STATE-OF-THE-ART EXPERIMENTAL STUDIES ON LOOP HEAT PIPES

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ABSTRACT

The present study provides a concise overview of the main feature evolution of Loop Heat Pipe (LHP) designs, in particular on the evaporator/compensation chamber, based on experimental studies published between 1998 and 2010. It presents a synthesis of results on LHP performances at steady-state, and observations related to the LHP dynamic behaviours, as start-up or oscillating behaviours. The objective of this review is to gather as much information from the LHP experimental studies so as to provide a database, and thus promote the development of tools for understanding and predicting the LHP behaviors. The review shows some omissions in the LHP description and in the results presentation that makes these data difficult to use for further studies. This review should promote some coordination for future experimental works on LHP.

Keywords: *LHP, review, two-phase heat transfer system, porous, capillarity.*

1. INTRODUCTION

A two-phase capillary pump loop, as Loop Heat Pipe (LHP), is an efficient heat transfer system based on the liquid-vapor phase change phenomena. A LHP consists of an evaporator, a condenser, a compensation chamber (also called reservoir), and some vapor and liquid transport lines (Fig. 1). Only the evaporator and eventually the compensation chamber contain wicks, while the other components could be made of smooth tubing. Like a conventional heat pipe, a two phase capillary loop uses capillary action to circulate the working fluid in a sealed enclosure and, in the process, to transfer waste heat from one location (heat source) to another (heat sink). Hence it contains no mechanical moving part to wear out or require electrical power to operate. Despite the apparent design simplicity and the theoretical operational robustness, a loop heat pipe is a complex system, whose operation involves multi-scale mechanisms, as thermo-hydraulic in porous media ($\sim 10 \mu\text{m}$), wettability and evaporation in thin film ($\sim 1 \mu\text{m}$), condensation two-phase flow ($\sim 1 \text{ mm}$), and thermal and hydrodynamic couplings at the system scale ($\sim 1 \text{ m}$). Detailed explanations on LHP operation are presented by Ku (1999) and Maydanik (2005).

Compared to conventional heat pipes, the originality of the LHP design assigns some additional advantages: (i) in term of operation against gravity with the use of wicks of very small pores (1 or 2 orders less than conventional heat pipes); (ii) in terms of capacity to transport large amounts of heat from the evaporator to the condenser, by decreasing the liquid and vapor fluidic resistance along the adiabatic zone (with the use of smooth tubing); (iii) in terms of heat transfer efficiency in the evaporator and the condenser, with the design of appropriated structure for evaporation and condensation phenomena; (iv) in terms of LHP integra-

tion in the electronic architecture, with the flexibility in the evaporator and condenser positions also due to the use of smooth tubing in the adiabatic zone (tubing which are easily constrained).

Another two-phase capillary pump loop, named Capillary Pumped Loop (CPL), has similar advantages as LHP compared to heat pipe. The basic distinction between a traditional CPL and a traditional LHP lies in the fluidic and thermal attachment of the compensation chamber to the evaporator. But seemingly this distinction has a large impact on the design and operation of the capillary loops (Maidanik and Fershtater, 1997; Nikitkin and Cullimore, 1998). In what follows, only studies on LHP are reviewed as they cover most of the publications since the 2000s.

The appearance of LHP was a response to the challenge connected with the specific request of the aerospace technology for highly efficient heat transfer devices. Currently, LHP are already part of the thermal control of satellites (Russian Granat, Obzor spacecraft, Boeing's HS 702 communication satellites). As demonstrated for spacecraft thermal control systems, LHP can likely be a high performance heat transport device for terrestrial electronic cooling applications. The thermal management of modern electronics has become a problem of significant interest due to the demand for power and reduction in packaging size. The requirements of next-generation microprocessors in terms of power dissipation and heat flux rate will certainly outgrow the capability of today's thermal control technology. The most versatile feature of using LHP is the wide variety of design embodiments that can be realized to take advantage of the available space around the electronic to be cooled. Computer technology, in particular the PC notebook, is a new dimension of the LHP application, which was revealed by the appearance of miniature and fairly efficient devices (Pastukhov *et al.*, 2003). Flexible, ramified, controllable,

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Fig. 1 Schematic of the main physical mechanisms in a LHP

reversible, miniature or cryogenic are some of the LHP designs that have been presented in the literature (Maydanik, 2005).

The first experimental studies on LHP (years 80-90) mainly focused on the characterization of the performance and the behaviors of the LHP when these ones are subjected to a protocol in connection with applications and conditions of space. For reasons of economic issues, the LHP are generally only briefly described. These studies, while important in the loop development process, do not allow people outside the project to make the most of these data for further reflection on the operation and optimization of such systems. While the space applications are a driving force for the development of LHP, extending their use in the thermal control of electronic components for terrestrial applications has generated additional interest on this topic to a wider community of researchers. Thus, from the 2000s, there has been an intensification of the literature on this subject. In light of these articles, this new community seems inclined to share more detailed information on the design and sizing of the LHP.

The objective of this review is to gather as much information from these experimental studies so as to provide a database, and thus promote the development of tools for understanding and predicting the LHP behaviors. Indeed, we find in the literature some transient models capable of reproducing fairly well the dynamic behavior of LHP (Launay *et al.*, 2007a; Vlassov and Riehl, 2008) but after adjustment of certain variables, either due to lack of information issued by the experimental study, or by the difficulty of characterizing some of these variables. This work also has a role to coordinate future experimental studies stressing the need to better describe a number of elements of the LHP in order to promote further the exploitation of such data.

2. SYNTHESIS OF LHP EXPERIMENTAL STUDIES

Tables 1 and in "Appendix A" summarize an exhaustive review of LHP experimental studies published between 1998 and 2010. Listed in the table are the papers for which the LHP characteristics and the results are substantial enough to generate a qualitative or quantitative analysis. This database, which includes more than fifty LHP of various characteristics, is a source of information to start a constructive reflection to the improvement of the LHP knowledge. Only the studies of LHP with relatively conventional design (with one evaporator and one condenser) are presented in these tables. In this review has been deliberately excluded

the LHP of cryogenic application because of their specificity (secondary capillary pump, high pressure tank). The LHP with multiple evaporators and/or multi-condensers are poorly represented in this table because of the difficulty of access to the geometry data and to the results formulation. The reader may refer to the articles of Hoang *et al.* (2003) and Mo and Liang (2006) for the cryogenic LHP, and to the article of Ku *et al.* (2009) for multiple LHP evaporators/condensers for more information on the specificities of these kinds of two-phase capillary loops.

In the Tables 1 and "Appendix A", the LHP studies are arranged chronologically. Table 1 provides a concise overview of the main feature evolution of the LHP designs, in particular on the Evaporator/Compensation Chamber (E/CC), as well as the objectives of the various studies. Although the E/CC of cylindrical design were initially preferred by the researchers (for various reasons of manufacturing processes and/or of the mechanical strength in connection with the use of ammonia as the working fluid), a number of new E/CC configurations have been proposed during the years 2009-2010. The diversity of these designs is a response to the use of LHP for the thermal control in the Earth's field (use of other working fluids, high flux densities, implementation challenge, etc.). The main design characteristics of the E/CC are developed in subsection 3.2.

It emerges from Table 1 that the objectives of most of the experimental investigations conducted on the various LHP are to study their thermal performance, or to study their behavior during start-up or a power cycling test. Some of these studies concentrate on analyze the influence of certain parameters, such as the fluid filling ratio, the working fluid nature, the relative position of the evaporator to the reservoir and the condenser, or the capillary wick characteristics on the LHP behavior. Readers, who are interested in analyzing the effect of parameters on the LHP operation, may refer to the review paper of Launay *et al.* (2007b). The present paper focuses more specifically on the summary of the results and observations related to steady-state and dynamic behaviors of the LHP, the start-up, and the oscillating behaviors. This analysis is presented in Section 4.

A more detailed summary of the LHP experimental studies is presented in Table "Appendix A". For each LHP, two parts can be distinguished. The first part presents the LHP specificity and the description (dimensions, materials) of each of these components. The second part summarizes the main objectives of the experimental study, the installed instrumentation and the thermal performance obtained for the LHP. We can note a lack of information in this table, particularly regarding the characteristics of the capillary wick, the hydraulic connections between the evaporator and reservoir, and the thermal resistance of the evaporator RE. In this table, it would be interesting to add the effective thermal conductivity of the porous wick, but this data is too infrequently reported in the literature.

It is observed that the development trend is the miniaturization of the LHP with proposals of various designs of E/CC. These proposals are a response to various kinds of criteria:

- New applications involving the selection of other working fluids (and then some other materials);
- Implementation in electronic module architecture: constraints in terms of volume and containment;
- Operation reliability: durability, thermal and environmental constraints;
- Economic: weight, cost of manufacturing and packaging;
- Thermal performance: efficiency of the heat transfer and the heat transport;
- Knowledge: choice of materials, and specific instrumentation.

How to define the transition between a conventional LHP and a miniature LHP? Various criteria have been found in the literature to answer this question. For Maydanik (2004), a LHP is considered as minia-

Table 1 Synthesis of LHP experimental studies.

		Evaporator/CC					Vapor grooves	Cond.	Study	Parameters of investigation or observations													
Authors	Year	NH3 working fluid cylinder	flat (CC in the length)	flat (CC in the thickness)	Bayonet	secondary wick	Porous wick	Evaporator wall	Air heat exchanger	Liquid heat exchanger	steady-state	transient	Performance	power cycle test	start-up	oscillations	Hysteresis	gravity	Fluid charge	NCG	Porous material	working fluid	Heat sink
Baker	1998	o	o										o	o									
Cheung	1998	o	o			o			o				o	o		o							
Wrenn	1999	o	o		o				o				o										
Pauken	2000	o	o						o						o			o				o	o
Ku	2001	o	o					o					o		o	o	o		o				
Rodriguez	2001		o						o		o	o		o	o								o
Delil	2002			o				o		o								o			o		
Chuang	2003	o	o		o				o		o	o				o		o	o				
Delil	2003	o	o		o				o		o			o				o					
Kaya	2003	o	o						o		o	o			o	o		o					
Ogushi	2003	o	o			o	o		o		o	o	o		o			o					
Pasthukhov	2003	o	o					o	o	o	o							o			o		o
Boo	2004			o	o			o		o	o	o	o	o				o	o			o	o
Lee	2004			o				o		o	o	o	o	o					o		o		
Mishkinis	2004	o	o						o		o	o	o			o							
Zhang	2004	o	o		o				o		o	o	o			o	o	o					
Hsu	2005			o					o		o	o	o					o			o	o	
Rhiel	2005		o		o		o		o		o	o	o	o	o								
Chen	2006	o	o		o		o		o		o	o	o	o		o	o		o				o
Nagano	2007	o	o		o				o		o			o				o					
Pasthukhov	2007		o	o				o		o			o										o
Rhiel	2007				o			o		o		o		o									o
Vershinin	2007a		o		o				o	o	o		o					o				o	o
Vershinin	2007b	o	o						o			o		o			o		o	o			
Bai	2008	o	o		o		o		o		o				o			o					
Entremont	2008	o	o		o				o		o	o	o			o		o					
Joung	2008			o		o		o		o	o	o	o			o			o				
Singh	2008				o			o		o		o	o	o	o	o							
Tanaka	2008		o				o		o			o		o									
Vasiliev	2008		o		o		o		o		o	o	o	o	o	o	o				o	o	
Yeh	2008	o	o		o				o		o		o								o		
Adoni	2009		o		o		o		o		o	o	o						o				
Huang	2009		o					o			o				o	o							
Maydanik	2009	o	o						o	o	o		o								o		o
Yeh	2009	o	o				o		o		o		o								o		
Becker	2010			o			o		o		o		o					o					o
Celata	2010				o		o		o		o	o	o		o	o		o	o				
Choi	2010				o		o		o		o	o			o								
Li	2010				o		o		o		o	o	o	o	o	o	o						
Lin	2010	o	o		o				o		o	o	o	o	o	o		o					o
Maydanik	2010			o					o		o		o										
Nagano	2010		o				o		o			o		o	o								
Pasthukhov	2010			o				o		o	o	o	o					o					o
Singh	2010				o		o		o		o	o	o		o					o			
Joung	2010a			o			o		o		o	o	o	o	o	o		o					o

ture if the evaporator outer diameter does not exceed 8 mm (with or without the saddle?), and if the diameters of the liquid and vapor lines are less than 3mm. Moreover, for flat evaporators, the ratio between the length and the thickness should not be less than 2. For Li *et al.* (2010a), the miniature LHP is defined only from the dimensions of the liquid and vapor line diameters (the diameter must be in between 1 - 3 mm).

3. DESCRIPTION OF LOOP HEAT PIPES

Maydanik (2005) proposed a LHP classification depending on the LHP configuration, the design of its components, the temperature level and the control strategy (Table 2). This classification is intended to show a general panel on the various types of existing LHP.

Table 2 Classification of LHPs (Maydanik, 2005)

LHP design	LHP dimensions	Evaporator shape	Evaporator design
<ul style="list-style-type: none"> Conventional (diode) Reversible Flexible Ramified 	<ul style="list-style-type: none"> Miniature All the rest 	<ul style="list-style-type: none"> Cylindrical Flat disk-shaped Flat rectangular 	<ul style="list-style-type: none"> One butt-end compensation chamber Two butt-end compensation chambers Coaxial
Condenser design	Number of evaporators and condensers	Temperature range	Operating-temperature control
<ul style="list-style-type: none"> Pipe-in-pipe Flat coil Collector 	<ul style="list-style-type: none"> One Two and more 	<ul style="list-style-type: none"> Cryogenic Low-temperature High-temperature 	<ul style="list-style-type: none"> Without active control With active control

In this section, we propose to describe in more detail each LHP components, and then, to make the connection between these components and the behaviors or the thermal performances of the LHP. The subsections are presented in a certain order that might look like a LHP design procedure for a given application. Obviously, the design procedure can not be linear and requires flow back and forth between the various components in the same way that there are many thermo-hydraulic couplings between the LHP components.

3.1. Working fluids

As part of a LHP design procedure for a targeted application, the selection of the working fluids is the first step and must satisfy a number of criteria:

- To be in liquid-vapor state at the minimum operating temperature, outside the storage phase.
- The saturation pressure of the fluid at the maximum operating temperature, $P_{sat}(T_{max})$, should be below the critical pressure P_c : In general, the maximum allowable pressure is less than P_c and is limited by the mechanical strength of the LHP elements. A decrease in the maximum pressure induces a reduction in the wall thickness of the materials that make up the system, and consequently, a reduction of the size and mass of the LHP.
- Respond to standards in terms of environmental, hazardous and harmful properties.

In a second step, other criteria, most likely to characterize the ability of the fluid to transfer heat for this type of system, have been proposed by Launay *et al.* (2010). These criteria, developed under the name of numbers of Merit, are linked with the critical mode of the LHP operation.

From this new selection, it is then necessary to address each of these fluids and the compatible materials to avoid chemical reactions. These chemical reactions would cause a degradation of the mechanical strength or the tightness of the LHP, or a generation of condensable gases which could alter the LHP operation. The materials covered are those that make up the structure of LHP (shell of the evaporator tank, liquid line and vapor condenser) and the capillary structure. This problematic has been extensively treated in the heat pipe development. Information on the heat pipe life test procedures may be found in Reay and Kew (2006). Mishkinis

et al. (2010) have performed a survey of possible solutions and selection of potential working fluids and LHP materials for envelope and wick.

According to the various criteria listed above, ammonia seems to be the most appropriate fluid for the thermal control in space applications. From Tables of Appendix A, it is the fluid that was the most used. In such applications, other alternative fluids such as propylene and acetone have rarely been tested. For ammonia, Stainless Steel is the most common material that has been used for the container, while the Nickel or Titanium materials have been used for the capillary structure.

The extension of the LHP application to ground and to some operating temperature ranges around 20 to 100 °C has guided researchers to other working fluids. Fluids such as methanol, acetone, ethanol and more unusually, pentane and R134a, were used for this type of application, water, according to its thermo-physical characteristics and its innocuous nature, is a working fluid that is increasingly used. The use of water as the working fluid partly explains the recent use of copper as the container material and the capillary structure of the LHP.

3.2. Evaporator/Compensation Chamber (E/CC)

While initially the E/CC of cylindrical shape was preferred, partly for reasons of mechanical strength due to the use of ammonia and to the spatial context, recent studies have shown a diversity of E/CC designs. The main categories of E/CC designs presented in the literature are shown in Fig. 2. The E/CC designs can be classified in three categories :

- The E/CC of cylindrical shape with the CC located in the extension of the evaporator (Fig. 2(a))
- The E/CC of flat shape with the reservoir located in the extension of the evaporator (Fig. 2(b))
- The E/CC of flat shape with the reservoir located in the thickness of the evaporator (Fig. 2(c)). For this case, the evaporators can be a square, rectangular or disc-shaped.

The proposal for these various geometries of E/CC is a response to expanding applications of the LHP with the issue of miniaturization for a better implementation (area, volume) within a 3D architecture of the electronic modules, and with the issue of higher dissipated heat flux rates.

What is the LHP problematic related to the miniaturization and to the increase of the heat flux rates? The operation and performance of a LHP are very sensitive to the thermal hydraulic link between the evaporator and the reservoir. The E/CC miniaturization tends to reduce the conductive thermal resistance between the evaporator and the compensation chamber, and thus to increase the part of parasitic heat flux on the heat transferred to the condenser by latent heat. It may be reminded here that the parasitic heat flux is compensated by the subcooling heat flux of the liquid flow at the reservoir inlet: increase in the parasitic heat flux tends to increase the liquid subcooling, and hence to increase the LHP operating temperature.

In the literature, various approaches are implemented to reduce the amount of parasitic heat flux:

- Using materials of medium thermal conductivity (Stainless Steel, Inconel) for the E/CC container;
- Reducing the wall thickness, up to 0.25 mm for a cylindrical E/CC in Stainless Steel (Pastukhov *et al.*, 2003). The wall thickness depends on the shape and the material of the evaporator, in particular to prevent a deformation of the container under the pressure force of the working fluid. The procedures of manufacturing or machining of the component can also define the allowed minimum thickness.
- Using a capillary structure of low or medium thermal conductivity (polyethylene, polypropylene, PTFE, Stainless Steel, Titanium);

- Reduction of the conductive section between the evaporator and the reservoir by selecting the appropriate design of the E/CC (Riehl and Siqueira, 2006)
- Using materials of various kinds for the evaporator compared to the reservoir (Adoni *et al.*, 2009)

In contrast to these trends, some copper miniature E/CC, which show high performance thermal characteristics, have been presented in the literature (Li *et al.*, 2010a). According to the analytical model of Launay *et al.* (2008), it could be noted that increasing the thermal resistance of the conductive wall (R_{wall}) or the porous wick (R_w) is not a sufficient condition to reduce the parasitic heat flux. For a miniature LHP, the role of the ratio R_E/R_{wall} is predominant and must be reduced. R_E characterizes the thermal resistance between the heat source and the vaporization front in the capillary wick, mainly including the conductive thermal resistances of the container wall at the evaporator and of the dry region in the capillary wick. If R_{wall} can be easily characterized globally, R_E is much more complex. Indeed, the thermal resistance R_E includes a set of coupled physical mechanisms, some of which are difficult to characterize, such as:

- The heat conduction in the thickness of the evaporator container;
- The thermo-mechanical contact between the evaporator container and the capillary wick. The quality of the contact depends on a number of parameters, such as the surface roughness (related to the manufacturing process and/or machining) of the elements in contact, the method of assembly/installation of the evaporator (mechanical clamping, brazing, soldering?). This contact quality can also be dependent on a consideration of cost. In the literature, there is little or no information regarding the implementation or the control of the wick/wall mechanical contact;
- The thermo-hydraulic phenomena in the capillary wick, including the conduction, the liquid and vapor flows, the vaporization, and the capillarity associating the phenomenon of the fluid/porous wettability. All these mechanisms contribute to the position of the evaporation front in the capillary wick. The thermo-hydraulic characteristics of the porous wick play a fundamental role.

Various experimental approaches have been implemented to improve the performance of the E/CC, especially by playing on:

- The characteristics of the capillary wick
- The characteristics of the vapor channel
- The hydraulic link between the evaporator and reservoir
- The use of a conductive heat spreader (saddle) between the heat source and the evaporator container.

Porous characteristics Figure 3 shows the values of the porosity ϵ and the permeability K as a function of the pore diameter for the capillary wicks, whose characteristics are specified by the authors of the various experimental studies on LHP (Tables of Appendix A). In Fig. 3, there is no distinction between the nature of the porous materials. There is a wide dispersion of ϵ and K for a given pore diameter. The porosity is between 30 and 75 % regardless of the pore diameter $20 \mu\text{m}$. The permeability is between 10^{-14} and $3 \cdot 10^{-13} \text{m}^2$ and tends to increase with the pore diameter. However, one must be cautious about analyzing this chart.

Fig. 3 Properties of common LHP wicks (Hoang and Ku, 2003)

Is the dispersion of these characteristics due to the material, the manufacturing process, or the definition or characterization of the wick pore diameter? If the porosity and permeability are generally well defined for a given porous material, the concept of pore diameter seems more difficult. For the ideal case of porous spherical particles with uniform diameter, the equation of Kozeny-Carman connects the particle diameter to the permeability and porosity ($K \sim \epsilon^3 * \phi_p^2$). What is it for polydisperse porous wicks that are more representative of the LHP capillary structures? What

(a) E/CC of cylindrical shape

(b) E/CC of flat shape with the reservoir located in the extension of the evaporator

(c) E/CC of flat shape with the reservoir located in the thickness of the evaporator

Fig. 2 Schematics of evaporator designs

size of pores must be considered in calculating the capillary limit in relation to the pore distribution? This information is rarely made explicit in the literature of LHP, some authors (Becker *et al.*, 2010) clarify the nature of the pore diameter by the expression of "breakdown pore radius".

Hoang and Ku (2003) have compiled some porous characteristics by distinguishing the materials. This compilation shows some dependence (increase) of the permeability with the pore diameter (Fig. 4). Concerning the nature of the porous materials used as LHP capillary structure, Nickel sintered porous is the most commonly encountered, followed by the use of polymers (polyethylene, polypropylene, PTFE), Titanium, Copper, Stainless Steel, and more marginally Iron Oxide, Brass, and Nickel-Chromium. The use of copper as a base material of the LHP capillary wick came out recently. The pore diameters of these various porous materials are between 1 and 20 μm , except for copper, which has larger pore diameters (between 20 and 100 μm). Results obtained by Li *et al.* (2010a) indicate that the use of wicks of large pore diameter is not necessarily a handicap.

Fig. 4 Properties of common LHP wicks (Hoang and Ku, 2003)

The selection of porous properties is critical for the performance of the LHP. According to Hoang and Ku (2003), the required properties for the capillary wick are: (i) small pore diameters to ensure a large capillary driving pressure, (ii) a large porosity and permeability for lower hydraulic resistance, (iii) a low thermal conductivity to minimize parasitic heat flux through the porous wick. An increase of the permeability with the pore diameter is observed experimentally and is predicted by theoretical models as well. Consequently, conditions (i) and (ii) may not be achieved simultaneously: the hydraulic performance of a capillary wick can be regarded as a balance between capillary forces and flow properties.

Li *et al.* (2010b) studied the effect of various parameters relating to the manufacturing process on the properties of a sintered-nickel porous. The studied parameters are the mechanical stress applied to the sample during manufacturing, the size and the concentration of microcrystalline cellulose added to the sintered-nickel porous. For an equivalent porosity between the various samples, the measured hydraulic performance (time evolution of the saturation level of water in the capillary structure) seems to be to the benefit of the porous wick of smaller diameter pores compared to those of higher permeability. Yeh *et al.* (2008) have developed a biporous wick structure of sintered nickel (3 μm in diameter) from inclusion of Na_2CO_3 space-holder (32–48 μm in diameter) (Fig. 5). The distribution of the pore diameters for the mono-porous (without inclusion) and biporous (with inclusion) are shown in Fig. 6. The average pore diameters and permeabilities are 10 μm and 10^{-12}m^2 , respectively, for the mono-porous, and 7 and 24 μm (2 average pore diameters) and 10^{-11}m^2 for the bi-porous. This example of manufacturing process shows the feasibility of raising the contradiction between the opposite effects of the pore diameter and the permeability on the hydraulic performance of the capillary structure.

Fig. 5 Schematic view of the process to produce bi-porous wicks (Yeh *et al.*, 2009)

Fig. 6 Measured pore size distribution of monoporous and bi-porous wicks (Yeh *et al.*, 2008)

What about the thermal properties of porous media, such as the thermal conductivity? The characterization of the porous thermal conductivity is as complex as the characterization of the pore diameter as it is highly dependent on the manufacturing process. This may explain why this characteristic is rarely provided by the researchers in LHP studies? In the literature, one can find different formulations of the equivalent thermal conductivity of porous k_{eff} according to the intrinsic thermal conductivity of each element (metal and fluid) and to the porosity (Mo *et al.*, 2006). A large dispersion of results for k_{eff} has been obtained depending on the formulations. It appears that the choice of one or the other formulations can be made retrospectively, by comparison with measurements. For a dry porous nickel wick, Li *et al.* (2010b) measure an effective thermal conductivity (k_{eff}) of 3 $\text{W/m} \cdot \text{K}$ for a wick porosity of 75 %, and Mo *et al.* (2006) measure a k_{eff} of 6 $\text{W/m} \cdot \text{K}$ for a wick porosity of 56 %. According to these measurements, the thermal conductivity of the nickel material, which is deduced from a parallel scheme formulation, is therefore up to 13 $\text{W/m} \cdot \text{K}$. This value, far from the theoretical conductivity of nickel (90 $\text{W/m} \cdot \text{K}$), can be explained by the discontinuity of the material at each point of contact of the nickel sintered beads. Moreover, Mo *et al.* (2006) observed a significant increase in the effective thermal conductivity of a porous media saturated with liquid (transition from 6 to 8 $\text{W/m} \cdot \text{K}$ without and with water, respectively) compared to the expected result with the formulations. This significant increase can be partly explained by a change in the quality of the contact between balls (replacement of air by liquid), thereby decreasing the thermal contact resistance.

This effect was also observed by Xin *et al.* (2010) for a monodisperse porous media in Nickel and a bidisperse porous media in Nickel and Copper (Nickel-Copper powder mixtures, $D_{Ni} = 2.5 \mu\text{m}$ and $D_{Cu} = 13 \mu\text{m}$). This study also reveals, that the effective thermal conductivity k_{eff} of the bidisperse porous media in Ni/Cu is much lower than the one

of pure monodisperse porous in Ni, and that Keff decreases when the concentration of copper increases. The authors justify the decrease of k_{eff} of the powder mixture by the formation of Ni-Cu solid solution.

Vapor channels The position and the shape of the vapor channels are other parameters that can influence the thermal resistance at the evaporator RE. Thus, among the LHP experimental studies, vapor channels have been machined either at the inner surface of the evaporator container, or at the outer surface of the porous wick. The hydraulic diameter and the channel spacing are variables (of the order of the millimeter) and some transverse channels (collectors) are sometimes added to the longitudinal ones (Figures of Appendix B). The choices of the position and the dimensions of the vapor grooves can respond to a criterion of heat transfer performance, or to a criterion related to the feasibility and to the cost of the groove machining.

The theoretical study of Altman *et al.* (2002) shows the effect of the design of the vapor channels on the heat transfer from the wall to the evaporation front position in the porous wick (Fig. 7). The characterization of the evaporation front shape in the porous wick and of the thermal performance that are derived for various sets of parameters is the goal of many modeling studies (Figus *et al.*, 1999; Kaya and Goldak, 2006).

Fig. 7 Design and technology versions and schemes of heat transfer of LHP evaporators (Altman *et al.*, 2002)

Thermal-hydraulic link between the evaporator and the compensation chamber To make the LHP more flexible, the E/CC should operate whatever its orientation (evaporator above or below the reservoir). The hydraulic link between the reservoir and the evaporator is then necessary to ensure the continuity of the liquid flow, and this, even in an unfavorable position (evaporator above the reservoir). The hydraulic link should promote a homogeneous hydration of the main capillary wick to avoid a local and premature drying of the evaporator.

Although the nature of the hydraulic link is rarely described in the literature, mainly three techniques have been implemented:

1. use of a capillary wick, called secondary wick. Its role may be double: (i) maintain the hydraulic connection between the fluid pool in the reservoir and the base of the main capillary wick, (ii) act as a thermal barrier between the evaporator and reservoir. Its required characteristics are a high permeability and a low thermal conductivity.
2. use of a bayonet that can transport the liquid from the liquid line directly to the evaporator core. This technique is generally used for cylindrical evaporators and maybe coupled with a secondary wick.
3. use of gravity effect by placing the reservoir above the evaporator. This is the general case of the flat evaporators operating at horizontal and with the reservoir above the evaporator.

Lin *et al.* (2010) have proposed an original configuration of E/CC, which includes a bayonet and two reservoirs located at the ends of the cylindrical evaporator. This configuration allows the LHP, without any secondary wick, to operate whatever its position.

Metal saddle at the evaporator A better distribution of heat flux from the heat source to the evaporator tends to homogenize the temperature of the heat source, while reducing the temperature for a given heat flux. Thus, the flat evaporators seem more convenient to evenly distribute the heat flux. In the case of cylindrical evaporators and flat heat sources, a metal saddle (usually in aluminum or copper) is inserted between the two components. The quality of the heat diffusion to the evaporation front position in the porous wick is then dependent on the geometry of the metal saddle and on its fixation with the cylindrical evaporator. The supplement of the metal saddle is not without any consequences on the integration of the evaporator in a 3D architecture of electronic modules, and on the LHP typical operation. In terms of integration, the saddle generates additional weight and thickness to the LHP evaporator. In terms of LHP operation, the thermal inertia of the saddle can significantly change the LHP dynamic behavior, especially on the fast transient behaviors as brutal startups at high heat fluxes. This point may be critical if the thermal diffusion in the saddle becomes the limiting heat transfer regarding the LHP startup. According to Ku and Rodriguez (2003), the thermal inertia of the saddle may also be one of the parameters responsible of the oscillating behaviors during steady-state operation, particularly on the low-frequency temperature oscillations.

A compromise should be found between the thermal inertia and the temperature homogenization at the heat source. In the LHP literature, the metal saddle is rarely described. However, this description is needed, particularly for the validation of transient LHP modeling.

3.3. Condenser

The role of the condenser is to evacuate the heat flux, which is dissipated by the heat source, to the surroundings.

The overall heat exchange coefficient at the condenser includes condensation, heat diffusion by conduction through the walls of the heat sink, and convective or radiative heat transfers with the surrounding environment that can be the air, the water or the space environment. In the literature, the diversity of the condensers can be represented by different combinations of the three element classes of Table 3. The most common condensers are the condensation in tube and a heat flux evacuation, either through a liquid cooled plate, or by a counter-current liquid flow heat exchanger (tube-in-tube), or by convective heat transfer of the ambient air over a finned surface heat exchanger.

Table 3 Classification of condenser elements.

Condensation cross-section	Thermal link	Modes of external heat transfer
- circular	- plate heat exchanger	- air forced convection
- annular	- finned heat exchanger	- air natural convection
	- tube-in-tube heat exchanger	- liquid forced convection
		- radiative heat transfer

Even if the condenser gives rise to less interest in terms of research compared to the evaporator, its design plays an essential role in the conception, design and performance of the LHP. In many cases, the heat sink is imposed by the surrounding environment of the system to cool (water circulation, air or space environment), and thus, it determines the type of heat exchanger. At the condenser, the heat transfer coefficients with the

surrounding are often small in comparison to the one of condensation, but are partially offset by an extension of the effective heat exchange surface area.

In general, the condenser is sized for the maximum heat flux to be transferred. Under these conditions, the two-phase flow length at the condenser is constrained by the total length of the condenser: the LHP operates in fixed-conductance mode. Therefore, the operating temperature and hence the performance of the LHP are mainly governed by the global heat transfer coefficient at the condenser (Launay *et al.*, 2008). The role of the condensers thermal inertia on the LHP dynamic operation, as start-up or oscillating behaviors, is shown very little attention in the literature.

The choice of a condenser configuration is generally guided by the constraints imposed by the LHP integration to the system architecture. If the increase of the heat exchanger external surface has the effect of improving the overall condenser conductance for a given external source and heat flux, it is necessary to ensure that this increase does not come at the expense of system integration. For example, the LHP operation in fixed-conductance mode at high heat flux implies that the reservoir capacity can at least contain the internal volume of the condensation region (tube, annular space) for an optimal use of the condenser. Therefore, reducing the condenser internal volume by adjusting the length or diameter of the tube has a direct effect on the miniaturization of the E/CC.

3.4. Vapor and liquid lines

The sizing of the liquid and vapor lines should meet various requirements:

- (i) The wall thickness depends on the tube material and is sized to accommodate the variations of the working fluid pressure;
- (ii) The tube lengths are generally defined or imposed by the positions of the heat source (evaporator) and the heat sink (condenser);
- (iii) The diameters of the liquid and vapor lines are then the only adjustable parameters. The sizing of the tube diameter in a contrary way acts on the performance of the LHP and on its integration to the system. Large diameters tend to reduce the pressure drop along the LHP, therefore, limit the parasitic heat flux between the evaporator and the reservoir, and thereby improving the LHP performance. Diameters of small dimensions contribute to a better integration of LHP by the flexibility of the tubes and by a miniaturization of the E/CC. Indeed, the reservoir must contain not only the volume of the condenser, but also the volume of the vapor line (Maydanik, 2004).

In the literature, the line length varies between 0.05 m and 5 m, with diameters ranging between 1.2 and 6.35 mm. For a LHP, the vapor line diameter is generally higher or equal to the liquid line one.

4. EXPERIMENTAL RESULTS

As indicated in Table 1, various designs of LHP were developed and tested over the past 10 years. It appears that the objectives of most of the experimental investigations are to characterize the LHP thermal performance, or to study the LHP thermal behaviors at the start-up, or during a test of power cycling. Some of these studies analyze the effect of certain parameters, such as the fluid charge, the working fluid nature, the position of the evaporator from the reservoir and the condenser, and the capillary structure on the LHP behaviors. A review of the parameter influence on the LHP operation has been presented by Launay *et al.* (2007b). In the following section, a synthesis of results on LHP performances at steady-state, and observations related to the LHP dynamic behaviours, as start-up or oscillating behaviours, are presented.

4.1. Instrumentation

The Characterization of the performance and the dynamic behavior of a LHP is determined from some sensors implemented on the system.

The heat flux dissipated by the heating element is obtained from the electrical measurements (current, voltage, resistance), while the one, which is transferred to the cold source, is rarely determined. The extracted heat flux was determined by Celata *et al.* (2010) from the measurements of mass flow rate and the temperature difference of the secondary fluid between the inlet and outlet of the heat exchanger.

The LHP are mostly instrumented by a number of non-intrusive thermocouples up to fifty (or more than 70 for studies on multi-evaporators and multi-condensers). On average, ten temperature measurements are implemented along the LHP. In the literature, there is different representations of the temperature measurements: (i) time evolutions of the temperature at various points of the LHP, (ii) temperature profiles along the LHP for various heat fluxes, (iii) temperature versus heat flux at various points of the LHP. The temperature profile along the condenser allows locating the transition between the condensation region and the region of liquid subcooling. The accuracy of this localization depends on the spacing of the temperature measurements.

In the experimental implementation, there are some measurements of absolute pressure (Ku *et al.*, 2001; Ogushi *et al.*, 2003; Lee *et al.*, 2004; Joung *et al.*, 2010b,a; Singh *et al.*, 2010), and more rarely some measurements of differential pressures (Wrenn *et al.*, 1999; Ku *et al.*, 2001). A few experimental studies in the literature provide direct observations of some parts of the LHP. Cimbala *et al.* (2004) used neutron radiographic images to visualize the liquid and vapor phase distribution along the LHP tubing. Wang and Nikanpour (2007) developed a special design LHP made of transparent windows in the compensation chamber and FEP clear tubes in order to view the local fluid flow and phase changes. d'Entremont and Ochterbeck (2009) used a borescope inserted through the compensation chamber to capture two-phase conditions into the evaporator core. Li *et al.* (2010a) used an IR metrology to deduce the temperature distribution at the E/CC component. Launay and Mekni (2010) have developed a LHP, which is specifically designed to follow the vapor and liquid flows all along the LHP, from the vapor line until the evaporation front position inside the capillary structure, based on visible and IR visualizations.

4.2. Steady-state operation and performance indicators

In steady-state, various LHP operating characteristics have been experimentally observed. They are usually defined by the operating temperature trends. The LHP operating temperature versus the heat load may be U-shaped, flattened, or may increase linearly with the heat flux. Two main LHP operating modes have been noted in the literature, cited as variable and fixed conductance modes. These modes are correlated to the condenser heat transfer distribution (Launay *et al.*, 2008), one part being used for condensation while the other for the liquid flow subcooling. The transition between these two operating modes is dependent on the heat transfer distribution at the E/CC and on the global heat transfer in two-phase flow region in the condenser. According to the experimental results, it is difficult to predict the shape of the LHP operating curve, because heat leak at the the evaporator/reservoir is strongly influenced by the E/CC design (wall materials and wick characteristics). U-shaped curves have been most often observed for metal capillary wicks, with ammonia as the working fluid (Maydanik, 2004; Chen *et al.*, 2006) and when the temperature difference between the sink and the ambient is large, whereas curves with flattened shapes have been usually observed for low-thermal-conductivity wicks with low-pressure working fluids such as methanol, ethanol, or acetone (Boo and Chung, 2004; Riehl and Dutra, 2005).

The LHP performances are usually characterized by the resistance

R_{LHP} defined by the equation:

$$R_{LHP} = \frac{T_E - T_{Sink}}{Q_{load}} \quad (1)$$

where T_E and T_{Sink} are the temperatures of the heat source and the heat sink, respectively, and Q_{load} is the total heat load.

R_{LHP} is the resistance of the overall heat transfer system between the heat source and the heat sink, which includes, in the general case, the heat conduction between the heat source and the evaporator through the saddle, the thermo-hydraulic mechanisms of the LHP, and the heat exchange at the cold source. According to the many possible configurations of the LHP integration as presented in Appendix A from the review of the LHP experimental studies, the LHP performances depend on a number of parameters that makes the comparison between them difficult.

In order to analyze the efficiency of the various thermal links of the LHP integration system, R_{LHP} may be decomposed into two terms:

$$R_{LHP} = \frac{T_E - T_v}{Q_{load}} + \frac{T_C - T_{Sink}}{Q_{load}} = R_E + R_{Sink} \quad (2)$$

where T_v and T_C are the temperatures of the fluid saturation at the evaporator and at the condenser, respectively, and assuming the temperature difference ($T_v - T_C$) small compared to other ones. R_E and R_{Sink} characterize the efficiency of the heat transfer in the evaporator (wall conduction and evaporation in the porous wick, see section 3.2) and of the condenser, respectively.

The inherent efficiency of the thermo-hydraulic mechanisms in the LHP (ie not including the external heat exchange of the heat sink), may be obtained from the measurement of working fluid temperature at the condenser output $T_{C,o}$. The value of $T_{C,o}$ provides information on the mode of the LHP operation. A value of $T_{C,o}$ near T_{sink} means that the LHP is operating in variable conductance mode. The LHP performances are then related to the thermal-hydraulic transfers within the LHP, and particularly at the E/CC. A value of $T_{C,o}$ above T_{sink} indicates that the LHP is operating in fixed conductance mode. Consequently, the LHP performances become dependent on the condenser efficiency.

The efficiency of the LHP can then be characterized by the equation:

$$R_{LHP_2} = \frac{T_E - T_{C,o}}{Q_{load}} = R_E + \frac{T_C - T_{C,o}}{Q_{load}} \quad (3)$$

R_{LHP_2} is equal to R_{LHP} in variable conductance mode, whereas it differs perceptibly in fixed conductance mode. R_{LHP_2} allows to compare the performances of the various design LHP independently from that of the heat sink. Thus, a minimum of five measures for the temperature (T_C , T_{sat} , T_{Sink} , $T_{C,o}$ and T_{Amb}) and one for the heat load Q_{load} are necessary to determine the resistance R_E , R_{LHP} et R_{LHP_2} in order to deduce the performance of the individual thermal links of the heat transfer system separately. It will be interesting to divide the LHP resistances by the evaporator surface area in order that the LHP performance takes into account the LHP miniaturization versus the heat load the LHP has to transfer.

According to the review of the experimental results of the LHP literature (See Tables of Appendix A), the resistance R_{LHP} is most of the time specified, whether from the value given by the authors, or deduced from the measures of the study. The resistance R_E is less frequently indicated, and most of the time, this value has been deduced from the presented measures. The measure of the working fluid at the condenser outlet $T_{C,o}$ is rarely given, except in some studies for which the authors present the temperature profile along the LHP for various heat fluxes. That's explain why the resistance R_{LHP_2} is rarely indicated in the Tables of Appendix A.

It would be rash to make a precise analysis of the LHP performance and attempt to identify the parameters (porous, vapor channel position, bayonet, secondary wick, etc.) that affect the performance with the only

value of R_{LHP} . Indeed, a "good" E/CC combined with a "bad" condenser would tend to the only conclusion that the LHP has lower performance. On the other hand, this resistance R_{LHP} would then be only slightly useful for developing knowledge about the LHP operation.

4.3. Analysis of experimental results in transient state

Start-ups As compared to the capillary pump loop (CPL), which requires pre-conditioning of the loop prior to start-up, the LHP can start directly by applying heat load to the evaporator: LHP is called a self-starting two-phase heat transport device. This specificity is achieved through the presence of a hydraulic link that provides continuous supply of liquid between the reservoir and the evaporator. However, self-start does not necessarily imply a quick and immediate start. The consequences of a delayed start of the LHP with respect to the heat flux input may induce stresses that can damage the operating system to cool. Under certain conditions, the LHP may not even start. It is therefore necessary to understand the various mechanisms that interact during the start-up step and to highlight the parameters that influence its characteristics.

Many studies have been conducted on the experimental characterization of the LHP start-up for various operating conditions. The LHP start-up is for most of the time identified by a rapid increase in the temperatures at the vapor line and, in particular, at the condenser inlet. The sharp temperature increase is the result of a shift of the vapor from the evaporator, where vapor is generated, to the condenser. It takes a certain delay between the heat flux input at the evaporator and the LHP start-up. The delay before the LHP start-up (onset) depends on a number of parameters that have been listed by Mishkinis *et al.* (2004): 1/ Liquid and vapour distribution in evaporator and compensation chamber (working fluid inventory, pre-test histories, orientation, etc.); 2/ LHP element geometry: vapor groove design, wick characteristics, etc.; 3/ Heat input power: start-up is usually more reliable and has small temperature overshoot at high input power; 4/ Heat sink temperature; 5/ Thermophysical and chemical properties of the working fluid: effect on the temperature overshoot; 6/ Ambient conditions and thermal insulation of LHP components; 7/ Non-condensed gas (NCG) presence and amount: NCG can influence boiling process in vapor grooves and promote start-up but NCG can also prevent start-up if it is collected in the wick or/and the evaporator wick core (Baumann *et al.*, 1999); 8/ Evaporator mass: high thermal mass may lead to higher overshoot prior to start-up.

This initial phase of heating before the onset of the LHP has not only an effect on the start-up characteristics, but also on the system behavior after the start-up conditions. By LHP startup characteristics, we note, in particular, 1/ the required minimum heat flux to induce the LHP start-up, and 2/ the overheating of the evaporator wall, compared to the ambient and heat sink temperatures, previously the LHP onset for various operating conditions. The maximum heat flux rate could also become a feature of startup characteristics because it could cause sudden dry-out of the thermo-capillary pump.

As a consequence of these different startup characteristics, various residual effects on the LHP operation in steady state have been reported in the literature (Ku, 1999; Zhang *et al.*, 2004; Huang *et al.*, 2009). For example, Huang *et al.* (2009) observed, following the LHP start-up for different heat flux, 4 modes of LHP behavior in steady-state: 1 / failure mode if the heat flux is too low, 2 / oscillating mode: continuous oscillation of the operating temperature, 3 / overshoot mode: peak of temperature at the onset of the LHP, 4 / and the normal mode: continuous increase in the operating temperature until steady state.

The main explanations of the LHP start-up characteristics are intimately related to the fluid distribution in the LHP, and especially in the E/CC, at the initial state. There are four possible situations of the liquid/vapor states inside the evaporator/compensation chamber prior to start-up, as shown in Fig. 8 (Maidanik *et al.*, 1995). Also shown in the figure are temperature profiles of the compensation chamber T_{CC} and the

evaporator T_E for each situation from the time the power is applied to the evaporator. It needs to be emphasized that the temperature profiles subsequent to start-up are functions of the heat leak from the evaporator to the compensation chamber and the initial liquid line temperature. Based on the experimental studies, three main factors affect the LHP start-up: 1/ the liquid charge in the vapor grooves: if the evaporator grooves are completely liquid filled, a superheat is required to initiate nucleate boiling; if the vapor grooves already contain vapor, liquid will evaporate as soon as power is applied without any superheat requirement. 2/ the liquid charge in the evaporator core: if the evaporator core is liquid filled, the heat leak from the evaporator to the compensation chamber is minimal since heat is transmitted by conduction through the evaporator container. However, if vapor is present in the evaporator core, heat is transmitted through the primary wick and reaches the vapor immediately. The evaporator core becomes a vapor extension of the compensation chamber. Heat leak therefore increases substantially. 3/ the initial power load: the evaporator heat flux also affects the start-up through interactions with the other two factors.

Fig. 8 Schematic of the main physical mechanisms in a LHP

The LHP requires a pressure difference across the wick in order to start. This pressure drop is related to the temperature differential across the wick in accordance with the Clausius-Clapeyron relation (Baumann *et al.*, 1999). With no flow in the system and no heat load on the evaporator, no pressure or temperature gradient will exist across the evaporator wick. As heat is applied to the evaporator wall, a temperature gradient will develop across the wick, thus allowing the system to start pumping. For very low heat loads into the evaporator, the conductive paths to the compensation chamber and through the wick become dominant and the temperature gradient required to develop a sufficient pressure gradient across the wick is elusive. One can conclude that there exists a minimum heat flux for which the LHP may start. Moreover, according to the experimental results of the literature, this demonstration must partly explain the ability of the cylindrical evaporators to start at a lower heat flux rate compared to the flat ones.

The role of the fluid distribution in the E/CC on the LHP startup has been demonstrated from some experimental studies by varying the relative position of the LHP components (the evaporator position relative to the reservoir and the condenser ones) (Boo and Chung, 2004; Chen *et al.*, 2006; d'Entremont and Ochterbeck, 2009; Bai *et al.*, 2009; Celata *et al.*, 2010; Joung *et al.*, 2010b; Lin *et al.*, 2010). In the main cases, the initial distribution of the fluid charge in the E/CC is estimated from the compo-

nent position. However, the assumption on the fluid charge distribution at the initial state and its role in the start-up seems supported by various visualization methods, which were implemented by different researchers. A particular design LHP, which is composed of two reservoirs with some clear windows, was used by Lin *et al.* (2010), a borescope was inserted into the evaporator core by d'Entremont and Ochterbeck (2009), and a visualization by neutron radiography was used by Cimbalá *et al.* (2004). d'Entremont and Ochterbeck (2009) observed no significant role in the fluid distribution on the LHP start-up for medium to high heat loads. The effect of the fluid distribution on the start-up has also been studied by Ku *et al.* (2001) by combining different component positions and fluid filling ratios. The tests were at the advantage of the highest filling ratio and for the favorable inclinations, for which the evaporator core has a higher probability of being filled. We must keep in mind that some start-up characteristics result from the coupling effects between the fluid distribution and the hydrostatic pressure, which may be induced by the change in position between the LHP components (Joung *et al.*, 2010a).

The pre-history and pre-test conditions are other parameters involved in the fluid distribution at the initial state, and therefore influencing the LHP start-up (Kaya and Ku, 2000; Mishkinis *et al.*, 2004). To this, we must add to the randomness of the superheat temperature for the onset of LHP (Fig. 9, Mishkinis *et al.* (2004)). Various techniques of LHP pre-conditioning were implemented to help them start-up, such as reservoir heating or cooling at a given time (Ref).

According to all these observations, start-up represents perhaps the most complex transient phenomenon in the LHP operation, and the reliable start-up of a LHP is a primary issue of the thermal control.

Fig. 9 Patterns of LHP operation during periodical LHP heating (Mishkinis *et al.*, 2004)

Oscillating behaviours Despite the design simplicity and the theoretical operational robustness, various LHP behaviors have been experimentally observed consequently to transient changes, such as start-up or variations in power load and/or sink temperature (Kaya and Ku, 2000). Sometimes, the LHP never really reaches a steady-state but instead displays an oscillating behaviors. These oscillating behaviors have been observed for various LHP designs of the literature (see table N) (Rodriguez and Na-Nakornpanom, 2001; Ku, 2003; Ku and Rodriguez, 2003; Mishkinis *et al.*, 2004; Chen *et al.*, 2006; Singh *et al.*, 2008; Celata *et al.*, 2010; Joung *et al.*, 2010b; Li *et al.*, 2010a). Three types of temperature oscillation regimes have been reported in the literature by Ku (2003) and Ku and Rodriguez (2003): i) ultra-high frequency (period of about one second or less) temperature oscillations, ii) high frequency (period of few seconds or minutes) low amplitude oscillations caused by the inability of the vapor front to find a stable position inside the condenser, iii) low frequency

(period of about hours) high amplitude oscillations, which may appear for specific conditions as, for example, large evaporator thermal inertia, low heat load and cold sink temperature compared to the ambient one (Rodriguez and Na-Nakornpanom, 2001; Ku, 2003). As the type i) is usually assumed to be caused by the formation of liquid slugs in the condenser or the vapor line, types ii) and iii) are typically induced by the original complex couplings of the thermal and hydrodynamic mechanisms involved by the interactions and feedback between the loop components.

Chen *et al.* (2006) studied the influence of the cold source temperature and the relative position of the LHP components (the evaporator position relative to the condenser and the reservoir one's) on the characteristics of the high frequency oscillations (type (ii)). Figure 10 shows, for various LHP configurations, the heat flux ranges for which a temperature oscillating behavior is observed (black surface areas on the histograms). The oscillation frequency varies between 0.01 and 0.05 Hz and it increases with the heat flux, and the temperature amplitude ranges between 1 and 10 K at the liquid line inlet, considering the flow direction. Figure 11 shows the temperature oscillation amplitude in various parts of the LHP and for different heat flux when the reservoir is above the evaporator and with a heat sink temperature of 25 °C. The maximum amplitude is about 6 K at the liquid line inlet (L-line1), and is about 1 K at the heat source (Evap).

Fig. 10 Heat load range under which the oscillations occur (Chen *et al.*, 2006)

Rodriguez and Na-Nakornpanom (2001) studied the influence of the temperature of the heat sink on the characteristics of the low frequency oscillations (type (iii)). The characteristics of the oscillations are shown in Figs. 12 and Fig. 13 for the amplitude and frequency, respectively. The amplitude varies between 4 and 34 K and it decreases as the imposed heat flux increases. The frequency varies between $5 \cdot 10^{-5}$ and $4 \cdot 10^{-4}$ Hz and it increases with the heat flux. In general, lower heat load and sink temperature are conditions that produce oscillations of higher amplitudes with increasing periods. According to the authors, this phenomenon is not affected by initial start-up conditions and it is very stable and repeatable.

5. CONCLUSION

In the present work, it has been made an inventory of more than 50 conventional LHP from experimental studies of the literature in the years 1998-2010. Although the evaporator/compensation chamber (E/CC) of cylindrical design were initially preferred by the researchers, a number of new E/CC configurations have been proposed during the years 2009-2010. A classification of E/CC designs in three main class are proposed: (i) cylindrical; (ii) flat with the reservoir located in the extension of the

Fig. 11 Amplitude of the temperature oscillations at various positions along the loop for different heat loads (Chen *et al.*, 2006)

Fig. 12 Transient temperature amplitudes with low frequency oscillations (Rodriguez and Na-Nakornpanom, 2001)

evaporator; (iii) flat with the reservoir located in the thickness of the evaporator. The diversity of these various geometries of E/CC is a response to expanding applications of the LHP with the issue of miniaturization for a better implementation within a 3D architecture of the electronic modules, and for the issue of higher dissipated heat flux rates. The detailed description of the LHP components is presented in a certain order that might look like a LHP design procedure for a given application.

From the review of the LHP experimental studies, it has been noticed that some useful LHP characteristics are sometimes missing, as the thermal-hydraulic link between the evaporator and the reservoir, the porous wick characteristics, or the vapor channel configuration. Questions arise concerning the definition or characterization of the pore diameter of the porous wicks. Some studies only focused on the characterization of the global LHP thermal resistance R_{LHP} . This resistance concerns the overall heat transfer system between the heat source and the heat sink, which includes, in the general case, the heat conduction between the heat source and the evaporator through the saddle, the thermal-hydraulic mechanisms of the LHP, and the heat exchange at the heat sink. In the present paper, other criteria of LHP performance are proposed in order

Fig. 13 Transient temperature period for low frequency oscillations (Rodriguez and Na-Nakornpanom, 2001)

better define the effect of each thermal link characterizing the LHP system.

Particular attention in LHP experimental studies to inform the different elements mentioned in this review, while maintaining protection of know-how, should promote the development of tools for understanding and predicting the LHP behaviors.

ACKNOWLEDGEMENTS

The authors would like to thank the French Research Group (GDR SYREDI) working on two-phase cooling systems for the fruitful discussions during meetings.

NOMENCLATURE

k	thermal conductivity ($\text{W/m} \cdot \text{K}$)
K	permeability (m^2)
P	pressure (Pa)
P_c	critical pressure (Pa)
q''	heat flux (W/m^2)
R	thermal resistance (K/W)
t	time (s)
T	temperature (K)

Greek Symbols

ϕ	diameter (m)
ϵ	porosity
ρ	density (kg/m^3)

Subscripts

0	initial condition
Amb	ambient
C	condenser
CC	compensation chamber
C,o	condenser outlet
E	evaporator
eff	effective
h	hydraulic
load	load
max	maximal
p	pore
R	reservoir

sat	sat
Sink	Sink
v	vapor
w	wick
wall	wall

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APPENDIX A: SYNTHESIS OF CHARACTERISTICS AND PERFORMANCES OF VARIOUS LOOP HEAT PIPES

APPENDIX A

Nom auteurs	Baker et al. (1998)	Cheung et al. (1998)	Wrenn et al. (1999)	Pauken and Rodriguez (2000)	Pauken and Rodriguez (2000)
Loop Heat Pipes					
Specificities (shape, components, applications ...)	space experiment	LHP for space application	macro LHP	macro LHP	Macro-LHP
Working fluid (mass charge)	ammonia	ammonia	ammonia	ammonia (96.4 g)	propylene (80 g)
Shape/design	cylindrical	cylindrical (L _r 427 mm)	cylindrical (L _r 444)	cylindrical	cylindrical
Evap body material	stainless steel + Al saddle	stainless steel	aluminium + saddle	aluminium + Al saddle	aluminium + Al saddle
Evap dimensions (mm)	L 300 / D 25	L 300 / OD 25	L 330	L 152 / D 24	L 152 / D 24
Heated surface area (cm ²)					
CC body material		stainless steel	inconel	stainless steel	stainless steel
CC dimensions (mm)		L 127 / OD 25	L 114 / OD 79		
Volume (cm ³)				115	115
Evap-CC hydraulic link	- / no 2nd wick	bayonet / 2nd wick	bayonet / -	- / no 2nd wick	- / no 2nd wick
Wick properties					
Materials	sintered Nickel	sintered Nickel	sintered nickel	sintered nickel	sintered nickel
Dimensions			L 330 / OD 19.05 / ID 9.91		
Pore characteristic length (um)	D 4.4	D 2.2	L 1.5	L 1.2	L 1.2
Porosity (%) - Permeability (m ²)	- / -	60 / -	65 / 0.75 10 ⁻¹³	60 / 4x10-14	60 / 4 10 ⁻¹⁴
Vapour groove location		-			
Lines					
Line material	stainless steel	stainless steel	stainless steel	stainless steel	stainless steel
Liquid line : L / OD / ID (mm)	4500 / - / 4.5	1524 / 4.76	5080 / - / 4.57	1040 / 4.5 / 2.5	1040 / 4.5 / 2.5
Vapour line : L / OD / ID (mm)	4500 / - / 4.5	1524 / 4.76	4267 / - / 4.57	1040 / 4.5 / 2.5	1040 / 4.5 / 2.5
Condensation and heat exchanger type	in tube mounted in a radiator plate	in tube in a pipe-in-pipe liquid heat exchanger	in tube mounted on a plate cooled by liquid heat exchanger	in tube mounted on a Al plate cooled by liquid convection	in tube mounted on a Al plate cooled by liquid convection
Materials		stainless steel	aluminium	aluminium	aluminium
Dimensions L / OD / ID (mm)	3700 / 4	2000 / 4.6 / -	6360 / - / 3.81	3800 / 5.5 / 4.1	3800 / 5.5 / 4.1
Condenser					
LHP operation conditions	steady-state / transient	steady-state / transient	transient	steady-state	steady-state/transient
LHP Instrumentation		45 thermocouples	50 thermocouples differential pressure sensor		
Power range (W) / qmax (W/cm ²)	25-200	2-700 / -	50-800 / -	20-800 / -	20-200 / -
T(Qmax) / Tsink (°C)	49 / -34 to 28	47 / -10	36 / -20 to 10	50 / -10 to 20	30 / -10 to 20
R _E / R _{LHP} / R _{LHP2} (K/W)	- / 0.1 / -	- / 0.08 / -	0.009 / 0.032 / -	- / 0.0058 / -	- / 0.022 / -
Specific effect studies	- power cycle tests - low power start-ups	- Power cycle tests, start-ups - Active control mode - Observations of temperature hysteresis phenomena and instabilities	Study of the LHP behavior during power and sink transient test	Effect of working fluid, and sink temperature on LHP thermal performance	- effect of working fluid, LHP elevation and sink temperature on LHP thermal performance - evaporator response at start-up in vertical position for various heat load

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_r: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Ku et al. (2001)	Rodriguez et al. (2001)	Delil et al. (2002)	Chuang (2003)	Delil et al. (2003)
Loop Heat Pipes					
Specificities (shape, components, applications ...)	Low power operation LHP	macro LHP	miniature-flat LHP	macro LHP	miniature LHP
Working fluid (mass charge)	ammonia (83 to 113 g)	propylene 99% (80 g)	ethanol	ammonia (147 g)	ammonia (1.9 g)
Shape/design					
Evap body material	cylindrical (L _r 43.2)	cylindrical (L _r 230)	flat disk with embedded CC in the thickness	aluminum + saddle	cylindrical
Evap dimensions (mm)	stainless steel + Al saddle L 30.5 / OD 25.4	aluminum + Al saddle L 152.4 / ID 24.2	copper OD 44 / H 22 / t 4.3	L 610 / OD 24.1	stainless steel + Al saddle L 38 / OD 8
Heated surface area (cm ²)		116	12		16
CC body material	stainless steel	stainless steel	stainless steel	stainless steel	stainless steel
CC dimensions (mm)	L 12.7 / OD 25.4	L=80,25 OD=43,94		L 160 / OD 69.3 / ID 68.3	
Volume (cm ³)		115		586	
Evap-CC hydraulic link	- / -	- / -	no bayonet / no 2nd wick	Al bayonet / no 2nd wick	- / no 2nd wick
Materials					
Dimensions	sintered nickel	sintered nickel	Ni-Cr (X20H80) / Nickel / Titane	Nickel	Titanium
Pore characteristic length (um)	D 4	L 1.2	D 39 / H1 3.7 - H2 7.9 - H3 7.3	L 610 / OD 19.1 / ID 9.5	
Porosity (%) - Permeability (m ²)	- / 1.4 10 ⁻¹⁴	60 / 4 10 ⁻¹⁴	D 12 / 2.4 / 10.4	D 3.2	- / -
Vapour groove location			36/66/57 - 2.4 10 ⁻¹³ / 2 10 ⁻¹⁴ / 2 10 ⁻¹³	60 / 0.2 10 ⁻¹³	
Wick properties					
Line material		stainless steel (304)	evaporator wall	evaporator wall	
Liquid line : L / OD / ID (mm)	2100 / - / 1.75	1074 / - / 4.52		alu/stainless steel	stainless steel
Vapour line : L / OD / ID (mm)	1940 / - / 3.34	1000 / - / 4.52		970 / 6.4 / 5.3	860 / 1.2 / -
Condensation and heat exchanger type					
Condenser	in tube mounted on a cooled Al plate	in tube mounted on a plate cooled by liquid convection	in tube in a liquid heat exchanger	in tube mounted on plate cooled by liquid convection	in tube mounted on a Al cooled plate
Materials		aluminium	copper		stainless steel
Dimensions L / OD / ID (mm)	1990 / - / 3.86	3810 / - / 4		3660-4650 / 6.4 / 4.6	770 / - / 1.2
LHP operation conditions					
LHP Instrumentation	transient 39 thermocouples 1 absolute pressure gauge 1 differential pressure transducer	steady-state/transient 18 thermocouples	steady-state 12 thermocouples	steady-state / transient 18 K-type thermocouples 1 wattmeter	steady-state
Power range (W) / q _{max} (W/cm ²)	5-200 / -	15-200 / 1.7	10-100 / -	10-700 / 4.8	2-20 / -
T(Q _{max}) / T _{sink} (°C)	- / -43 to 20	30 / -30 to 0		25 / 5	- / -20 to 20
R _E / R _{LHP} / R _{LHP2} (K/W)		- / 0.05-0.065 / -	- / 0.62 / -	- / 0.03 / -	- / 1.11 / - (at T _{sink} =20°C)
Specific effect studies	- effect of fluid inventory, relative tilt between Evap and CC on LHP start-up, operating temperature and capillary limit - effect of additional pressure drops (using a valve) on the LHP behavior	- effect of power load and sink temperature on the temperature oscillation characteristics	Effect of the wick characteristics and the LHP elevation on the evaporation heat transfer and LHP performance.	- Study of low power start-ups - Effect of LHP elevation on LHP performance - Observation of hysteresis phenomena	Effect of the LHP elevation, the sink temperature, the evaporator design on the LHP performance

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_r: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Delil et al. (2003)	Kaya and Ku (2003)	Ogushi et al. (2003)	Pastuhkhov et al. (2003)	Pastuhkhov et al. (2003)
Loop Heat Pipes					
Specificities (shape, components, applications ...)	miniature flat LHP	miniature LHP	Figure 22 - Annexe B		
Working fluid (mass charge)	ammonia (3 g)	ammonia	Polymer wick LHP under paraboloid flight conditions	miniature LHP	miniature LHP
Shape/design	flat disk with embedded CC in the thickness	aluminium + copper saddle	ammonia	ammonia	ammonia
Evap body material	stainless steel	L 120 / OD 13 / ID 10	cylindrical (L _r 250)	cylindrical (L _r 80)	cylindrical
Evap dimensions (mm)	OD 28	6	stainless steel	stainless steel + Al saddle	stainless steel + Al saddle
Heated surface area (cm ²)	6	aluminium	L 250 / ID 25.4	L 40 / OD 6 / ID 5.5	L 10 / OD 5 / ID 4.4
CC body material	stainless steel	L 30 / OD 13 / ID 10	stainless steel	12	1.1
CC dimensions (mm)	- / no 2nd wick	2.3	- / Polyethylene 2nd wick	stainless steel	stainless steel
Volume (cm ³)	- / no 2nd wick	- / no 2nd wick	- / no 2nd wick	- / no 2nd wick	- / no 2nd wick
Evap-CC hydraulic link	- / no 2nd wick	- / no 2nd wick	- / no 2nd wick	- / no 2nd wick	- / no 2nd wick
Wick properties					
Materials	Nickel-Titanium	sintered Nickel	PTFE	Titanium	Stainless Steel
Dimensions	L 120 / OD 10 / ID 4	D 2.4	D 1.2	- / -	- / -
Pore characteristic length (um)	- / -	60	- / -	- / -	- / -
Porosity (%) - Permeability (m ²)	- / -	- / -	porous wick	- / -	- / -
Vapour groove location	- / -	- / -	- / -	- / -	- / -
Wick properties					
Line material	stainless steel	aluminium	stainless steel	stainless steel	stainless steel
Liquid line : L / OD / ID (mm)	860 / 1.2 / -	130 / 3.8 / 2.5	730 / - / 2	340 / 1.8 / 1.3	24 / 2 / 1.2
Vapour line : L / OD / ID (mm)	860 / 1.2 / -	160 / 3.8 / 2.5	1770 / - / 2	210 / 1.8 / 1.3	52 / 2 / 1.2
Condensation and heat exchanger type	in tube mounted on a Al cooled plate	in tube mounted on a cooled Cu plate	in tube coupled with a water tube heat exchanger	in tube in a finned air heat exchanger	in tube in a finned air heat exchanger
Materials	stainless steel	aluminium	stainless steel	stainless steel	stainless steel
Dimensions L / OD / ID (mm)	770 / - / 1.2	100 / 3.8 / 2.5	1400 / - / 2	600 / - / 1.3	120 / - / 1.3
Condenser					
LHP operation conditions	steady-state	steady-state/transient	steady-state/transient	steady-state	steady-state
LHP Instrumentation	-	50 thermocouples	20 K-type thermocouples, clear window on CC, 2 pressure sensors	-	-
Experimental study					
Power range (W) / q _{max} (W/cm ²)	2-110 / -	5-250 / -	160-386 / -	2-40 / 3.3	2-30 / 27.2
T(Q _{max}) / T _{sink} (°C)	- / -20 to 20	47 / -40 to 0	120 / 20	100 / 24	85 / 22
R _E / R _{LHP} / R _{LHP2} (K/W)	- / 0.13 / -	- / 0.35 / -	0.16 / 0.26 / -	- / 1.7-2.4 / -	- / 1.9-4 / -
Specific effect studies	Effect of the LHP elevation, the sink temperature, the evaporator design on the LHP performance	- effect of LHP elevation and orientation on thermal performance - effect of spin acceleration on LHP start-up (effect on superheat prior to boiling)	- visualization of fluid distribution in the Evap. core - no significant effect of gravitational acceleration on LHP behavior - effect of singular pressure drop (valve) on LHP performance	Effect of LHP elevation, Evap design (cylindrical, rectangular) and condenser design (cold plate, fins) on the LHP thermal performance	Effect of LHP elevation, Evap design (cylindrical, rectangular) and condenser design (cold plate, fins) on the LHP thermal performance

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_r: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Boo et al. (2004)	Lee et al. (2004)	Mishkinis et al. (2004)	Rhiel et al. (2004-2005)	Zhang et al. (2004)
Loop Heat Pipes					
Specificities (shape, components, applications ...)	Figure 15 - Annexe B				
Working fluid (mass charge)	miniature LHP + polymer wick	miniature LHP	macro LHP	mLHP	macro LHP
Shape/design	water/methanol/acetone/ethanol	water (variable charge)	ammonia	acetone	ammonia (29.9 g)
Evap body material	rectangular with embedded CC in the thickness	flat rectangular with embedded CC in the thickness	cylindrical	cylindrical	cylindrical
Evap dimensions (mm)	stainless steel	copper	SST and Al + Cu saddle	stainless steel + Al saddle	stainless steel
Heated surface area (cm ²)	L 50 / W 40 / H 30	L 58 / W 58 / H 28 / t 3	L 200	L 67 / OD 19 / ID 16.5	L 175 / OD 18 / ID 16
CC body material	12.25	stainless steel / copper	stainless steel and aluminium	17.5	17.5
CC dimensions (mm)	stainless steel	stainless steel / copper	stainless steel	stainless steel	stainless steel
Volume (cm ³)	bayonet / no 2nd wick	21	- / Nickel 2nd wick	20	20
Evap-CC hydraulic link	bayonet / no 2nd wick	- / no 2nd wick	- / Nickel 2nd wick	bayonet / 2nd wick	bayonet / no 2nd wick
Wick properties					
Materials	Polypropylene	sintered brass / sintered SST	sintered nickel	polyethylene (UHMW)	nickel
Dimensions	- / - / H 5	L 52 / W 52 / H 10	L 52 / W 52 / H 10	L 67 / OD 16.5 / ID 7	L 125 / OD 16 / ID 8
Pore characteristic length (um)	L 0.5-25	D 9.6 - 19.1	D 1.8	D 12	D 2.2
Porosity (%) - Permeability (m ²)	40-50 / -	40-53 / -	- / -	50 / 10 ⁻¹³	- / 5 10 ⁻¹⁴
Vapour groove location	evaporator wall	evaporator wall	evaporator wall	porous wick	porous wick
Lines					
Line material	stainless steel	stainless steel	stainless steel	stainless steel	stainless steel
Liquid line : L / OD / ID (mm)	500 / 2	500 / 3.2 / -	500 / 3 / 2	850 / 2.85 / -	2500 / 3 / 2.2
Vapour line : L / OD / ID (mm)	500 / 4	500 / 6.35 / -	500 / 3 / 2	550 / 2.85 / -	2800 / 3 / 2.2
Condenser					
Condensation and heat exchanger type	in 10 parallel tubes mounted on plate cooled by liquid convection	in tube in a pipe-in-pipe liquid heat exchanger	in tube mounted on a plate cooled by liquid convection	in tube mounted on a plate cooled by liquid heat exchanger	in tube mounted on Al plate cooled by liquid heat exchanger
Materials	plate cooled by liquid convection	heat exchanger	stainless steel	stainless steel	stainless steel
Dimensions L / OD / ID (mm)	50 / -	50 / -	250 / 3 / 2	1000 / 4.85 / 2.85	2000 / 3 / 2.2
Experimental study					
LHP operation conditions	steady-state / transient	steady-state / transient	steady-state/transient	steady-state/transient	steady-state / transient
LHP Instrumentation	21 T-type thermocouples	19 thermocouples 1 pressure sensor	16 T-type thermocouples thermal vacuum chamber	20 T-type thermocouples	20 T-type thermocouples
Power range (W) / qmax (W/cm ²)	10-80 / 6.5	40-160 / 6	5-400 / -	2-80 / 4.6	5-300 / -
T(Qmax) / Tsink (°C)	90 / 20	100 / 20	62 / 13	80 / 5	7.5 / -20
R_E / R_LHP / R_LHP2 (K/W)	0.3 / 0.85 / - (methanol)	0.16 / 0.5 / -	- / 0.12 / -	- / 0.8-0.93 / -	- / 0.09 / -
Specific effect studies	LHP behaviors depending on: - working fluid and charge - LHP elevation (gravity effect) - wick pore size - sink temperature	- effect of the fluid charge on LHP behavior - effect of the fluid charge and the wick characteristics on LHP thermal performance	- steady-state and transient LHP performance during periodic heating cycles - temperature overshoot at start-up depending on pre-conditioning situations	- start-ups, power test cycle - effect of the Evap/CC design on the LHP thermal performance - observations: temperature oscillations after start-up, reverse flow, mainly at low heat load	- effect of start-up on the LHP operating temperature profile - observations: temperature oscillations after start-up, reverse flow, mainly at low heat load

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_T: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Hsu et al. (2005)	Chen et al. (2006)	Nagano and Ku (2007)	Pastukhov et al. (2007)	Pastukhov et al. (2007)
Loop Heat Pipes		Figure 18 - Annexe B			
Specificities (shape, components, applications ...)	Silicon/glass micro LHP	miniature LHP	multi-Evap and Cond LHP		
Working fluid (mass charge)	water/methanol	ammonia	anhydrous ammonia (29 g)	water	water
Shape/design	flat rectangular with embedded CC along the length	cylindrical (L _r 66)	cylindrical (L _r 125)	flat rectangular with embedded CC along the length	flat oval with embedded CC along the length
Evap body material	silicon/7740 glass	stainless steel / copper saddle	aluminium	copper	copper
Evap dimensions (mm)	L 10 / W 8.5 / H 0.8 (x2)	L 29 / OD 5 / ID 4.6	L 52 / OD 9.65	L 32 / W 25 / H 7	L 32 / W 42 / H 7
Heated surface area (cm ²)	1.7	4.7		10	10
CC body material	silicon/glass	stainless steel	stainless steel	copper	copper
CC dimensions (mm)	L 20	L 37 / OD 5 / ID 4.6	L 72.4 / OD 22.2	L 29 / W 25 / H 7	L 32 / W 42 / H 7
Volume (cm ³)		0.6			
Evap-CC hydraulic link	- / no 2nd wick	bayonet / no 2nd wick	bayonet / -	- / no 2nd wick	- / no 2nd wick
Materials	silicon	Titanium	Titanium		
Dimensions		L 29 / OD 4.6 / ID 2	OD 6.35 / ID 3.18		
Pore characteristic length (um)	V-grooves Dh=47/67/83 um	D 34.4	L 1.5		
Porosity (%) - Permeability (m ²)	- / -	78 / 11.5 10 ⁻¹²	- / 0.2 10 ⁻¹³	- / -	- / -
Vapour groove location		porous wick			
Line material	silicon/glass	stainless steel	stainless steel	copper	copper
Liquid line : L / OD / ID (mm)	71.8 / 1	240 / 2 / 1.5	- / 1.59 / -	- / 3 / 2.	- / 3 / 2.
Vapour line : L / OD / ID (mm)	40 / 1	188 / 2 / 1.5	- / 2.38 / -	- / 4 / 3.	- / 3 / 2.
Condensation and heat exchanger type	in square cross-section channel cooled by liquid heat exchanger	in annular cross-section in a finned liquid heat exchanger	in tube in a pipe-in-pipe liquid heat exchanger	in parallel tubes mounted on a finned air heat exchanger	in tube mounted on a cooled finned plate
Materials	silicon/glass	stainless steel	aluminium	copper	aluminium
Dimensions L / OD / ID (mm)	31.8 / - / 1	60 / 4.6 / 4	2540 / 2.38 / -		- / 3-4 / -
LHP operation conditions	steady-state / transient	steady-state / transient	steady-state/transient	steady-state	steady-state
LHP Instrumentation	9 thermocouples	7 K-type thermocouples, 3 PT100	72 thermocouples, flow regulator thermoelectric converter	T-type thermocouples	T-type thermocouples
Power range (W) / qmax (W/cm ²)	2-16 / 8.2	5-70 / 14.8	0-2x80 and 0-110 / -	5-140 / 14	10-150 / 15
T(Qmax) / Tsink (°C)	85 / 17	80 / 5 to 50	32 / 0	85 / 22	85 / 22
R _E / R _{LHP} / R _{LHP2} (K/W)	- / 5 / - (with water at 14 W)	- / 0.2 / -	- / 0.2 / -	- / 0.45 / -	- / 0.43 / -
Specific effect studies	Effect of the working fluids, the presence of NCGs, and groove hydraulic diameter on the LHP performance and start-up (best performance for small capillary size)	- Study of LHP oscillation characteristics depending on LHP orientation and Tsink - Start-ups	- effect of the LHP elevation on the capillary limit - study of LHP behavior under power test cycle on each evaporator - active control of the CC set point temperature by TEC	- effect of Evap design (flat, cylindrical) and condenser design (thermosyphon, cold or finned plate on LHP thermal performance - proposition of LHP integration in computer	- effect of Evap design (flat, cylindrical) and condenser design (thermosyphon, cold or finned plate on LHP thermal performance - proposition of LHP integration in computer

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_r: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Rhiei (2007)	Vershinin and Maydanik (2007a)	Vershinin and Maydanik (2007a)	Vershinin and Maydanik (2007b)
Loop Heat Pipes				
Specificities (shape, components, applications ...)	LHP using nanofluid	Flat disk LHP	LHP	miniature LHP
Working fluid (mass charge)	water + (40 nm-NiO particles)	ammonia	pentane/acetone	ammonia
Shape/design				
	flat disk with embedded CC in the thickness	flat disk with embedded CC in the thickness	cylindrical	cylindrical (L _r 66)
Evap body material	stainless steel (316)			Nickel + saddle
Evap dimensions (mm)	- / - / H 8	D 30 / H 13	L 80 / D 8	L 20 / D 6
Heated surface area (cm ²)	3			4
CC body material	stainless steel (316)			
CC dimensions (mm)				L 46 / D 6
Volume (cm ³)	6			
Evap-CC hydraulic link	- / -	- / -	- / -	
Materials				
	sintered nickel			Nickel
Dimensions				
Pore characteristic length (um)	L 3			D 20
Porosity (%) - Permeability (m ²)	75 / -	- / -	- / -	65 / -
Vapour groove location	evaporator wall			
Lines				
Line material	stainless steel (316)			stainless steel
Liquid line : L / OD / ID (mm)	300 / 3 / 1.5	410 / 2	750 / 4	190 / 2
Vapour line : L / OD / ID (mm)	200 / 3 / 1.5	390 / 2	500 / 5	180 / 2.5
Condensation and heat exchanger type				
	in tube mounted on a plate cooled by liquid convection	in tube mounted on a cold plate	in tube in a pipe-in-pipe liquid heat exchanger	in annular cross-section in a finned air heat exchanger
Materials				
	stainless steel			stainless steel
Dimensions L / OD / ID (mm)				
	500 / 3 / 1.5			59 / 5 / -
LHP operation conditions				
	steady-state/transient	steady-state	steady-state	transient
LHP Instrumentation				
	10 T-type thermocouples	4 thermocouples (minimum)	4 thermocouples (minimum)	7 thermocouples (minimum)
Experimental study				
Power range (W) / q _{max} (W/cm ²)	10-40 / 13.3	0-180 / -	25-350 (pent.) 75-500 (acet.)	5-110 / 27.5
T(Q _{max}) / T _{sink} (°C)	100 / 5 and 10	62 / 20	85 / 25	95 / 24
R _E / R _{LHP} / R _{LHP2} (K/W)	- / 2.25 / -	- / 0.23 / -	- / 0.17 (pent.) - 0.12 (acet.) / -	0.14 / 0.65 / -
Specific effect studies	- LHP thermal performance and Evap heat transfer coefficient using nanofluid as working fluid - preliminary results: reduced thermal performance for nanofluid as compared to pure water	- study of hysteresis phenomena for various Evap and condenser designs and various working fluids	- study of hysteresis phenomena for various Evap and condenser designs and various working fluids	Effect of amount of working fluid, LHP orientation, condenser cooling conditions on operating temperature oscillations

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_T: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	d'Entremont et al. (2008)	Joung et al. (2008a)	Singh et al. (2008)	Tanaka (2008)	Vasiliev et al. (2008)
Loop Heat Pipes					
Specificities (shape, components, applications ...)	Liquid core visualization	Flat bifacial evaporator LHP	miniature LHP	copper-LHP	High power LHP
Working fluid (mass charge)	ammonia	methanol (19 to 60 % of Vtot)	water	methanol	water or acetone
Shape/design					
Shape/design	cylindrical (L _r 250)	flat rectangular with embedded CC along the length	flat disk with embedded CC in the thickness	cylindrical	cylindrical (L _r 334)
Evap body material	stainless steel	copper	copper	copper	stainless steel
Evap dimensions (mm)	L 180 / OD 24	L 50 / W 54 / H 6 / t 1	D 30 / H 10	L 95 / OD 19 / ID 12.7	L 280 / OD 18 / ID 16
Heated surface area (cm ²)		25 x 2	3.25		
CC body material	stainless steel	copper	copper		stainless steel
CC dimensions (mm)		L 26 / W 54 / H 6 / t 1	D 30		L 45 / D 30
Volume (cm ³)	39	4			
Evap-CC hydraulic link	bayonet / no 2nd wick	SSTI bayonet / no 2nd wick	no bayonet / no 2nd wick	- / no 2nd wick	- / no 2nd wick
Materials					
Materials	Iron oxyde	stainless steel	sintered nickel	sintered copper	sintered Nickel or Titanium
Dimensions	L 180 / OD 22 / ID 12	L 50 / W 50 / H 1.5 (x2)	D 30 / H 3	L 60 / OD 12.7 / ID 6	L 280 / OD 16
Pore characteristic length (um)	D 1.6	L 2	D 6-10	D 100	D 5
Porosity (%) - Permeability (m ²)	33 / -	- / -	75 / -	- / -	55 / -
Vapour groove location		evaporator wall	evaporator wall	porous wick	porous wick
Wick properties					
Line material	stainless steel	copper	copper	copper	stainless steel
Liquid line : L / OD / ID (mm)	1200 / - / 1.75	165 / 3.18 / 1.76	290 / - / 2	250 / 3.5 / 2.5	1500 / 4 / 3
Vapour line : L / OD / ID (mm)	2000 / - / 4.57	160 / 3.18 / 1.76	150 / - / 2	250 / 5 / 4	1500 / 6 / 5
Condenser					
Condensation and heat exchanger type	in tube in a pipe-in-pipe liquid heat exchanger	in tube in a pipe-in-pipe liquid heat exchanger	condensation in tube in a finned forced air heat exchanger	in tube mounted on a plate cooled by air convection	in tube in a pipe-in-pipe liquid heat exchanger
Materials	stainless steel	copper	copper	copper	
Dimensions L / OD / ID (mm)	1000 / - / 4.57	100 / 3.18 / 1.76	50 / - / 2	600 / 3.5 / 2.5	560 / 5
LHP operation conditions					
LHP Instrumentation	steady-state / transient 6 thermocouples, endoscope, absolute pressure transducer	steady-state/transient 21 T-types thermocouples	steady-state/transient 14 thermocouples	steady-state 16 thermocouples 1 pressure transducer	steady-state 8 T-type thermocouples
Experimental study					
Power range (W) / q _{max} (W/cm ²)	25-400 / -	10-78 / 1.56	5-70 / 11.2	10-50 / -	75-900 / -
T(Q _{max}) / T _{sink} (°C)	41 / 9.3	130 / 25	100 / 22	100 / 22	96 / 20 (water) - 97/ 35 (acet.)
R _E / R _{LHP} / R _{LHP2} (K/W)	- / 0.075 / -	0.13 / 1.27 / -	0.15 / 1.1 / -	0.2-0.6 / 1.53 / -	- / 0.08 (water) - 0.3 (acet.) / -
Specific effect studies	Visualization of the fluid distribution in the evaporator core during transient tests as start-ups or heat load steps - effect of the LHP elevation and orientation	- effect of the fluid charge on the LHP performance - start-up - effect of power load on the LHP start-up behavior - characteristics of the temperature oscillations at different heat loads	- effect of power load on the LHP start-up behavior - characteristics of the temperature oscillations at different heat loads	- characterization of LHP performance	- effect of wick characteristics, vapor groove design and working fluid on LHP performance - start-up test - observation of oscillating behaviors

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_r: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Yeh et al. (2008)	Adoni et al. (2009)	Bai et al. (2009)	Huang et al. (2009)	Maydanik and Vershinin (2009)
Loop Heat Pipes	Figure 21 - Annexe B		Figure 20 - Annexe B		
Specificities (shape, components, applications ...)	biporous wick-LHP	Polymer wick LHP	Dual compensation chamber	miniature LHP	miniature LHP
Working fluid (mass charge)	ammonia (37 g)	R 134a (189 g)	ammonia	acetone	ammonia
Shape/design	cylindrical (L_T 183)	cylindrical with reduced thermal link between Evap and CC aluminium	cylindrical	cylindrical (L_T 90)	cylindrical
Evap body material	stainless steel		stainless steel	copper + Al saddle	Nickel + Copper saddle
Evap dimensions (mm)	L 65 / OD 16 / ID 11	L 200 / OD 45 / ID 25.4	L 190 / OD 18 / ID 16	L 40 / OD 14	L 20 / OD 5
Heated surface area (cm ²)	8			18	4
CC body material	stainless steel	stainless steel	stainless steel	copper	Nickel
CC dimensions (mm)	L 118 / OD 29 / ID 24	L 62.5 / ID 45		L 50 / OD 14	
Volume (cm ³)	53.4	98.4	2 x 24.7		
Evap-CC hydraulic link	bayonet / -	bayonet / no 2nd wick	bayonet / no 2nd wick	- / no 2nd wick	- / no 2nd wick
Materials	sintered nickel	Polyethylene (UHMWPE)	Nickel	Nickel powder	Titanium
Dimensions	L 40 / OD 11 / ID 8	L 199 / OD 25.4 / ID 11	L 180 / OD 16 / ID 6		
Pore characteristic length (um)	D 6-60	D 7-12	D 2.2	5.-10	D 19
Porosity (%) - Permeability (m ²)	78-86 / 10 ⁻¹¹ - 10 ⁻¹²	35 / -	55 / 5 10 ⁻¹⁴	60 / -	70 / -
Vapour groove location		evaporator wall	porous wick		
Lines					
Line material	stainless steel	stainless steel	stainless steel	copper	
Liquid line : L / OD / ID (mm)	583 / 6 / 3	2310 / 6.35 / 4.57	2600 / 3 / 2	- / 4 / 2.3	224 / 2 / -
Vapour line : L / OD / ID (mm)	470 / 6.4 / 5	1760 / 6.35 / 4.57	2200 / 3 / 2	- / 4 / 2.3	224 / 2 / -
Condenser					
Condensation and heat exchanger type	in tube in a pipe-in-pipe liquid heat exchanger	in tube in a pipe-in-pipe liquid heat exchanger	in tube mounted on Al plate cooled by forced air convection	in tube mounted on plate cooled by variable forced air convection	in tube in a finned air or water heat exchanger
Materials	stainless steel	stainless steel	aluminium	copper	nickel
Dimensions L / OD / ID (mm)	800 / 6.4 / 5	1260 / 6.35 / 4.57	2000 / 3 / 2.6	- / 4 / 2.3	63 / 2 / -
LHP operation conditions					
LHP Instrumentation	steady-state 12 T-type thermocouples	steady-state / transient 29 T-types thermocouples absolute pressure transducer	transient 19 T-type thermocouples	transient 6 T-types thermocouples fan with adjustable speed	steady-state
Experimental study					
Power range (W) / q _{max} (W/cm ²)	25-200 / -	50-150 / -	5-20	5-100 / -	0-120 / 30
T(Q _{max}) / T _{sink} (°C)	73 / 10	85 / -10	- / Tamb	- / 25	- / 20 (water-air)
R _E / R _{LHP} / R _{LHP2} (K/W)	- / 0.31 / -	0.09 / 0.63 / 0.63		- / 0.6 / -	0.18 / 0.62 (water) - 1 (air) / -
Specific effect studies	- characterisation of mono-porous and bi-porous wick - effect of the pore size distributions of bi-porous wicks on the LHP performance - comparison between mono-porous and bi-porous wick	Incidence of the Evap/CC charge distribution on the LHP temperature - characterisation of mono-porous and bi-porous wick - effect of the pore size distributions of bi-porous wicks on the LHP performance - comparison between mono-porous and bi-porous wick	Start-up: - effect of initial heat load distribution on the LHP position depending on LHP position	Start-ups behavior study depending on heat load: - minimal heat loads - observations of smooth, overshoot or oscillating modes	- effect of condenser heat exchanger (water/air) and T _{sink} on LHP performance - comparison of various LHP thermal performance (condenser design, wick characteristics ...)

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_T : Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Maydanik and Vershimin (2009)	Maydanik and Vershimin (2009)	Yeh et al. (2009)	Becker et al. (2010)	Celata et al. (2010)
Loop Heat Pipes					
Specificities (shape, components, applications ...)			Figure 21 - Annexe B	Figure 19 - Annexe B	Figure 16 - Annexe B
Working fluid (mass charge)	miniature LHP ammonia	miniature LHP ammonia	biporous wick-LHP ammonia (38 g)	miniature-flat LHP water	miniature-flat LHP water (40-80% of $V_{tot} = 18\text{cm}^3$)
Shape/design	cylindrical	cylindrical	cylindrical (L_T : 183)		flat oval with embedded CC along the length the length of copper
Evap body material	Nickel + Copper saddle	Nickel + Copper saddle	aluminium + Cu saddle		stainless steel
Evap dimensions (mm)	L 20 / OD 5	L 20 / OD 6	L 65 / OD 15.5 / ID 12.5	L 85 / W 42 / H 7 / ± 0.5	OD 82 / ID 50 / H 20
Heated surface area (cm^2)	4	4	19.5	16	18
CC body material	Nickel	Nickel	aluminium	copper + finned radiator	stainless steel
CC dimensions (mm)			L 118 / OD 29 / ID 24	L 20 / W 42 / H 7 / ± 0.5	D 50
Volume (cm^3)					4.15
Evap-CC hydraulic link	- / no 2nd wick	- / no 2nd wick		- / no 2nd wick	- / no 2nd wick
Materials	Titanium/Nickel	Nickel	sintered nickel		Stainless Steel
Dimensions			L 40 / OD 12.5 / ID 9	L 55 / W 41 / H 6	D 50 / H 4
Pore characteristic length (um)	D 17.2/13.5	D 10/13.5	D 3-34	D 16	D 20
Porosity (%) - Permeability (m^2)	78-71 / -	65-65 / -	70-85 / 10^{-12} - 10^{-13}	69 / -	32 / 2.5 10^{-14}
Vapour groove location			porous wick	porous wick	evaporator wall
Line material			stainless steel	copper	stainless steel
Liquid line : L / OD / ID (mm)	210-224 / 2 / -	190 / 2 / -	583 / 6 / 4	765 / 3	900 / - / 2
Vapour line : L / OD / ID (mm)	163-168 / 2 / -	184 / 2.5 / -	470 / 6.4 / 5	358 / 4	450 / - / 2
Condensation and heat exchanger type	in annular cross-section in a finned air or water exchanger nickel	in annular cross-section in a finned air or water exchanger nickel	in tube in a pipe-in-pipe liquid heat exchanger stainless steel	in tube mounted on plate cooled by liquid convection	in tube in a pipe-in-pipe liquid heat exchanger stainless steel
Materials					
Dimensions L / OD / ID (mm)	61.5 / 5 / -	61 / 5 / -	800 / 6.4 / 5	562 / - / -	170 / 8 / 6
LHP operation conditions	steady-state	steady-state	steady-state	steady-state	steady-state / transient
LHP Instrumentation			12 T-type thermocouples	18 K-type thermocouples	15 K-type thermocouples pressure transmitter
Power range (W) / q_{max} (W/ cm^2)	0-140 / 82-97	0-170 / 100	50-570 / -	20-100 / 6.25	20-70 / 3.9
T(Q_{max}) / Tsink ($^{\circ}\text{C}$)	- / 35	68 / 35-45	85 / 10	90 / 20 and 55	144 / Tamb
R _E / R _{LHP} / R _{LHP2} (K/W)	0.11 / 0.3 (water) - 0.66 (air) / -	0.06 / 0.14 (water) - 0.61 (air) / -	0.025 / 0.13 / -	- / 0.2-2.7 / -	0.27 / 1.7 / 1.7
Specific effect studies	- effect of condenser heat exchanger (water/air) and Tsink on LHP performance - comparison of various LHP thermal performance (condenser design, wick characteristics ...)	- effect of condenser heat exchanger (water/air) and Tsink on LHP performance - comparison of various LHP thermal performance (condenser design, wick characteristics ...)	- characterisation of mono-porous and bi-porous wick - effect of the pore size distributions of bi-porous wicks on the LHP performance - comparison between mono-porous and bi-porous wick	- LHP performance depending on: LHP orientation - sink and ambient temperature level (Tsink=Tamb)	- LHP start-ups at various heat loads and Evap/CC orientation - Performance depending on fluid charge, Evap/CC orientation Observation of temp oscillations

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_T : Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Choi et al. (2010)	Joung et al. (2010a)	Joung et al. (2010b)	Li et al. (2010)	Lin et al. (2010)
Loop Heat Pipes					
Specificities (shape, components, applications ...)	miniature-flat LHP	Flat bifacial evaporator LHP	Flat bifacial evaporator CPL	High heat flux LHP	Dual compensation chamber
Working fluid (mass charge)	-	acetone	water (70% Vtot)	distilled water	ammonia (50 g)
Shape/design	flat disk with embedded CC in the thickness	flat rectangular with embedded CC along the length	flat rectangular with low thermal link between Evap and CC	flat rectangular with embedded CC in the thickness	cylindrical
Evap body material	copper	stainless steel	stainless steel	copper	stainless steel
Evap dimensions (mm)	ID 38	L 52.4 / W 56 / H 20		L 30 / W 30 / H 15	L 190 / OD 18 / ID 16
Heated surface area (cm ²)	4	25 x 2		6.25	
CC body material	stainless steel	stainless steel	copper	copper	stainless steel + glass
CC dimensions (mm)	ID 38 / H 7				
Volume (cm ³)	80	1/2 Vtot	1/2 Vtot		24.7 x 2
Evap-CC hydraulic link	no bayonet / SS 2nd wick (H 8)	- / no 2nd wick	tube / no 2nd wick	- / no 2nd wick	bayonet / no 2nd wick
Materials	Nickel	SST 304	SST 316L	copper powder	Nickel
Dimensions	ID 38 / H 3	L 52.4 / W 52.4 / H 5	L 52.8 / W 52.8 / H 3	L 25 / W 25 / -	L 160 / OD 16 / ID 6
Pore characteristic length (um)		L 8.4	L 5	D 130	D 2.2
Porosity (%) - Permeability (m ²)	- / -	- / 1.5 10 ⁻¹³	50 / -	- / -	55 / 5 10 ⁻¹⁴
Vapour groove location	evaporator wall	evaporator wall	evaporator wall	porous wick	
Line material	Tygon	stainless steel 316	stainless steel	copper	stainless steel
Liquid line : L / OD / ID (mm)	300 / - / -	1350 / 6.35 / -	460 / 6.35 / -	120 / - / 5	2500 / 3 / 2.
Vapour line : L / OD / ID (mm)	250 / - / -	1350 / 6.35 / -	600 / 6.35 / -	120 / - / 5	2500 / 3 / 2.
Condensation and heat exchanger type	in tube in a finned forced air heat exchanger	in tube in a pipe-in-pipe liquid heat exchanger	in tube in a pipe-in-pipe liquid heat exchanger	in tube in a finned forced air heat exchanger	in tube in a pipe-in-pipe liquid heat exchanger
Materials	copper	copper	copper	copper	
Dimensions L / OD / ID (mm)	80 / - / 3.2	800 / 6.35 / -	800 / 6.35 / -	500 / - / 5	1800 / 3 / 2.
LHP operation conditions	steady-state / transient	steady-state/transient	steady-state/transient	steady-state/transient	steady-state/transient
LHP Instrumentation		30 T-types thermocouples, absolute pressure sensor	18 T-type thermocouples 2 absolute pressure transducers	8 T-type thermocouples IR thermography	12 T-type thermocouples, clear windows on condenser and CC
Power range (W) / qmax (W/cm ²)	10-110 / 27.5	20-260 / 5.2	100-220 / 4.4	10-600 / 96	5-400 / -
T(Qmax) / Tsink (°C)	80 / Tamb	- / 5 to 25	67 / 25	120 / Tamb	39 / -10 to 10
R _E / R _{LHP} / R _{LHP2} (K/W)	0.18 / 0.66 / - (model LHP_1)	- / - / 0.2-0.4	- / 0.2 / - (T _{CC} =40 °C)	0.033 / 0.162 / -	- / 0.08 / -
Specific effect studies	Effect of the presence of a 2nd wick (link between Evap and CC) on LHP start-up and thermal behavior	- effect of LHP elevation and heat load on LHP start-ups (role of the initial fluid distribution) - effect of LHP elevation and sink temperature on LHP performance - observation of oscillations behavior	- control of the CC operating temperature at setting point - effect of the non-condensable gas - observation of oscillating behavior	- vertical position (CC on top) - effect of Power load on LHP Start-up behavior - observation of oscillating behavior and small hysteresis phenomena at Low heat flux	- visualization of the fluid distribution in the Evap/CC - effect of initial fluid distribution on LHP start-up - effect of Tsink on thermal perf - observations of temperature fluctuations and hysteresis

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_T: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX A

Nom auteurs	Maydanik and Vershinin (2010)	Nagano et al. (2010)	Pastukhov et al. (2010)	Pastukhov et al. (2010)	Singh et al. (2010)
Loop Heat Pipes					Figure 14 - Annexe B
Specificities (shape, components, applications ...)	High heat flux LHP	Polymer wick LHP	miniature LHP	miniature LHP	miniature LHP
Working fluid (mass charge)	water	acetone (10.5 g)	water	water	water
Shape/design	flat oval with embedded CC along the length copper	cylindrical (L _r 117) stainless steel + Al saddle	flat oval with embedded CC along the length copper + Cu saddle	flat oval with embedded CC along the length copper + Cu saddle	flat disk with embedded CC in the thickness copper
Evap body material	L 30 / W 42 / H 7 / t 0.5	L 77 / OD 12.7 / ID 9.3	L 50 / W 42 / H 7	L 42 / W 42 / H 7	D 30 / H 10
Evap dimensions (mm)	9		25	17.6	3.75
Heated surface area (cm ²)					
CC body material	copper	stainless steel	copper	copper	copper
CC dimensions (mm)	L 50 / W 42 / H 7 / t 0.5	L 40 / OD 20	14 / W 42 / H 7	29 / W 42 / H 7	D 30
Volume (cm ³)					
Evap-CC hydraulic link	- / -	- / no 2nd wick	- / no 2nd wick	- / no 2nd wick	no bayonet / no 2nd wick
Materials	copper	PTFE	copper	copper	copper
Dimensions		L 50 / OD 9.5 / ID 5			D 28 / H 3
Pore characteristic length (um)		D 4	D 20	D 20	D 24-30
Porosity (%) - Permeability (m ²)	- / -	50 / -	70 / -	70 / -	45 / 7.2 10 ⁻¹²
Vapour groove location		porous wick			evaporator wall
Line material	copper	stainless steel	copper	copper	copper
Liquid line : L / OD / ID (mm)	180 / 4 / 3.4	800 / 3.3 / 1.8	283 / 4 / 3	206 / 4 / 3	290 / 2
Vapour line : L / OD / ID (mm)	120 / 4 / 3.4	800 / 4.7 / 3	311 / 4 / 3	230 / 5 / 4	150 / 3 / 2.
Condensation and heat exchanger type	in tube in a pipe-in-pipe liquid heat exchanger	in tube mounted on a Cu plate cooled by liquid convection	in tube mounted on a finned heat exchanger cooled by forced air	in tube mounted on a finned heat exchanger cooled by forced air	in tube in a finned forced air heat exchanger
Materials	copper	stainless steel	copper	copper	copper
Dimensions L / OD / ID (mm)	335 / 4 / 3.4	673 / 3.3 / 1.8	730 / 4 / 3	730 / 4 / 3	50 / - / 2
LHP operation conditions	steady-state	steady-state/transient	steady-state/transient	steady-state/transient	steady-state/transient
LHP Instrumentation		40 T-type thermocouples	7 thermocouples	7 thermocouples	17 T-type thermocouples 1 pressure transducer
Power range (W) / q _{max} (W/cm ²)	20-900 / 100	5-130 / -	0-350 / 14	0-400 / 22.7	5-60 / 16
T(Q _{max}) / T _{sink} (°C)	118 / 21	100 / 0 to 15	115 / 22	90 / 22	100 / 25
R _E / R _{LHP} / R _{LHP2} (K/W)	0.014 / 0.11 / 0.088	- / 0.8 / -	0.05 / 0.27 / -	0.05 / 0.17 / -	- / 1.25 / -
Specific effect studies	- characterization of LHP thermal performance - effect of the heat load concentration on the evaporator heat transfer coefficient (h _{max} = 113700 W/m ² .K)	- start-up at low power - study of LHP behavior during power cycle test	- effect of LHP orientation on LHP performance - effect of condenser heat exchanger design - performance comparison between LHP with cooled and uncooled CC	- effect of LHP orientation on LHP performance - effect of condenser heat exchanger design - generation of non-condensable gas during the first initial runs - effect of NCG on LHP start-up and thermal performance - effect of NCG at low heat load - observation: surface oxidation due to NCG	- generation of non-condensable gas during the first initial runs - effect of NCG on LHP start-up and thermal performance - effect of NCG at low heat load - observation: surface oxidation due to NCG

D: Diameter - OD: Outer Diameter - ID: Inner Diameter - L_r: Total length - L: Length - W: Width - H: Height - t: thickness

APPENDIX B: LHP EVAPORATOR DESIGNS OF THE LITERATURE

Fig. 14 : Singh et al. (2010)

Fig. 16 : Celata et al. (2010)

Fig. 15 : Boo et al. (2004)

Fig. 17 : Li et al. (2010)

APPENDIX B: LHP EVAPORATOR DESIGNS OF THE LITERATURE

Fig. 18 : Chen et al. (2006)

Fig. 19 : Becker et al. (2010)

Fig. 20 : Bai et al. (2009) – Lin et al. (2010)

Fig. 21 : Yeh et al. (2008-2009)

Fig. 22 : Ogushi et al. (2003)