Thermal behavior of a cryogenic loop heat pipe for space application

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A R T I C L E   I N F O

Article history:
Received 28 November 2010
Received in revised form 9 February 2011
Accepted 27 April 2011
Available online 5 May 2011

Keywords:
Nitrogen
Heat transfer
Level detection
Space cryogenics

A B S T R A C T

This paper discusses a prototype of cryogenic loop heat pipe (CLHP) working around 80 K with nitrogen as the coolant, developed at CEA-SBT in collaboration with the CAS/TIPC and tested in laboratory conditions. In addition to the main loop it features a pressure reduction reservoir and a secondary circuit which allow cooling down the loop from the room temperature conditions to the nitrogen liquid temperature and transferring the evaporator heat leaks and radiation heat loads towards the condenser. The general design, the instrumentation and the experimental results of the thermal response of the CLHP are presented, analyzed and discussed both in the transient phase of cooling from room temperature (i) and in stationary conditions (ii). During phase (i), even in a severe radiation environment, the secondary circuit helped to condense the fluid and was very efficient to chill the primary evaporator. During phase (ii), we studied the effects of transferred power, filling pressure and radiation heat load for two basic configurations of cold reservoir of the secondary circuit. A maximum cold power of 19 W with a corresponding limited temperature difference of 5 K was achieved across a 0.5 m distance. We evidenced the importance of the filling pressure to optimize the thermal response. A small heating power (0.1 W) applied on the shunted cold reservoir allows to maintain a constant subcooling (1 K). The CLHP behaves as a capillary pumped loop (CPL) in such a configuration, with the cold reservoir being the compensation chamber of the thermal link. The radiation heat loads may affect significantly the thermal response of the system due to boiling process of liquid and large mass transfer towards the pressure reduction reservoir.

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1. Introduction

Operation of instruments and electronics onboard satellites and spacecrafts requires efficient cooling systems. When high performance is needed, either for the electronics of control systems or for measuring units or detectors, cryogenic temperature may also become necessary. Thus, original cryocoolers have been developed at Commissariat à l’Énergie Atomique – Service des Basses Températures (CEA-SBT) for many years, within the scope of Technical Research Programs of European Space Agency. These include the development of systems such as pulse tube cold fingers, sorption coolers and continuous adiabatic demagnetization refrigerators. However as heat removal across large distances (meter or more) may cause significant temperature gradients, the development of efficient thermal links at cryogenic temperatures which must work in a warm radiation environment is also considered as a new technical challenge.

Loop heat pipes are extensively used around 300 K as passive thermal links between the cold source and control systems. Their ability to work without sensitivity to gravity relies on the circulation of a coolant that is driven by capillary forces from the cold part to the hot part where enthalpy is removed by evaporating some liquid. This technology showed many advantages [1] compared with solid conduction bars (too heavy), with single phase circulators (too large pipe diameter) and with mechanically pumped two-phase circulators (less reliable). This is the reason why this concept is currently used in space applications as a passive thermal control device at ambient temperature, mainly with ammonia as working fluid. At cryogenic temperatures this concept has already been tested using different designs and cryogenic working fluids such as nitrogen, oxygen, ethane, hydrogen, neon, propane [1–8] and more recently by CEA-SBT [9,10] with a prototype of cryogenic loop heat pipe (CLHP), based on the concept invented by Swales [5]. This prototype uses nitrogen as the working fluid and has been designed, manufactured and tested at around 80 K, in collaboration with the Technical Institute of Physics and Chemistry of the Chinese Academy of Science (CAS/TIPC). This paper presents detailed selected experimental results, physical analysis and a discussion about its thermal behavior. For two basic configurations, we firstly discuss the transient cool down of the system from room temperature in a warm radiation environment. Then secondly the thermal performance in stationary cold operation is detailed for cold and warm environments.
2. General considerations on capillary pumped two-phase cryogenic thermal links

In comparison with the well known Loop Heat Pipes (LHPs) or Capillary Pumped Loops (CPLs) working with a two phase fluid at room temperature, such as ammonia, the operation of such a thermal link at cryogenic temperatures requires considering additional aspects for an operation in a realistic micro gravity warm environment.

The filling pressure at room temperature may become dramatically large (typically in the range of 20–80 MPa) if no pressure reduction reservoir (PRR) is used (Fig. 1). The high level of pressure is the consequence of the temperature change of the constant volume system from the cold two phase liquid–vapour state to the warm super critical state. To solve the problem of over-pressure to be supported by the system, the volume of the PRR must be much larger than the cold part section and sufficiently large so as to reduce the filling pressure at room temperature below the critical pressure. This last precaution allows a faster cool down of the system and favors the condensation of vapour in the cold parts of the thermal link.

The warm environment of the future space application may cause significant radiation heat load on the system. This aspect must be absolutely addressed.

Finally the cool down from initial room temperature without gravity assistance must be considered. Prior to start up, the evaporator must be chilled and its wick full of liquid to start the fluid circulation in the loop. This is a difficulty to be considered.

3. Presentation of the tested CLHP and experimental set up

An instrumented prototype of CLHP, has been built and tested in horizontal position by CEA-SBT. The tested CLHP (Figs. 2 and 3) consists of a primary loop for heat transfer, a secondary circuit and a PRR. The primary loop features a primary evaporator and a primary condenser linked by the liquid and vapour lines. The evaporation of liquid in the capillary structure of the primary evaporator makes the fluid circulation owing capillary forces at the liquid–vapour interfaces. The primary evaporator is a three port type. It features a cylindrical body with machined grooves in the inside for the vapour flow. The fins of the body are in mechanical contact with a tubular wick. A bayonet located on its axis allows the liquid to penetrate the core. A Kapton backed foil heater is bonded to the body to simulate the heat load power Q\textsubscript{L}. The primary evaporator has neither a secondary wick nor a collocated compensation chamber which is replaced by a cold reservoir that is thermally linked to a cold plate and hydraulically connected to the primary evaporator by the secondary liquid line. The secondary circuit has two important functions. Firstly, the cool down of the primary evaporator is performed by means of a heating power applied on the secondary evaporator just after being chilled. Liquid coming from the condenser feeds the primary evaporator and the formed vapour returns towards the secondary condenser through the secondary liquid line. Secondly, during cold operation the vapour flow induced by the heat leak of the primary evaporator and the radiation heat load is transported towards the secondary condenser in forced flow owing to capillary pressure of the secondary evaporator which is permanently heated. Table 1 lists the dimensions and characteristics of the components used to build the presented prototype, whereas some aspects of the sizing are given in Ref. [11]. The cold reservoir can be heated (Q\textsubscript{CR}) and be a removable thermal shunt (thermal resistance ~10 K/W) can be mounted between the cold reservoir and the cold plate. In the first configuration, the cold reservoir is directly clamped to the cold plate and their temperatures are close. In the second configuration, with thermal shunt, the main idea is to create a two-phase cold reservoir in order to control its temperature using a small heating power applied on the cold reservoir. The two condensers are serpentine tubes brazed to the cold plate. The cold parts of the CLHP are mounted inside a 354 mm OD × 750 mm Long vacuum vessel. The cold plate is bolted to the first stage of a Gifford McMahon refrigerator which produces the cold power (Fig. 5). The cold plate temperature is kept constant at around 80 K owing to a heater and PID regulation. The PRR (1 l or 4 l) is located outside of the vacuum vessel at room temperature and is connected to the cold parts through a 3 mm OD CuNi pipe inside the cryostat then a 6 mm OD SS pipe outside.

A copper thermal shield surrounding the cold parts was installed and either cooled close to 80 K by liquid nitrogen or kept to room temperature using an electrical heater to compensate the cold radiation heat from the cold parts and from the cryo cooler cold finger. So, the two extreme situations, warm and cold thermal shield, can be investigated.

Last, a removable multi layer insulation protection (MLI) was installed on the liquid line, on the secondary liquid line and on the cold reservoir to address this issue. This protection features ten layers and allows to reduce twice as low the 300 K radiation heat loads.

The instrumentation is presented in Fig. 4. Two pressure sensors allow to measure the absolute pressure just at the outlet of the primary evaporator, within an accuracy of ±0.002 MPa in the range 0–0.3 MPa abs (P\textsubscript{2}) and ±0.02 MPa in the range 0.3–3 MPa (P\textsubscript{1}). Pt100 miniature thermometers (PTFD102A) are glued on the external walls of the CLHP. To reach the accuracy of ±0.2 K, offset corrections are made on the temperature measurements. These offsets are regularly determined during calibration operation with the CLHP in steady state isothermal behavior at T\textsubscript{sat}(P\textsubscript{2}) with a small two-phase flow circulation in the loop and negligible radiation heat losses (thermal shield at 80 K). In addition, a particular effort is devoted to the internal instrumentation of the primary evaporator. Its bayonet is equipped with a thermometer (T\textsubscript{BAY}) and a foil capacitive sensor [12] respectively located at its end and on its external cylindrical surface. These sensors give valuable information about the temperature and the vapour fraction on the sensor surface. The measured capacitance (capu) is 5 pF and 7.3 pF for the sensor totally immersed respectively in vapour and liquid. In particular, we measured in many situations an intermediate value of 6.2 pF corresponding to the situation where the annular core, which is horizontal, is half full of liquid, i.e. a liquid level exists in it. The heating powers Q\textsubscript{L}, Q\textsubscript{CR} and Q\textsubscript{HEAT} are measured within an accuracy of ±3% of the measurement. The maximum allowed

![Fig. 1. Room temperature filling pressure of a cryogenic two phase thermal link without any PRR, assuming 50% vapour and 50% liquid volumes and perfect gas at room temperature, for common cryogenic fluids.](image-url)
working pressure in the cold parts is 0.7 MPa, thus a safety valve between the cold parts and the PRR (Fig. 5) is mounted to prevent over pressure during the charging process. This valve is always kept open during the tests.

4. Filling pressure consideration

The thermal link being a closed circuit it must be charged at room temperature at the right filling pressure, i.e. at the right mass inventory in the cold parts. Using the volume of all components and a reference temperature (81.5 K) for calculating the liquid and vapour densities, a value of 2.2 MPa was calculated for the filling pressure with a 1 l PRR to obtain a “full” CLHP, i.e. all cold parts are in pure liquid except the two vapour lines and the vapour channels of evaporators. If the CLHP is over-charged, i.e. its filling pressure greater than 2.2 MPa, the excess of mass cannot remain in the vapour parts. It is pushed towards the PPR and increases its pressure increases significantly. This phenomenon is illustrated on Fig. 6 which presents the cold plate subcooling as function of filling pressure for an ideal configuration with negligible radiation heat load and no electrical heating power except a small amount (1 W) applied on the secondary evaporator to get a reduced circulation in the loop.

5. Transient operation during cool down

The 4 l PRR was used in the tests to limit the pressure to about 0.7 MPa at room temperature. This filling pressure corresponds to a
“full” CLHP in cold operation. Results are presented with a severe radiation heat load environment (warm thermal shield), which is deliberately an extreme situation compared to the actual case. At the beginning the CLHP is isothermal at room temperature. The tests start when the cryocooler is switched on. The cold plate temperature set point is fixed to 81.5 K.

5.1. Cool down transient without thermal shunt

Detailed experimental results of a typical cool down test of the CLHP without thermal shunt are presented in Fig. 7. For this test the CLHP has no particular MLI protection. The cool down of the primary evaporator is achieved within 4h30 when making use of the secondary evaporator. During the first period of time (0–2 h) the cold plate temperature decreases gradually under the action of the cryocooler. The cold reservoir temperature follows it because of the good thermal contact between both of them. At time 1 h30 the cold plate temperature reaches the saturation temperature. The hot gas coming from the PRR enters the vapour lines and flows towards the primary condenser. Condensation process

<table>
<thead>
<tr>
<th>Components</th>
<th>Characteristics and dimensions in mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary evaporator body</td>
<td>22 OD × 64 Long, Cu</td>
</tr>
<tr>
<td>Ten micron pore-radius wick</td>
<td>17 OD × 50 Long × 2.1 thick, SS 316L 30% measured porosity, 2 × 10⁻¹² m² measured permeability, 3.2 W/mK measured effective thermal conductivity (vacuum environment at 80 K), 900 mm² theoretical surface area in contact with the fins of the body</td>
</tr>
<tr>
<td>Serpentine primary condenser</td>
<td>3.5 OD × 1502 Long, Cu</td>
</tr>
<tr>
<td>Liquid line</td>
<td>3.0 OD × 328 Long, CuNi</td>
</tr>
<tr>
<td>Vapour line</td>
<td>3.0 OD × 567 Long, CuNi</td>
</tr>
<tr>
<td>Serpentine secondary condenser</td>
<td>3.5 OD × 528 Long, Cu</td>
</tr>
<tr>
<td>Secondary liquid line</td>
<td>3.0 OD × 330 Long, CuNi</td>
</tr>
<tr>
<td>Cold reservoir</td>
<td>19 OD × 58 Long, 8.5 cc, Cu</td>
</tr>
<tr>
<td>Secondary vapour line</td>
<td>3.0 OD × 747 Long, CuNi</td>
</tr>
<tr>
<td>PRR (1 l) or (4 l)</td>
<td>88.9 OD × 277 Long or 102 OD × 678 Long, SS 304L</td>
</tr>
<tr>
<td>Cold plate</td>
<td>192 Long × 101 wide × 15 thick, Cu</td>
</tr>
</tbody>
</table>

Fig. 4. Instrumentation of the tested CLHP.

Fig. 5. Test cryostat.

Fig. 6. Measured cold plate sub cooling $T_{Sat(P2)} - T_{CP}$ as a function of the filling pressure for the following conditions: $Q_s = 1$ W, $Q_{CR} = 0$ W, $T_{CP} = 81.5$ K, $T_{TS} = 80$ K.

Table 1. Components, characteristics and dimensions of the tested CLHP.
occurs first in the condensers and then in the cold reservoir. At time 1 h45, the secondary evaporator is chilled. This condensation process induces temporary fluid circulation in the loop. In particular the liquid line and the primary evaporator see some liquid arrival. The primary evaporator temperature drops dramatically down to 240 K and then its temperature is stable. From the time 3 h a heating power is gradually applied on the secondary evaporator up to 3 W. So a forced circulation starts in the secondary loop and liquid feeds the primary evaporator which is gradually chilled. It has to be noted that the severe radiation environment causes boiling of liquid in the liquid line and permanent heat load to be supported by the primary evaporator. This phenomenon reduces significantly the speed of the cool down of the primary evaporator. At time 4 h30, the primary evaporator is completely chilled with its annular core half full of liquid (capacitance ~6.2 pF). The loop is ready to be started.

5.2. Cool down transient with thermal shunt

The cool down process of the CLHP equipped with a thermal shunt is much longer. Indeed the filling of the cold reservoir with liquid is significantly retarded due to the large thermal resistance of the thermal shunt through which significant energy is transferred to the cold plate. Without any MLI protection the secondary evaporator is hardly chilled after 25 h. In contrast, when MLI covers the cold reservoir, the liquid line and the secondary liquid line become cold after 10 h and the cool down time is roughly five times longer in such a configuration (Fig. 8).

As a conclusion, for both configurations, with and without thermal shunt, the secondary circuit is efficient to cool down the system in a severe radiation heat load environment.

6. Thermal behavior in stationary cold condition with reduced radiation heat load

Let us consider first the ideal situation of the CLHP working with reduced radiation heat load (cold thermal shield) at different levels of primary heating power. The CLHP cold plate temperature is kept constant at around 81.5 K. During all the tests a constant electrical heating power is applied on the secondary evaporator ($Q_P = 1$ W).

6.1. CLHP without thermal shunt

The filling pressure has a huge effect on the performance of the CLHP as shown in Fig. 9. The effect is observed on the system...
temperature difference \( T_{\text{EVAP}} - T_{\text{CP}} \) and on the capability of the thermal link, i.e. the power limit corresponding to the CLHP depriming. For the over-charged CLHP (2.56 MPa), some liquid is probably locked by capillary forces in the vapour channels of the primary evaporator before applying the heating power. As soon as the heat is applied, the liquid is pushed out of the primary evaporator and is transferred towards the PRR. This phenomenon explains the sharp 5 K step on the system temperature difference at low primary heat power (1 W). Then the temperature difference increases almost linearly with the primary heating power, because the vapour expansion inside the primary condenser induces a mass transfer towards the PRR and consequently a pressure increase. A significant response is therefore measured on the system temperature difference, which reaches 13 K. However, the over-charged CLHP allows getting the best capability (21 W), because of reduced pressure drops in the primary condenser which works with large cold plate subcooling.

For the under-charged CLHP (1.77 MPa), the system temperature difference is very small. The cold plate subcooling is therefore very small too, so the two phase pressure drops in the primary condenser are large and consequently the capillary limit is reached for very small primary heating power (5 W). The best compromise is obtained with 2.15 MPa with a large achieved capability (19 W) and a limited temperature difference (5 K).

The detailed thermal response of the CLHP at this filling pressure is presented in Fig. 10. The following remarks can be drawn:

- Below 7 W the CLHP is isothermal and close to saturation temperature, which is almost the same around the loop because of a very low capillary pressure which corresponds to a maximum 0.1 K step of saturation temperature. The annular core is half full of liquid (capacitance of 6.2 pF) at very low primary heating power. It is being filled gradually when the power increases. It is completely full with liquid for 7 W. Two phase flow is consequently also present in the secondary liquid line. The line and the core behave as a compensation chamber. The pressure hence the saturation temperature are constant. The cold reservoir, which is in good thermal contact with the cold plate, is probably full of liquid. It does not play any functional role.

- Above 7 W the annular core, the secondary liquid line and the cold reservoir are subcooled, i.e. totally full of liquid. The heat leak of the primary evaporator induces heating of liquid entering the primary evaporator. So higher temperature in the annular core \( T_{\text{BAY}} \) and in the secondary liquid line \( T_{\text{SLL}} \) are measured. The power increase initiates the vapour expansion in the primary condenser. This induces some mass transfer towards the PRR. The pressure, hence the saturation temperature increase gradually with the power. The cold plate subcooling also increases. It tends to limit the vapour expansion in the primary condenser and the pressure drop of the loop. At 17 W the capillary limit is almost reached and the loop is close to its capability (19 W).

- The heat transfer in the evaporators: the temperature of the secondary evaporator remains very close to the saturation. Its wick is therefore completely full with liquid. On the contrary, above 7 W, a temperature difference \( T_{\text{EVAP}} - T_{\text{Sat}(P_2)} \) appears gradually in the primary evaporator which starts to support large heat fluxes. This difference is small and reaches 1.3 K for 17 W transferred power. It can be explained by a small penetration of vapour inside the wick, which can be calculated using the measured effective thermal conductivity and the theoretical surface area of the wick in contact with the fins (Table 1).

- Cold reservoir: because of the good thermal contact with the cold plate, the cold reservoir is always full of liquid. It cannot act as the compensation chamber.

6.2. CLHP with thermal shunt

The presence of a thermal shunt allows getting vapour, i.e. a liquid level, in the cold reservoir owing to a small amount of power \( Q_S \) on it. Fig. 11 presents detailed experimental results of such a configuration with 2.15 MPa filling pressure and 1 W and 0.1 W applied respectively on the secondary evaporator and on the cold reservoir. The results are presented for limited transferred powers \( Q_P < 10 \) W. This limitation is explained by additional pressure drops in the liquid circuit, which suddenly appeared during the test program. On the figure, as expected, the cold reservoir and the secondary evaporator are at saturation temperature. The liquid line and the core of the primary evaporator and the secondary liquid line are subcooled, therefore they are full of liquid only. This is the reason why no vapour is detected by the capacitance sensor which remains at 7.2 pF. The small heating (0.1 W) on the cold reservoir makes the system temperature difference almost constant at around 1 K whatever the applied power \( Q_P \). The thermal link behaves as a CPL with the two phase heated cold reservoir being the master who determines the pressure and therefore the temperature of the primary evaporator. The cold reservoir is therefore the compensation chamber and negligible mass is exchanged with the PRR whose pressure is almost constant. So when the heating power on the cold reservoir is changed the system temperature difference changes too as shown in Fig. 12. This demonstrates that it is possible to adjust the temperature of the primary evaporator...
by means of a small heating power applied on the cold reservoir. On the other hand the filling pressure can be reduced down to 1.7 MPa without affecting the thermal response and the capability of the CLHP, because only the mass inventory of the cold reservoir is reduced with no other change in the loop. This configuration is therefore less sensible to the filling pressure. This is not the case for the other configuration without thermal shunt.

7. Thermal performance in stationary cold condition with severe heat load

7.1. Role of radiation heat load

Warm environment is a more realistic situation for any space application and radiation heat loads exists in any cases, depending upon the kind of protection (for instance active shields or gold plating of surfaces). So an experimental program has been conducted to investigate effects on the thermal behavior of the CLHP. A warm shield deliberately maintained at room temperature was used for the tests. Some tests were conducted without any MLI protection, obviously not realistic, to investigate the effect of large radiation heat loads. In that case the CLHP must support around 5 W. This power is mainly located on the cold plate which offers a very large external surface area. However, the cold parts where liquid is present, i.e. the liquid line, the secondary liquid line and the cold reservoir, only support small amount of this power, about 0.2 W for each, because of their limited external surface area. Like for cold tests without radiation heat loads the CLHP cold plate temperature is kept constant and close to 81 K and during all the tests a constant electrical heating power is applied onto the secondary evaporator \( Q_s = 1 \text{ W or 4 W} \) to maintain a permanent flow in the secondary circuit. Each test series consists of a gradual variation (4 W/h) of the primary heating power.

7.2. CLHP without thermal shunt

The radiation heat load affects significantly the system temperature difference (4 K for 8 W instead of 0.5 K without radiation) as shown in Fig. 13. The CLHP being “full” (filling pressure of 2.15 MPa), boiling process of liquid mainly located in the liquid line and in the secondary liquid line is responsible for this effect. It raises the vapour production of the CLHP. As a consequence a significant mass is transferred towards the PRR and leads to large pressure hence large saturation temperature. On the other hand

![Fig. 11. Detailed thermal response of the CLHP with thermal shunt at different cold reservoir heating power, for the following conditions: FP = 2.15 MPa, Q_s = 1 W, Q_CR = 0.1 W, T_CP = 81.5 K, T_TS = 80 K.](image1)

![Fig. 12. System temperature difference of the CLHP with thermal shunt at different cold reservoir heating power, for the following conditions: FP = 2.15 MPa, Q_s = 1 W, T_CP = 81.5 K, T_TS = 80 K.](image2)

![Fig. 13. General thermal response of the CLHP without thermal shunt, for the following conditions: FP = 2.15 MPa, Q_s = 1 W, Q_CR = 0 W, T_CP = 81.5 K, T_TS = Troom.](image3)
unstable thermal behavior is measured at low power. The heating power applied onto the secondary evaporator has to be significantly put up to 4 W to get the stable thermal response shown in Fig. 14.

7.3. CLHP with thermal shunt

Effect of thermal shield temperature on the system temperature difference is presented in Fig. 15, with different configurations – with and without any MLI protection – and for two filling pressures (2.15 MPa and 2.00 MPa). Without any MLI protection warm shield leads to very large system temperature difference (8 K), in comparison with the result for 80 K thermal shield. In such a situation the cold reservoir is subcooled and is no more predominant. The pressure is determined by the PRR which sees significant mass arriving from the liquid line and secondary liquid line where boiling process occurs. The MLI protection of the theses cold parts and also the cold reservoir limits this phenomenon and reduces significantly the system temperature difference. However the 2.15 MPa filling pressure is too large. It must be reduced down to 2.00 MPa to get the desired system temperature difference (1 K). Fig. 16 shows the detailed thermal response of the CLHP for such a filling pressure. On this figure the desired 1 K system temperature difference is
observed under 5.5 W transferred power. The liquid cold parts except the liquid line are at saturation temperature, in particular the cold reservoir which plays its role of compensation chamber. At larger transferred power ($Q_T > 5.5$ W) it becomes suddenly subcooled and therefore full of liquid. The system temperature difference is consequently no more constant. The pressure is determined by the PRR. It is therefore necessary to reduce again the filling pressure to get a complete CPL behavior of the thermal link.

8. Conclusion

In 2007, CEA-SBT has designed, built and tested a prototype of CLHP using nitrogen as working fluid. The experimental program, conducted in 2007, 2008 and 2009 in collaboration with CAS/TIPC, allowed characterizing in details the thermal behavior of the thermal link for two basic configurations without and with thermal shunt associated with a heated cold reservoir. The internal instrumentation, vapour fraction capacitive sensor and thermometers, located in the core of the primary evaporator, associated to external temperature sensors and pressure sensors allowed understanding the physical phenomena, in particular identifying the phase repartition in the loop. The following conclusions can be drawn:

- With an optimized filling pressure, the CLHP allows transferring a maximum of 19 W with a limited temperature difference (5 K) over a large distance (0.5 m)
- The heat transfer in the primary evaporator is very efficient with limited internal temperature gradient which can be explained by small vapour penetration in the wick. Some situations with vapour present in the core have been observed.
- The secondary circuit is confirmed to be efficient to cool down the primary evaporator even with severe radiation heat loads. The cool down time is larger in the configuration with thermal shunt. In stationary conditions it allows to get a permanent circulation in the loop for thermal stability and for removal of the heat leak.
- The cold reservoir is always full of subcooled liquid when directly clamped to the cold plate without any thermal shunt. It plays no role of compensation chamber in such a configuration. On the contrary, with an optimized filling pressure and with a shunted and heated cold reservoir, it acts as the compensation chamber with a liquid level inside. The thermal link behaves as a CPL. This configuration allows getting a constant and limited system temperature difference (1 K) by means of a limited heating power (0.1 W) on the cold reservoir. It is therefore possible to drive the evaporator temperature using the cold reservoir heater. This could be useful to cool an object producing a variable heat load.

- The PRR is obviously necessary to reduce the pressure when the loop is at room temperature. In case of an over-charged system, significant mass and energy can be exchanged with the cold parts of the loop, with significant effect on the system temperature difference. A variable volume unit would certainly reduce this effect.
- The radiation heat loads may affect significantly the thermal response of the system due to boiling process of liquid and large mass transfer towards the pressure reduction reservoir.

Acknowledgements

The authors gratefully acknowledge D. Garcia, A. Four and all the technical support staff of CEA-SBT for their help for manufacturing and commissioning of the cryostat and testing the CLHP. This work is performed in the frame of a collaboration agreement between CEA and CAS. The Chinese participation is partly supported by the National Natural Science Foundation of China (Grant Nos. 50676098 and 50906095).

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