Nitrogen Cryogenic Loop Heat Pipe: Results of a First Prototype

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ABSTRACT

Efficient thermal links are needed to ease the distribution of the cold power in satellites. Loop heat pipes are currently used at room temperature as passive thermal links based on a two-phase flow generated by capillary forces. Transportation of the cold power at cryogenic temperatures requires a specific design. In addition to the main loop, the cryogenic loop heat pipe (CLHP) features a hot reservoir and a secondary circuit that allows the loop to be cooled down from room-temperature conditions. A first prototype of a CLHP working with nitrogen around 80K has been developed at CEA-SBT and tested in collaboration with the CAS/TIPC. The general design, the instrumentation, and the performance results are presented. The effects of mass inventory and the influence of the secondary circuit are presented and discussed. It is shown that, owing to an appropriate design of the prototype, a cold power of 19W with a limited temperature difference of 5K is obtained across a 0.46m distance.

INTRODUCTION

Operation of instruments and electronics onboard satellites and spacecrafts requires efficient cooling systems. Original designs of cryocoolers have been developed at Commissariat à l'Energie Atomique – Service des Basses Températures (CEA-SBT) for many years, within the scope of Technical Research Programs of the European Space Agency. These include the development of systems such as pulse tube cold fingers, sorption coolers, and continuous adiabatic demagnetization refrigerators. As distribution of cold power across large distances (1m for instance) may cause significant temperature gradients, the development of efficient thermal links to distribute it is simultaneously considered as a new technical challenge.

Capillary pumped two-phase cryogenic fluid circulators have many advantages[1] in comparison 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. At cryogenic temperatures, this concept has been already studied using different designs and cryogenic working fluids such as ethane, oxygen, nitrogen, neon, and hydrogen.[2,3,4,5,6]

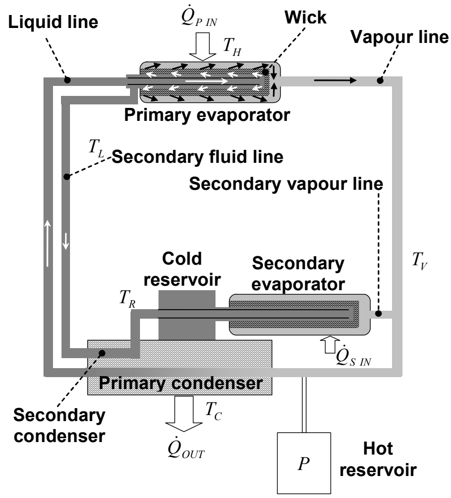

Figure 1 CLHP Flow diagram.

The "Cryogenic Loop Heat Pipe"(CLHP) system invented by Swales Aerospace[4] has been chosen to be evaluated by CEA-SBT. A first laboratory prototype has been designed and tested at around 80 K and in 1W-21W cold power range using nitrogen as the working fluid. In this paper the design principles are presented, the prototype is described and the results are discussed.

DESIGN

The CLHP consists of a primary loop for heat transfer and a secondary circuit for heat leak management and cool-down of the thermal link. Figure 1 and Figure 2 show the operating principle and the layout of the design. Table 1 lists the dimensions of the components. Some aspects of the sizing are given elsewhere.[7]

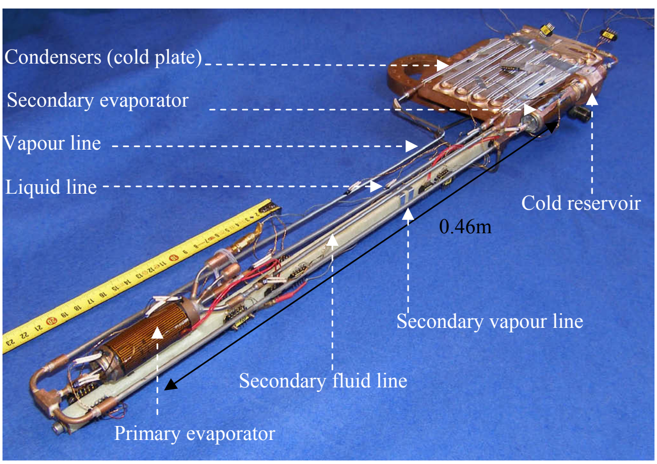

Figure 2 Layout of CLHP prototype.

Table 1. Components and dimensions of the CLHP.

Components	Dimensions in mm
Primary evaporator	22ODx64L
7.5 micron pore-size wick	35% porosity, 2 10^{-13} m^2 permeability
Serpentine primary condenser	3.5ODx1502L
Liquid line	3.0ODx328L
Vapour line	3.0ODx567L
Serpentine secondary condenser	3.5ODx528L
Secondary fluid line	3.0ODx330L
Cold reservoir	19ODx58L
Secondary vapour line	3.0ODx747L
Hot reservoir (1liter) or (4 liters)	88.9ODx277L or 102ODx678L

The primary evaporator is a three port type. It features a cylindrical body made of good conductivity copper, with machined grooves in the inside for the vapour flow. A Kapton backed foil heater is bonded to the body to simulate the heat load $\dot{Q}_{P\,IN}$. The wick is tubular and made of 316L stainless steel. A bayonet located on its axis allows the liquid to penetrate the liquid core. The primary evaporator has neither a secondary wick nor a collocated compensation chamber which is replaced by a cold reservoir thermally linked to the cold plate and hydraulically connected to the primary evaporator by the secondary fluid line.

The secondary circuit consists of the fluid line, the cold reservoir, the evaporator and the vapour line. The geometry of the secondary evaporator is identical to the primary one and equipped with the same heater to generate a secondary heat load $\dot{Q}_{S\,IN}$. For the test program the cold reservoir is directly clamped to the cold plate without any thermal shunt and is not equipped with a heater. The volume of the cold reservoir is of the same order as that of the primary condenser.

The two condensers are copper serpentine tubes brazed to the same thick copper cold plate to get an uniform temperature field. No particular effort is made to minimize the mass of the thermal link

TEST SET UP

All tests were conducted in a 354 mmOD x 750mmL vacuum chamber at CEA-SBT. A Gifford McMahon refrigerator was used as a cold source and bolted to the cold plate which is regulated at a temperature T_C of 81.5K. All cold parts of the CLHP were on the same horizontal plane and were surrounded by a 77K thermal shield cooled by liquid nitrogen (Figure 3). The hot reservoir is connected to the cold parts by means of a small pipe equipped with a valve. Two sizes of hot reservoir were used. The larger one is used for cool down tests and the smaller for cold steady state tests.

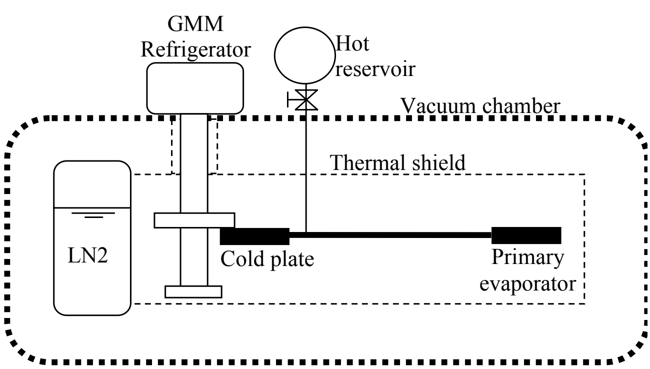

Figure 3 Experimental set-up

The CLHP is equipped with two pressure transducers connected to the hot reservoir and with Lakeshore Pt100 thermometers on all components. In addition, a particular effort is devoted to the internal instrumentation of the primary evaporator. Its bayonet is equipped with a capacitive sensor[8] to measure the vapour fraction in this region. This sensor gives valuable information about possible vapour production phenomena due to back conduction heat leak in the wick. This specific instrumentation limited the maximum allowable pressure to 7 bar.

The pressure and temperatures are measured respectively within an accuracy of +/-0.02 bar in the range 0-3bar abs, +/-0.2bar in the range 3-30bar, and +/-0.2K. The heating powers $\dot{Q}_{PIN}$ and $\dot{Q}_{SIN}$ are measured within an accuracy of +/-3% of the measurement, and the capacitive sensor allows measuring the vapour fraction within an accuracy of +/- 2%.

TESTING

Test goals

For this prototype, the test goals were:

1. Start up from room temperature conditions
2. Study the effect of the fill pressure *FP* (17.7bar – 25.6bar) and of the secondary heat load $\dot{Q}_{SIN}$ (0W – 11W)

Test procedure

Each test series starts with the cool down of the thermal shield by filling the liquid nitrogen tank.

Cool down test: The isolation valve is left open. The CLHP and its hot reservoir are filled up to a required pressure (about 7 bar) and the cryocooler is turned on to decrease the cold plate temperature down to its set point (81.5K). Once the secondary evaporator is cold owing to gravity liquid transfer coming from the cold reservoir, its heater is switched on at 5W and vapour is pushed through the secondary vapour line towards the primary condenser where it is condensed. Therefore, liquid is pushed towards the primary evaporator, which is consequently cooled down. The fluid returns through the secondary fluid line and is condensed before returning to the secondary evaporator. The test is finished when the primary evaporator is cold.

Steady state cold test series: The isolation valve is initially closed. The hot reservoir is filled up to a required fill pressure, and the isolation valve is manually and gradually opened to avoid over pressurization of the cold parts. The warm gas flows towards the cold parts and is condensed in the cold reservoir and in the two condensers. Then, the same procedure as for the cooldown tests is used to cool the primary evaporator. The test series is then started at constant $\dot{Q}_{SIN}$, and the primary heat load $\dot{Q}_{PIN}$ is increased step by step. The test series is finished when depriming is observed.

Experimental Results and Analysis

Figure 4 presents a typical cooldown sequence of the CLHP for 6.93 bar fill pressure. It demonstrates the efficiency of the cooldown. The capacitive sensor, which gives the vapour fraction around the bayonet, detects some liquid when cold flow starts in the liquid line and in the secondary fluid line. This occurs first, due to gas condensation in the two condensers and the cold reservoir, and second, due to the secondary vapour forced flow of the secondary heated evaporator. The primary evaporator is completely cold when a sharp decrease of vapour fraction down to about 50% is observed. At this time, the wick is full of liquid, the liquid core—which is horizontal—is half full of liquid, and the CLHP is ready to be primed.

Figure 5 presents a typical test series sequence for 21.5 bar fill pressure, 1 W secondary heat load $\dot{Q}_{SIN}$, and a cold plate temperature T_C maintained at 81.5 K. The heat load $\dot{Q}_{PIN}$ is gradually increased step by step up to 19 W, which corresponds for these conditions to the maximum capability of the CLHP. The next step (21 W) leads to depriming of the CLHP, which is characterized by a sharp increase in the vapour fraction around the bayonet and of the evaporator temperature. During the test series, the system temperature difference ΔT between the primary evaporator and the condensers increases with the heat load and reaches limited values of around 5 K for 19 W.

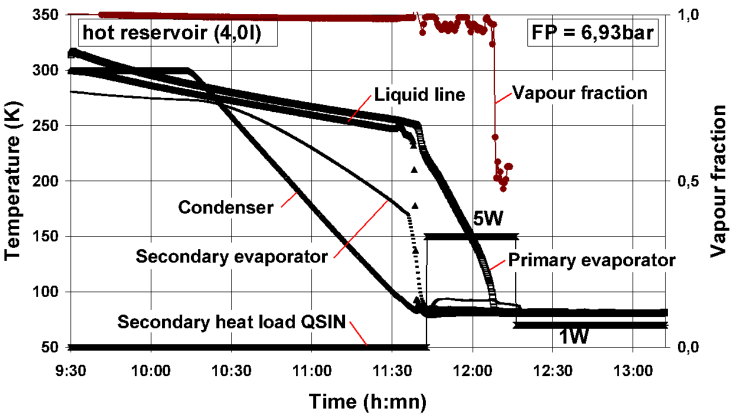

Figure 4. Cooldown sequence of the primary evaporator.

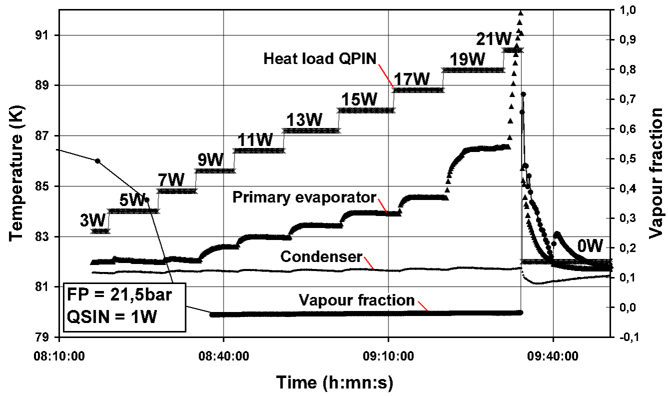

Figure 5. Typical performance test series.

The effect of fill pressure is presented in Figures 6, 7 and 8. A sharp dependency on the system temperature difference ΔT is measured (Figure 6). This effect is not observed on the evaporator temperature difference where the heat transfer is excellent (Figure 7). This effect is mainly observed on the coldplate subcooling (Figure 8), because for any fill pressure the cold reservoir is completely full of subcooled liquid. A 21.5 bar fill pressure which corresponds to less than 2 K is chosen.

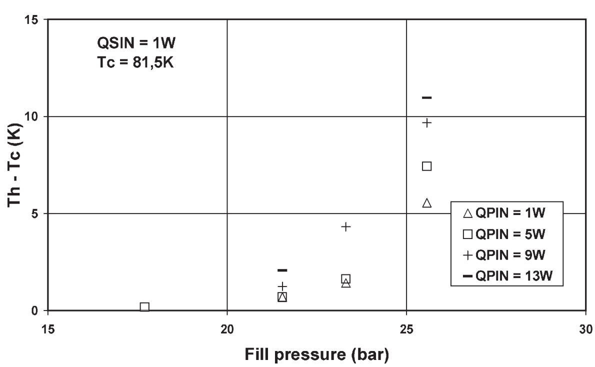

Figure 6. System temperature difference ΔT between evaporator T_H and cold plate T_C as a function of fill pressure.

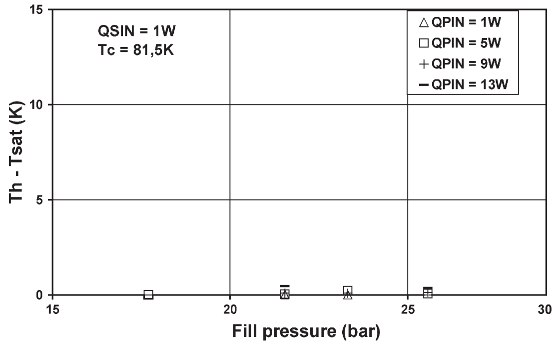

Figure 7. Evaporator temperature difference between evaporator T_H and saturation T_{Sat} as a function of fill pressure.

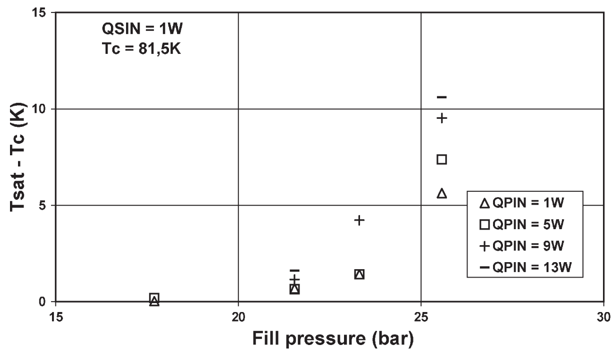

Figure 8. Coldplate subcooling as a function of fill pressure.

Larger fill pressure, and therefore larger subcooling, is not necessary because the thermal shield is cooled at 77 K, and the parasitic heat loads are negligible.

The fill pressure has also an effect on the capability of the CLHP (Figure 9). This figure shows that the fill pressure must not be too low. Indeed, at low fill pressure (17.7 bar), the cold plate subcooling (Figure 8) is negligible, and therefore the pressure drop in the condenser is very large due to very large two-phase length, and the maximum capillary pressure is reached at very low heat loads $\dot{Q}_{PIN}$.

On the other hand, Figure 9 shows that it is not necessary to operate the CLHP with a large secondary heat load $\dot{Q}_{SIN}$, because the secondary vapour flow produced by the secondary evaporator induces additional pressure drops in the vapour line and in the primary condenser. A 1 W power

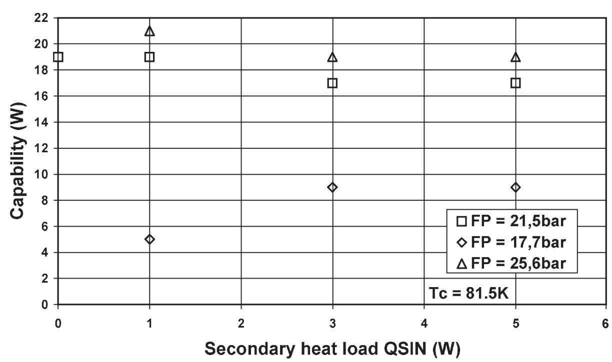

Figure 9. Capability as function of secondary heat load.

seems to be sufficient, but for 21.5 bar fill pressure and at very low heat loads $\dot{Q}_{PIN}$, the available cold power due to the sub cooling of the liquid entering the primary evaporator is not enough to compensate for the back conduction heat leak through the primary wick. This is the reason why vapour is observed in the liquid core (Figure 5). This apparently does not affect the operation of the primary evaporator.

CONCLUSION

In 2007, CEA-SBT designed, built and tested a first prototype of a CLHP. The experimental program allowed characterizing the thermal performance of the thermal link. It shows the excellent efficiency of the secondary circuit to cool down the system and the low thermal resistance of the system. Optimum fill pressure and secondary heat load are found, which lead to a capability to transport a cold power of 19 W at 80 K with a limited temperature difference of 5 K across a 0.46 m distance. The specific instrumentation installed in the primary evaporator has shown some situations where vapour can be present in the liquid core of the primary evaporator without affecting the operation of the link.

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