DEVELOPMENT OF PROPYLENE LHP FOR EUROPEAN MARS ROVER APPLICATIONS

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Abstract

This paper presents the modeling and experimental results during the development and investigation of Passive Variable Thermal-Conductance Device for Mars Rover Applications. The device is composed by a Loop Heat Pipe and a pressure regulating valve. Both function as a heat switch and temperature control element. Propylene was selected as a working fluid due to challenging Mars environmental conditions. Excellent agreement between EcosimPro model and test data was found. Special investigation with design optimization was conducted to demonstrate LHP functionality at any arrangement with respect to gravity vector, including worst possible orientation and environmental operational scenarios. High robustness and effectiveness of LHP with pressure regulating valve as a heat switch for Mars Rover thermal control applications were fully validated through extensive test campaign.

KEYWORDS

Loop Heat Pipe, Pressure Regulator Valve, Heat Switch, Variable Thermal-Conductance Device, EcosimPro, Thermal Control, Mars Rover, Secondary Wick

INTRODUCTION

The main objective of a thermal control system is to maintain some components within some appropriate temperature limits. This objective usually only implies the need of thermal power removal to prevent electronics to exceed maximum temperature upper limit. But for some applications where the heat dissipating units operation modes and/or the external environmental conditions vary in a wide range, there is also the need for some periods to introduce thermal power in the system to keep it above a temperature lower bound. A constantly running heat power pumping system will require much more heat power injection in the situations where there is not need to remove heat. For space applications, the availability of power is limited. In such circumstances, a kind of heat or thermal "switch", in which the pumping capacity of the thermal control system may be turned ON/OFF, is of great interest for power (and mass) optimization.

A clear example of such applications is the thermal control system for a Mars rover. During the Martian day, the environment is relatively hot (~ 10 ºC) and the on-board instruments are usually in operation; heat power must be dissipated. In night time, the ambient conditions are well below -100ºC and the payload units are normally off; heat power must be applied to counterbalance the heat leaks to the very cold environment.

Capillary Pumped Loops are providing very effective thermal control solution for many current and ongoing space missions. Current thermal demands (or rather challenges) of onboard electronic are leading to very extensive use of two-phase heat transfer systems. Especially it is related to planetary exploration missions as soon as no other technology today can provide such degree flexibility as two-phase capillary pumped loops taking into account its high ratio between heat transfer capacity and mass, energy-independence and self-regulation.

In the frame of its Exploration Program, ESA is promoting and carrying out number of activities to support and prepare future robotic and human mission to the Moon and Mars. Technological Research Program "Passive Variable Thermal-Conductance Device (VTCD) for Mars Rover Applications" is one of such activities. The objective of the TRP work is the development and experimental validation of a VTCD (or Heat Switch) for passively controlling the temperature of a dissipating unit. Industrial team on the

strength of IberEspacio and Carlo Gavazzi Space was established for the development, design, manufacturing and testing of this Heat Switch, to demonstrate its performance at breadboard model level.

ESA have specified following governing requirements for Mars Rover Heat Switch:

- Variable thermal conductance link with:
  - Global thermal conductance (defined between evaporator and radiator) in ON conditions ("closed" switch): >0.75 W/K;
  - Global thermal conductance in OFF conditions ("open" switch): <0.015 W/K;
  - Transition between ON and OFF conditions: < 20 K;
  - Transition Temperature set point: TBD, in the range from –40 to +40 ºC;
- Maximum/minimum operational temperatures (evaporator side) +50/-40 ºC;
- Maximum/minimum non-operational temperatures (evaporator side) +60/-50 ºC;
- Maximum heat transfer capacity: ~40 W;
- Passive and self-regulating;
- Minimum mass and volume;
- The design aim shall be the correct operation of the device in any orientation under 1g. Design features, which preclude reaching this aim, shall be fully justified and subject to ESA approval.

To address the thermal control system issue, a passive thermal switch is proposed. It is based on a conventional loop heat pipe (LHP) coupled with a valve which can bypass the condenser when there is no need of pumping heat. Analogous thermal scheme was successfully implemented in a number of satellites [1-4] as a temperature control system and it was tested and qualified in Lavochkin Association/TAIS for MARS96 mission [5].

HEAT SWITCH DESIGN

After extensive trade-off of avilable technologies (mainly Phase Change Material and Two-Phase systems were analyzed [6]) the design based on the LHP technology was preferred. Pressure regulating valve was selected as a heat flow controlling element because this solution is totally passive and self-regulating. The selected working fluid is propylene to avoid freezing problem in the condenser during Mars night time cold temperatures.

General view of Heat Switch is shown on Figure 1

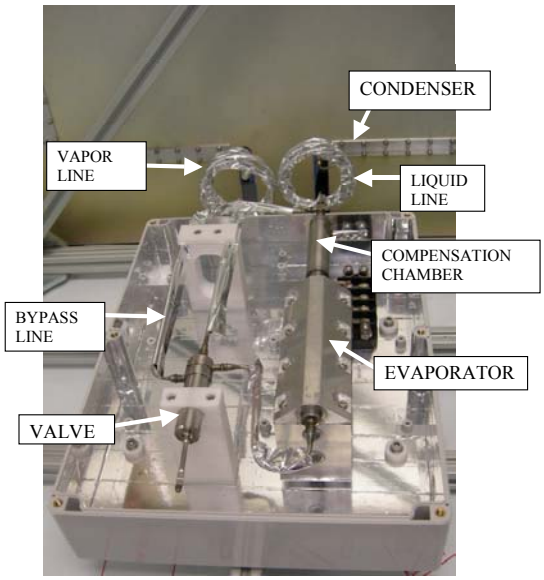


Fig. 1. General view of Heat Switch (evaporator Model I)

The main characteristics of the design are:

- Condenser line: 2.3 m, Ø3 mm;
- Vapor line: 0.5 m, Ø3 mm;
- Liquid line: 0.8 m, Ø2 mm;

- Evaporator: 120 mm, Ø12 mm;
- Sintered nickel primary wick with 1.4 μm pore diameter, 60% porosity;
- Compensation chamber: 65 mm, Ø17 mm;
- Radiator: aluminum honeycomb panel 0.65x0.6x5.9 mm;
- Condenser tube attached to the radiator with aluminum saddles;
- Propylene charge: 9.5 g;
- LHP mass without radiator nor saddles: 254 g;
- LHP mass with radiator and saddles: 2259 g.

The “switch” capacity of the LHP is provided by the valve, which passively splits the evaporator flow to the bypass line or the condenser in accordance with the working fluid pressure. This pressure is directly related with the working fluid temperature given by the saturation line dependency. For low temperatures, the flow is completely diverted to the bypass line (OFF mode). When the temperature reaches a given set point, the flow goes completely to the condenser (ON mode).

The valve provides also a “temperature regulation” capacity when it allows the flow to be shared between bypass line and condenser (intermediate ON/OFF mode). Effectively, for thermal power under a certain limit, the valve reaches an intermediate position, which allows a suitable cold/hot flow mixing ratio to keep the evaporator around the required set point.

The valve set point is established, in accordance to the application requirements, by an adjustable back pressure in the valve. This backpressure is obtained by argon (or others) gas charging. For this application, a 17.5 ºC set point has been established, with a corresponding argon pressure (at room temperature) of 11.1 bar. The transition temperature range from OFF to ON modes is 8.5-17.5 ºC.

When in OFF mode, this design provides an excellent thermal insulation from the radiator, being the heat leak due mainly to thermal conduction through the small transport lines cross section.

Due to the great variety of design cases combining possible rover orientations and environmental scenarios a second LHP evaporator (model II) was manufactured to optimize the design of the secondary wick (SW).

General view of model I evaporator with technological condenser is presented on Figure 2.

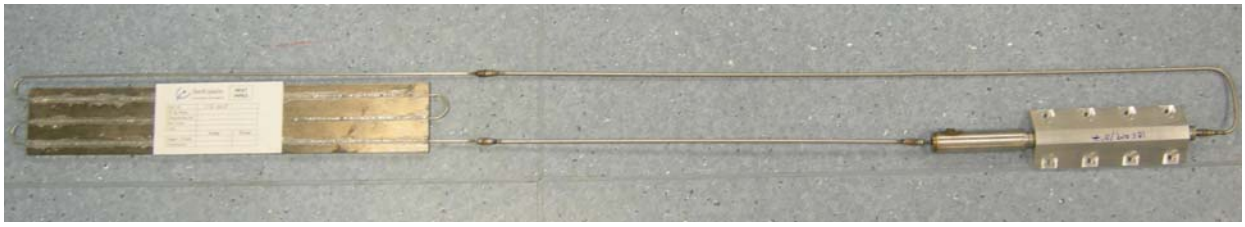


Fig. 2. Model I evaporator LHP with technological condenser

Second LHP have following characteristics:

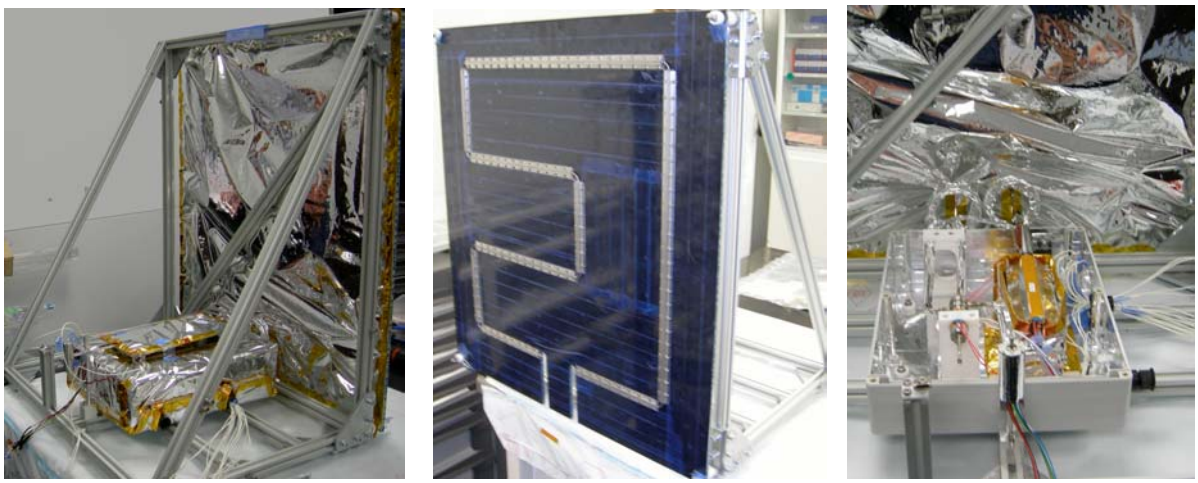
- Condenser line: 1.1 m, Ø2 mm;
- Vapor line: 1.2 m, Ø3 mm;
- Liquid line: 0.55 m, Ø2 mm;
- Evaporator: 120 mm, Ø12 mm;
- Sintered nickel primary wick with 1.4 μm pore diameter, 60% porosity;
- Propylene charge: 9.2.

EXPERIMENTAL PROGRAM

Test Approach

Two LHP prototypes were manufactured and tested. First prototype (Evaporator Model I) with standard evaporator was breadboard model of Heat Switch (see Figure 3). This model was investigated in ambient and vacuum environment according to prepared test program in frame of ESA TRP. The objective was to perform a detailed characterization of the unit as a Heat Switch.

An extensive 2 months test campaign has been accomplished to demonstrate heat switch design compliance with the requirements. The test campaign has been performed mainly in vacuum conditions. Specific test plan has been prepared to test the following features:



a) Test setup; b) Radiator c) LHP Evaporator and valve layout

Fig. 3. Views of Heat Switch assembly for testing in vacuum chamber

- Minimum start-up power for several controlled LHP initial conditions;
- Thermal power capability;
- Thermal cycling to check regulating valve actuation;
- LHP response under sharp thermal power variations;
- LHP characteristics for adverse orientations under 1g;
- Conductance in ON and OFF modes;
- Behavior for a Mars sol representative cycle (in terms of thermal power and ambient conditions);
- Heat Switch behavior with a very cold sink (-120 ºC) obtained with LN2, performed in ambient conditions.

Second prototype (Evaporator Model II) had optimized design of SW for LHP operation at any possible orientation in gravity field. Internal IberEspacio test plan was developed and issued to verify LHP performance at different tilts and stressful transient regimes (when SW properties are critical for proper operation of LHP). The LHP was well thermally insulated and all tests were conducted in ambient. The thermocouple layout is shown in Figure 4.

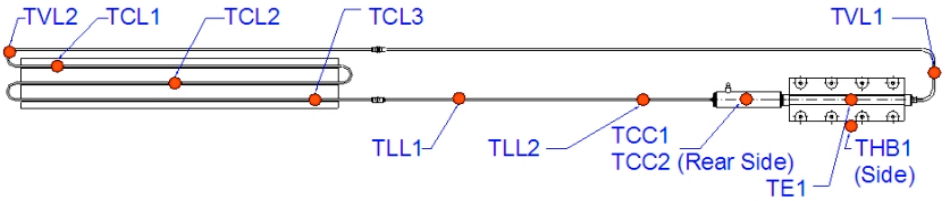


Fig. 4. Thermocouple layout of Model II evaporator LHP

Model I Evaporator Experimental Investigation

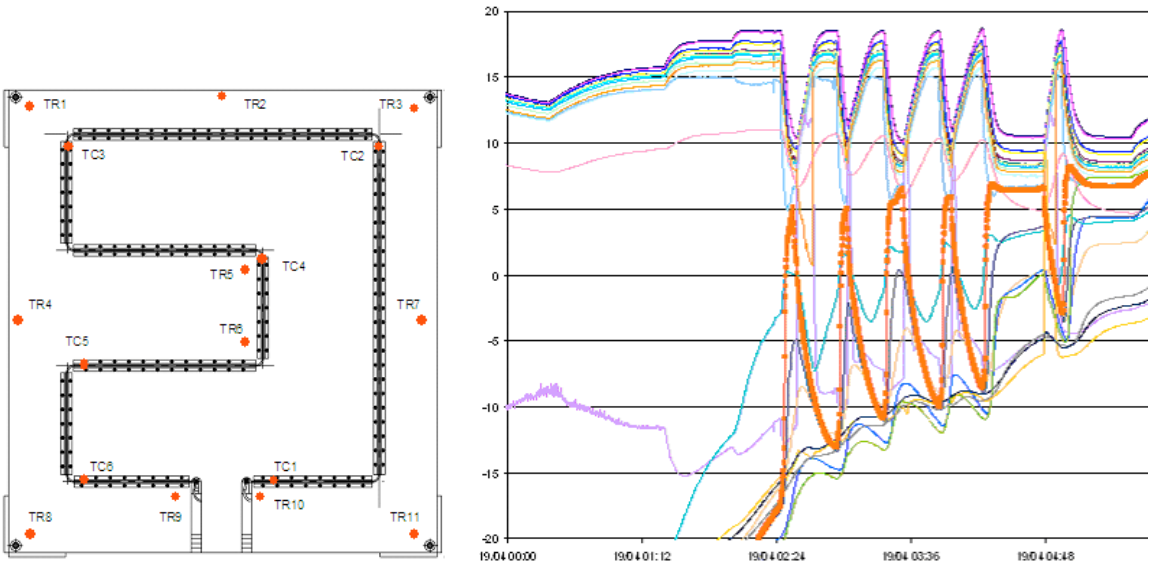
The Heat Switch tests were conducted by Carlo Gavazzi Space with IberEspacio support. The general conclusion is that the “performance of the Heat Switch exceeds the requirements, and shows excellent conductance values and heat transport capabilities”:

- Global thermal conductance in ON conditions: 5 W/K versus 0.75 W/K required;
- Global thermal conductance in OFF conditions: 0.009 W/K versus 0.015 W/K required;
- Transition between ON and OFF conditions: ~6 K versus < 20 K required;
- Demonstrated maximum heat transfer capacity: ~55 W versus 40 W required

(the limit being constituted by the reaching of the maximum operational temperature for given radiator area at vacuum chamber cold temperature -55 °C. Therefore is not intrinsic to the Heat Switch and heat power could be still raised if a colder sink would be provided).

The valve regulation behaviour was "extremely satisfactory, preventing the system to cool down whenever the power levels or the external temperatures bring the system below the temperature threshold".

In the power range for which the valve provides a regulating operation, the valve usually reaches a stable position within its stroke. Some valve oscillations between the ON and OFF positions have been observed during the tests, for certain combinations of low powers and shroud temperatures. These are clearly explained by the gravity effect in the condenser. The geometrical layout of the loop with parts of the condenser 0.6 m above the evaporator can make the gravity to assist the circulation of fluid through the loop with the corresponding effect being very dependant on the position of vapor front. Gravity assistance is maximal when the front is at maximum height in the radiator, while it is negligible when it is leveled with the evaporator. Gravity assistance can be so high that the hydrostatic head in the condenser becomes greater than the pressure losses in the loop. In this case the loop can work as thermosyphon. Then, if the valve tries to open the bypass to compensate a temperature decrease backward flow will occur in the bypass, the regulation effect of the valve will be lost and oscillations will arise. The phenomenon is illustrated on Figure 5.



a) Temperature Sensors Layout on Radiator b) Experimental data. TC2 – dotted line

Fig. 5. Temperature oscillations caused by gravity: liquid to vapor oscillations in vertical inlet tube of Heat Switch condenser (part TC1-TC2)

It is also interesting to point out that the situation of the condensation front depends on the power and the shroud temperature. High power and/or shroud temperature push this front towards the condenser end. We can observe that the condensation front is around TC2 position with only 10 W when the shroud temperature is high (-10 ºC). For attain the same position when the shroud is colder (< -20 ºC), it is needed more power (15 – 20W).

These oscillations may be avoided in a final flight model with a more appropriate condenser design and relative positioning of the liquid and vapor lines.

Model-II Evaporator Experimental Investigation

There are two cases of LHP orientations in gravity field have to be considered in general which are governed by relative positioning of:

1. LHP Evaporator and Condenser,

2. LHP Evaporator and Compensation chamber.

Very often when the “LHP operation in any orientation” statement is clamed it is related to the case No. 1. It means, that condenser can be installed anywhere around evaporator and primary capillary pump will be capable to overcome gravity constraints if condenser will be underneath. In some LHP designs SW is not presented at all and it is suggested that the compensation chamber will be always above the evaporator or at least at the same level.

During the Heat Switch TRP program realization special attention was given to orientation case No. 2. Heat Switch Model-I has demonstrated successful start-up and operation for all cases then angle between evaporator axis and horizon line (compensation chamber below capillary pump) does not exceed 30 °C. Attempt to start LHP at 60° tilt was not successful. Taking into account that gravity on Mars is less than on Earth (0.38 g) the maximum angle will be higher in Martian conditions. However internal research activity was conducted by IberEspacio to overcome this operational limitation of Heat Switch.

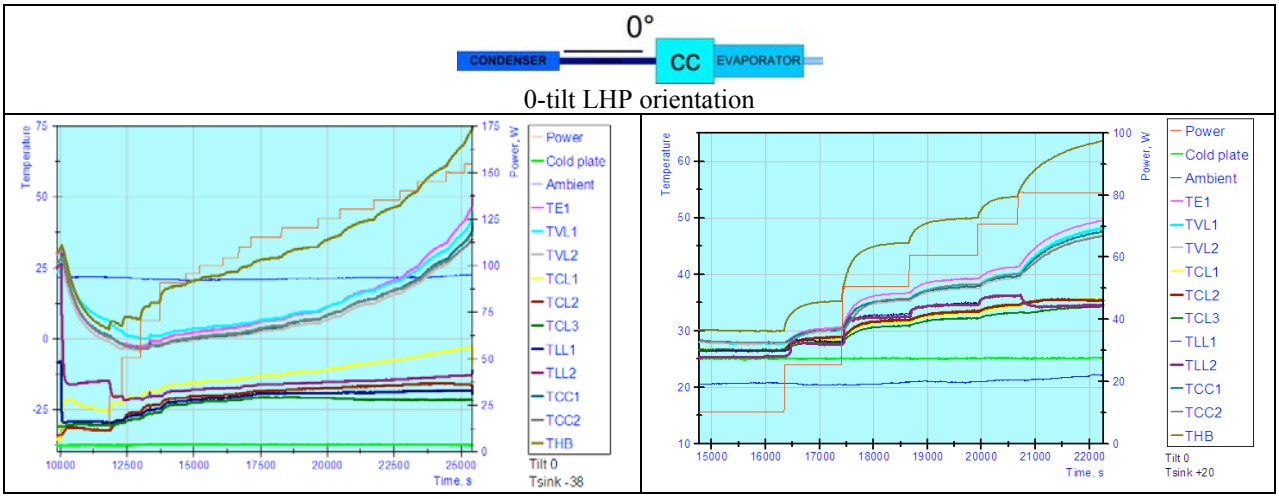
After certain optimization of SW design the Model-II evaporator was manufactured and IberEspacio have prepared and conducted number of test to demonstrate full compliance to the “operation in any orientation’ requirement. This requirement is not critical as soon as main thermal characteristics of Heat Switch exceed the specification but it gives more flexibility for Mars rover system level design, Heat Switch integration and Earth testing campaign.

The SW is one of the critical elements of LHP and often it is responsible on LHP functionality at start-up and stressful transient regimes. The main functions of the secondary wick can be bulleted as follows [7]:

- SW wick ensures that primary wick is wetted before start-up;
- SW must maintain mass balance on primary wick (operation);
- SW must replace vapor condensed in reservoir with liquid supplied to primary wick (vapor could be generated in evaporator core or returned from condenser);
- SW must compensate for differences in flow rate in vapor line and liquid line during transients

Test program of Model-II evaporator was based on analysis, which was made in [7] and comprises following main steps:

- LHP characterization: heat transfer performance at 0° tilt;
- Worst orientation scenario: heat transfer performance at 90°-tilt (evaporator is above compensation chamber and compensation chamber is above condenser);
- Worst operational scenario: sink temperature/power cycling at 0°-tilt;
- Worst orientation and operational scenario: sink temperature/power cycling at tilt angles 30-60-90°;



a) Sink temperature -38°C b) Sink temperature 20°C

Fig. 6. Heat Transfer Performance of Model II LHP in horizontal layout for different sink temperatures

Heat Transfer Performances of Model II LHP at 0°-tilt for two sink temperatures –38 and 20 °C are shown on Figure 6. The thermocouple positioning in all figures of this sections corresponds to Figure 4. LHP demonstrates typical behaviour and maximum heat transfer capacity is around 140 W. The dry-out was reached in the case of the cold sink (Figure 6 a). At sink temperature 20 °C (Figure 6 b) the maximum power is 80 W but the test was limited by maximal allowable evaporator temperature (~50°C). The power values are significantly higher then the Heat Switch requirement 40 W.

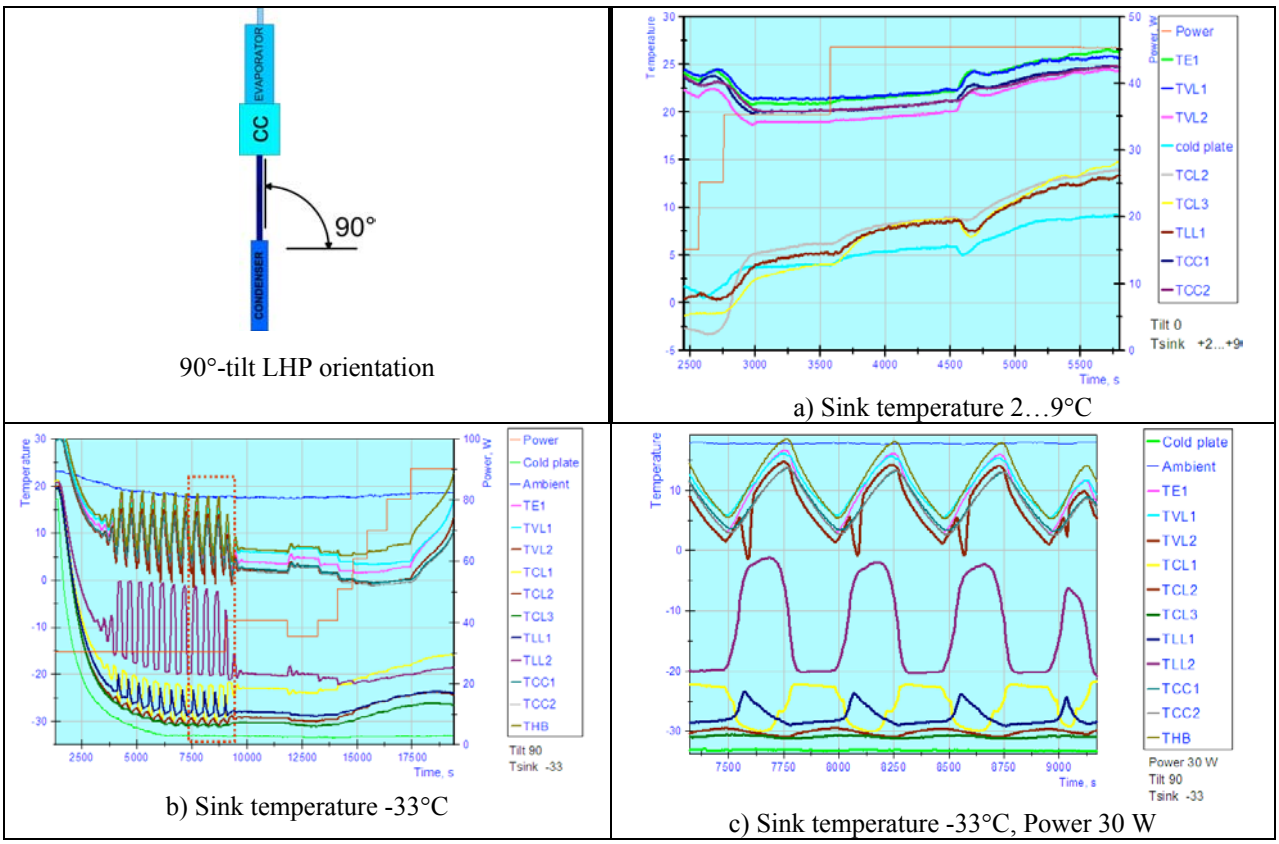


Fig. 7. Heat Transfer Performance of Model II LHP in vertical layout for different sink temperatures

Results of LHP testing for the worst orientation scenario are presented on Figure 7. There are no anomalies were observed at the positive sink temperatures 2-9 °C (Figure 7 a). In this case large part of the condenser was open and liquid level in compensation chamber was high.

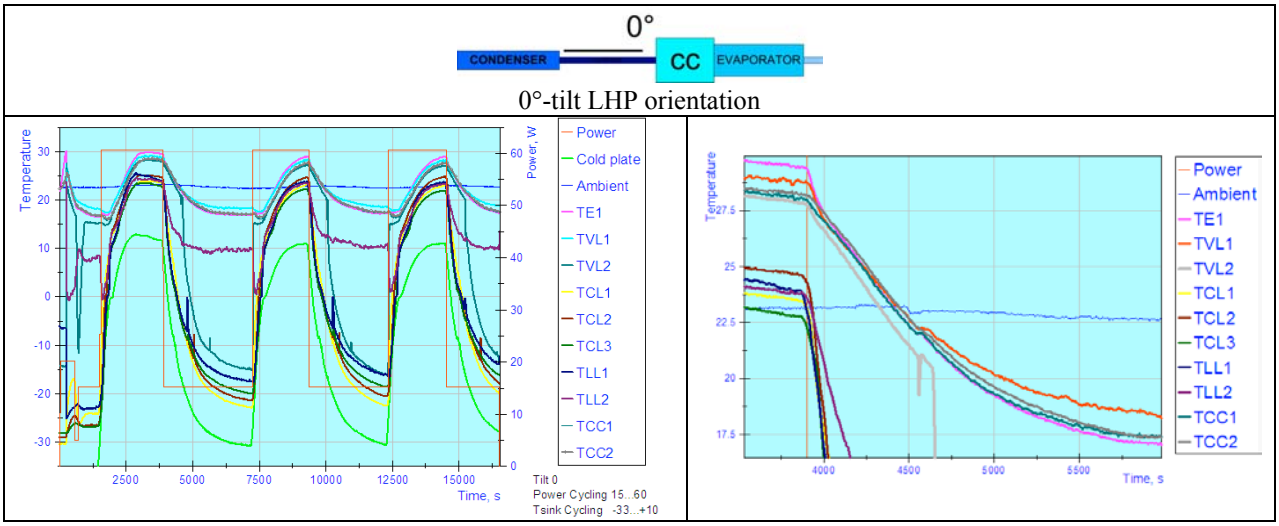
For low sink temperature (Figure 7, b, c) the temperature oscillations were detected at 30 W. With further power increase the LHP behavior stabilized. It means that the oscillation regime is a characteristic feature of the low power and low temperature LHP operation. Maximum reached power before dry-out took place is 80 W. It is on 40 % less than power which was reached in horizontal orientation.

It is interesting to analyze the oscillating regime in detail (see Figure 7 c). When evaporator temperature (TE1) grows up the inlet of the condenser (TCL2) getting colder but liquid line outlet (TLL2) is getting hotter. In the beginning of this mode the temperature of vapor line outlet (TVL2) goes down but in short time it returns back to rising evaporator (TE1), heater block (THB) vapor line inlet (TVL1) and compensation chamber (TCC1 and TCC2) temperatures. The opposite behavior of the described TVL2 point is demonstrated by the liquid line outlet (TLL1).

The possible explanation is following: in temperature growing phase there is no circulation in the loop. The liquid column in liquid line disconnects from primary wick and drops down into condenser. Vapor is presented and generated on both sides of primary wick. Thus, vapor slugs are in vapor and liquid lines. Central core of evaporator is operating as an anti-gravitational heat pipe: propylene evaporates from inner part of the primary wick and condenses in the compensation chamber. The primary wick does not dry out because SW returns liquid back and wets primary wick. In parallel the evaporation from outer surface also takes place. Corresponding saturation pressures grow on both sides of primary wick and in certain moment

pressure in vapor line is capable to push liquid out of condenser (action against gravity) and restore circulation in the LHP. All temperatures go down (only condenser inlet temperatures TCL1 and TCL2 are increasing, which is logical as soon as LHP operates). However because of low sink temperature and low mass flow rate the bubbles, which are generating in central core of capillary pump due to the heat leak, are not evacuated into compensation chamber but migrate into the upper part of wick central core (buoyancy effect). These bubbles can agglomerate and finally to block liquid access from liquid line to primary wick. As soon as liquid path is broken the capillary forces are not capable to support the liquid column and the column drops down. The situation repeats.

To show that SW is really prevents the dry-out of primary wick the simple estimation can be performed. The time of evaporator temperature rise is around 300 s. It means, that total amount of heat generated during off-circulation phase is 30W*300 s = 9000 J. Mass of evaporator with compensation chamber (including SS, Ni elements and propylene) is around 130 g. Mass of aluminum alloy evaporator saddle with heater is 140 g. Taking into account that temperature increase during this phase is ~12 °C the amount of energy which goes for evaporator block heating can be calculated and it is 2040 J. Therefore 9000-2040=6960 J are used for propylene evaporation (with following condensation in different parts of LHP). Average latent heat of propylene for given case is 362 J/g. The amount of evaporated liquid is 6960 / 362 =19 g. It is in ~2 times higher than total charge of LHP and in 6 times more than amount of liquid, which can be stored in primary capillary wick. If SW does not provide effective wetting of primary wick by liquid return from compensation chamber the dry-out will happen quickly and circulation in LHP will never be re-established. 19g of propylene are not couple of drops even in American System of Units.



a) Three compete cycles; b) Temperature behaviour at step down mode;

Fig. 8. Heat Transfer Performance of Model II LHP at Power/Sink Temperature Cycling (30°-tilt)

To verify stability of LHP operation in transient regimes during sharp changes of sink temperature and power series of tests were conducted. Such regimes are very important for SW characterization. “Transients which are particularly problematic are rapid decreases in power and/or sink temperature that can temporarily stagnate flow in the liquid line, or rapid increases in power and/or sink temperature that can force vapor through the condenser into the reservoir”[7]. That is why rapid simultaneous up-down cycling of power (from 15 to 60 W) and sink temperature (from -33 to 10 °C) was proposed as worst operational scenario as a most stressful for SW. It is important, that power step up/down and sink temperature increase/decrease are applied at same time. As it is shown on Figure 8a LHP demonstrates very robust operation during cycling at 0°-tilt. From close view of one of the step down mode (Figure 8b) it is clear that compensation chamber is becoming hottest element of LHP during approx. 8 min (in time interval between 4000 and 4500 s).

Three cycles were conducted with evaporator elevation above compensation chamber at 30°-tilt (Figure 9). LHP operates even though low power, low temperature oscillations were observed. The explanation of

this phenomenon has been made above. The oscillations are stable and repeatable and they disappeared as soon as power was increased although the sink temperature is still -33°-tilt (see the temperature stabilization in the right part of Figure 8).

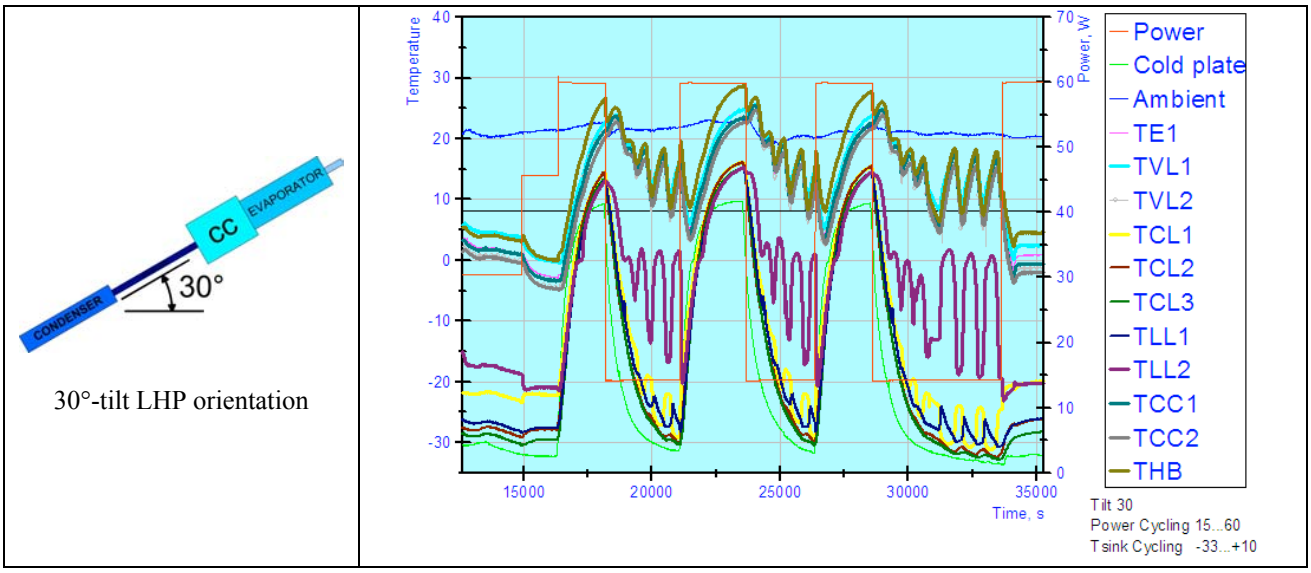


Fig. 9. Heat Transfer Performance of Model II LHP at Power/Sink Temperature Cycling (30°-tilt)

The experiment was continued and orientation of LHP was changed from 30°-tilt to 60°-tilt and finally to 90°-tilt (Figure 10). The amplitude of oscillations was increased, but LHP was still operative and no evidence of dry-out was observed. The evaporator with optimized design of SW is capable to provide the operation of Heat Switch at any orientation and position of compensation chamber and condenser relatively evaporator in Earth/Mars gravity fields.

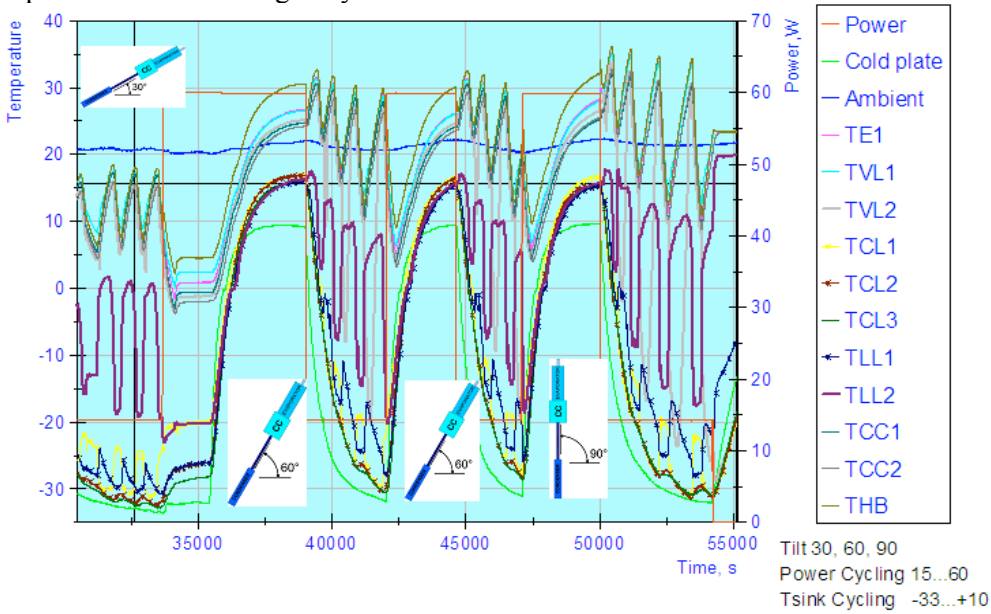


Fig. 10. Heat Transfer Performance of Model II LHP at Power/Sink Temperature Cycling (30°-tilt)

NUMERICAL MODELING

LHP Library of EcosimPro

To design the LHP and verify its performances, a model has been developed using the EcosimPro solver [8]. The mathematical model has modular architecture. It means that the complete LHP model has been created connecting the standard LHP components: evaporator, compensation chamber, transport

lines, condenser, pressure regulator valve, etc. Modular character of EcosimPro code allows quickly introduce new elements or upgrade existing ones and analyze complex two-phase thermal systems including multi-evaporator and multi-condenser schemes [9, 10]. The following main assumptions have been made for the modelling:

- The 1D fluid flow model is a homogeneous equilibrium model. It is considered that the two phases are in equilibrium assuming equal phase velocities, temperatures and pressures.
- The calculated thermodynamic properties correspond to the two-phase mixture and they are obtained by interpolation using the tables built from NIST routines.
- The LHP components can exchange heat with the environment by convection and radiation, and they can exchange heat with other external components (such as saddles) by conduction.
- The fluid model is based on the one-dimensional fundamental conservation equations (mass, momentum and energy) applied to control volumes. The fluid part of each LHP component is sub-divided into individual control volumes.
- The compressibility and transient effects are taken into account. The viscous effects are taken into account through the pressure drop calculations, which include empirical correlations for the pressure drop in a porous media.
- The gravity effects due to the different orientations of the LHP have been taken into account in the formulation.

Full explanation of the general mathematical model and the main hypothesis made for the capillary pump and the LHP lines can be found in references [9, 10]. More detailed description of the valve modeling is provided below.

Pressure Regulating Valve Modeling

The mass and energy conservation equations are applied for the fluid in the valve in order to obtain the density and the internal energy at any time. To fix the valve set point, the model requires an input temperature value (T_{close}). The saturation pressure corresponding to this temperature (P_{close}) is obtained by interpolation using the tables from NIST and represents the point where the valve starts to open the bypass line. Taking into account the properties of the valve bellows, a new pressure (P_{open}) is calculated to define the point where the valve is completely open and, consequently, the radiator branch is completely closed.

The valve position is calculated depending on the relation between the fluid pressure and these values of P_{open} and P_{close} as follows:

$$pos = 0 \qquad P_{fluid} \geq P_{close}$$

$$0 < pos < 1 \qquad P_{close} > P_{fluid} > P_{open}$$

$$pos = 1 \qquad P_{fluid} \leq P_{open}$$

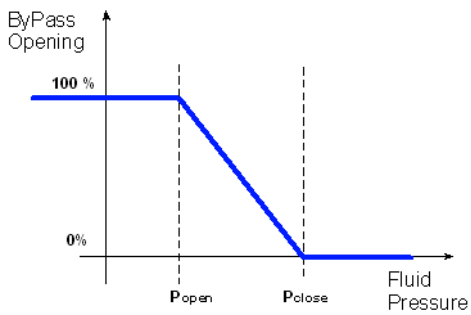


Fig. 11. Heat Transfer Performance of Model II LHP at Power/Sink Temperature Cycling (30°-tilt)

Therefore, in regulation conditions, it is assumed that the valve reaches a steady state in an intermediate position between 0 and 1 (Figure 11.).

Finally, the momentum equation is solved for each of the branches (direct and bypass) taking into account the value of the valve position.

LHP Model for the Heat Switch Design

To design the Heat Switch for Mars Rover application, an LHP model has been developed using the previous mathematical formulation and assuming the following conditions:

- The LHP components do not exchange heat with the ambient.
- The electronic unit (heat source) is a single thermal diffusion node.
- An equivalent sink temperature is considered for the thermal environment of the radiator.
- The radiator is modeled as a single thermal diffusion node.
- The radiative coupling between the radiator and the sink has been calculated taken into account a radiator efficiency of 0.89 and the following emissivities: 0.85 for the radiator and 0.49 for the chamber test.
- The valve set point has been set in accordance with the Heat Switch design. That is, a T_{close} value of 17.5 ºC has been introduced in the model as input data.

The LHP model schematic is presented in the Figure 12.

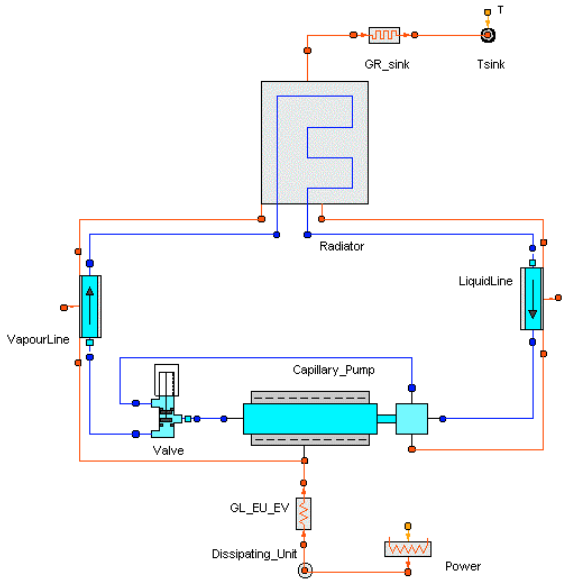


Fig. 12. LHP Model Schematic

COMPARISON OF HEAT SWITCH TEST RESULTS WITH ECOSIMPRO PREDICTIONS

Although an extensive test campaign has been carried out for this LHP, only the most representative cases are presented in this paper. To show the correlation between the thermal model and the experimental results, three steady states and one transient in horizontal conditions are discussed below.

For each of the cases, the power applied to the evaporator and the sink temperature have been introduced as boundary conditions in accordance with the experimental data.

Comparison of Steady State Regimes of Heat Switch Operation

Three different tests have been selected to verify the agreement between the EcosimPro model predictions and the test results. The set of boundary conditions of power and sink temperatures are different for each case. The calculated and experimental temperatures in steady conditions are presented in the Table 1.

Since a high power has been applied in the Case 1, the valve does not regulate the temperatures. That means, that bypass path is completely closed and all the fluid flows through the radiator and the liquid line up to reach the compensation chamber.

For the model correlation, a maximum temperature difference of ± 5ºC between the calculated and the experimental temperatures was required. As it shown, all the temperature differences are lower than the specified range for Cases 1 and 3. For these two cases the largest temperature differences are for the components at the hot side. Additionally, the experimental relative temperature differences between the heater block and the evaporator saddle and between the evaporator saddle and the evaporator are different from the calculated ones. In fact, the experimental temperature difference between the heater block and the evaporator saddle is lower than the calculated one and the difference between the

evaporator saddle and the evaporator is higher. This is because of the values of the thermal contact couplings introduced in the model. It would be possible to fit better the results by adjusting the values of these parameters. From one side, the thermal conductance between the heater block and the evaporator saddle can be increased since the real thermal contact conductance of the interfiller is better than the model value. On the other side, the conductance between the evaporator saddle and the evaporator case shall be decreased if the conduction across the saddle is taken into account. Since the updating of the contact thermal conductances will not affect importantly the results and the current model fits quite well with the experimental results (± 5 ºC), the conductance values was not updated. These values are intrinsic for the model, and they are not affected by the power or the sink temperature. Therefore, this comment is also applicable for all the following comparisons.

Table 1. Test and Model Temperatures Comparison for 3 representative cases

Component	**Temperature, (ºC)**								
	Case 1: Power = 30 W, Sink Temperature = -30 ºC			**Case 2: Power = 10 W, Sink Temperature = -37 ºC**			**Case 3: Power = 30 W, Sink Temperature = -6.5 ºC**		
	Test	**Model**	**ΔT**	**Test**	**Model**	**ΔT**	**Test**	**Model**	**ΔT**
Heater Block	22.4	25.6	3.2	17.8	18.1	0.3	35.7	39.5	3.9
Evap Saddle	20.8	22.9	2.1	17.2	17.2	0.0	34.1	36.8	2.8
Evaporator	19.8	22.8	3.0	16.9	17.2	0.3	33.1	36.8	3.7
VL Inlet	19.0	22.2	3.2	16.5	17.0	0.5	32.3	36.2	3.9
VL Medium	19.3	22.2	3.0	16.8	17.0	0.2	32.5	36.2	3.7
VL Outlet	18.7	22.2	3.5	16.0	16.8	0.9	32.1	36.2	4.1
Compensation Chamber	18.5	22.0	3.5	16.1	16.9	0.7	31.6	35.9	4.3
Cond 1	18.7	20.2	1.5	9.0	-9.8	-18.8	31.9	34.4	2.5
Cond 2	17.7	19.6	1.9	-19.8	-16.6	3.1	30.9	34.0	3.1
Cond 3	17.6	19.1	1.4	-21.4	-16.7	4.7	30.9	33.8	2.9
Cond 4	18.4	18.2	-0.2	-17.7	-16.7	1.0	31.6	33.4	1.8
Cond 5	18.1	17.6	-0.5	-18.3	-16.7	1.6	31.3	32.9	1.6
Cond 6	17.7	17.6	-0.2	-17.8	-16.7	1.1	31.0	32.4	1.5
LL Inlet	16.2	17.6	1.4	-16.5	-16.7	-0.2	31.2	32.7	1.5
LL Outlet	15.2	17.6	2.4	-10.9	-16.3	-5.4	29.7	32.8	3.0
Radiator	16.6	17.5	0.9	-16.8	-16.7	0.1	30.0	32.3	2.3
Sink	-30.1	-30.1	0.0	-37.0	-37.0	0.0	-6.5	-6.5	0.0

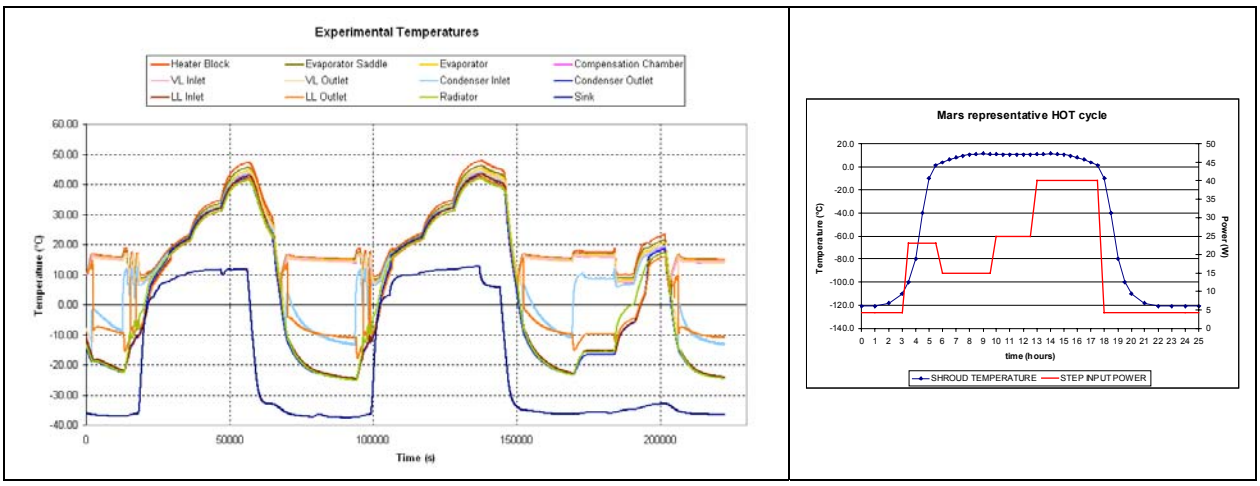
For Case 2, the power applied is relatively low as well as the sink temperature. Therefore, the valve needs to open partially the bypass line to keep the temperatures above its set point. As it is shown, the vapor temperatures are about 17 ºC. That means that the valve is in an intermediate position regulating in its upper range to reach this steady state.

According to the table, most of the calculated temperatures are within ± 5 ºC. However, there are two temperatures in Case 2 which differs more from the experimental ones. The calculated temperature at the condenser inlet is much lower than the experimental one. This is due to the fact that the thermocouple TC1 was located at the very inlet of the condenser. On the contrary, in the model this value corresponds to the temperature at the first node which is located farther from the condenser inlet. That means that in the model, the calculated value corresponds to a point which has much subcooling than experimental. The difference is so large because of the low power applied. Then the mass flow through the condenser is very small and the different locations of the real thermocouple and the calculated point affect notably to the results. The second temperature which differs more than 5 ºC from the experimental, is the one

calculated at the liquid line outlet. In experiments the thermocouple was located before the joint between the liquid line and the bypass line and it can be influenced by the vapor temperature in bypass line. However, in the model, it was considered that the vapor from the valve flows directly to the compensation chamber (not to the liquid line). For this reason, the calculated value is much lower than the experimental.

Comparison of Transient Regimes of Heat Switch Operation

In one of the tests the sink temperature and the applied power are varying continuously in order to represent Mars scenario (Figure 13). The temperatures obtained experimentally are presented in the Figure 13 a.



a) Experimental Temperatures b) Mars Representative Hot Cycle

Fig. 13. Mars Scenario Case

The sink temperature and the power have been introduced as boundary conditions in accordance to the testing values in the model. The calculated temperatures are presented in Figure 14.

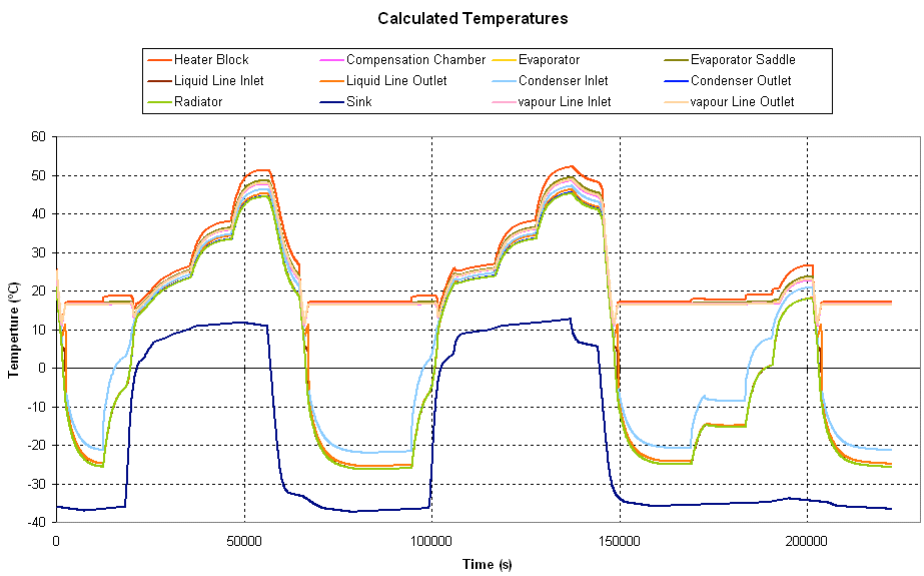
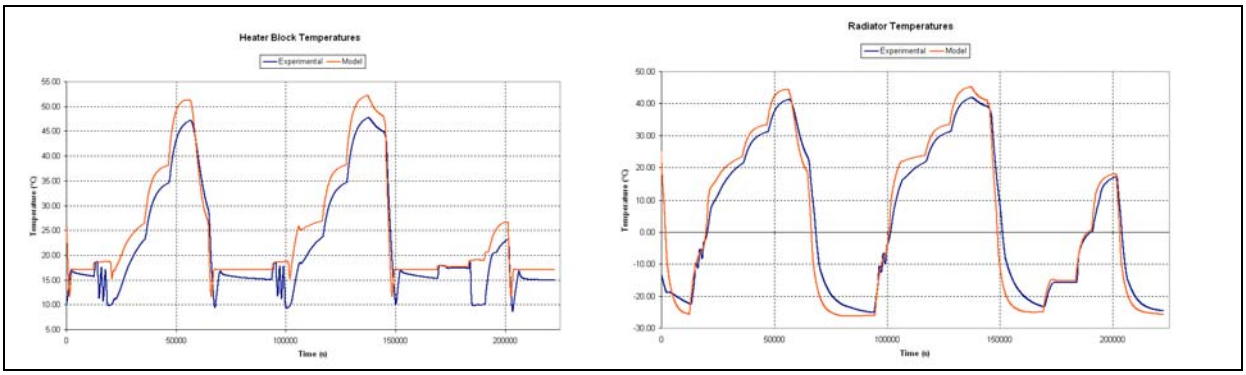


Fig. 14. Mars Scenario Case – Calculated Temperatures

When the sink temperature is low, the valve is regulating the temperature and the LHP temperatures are decoupled. In fact, a temperature difference higher than 30 ºC can be found between the capillary pump temperatures and the condenser ones. On the contrary, when the sink temperature is higher, the valve is not acting, that is, the bypass path is completely closed and the relative differences between the LHP temperatures are small.

As it is shown, the calculated temperatures are very close to the experimental ones even in transient conditions. For direct comparisons, the heater block and radiator temperatures are presented as examples in Figure 15.



a) Heater Block Temperatures b) Radiator Temperatures

Fig. 15. Mars Scenario Case – Temperatures Comparison of Heat Switch Elements

According to these figures, practically all the temperatures differences between the calculated and experimental values are within the specified range for both components. Exceptions are the regions were gravity caused temperature oscillations were observed. This effect is connected with position of liquid front in condenser. Then the valve is not capable to find an equilibrium positions, but oscillated between the two limiting points (transition close/open valve temperature range was 8.5-17.5ºC) as it was explained before. Some improvements of model are necessary to predict this transient behaviour. Homogenous condensation model is not fully realistic and flow mapping is not precise, especially with propylene. Also the condenser is a critical item and accuracy in monitoring of temperature sensors location in test with regard to model is very important.

CONCLUSIONS

Thermal Control System - Heat Switch for European Mars Rover was developed on the basis of Loop Heat Pipe technology. Passive bypass regulating valve was used as a heat flow control element.

Two models of evaporators were manufactured and tested.

Evaporator Model I was integrated in Heat Switch prototype with radiator and extensively tested in ambient and vacuum conditions. The device has demonstrated outstanding thermal performance. Corresponding thermal conductances in ON and OFF Heat Switch modes are 5W/K (0.75W/K ON required) and 0.009 W/K (0.015W/K OFF required). Observed oscillations of temperatures at certain combinations of power and sink conditions are connected with present design of condenser (~0.5 vertical tube above evaporator at vapor input) and can be avoided by different routing of LHP vapor/liquid/condenser lines. It has to be mentioned that the oscillations are very stable, and inside the operational range. Thus, in general case this effect will not have impact on functionality of controlled objects.

Evaporator Model II with optimized secondary wick design was integrated with representative technological condenser and has been tested in worst orientation (evaporator on the top of compensation chamber and compensation chamber on the top of condenser) and worst operational (rapid simultaneous decrease or increase of sink temperature and input power) conditions. LHP have operated in all the tests. Oscillating regime of heat transfer was realized at most stressful conditions (low power and low sink temperature). High performance of secondary wick provides LHP functionality at any orientation in gravity field.

EcosimPro model of Heat Switch was developed. The experimental results obtained from the tests have been analyzed and compared to the temperatures calculated by the thermal model. The results agree within the ± 5 ºC range required for steady conditions. Moreover, very good agreement was achieved for the transient cases also.

Acknowledgments

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ACRONYMS

CC – Compensation Chamber
Cond - Condenser
CPL – Capillary Pumped Loop
ESA – European Space Agency
Evap – Evaporator
HB – Heater Block
HP – heat pipe
LL – Liquid Line
SS – Stainless Steel
SW – Secondary Wick
T – Temperature, Thermocouple
TRP – Technological Research Program
TCS – Thermal Control System
VL – Vapor Line
VTCD – Variable Thermal-Conductance Device

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