavior both in ground and Space conditions. The accepted general approach is that prior to use a LHP as a component of thermal control of satellite and thus gather real operation conditions data in Space, the model should be properly validated using ground data under controlled conditions in laboratory. The experimental results obtained with the continuous development of a LHP have been used to validate the model, which objective is to use it as future designing tool for space applications as well as ground applications.

The conception of the LHP integral model is the following. The assembling radiator plate-condenser should be described mathematically as detailed as possible. Such a detailing is important because the temperature of the attached-to-evaporator equipment, which is a key temperature in thermal design, is defined mainly by conditions over the radiator and the design parameters of radiator plate and condenser tube. Such a detailing is possible because the LHP condenser is just a small-diameter tube, and the assumptions of annular pattern and Poiseuille flow in condensing film are applicable here under quasi steady-state approach of hydraulic processes.

As for the assembling evaporator-reservoir (CC), the approach is different. This assembling is represented by a coarse transient sub-model with few uncertain coefficients whose value should be afterward identified by ground test results. A detailed evaporator modeling is a sophisticated problem, whereas a proposed approach of the coarse sub-model yields a practical solution with a model adequacy acceptable for preliminary design purposes.

The radiator-condenser assembling is divided by 110 isothermal transient nodes as it is shown in Fig. 6.

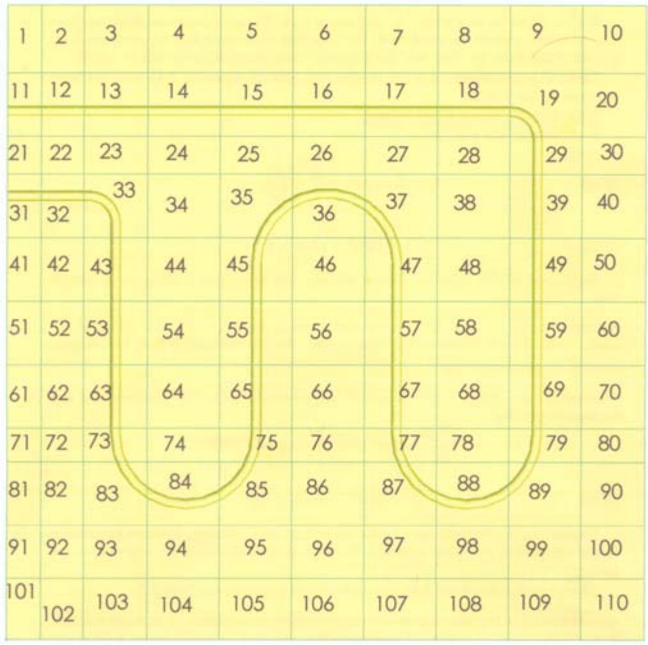


Figure 6. Diagram of nodal layout of the radiator with attached condenser tube.

The radiator plate model is based on the discretization of the following energy balance equation:

$$C\rho\delta_w \frac{\partial T_w}{\partial t} = \delta_w k_w \frac{\partial^2 T_w}{\partial x^2} + \delta_w k_w \frac{\partial^2 T_w}{\partial y^2} + q_w(t,x,y) - h(T_w - T_a(t)) - \varepsilon\sigma T_w^4, \tag{1}$$

where $T_w \equiv T_w(t,x,y)$ is the radiator plate (wall) temperature and the incoming flux $q_w(t,x,y)$ is defined from the condenser sub-model.

The axial momentum conservation for viscous flow in a liquid film is done in the suggestion that the gravity does not disturb the condensing film. Thus

$$\frac{dP'}{dz} + \rho' g \Delta\bar{y}_g = \mu' \left(\frac{\partial^2 u'}{\partial r^2} + \frac{1}{r}\frac{\partial u'}{\partial r} \right), \tag{2}$$

for the following boundary conditions:

$$u'(R) = 0, \tag{3}$$

$$\left.\frac{\partial u'}{\partial r}\right|_{r=R-\delta} = -\frac{1}{\mu'}\tau'_\delta. \tag{4}$$

The liquid axial mass flow rate is obtained by integration over the condenser tube cross-section, which results the expression:

$$\dot{m}' = 2\pi\rho' \int_{R-\delta}^{R} r u(r) dr = -\frac{\pi\rho'\delta^2(2R-\delta)^2}{8\mu}\frac{\partial P'}{\partial x}. \tag{5}$$

From the vapor phase, the tangential stress at vapor-liquid surface is defined as

$$\tau''_{\delta} = \xi \frac{\rho''(\bar{u}'' - u'_{\delta})^2}{8}. \tag{6}$$

For the Eq. (6), the condition of equality of stresses at the interface is used. At each node in the condensation plate, the local energy balance is represented as:

$$\lambda_i \dot{m}_{ci} = G_{\lambda wi}(T_{\delta i} - T_{wi}) - G_{\lambda li}(\bar{T}'_{i-1} - T_{wi}). \tag{7}$$

Evaporator and compensation chamber are both described by a few transient nodes. The capillary evaporator/CC schematically assembly is shown in Figure 8. Temperature definitions and main nodes of the model are also shown as follows.

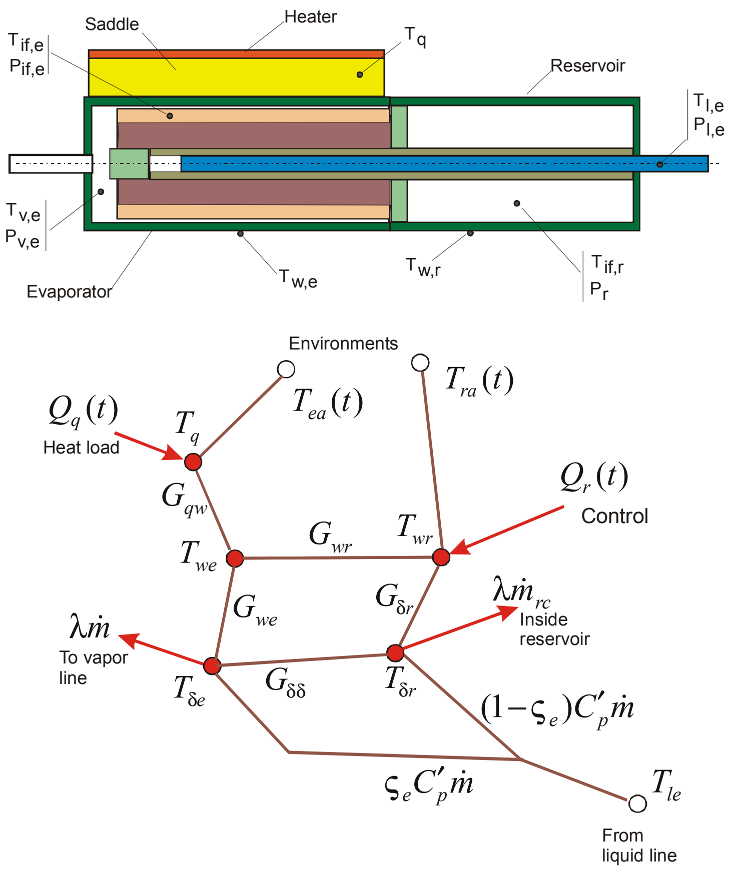


Figure 7. Diagram of nodal layout of the evaporator-reservoir

Note that, in this figure, $T_{if,e} \equiv T_{\delta e}$; $T_{if,r} \equiv T_{\delta r}$.

The basic system of equations, representing the energy balance for the evaporator and reservoir is presented as:

$$Cm_q \frac{dT_q}{dt} = Q_q(t) - G_{qw}(T_q - T_{we}) - G_{he}(T_{wr} - T_{ea}(t)), \tag{8}$$

$$Cm_{we} \frac{dT_{we}}{dt} = G_{qw}(T_q - T_{we}) - G_{we}(T_{we} - T_{\delta e}) - G_{wr}(T_{we} - T_{wr}), \tag{9}$$

$$Cm_e \frac{dT_{\delta e}}{dt} = G_{we}(T_{we} - T_{\delta e}) - \lambda \dot{m} - \varsigma_e C'_p \dot{m}(T_{\delta e} - T_{le}) - G_{\delta\delta}(T_{\delta e} - T_{\delta r}), \tag{10}$$

$$Cm_{wr}\frac{dT_{wr}}{dt}=G_{wr}(T_{we}-T_{wr})-G_{\delta r}(T_{wr}-T_{\delta r})+Q_r(t)-G_{ra}(T_{wr}^4-T_{ra}^4(t))-G_{hr}(T_{wr}-T_{ra}(t)),\qquad(11)$$

$$\left(C'_pV'_{res}\rho'+\frac{1}{2}Cm_{wr}\right)\frac{dT_{\delta r}}{dt}=\lambda\dot{m}_{rc}+G_{\delta r}(T_{wr}-T_{\delta r})-(1-\varsigma_e)C'_p\dot{m}(T_{\delta r}-T_{le})+G_{\delta\delta}(T_{\delta e}-T_{\delta r})\qquad(12)$$

The energy balance is written for the saddle (temperature T_q), evaporator wall (T_{we}), zone of vapor-liquid interface in the evaporator ($T_{\delta e}$), CC wall (T_{wr}) and zone of vapor-liquid interface in CC ($T_{\delta r}$).

The main generalized parameters that should be adjusted by tests are the following conductances: G_{we} (between evaporator wall and vapor-liquid interface in the evaporator), G_{wr} (between evaporator wall and CC wall), $G_{\delta r}$ (between CC wall and vapor-liquid interface in the evaporator), $G_{\delta\delta}$ (between vapor-liquid interfaces in the evaporator and CC, through the path: secondary wick – primary wick). One more important parameter is ζ, which defines the fraction of the enthalpy of the incoming cold fluid directed to the evaporator core; the remainder fraction ($1-\zeta$) is directed to the CC.

In the transport lines sub-models the fluid temperature, pressure, enthalpy together with vapor quality are tracked, making the model tolerant to eventual condensation on inner wall surface in the vapor line or vapor migration into the liquid line. The model is closed by the loop overall mass balance. The LHP liquid inventory is an important input parameter of the model. Interaction between interfaces in the compensation chamber and condenser is the driven mechanism for the determination of the compensation chamber rate of liquid occupancy and the rate of condenser being used.

The mass balance in a LHP is based on current time calculation of the amount of fluid present in the tube lines, condenser and evaporator. Therefore, having in hand the liquid inventory used to charge the LHP, the current amount of fluid in the CC is calculated by the loop balance as

$$M_{fres}=M_{charge}-M_{evap}-M_{cond}-M_{vline}-M_{lline}.\qquad(13)$$

The two-phase mixture in the CC is suggested to be in the saturation conditions. The vapor density is available from the table of thermophysical properties. The current volumes and masses are defined through the basic equations as follows:

$$V_{vres}=\frac{\rho' V_{res}-M_{fres}}{\rho'-\rho''},\qquad(14)$$

$$V_{lres}=\frac{M_{fres}-\rho'' V_{res}}{\rho'-\rho''},\qquad(15)$$

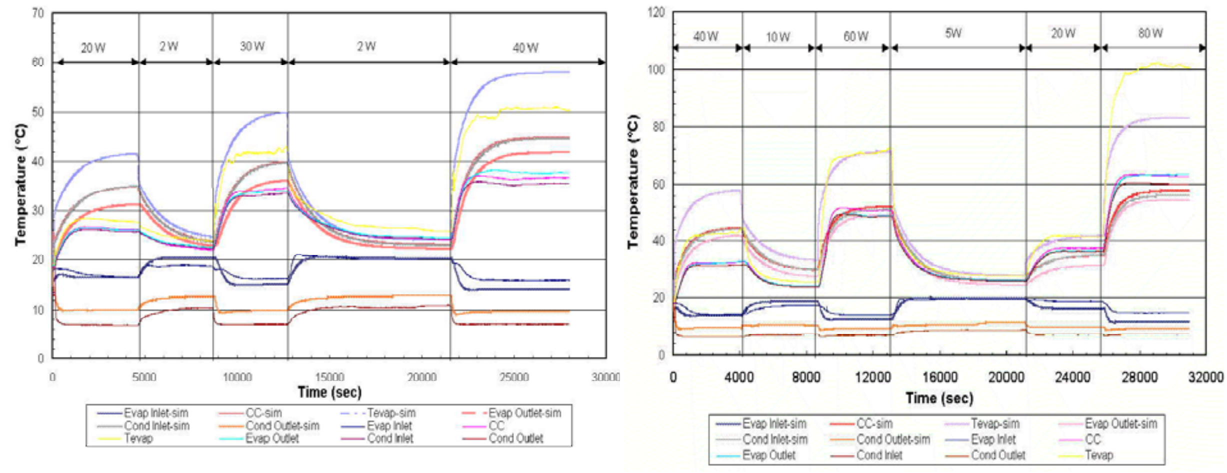
$$M_{vres}=\frac{\rho''(\rho' V_{res}-M_{fres})}{\rho'-\rho''},\qquad(16)$$

$$M_{lres}=\frac{\rho'(M_{fres}-\rho'' V_{res})}{\rho'-\rho''}.\qquad(17)$$

A special numerical procedure was developed to tackle all equations together yielding safe convergence. The mathematical model presented here was solved in order to be correlated to experimental data used to validate the model. An experimental test program for LHPs has been developed and the results have been used to adjust the mathematical model.

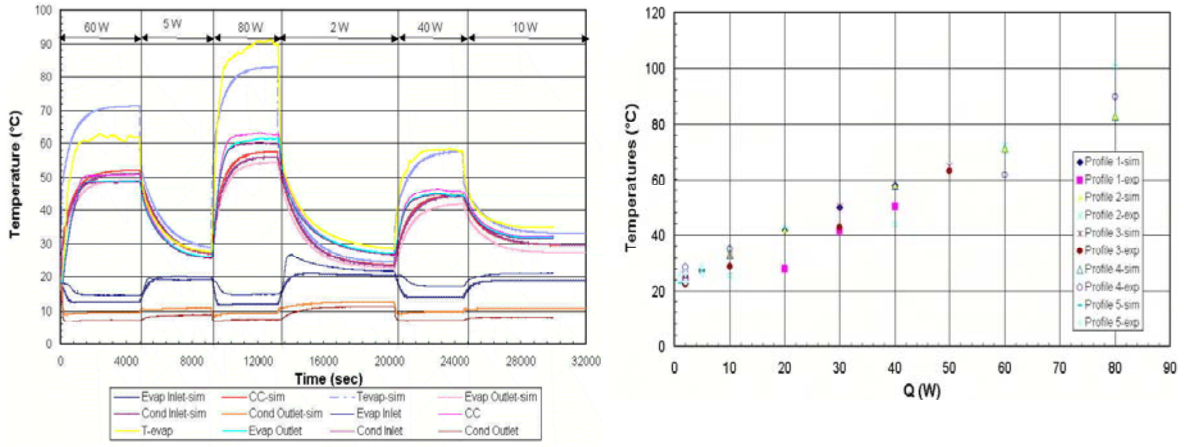
Model validation and results

The mathematical model was validated using the experimental results obtained by the LHP as presented previously. This validation is important to check the model sensitivity in regard to all variables involved during the LHP operation as well as to have it as an important tool for future LHP design for space applications. Using the same geometric characteristics presented on Table 1 and applying the same operating conditions on the mathematical model, for sink temperature of 5 °C, the results obtained on both transient and steady state mode has been achieved as presented by Figs. 8 and 9 where the profiles tested could be calculated as well as the heat source temperatures could be compared. From the comparisons, it can be observed that there is an average error of ± 7% on the calculated results when compared to the experimental, which certifies the model as reliable for the operation of a given LHP. Even when the LHP has to undergo sudden changes on the power applied to the capillary evaporator, just as what has been done during the experiments, the model showed prompt response as the operation conditions were changed and had to seek the steady state again under different conditions.. The reliability of the model in regard to the changes during the operation of the LHP shows its potential and reliability to be applied for design in both ground and space applications as the model can be extrapolated for such conditions.



(a) Profile 1 (b) Profile 2

Figure 8. Comparison between the experimental and simulated results.



(a) Profile 4 (b) Heat source temperature

Figure 9. Profile test and heat source temperature comparisons.

The results related to the heat source temperature present an average error of ± 7% which are considerably good for this kind of application, specially when evaluating the transient behavior of the mathematical model when compared to the experimental results. The model is able to present an expected behavior when the heat source has undergoing changes on the heat load with accurate prediction of the LHP overall performance. Thus, the model can be validated with the presented results and can potentially be used to predict LHPs thermal behavior in space conditions for given operating parameters.

CONCLUSION

This paper presents the development of LHPs destined for both ground and space applications where performance life tests have been performed in order to better define design parameters for such devices. An experimental program has been established to design, build and test LHPs operating with acetone as working fluid with a wish to substitute the so-used ammonia in passive thermal control devices.

Performance life tests on LHPs (classical and reversible configurations) have been made where they show the capability of these devices on transporting heat according to their design parameters. Continuous operation in both heat load profile tests and power steps could be achieved showing the potential in using high grade acetone as working fluid. The continuous operation of these devices have also presented negligible influence of non-condensable gases and reliable heat management for the power range applied to the heat sources.

A mathematical model, developed to determine the thermal behavior of LHPs in ground with aim for space applications, has been derived. For the geometric characteristics of the LHP tested and applying its operating conditions, the model could predict the LHP behavior with reliable accuracy. Both the transient and steady-state operation mode of the proposed mathematical model ensure its validation and an extrapolation on its parameters can surely attribute to the model capability in applying it for space applications.

Nomenclature

A	area (m^2)	***Greek Symbols***	
C	specific heat (J/kg K)	δ	Thickness (m)
G	thermal conductance (W/K)	ε	Emissivity
h	heat transfer coefficient (W/m^2 K)	λ	latent heat of evaporation (J/kg)
k	thermal conductivity (W/m K)	μ	dynamic viscosity (Pa.s)
L	length (m)	ρ	Density (kg/m^3)
M	Mass (kg)	σ	surface tension (N/m)
$\dot{m}$	mass flow rate (kg/s)	σ	Stefan-Boltzman constant
P	pressure (Pa)	ξ	friction factor
Q	heat flux (W)	τ	tangential stress
q	heat flux density (W/m2)	***Subscripts***	
r	Radius (m)	e	Evaporator
R	internal radius of tube (m)	f	Fluid
s	coordinate, either x or y	h	hydraulic or convective
T	Temperature (°C)	l	Liquid
V	Volume (m^3)	sat	Saturation
u	velocity along x-axis (m/s)	q	saddle (at interface to equipment)
Δy	elevation (m)	r, res	reservoir
x	coordinate or mass fraction of vapor	w	wall
Superscripts		δ	if at interface
''	vapor phase		
'	liquid phase		

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