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THERMAL PERFORMANCE OF MINI LOOP HEAT PIPE: INFLUENCE OF COOLING SYSTEMS

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Abstract. *Loop Heat Pipe (LHP) is a closed device containing a working fluid that transfers heat through the phase-change phenomenon from a hot to a cold region. According to the application, the heat transfer device must attend to some requirements, such as geometry, weight, heat transfer capacity and the available cooling system. In this regard, a flat LHP of 76 x 60 x 1.6 mm was developed for the thermal management of electronic gadgets. Sintering of copper powder and diffusion bonding was used to manufacture the device. Ethanol was used as the working fluid. An experimental bench to simulate the operating condition and the geometric characteristics of electronics were developed. The LHP was tested in the horizontal orientation (without gravity assistance) and gravity-assisted position. Two cooling systems were used: natural air convection and forced water convection. The main objective of this paper is to evaluate the effect of the cooling system on the thermal performance of the LHP. The LHP started working with 2 W/cm² for all conditions and reached a maximum heat flux of 16 W/cm². However, in forced water convection, the device could not operate successfully as an LHP once the vapor didn't achieve the entire condenser.*

Keywords: *loop heat pipe, diffusion bonding, thermal performance, electronics application, cooling systems*

1. INTRODUCTION

Loop heat pipe (LHP) is a highly effective device that uses the phase change of a working fluid to transfer heat from a heat source to a heat sink. The device operates due to a capillary force generated in a porous wick structure that circulates the fluid inside the closed loop.

The first report of LHP is dated 1982 by Russian scientists from the Lavochkin Association in cooperation with the Institute of Thermal Physics (Ural Branch of the Russian Academy of Sciences), developed by Gerasimov Yu. F. with his colleagues Maydanik Yu. F. and Kiseev V. M (Goncharov and Barantsevich, 2005). Since then, LHPs have been extensively researched due to some advantages: they can transport heat through short and long distances, have high flexibility, have large heat transfer capacity and their operation is not affected by the gravitational field (Chen et al., 2016). Owing to the promising technology of LHPs, they have been widely applied in the thermal control of electronics in gravity and microgravity environments (Domiciano et al., 2022; Zhao et al., 2017; Zhou et al., 2016), heat recovery (Liu et al., 2022), water heaters (Zhao et al., 2010), solar energy (Liao et al., 2018) and other applications.

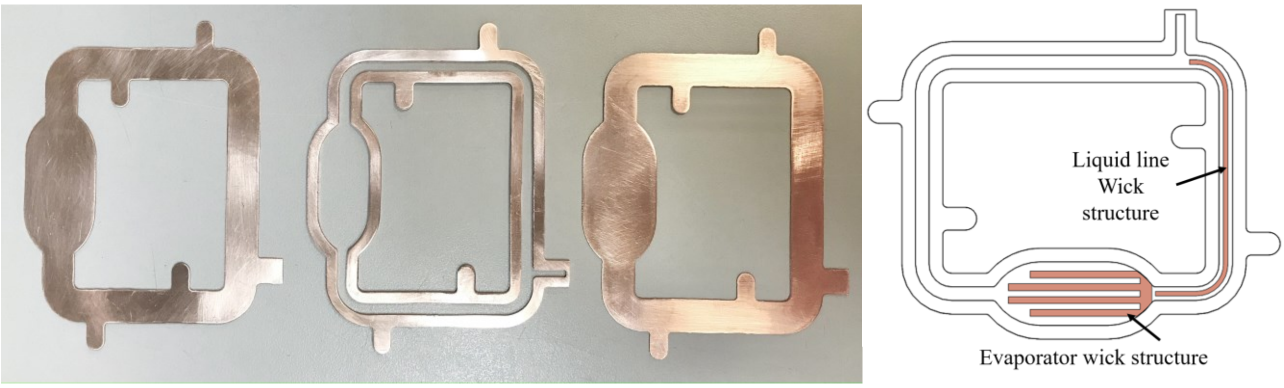
Recently, the development of electronics, driven by the world's technology consumption aim, lead to large heat generation inside small gadgets, such as laptops, CPU computers, tablets, smartphones and others. Flat LHPs can meet low volume and high heat power dissipation capacity requirements, receiving, therefore, special attention as thermal control technology in these applications.

Maydanik et al. (2010) developed an LHP that employed cylindrical evaporator and ammonia as the working fluid, to help the thermal management of central processing units (CPUs) and graphics processing units (GPUs). The LHP was designed to reject heat in the condenser by water forced convection. Their device was able to transfer heat loads between 100 and 320 W with a temperature of 40 °C to 70°C and a thermal resistance from 0.13 to 0.33 °C/W. Zhou et al. (2019) studied an flat LHP of 1.2 mm thick to remove heat from laptops using forced air convection. Their heat transfer device could dissipate up to 30 W, before reaching the temperature of 100 °C. They pointed out that the room temperature affects the thermal performance of LHPs. With an environment temperature of 23 °C, the minimum thermal resistance was 2.17 °C/W. When the room temperature was increased to 35 °C, the thermal resistance reduced to 0.31 °C/W. Domiciano et al. (2022) designed flat LHPs with different thickness (1.56 mm and 0.92 mm) to be applied on the thermal management of mobile phones. Because of this application, natural air convection was used to dissipate the excess of heat in the condenser section. A minimum thermal resistance of 0.2 °C/W and 0.72 °C/W was achieved for the larger and smaller devices, respectively.

In this context, according to each application, the heat transfer device must attend to some requirements, such as geometry, weight, heat transfer capacity and the available cooling system (natural air, forced water and forced air convection). Therefore, in the present work, an LHP of 76 x 60 mm and 1.6 mm in thickness, aimed to cooling electronic components, was developed and tested successfully. An experimental bench, designed to simulate the operating condition and the geometric characteristics of electronics, was accomplished, so that the LHP can be tested under several heat fluxes in the horizontal orientation (without gravity assistance) and gravity-assisted position. Two cooling systems were used: natural air convection and forced water convection. The main objective of this paper was to fully investigate the thermal characteristics of the LHP under both cooling systems in order to help better understand the influence of cooling system in the thermal performance of these novel heat transfer devices.

2. FABRICATION OF THE LOOP HEAT PIPE

As the main application of the designed mini LHP is to remove concentrated heat from small devices, such as chip processors of smartphones and tablets, the external dimensions of the LHP are 76 x 60 mm and 1.6 mm in thickness. Three copper sheets machined with a water jet provided the LHP geometric characteristic shown in Figure 1. The sintering process of a copper powder provided the wick structures, one located in the evaporator and one in the liquid line. The evaporator wick structure produces the capillary force needed to suppress the total pressure drop of the mini LHP. The liquid line wick structure promotes the working fluid from the condenser to the heated side. Lastly, the three copper sheets were diffusion bonded in a sandwich arrangement. Table 1 presents the main specifications of the mini LHP. For more details about the LHP manufacturing procedure, see Domiciano et al. (2022).



(a) (b)

Figure 1. (a) LHP copper sheets and (b) internal view of the LHP

Table 1. Main specifications of the mini LHP.

LHP	Parameter	
Evaporator [mm]	Size (LxW)	37.5 x 20
	Heating area (LxW)	10 x 10
Evaporator wick [mm]	Overall dimensions (LxW)	23.5 x 1.5 / 29 x 1.5
Liquid line wick [mm]	Size (LxW)	70.49 x 1
Wick structure porosity [%]	53.46 ± 3.87	
Wick structure permeability [m²]	$1.99 \times 10^{-12} \pm 1.02$	
Working fluid	Ethanol	

Abbreviations: L – Length; W – Width.

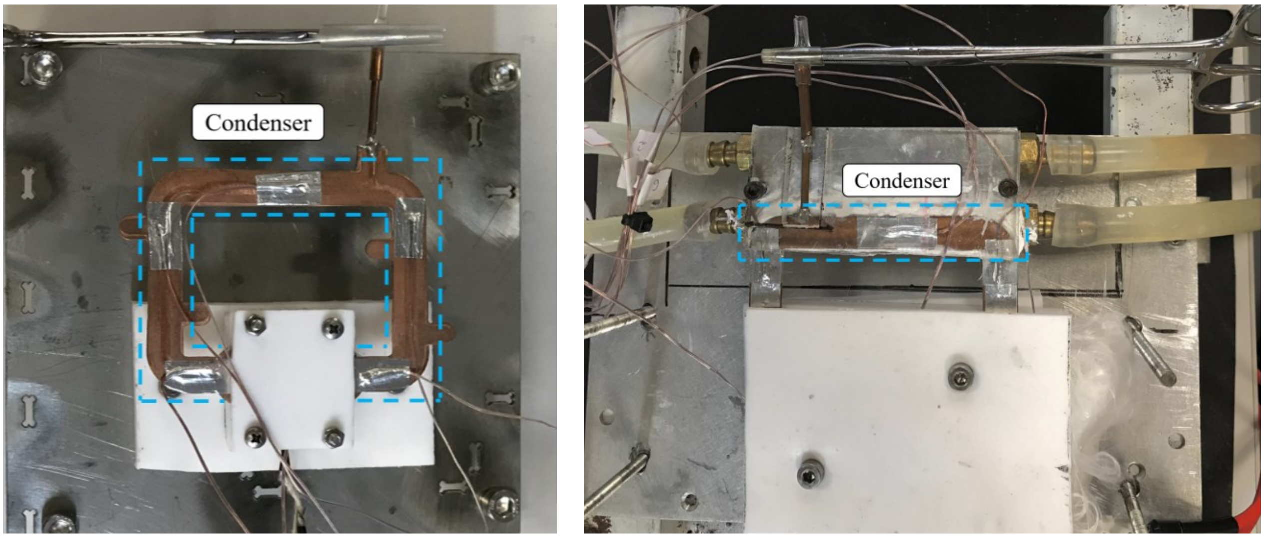
3. EXPERIMENTAL SETUP

A workbench, developed to simulate the operating condition and the geometric characteristics of electronics components, was used. The evaporator is heated by means of an electrical resistor (maximum output of 32 W with a diameter of 3.2 mm and a length of 25 mm) inserted into a copper block with a small surface area of 1 cm^2, to simulate an electronic device application.

According to each application, the heat transfer device must attend to some requirements, such as geometry, weight, heat transfer capacity and the available cooling system (natural air, forced water, or forced air convection). Therefore, to evaluate the LHP operational characteristics for distinguished applications, two cooling methods, natural air convection (condenser area of 34.62 cm^2) and forced water convection (condenser area of 7.05 cm^2) with prescribed temperature of 20 °C and 36 °C, were used in the LHP condenser. With the LHP cooled by the natural system, polytetrafluoroethylene (PTFE polymer) isolates the evaporator section from the external environment (see Figure 2a). On the other hand, when the LHP is cooled by forced water convection, the entire LHP is insulated, as shown in Figure 2b.

The experimental setup consists of a power supply unit, a data acquisition system (DAQ-NI™ SCXI-1000), seven T-type thermocouples (Omega Engineering™) and a computer. These temperature sensors measure the external surface of the LHP, fixed by thermosensitive adhesive strip of Aluminum and Kapton™. One thermocouple (T_1) evaluates the temperature of the heated area. Thermocouple T_2 estimates the evaporator exit temperature. Thermocouples T_3, T_4 and T_5 measure the temperature of the condenser inlet, intermedium and outlet. T_6 monitors the evaporator entry temperature and T_{amb} measures the environment temperature.

The LHP was tested in the horizontal orientation, where the device works in non-assisted gravity conditions, and in the gravity-assisted orientation (condenser above the evaporator). The applied heat loads increase in steps of 0.5, 1, or 2 W until the evaporator temperature reaches approximately 100 °C, considered the maximum temperature for a safe operating temperature of electronics (Maydanik et al., 2011). Each power input remained for 1200 s to the natural air condition and 900 s to the forced water convection, ensuring a steady-state operation (considered a maximum temperature variation of 0.1 °C in 1 minute). Thus, an acquisition rate of 1 sample/second was used.



(a) (b)

Figure 2. Experimental apparatus of the LHP thermal tests cooled by (a) natural air convection (b) forced water convection.

Previous experimental tests showed that ethanol is a good choice as a working fluid. For each cooling system, the LHP showed the lowest thermal resistance for different filling ratios (FR). Therefore, the LHP cooled by natural air convection and forced air convection (20 °C and 36 °C) presented the best tested FR of 39% and 49%, respectively. The FR, defined as the ratio between the working fluid and the total volume. is given by:

$$FR = \frac{V_l}{V_t} \times 100\% \tag{1}$$

where V_l is the liquid volume inserted into the LHP and V_t is the total void volume of the LHP, which is 0.82 ml for the studied heat transfer device.

The overall thermal resistance is used to quantify the thermal performance of the LHP, i.e., the ability to transport heat from the hot to the cold side. This parameter is calculated and defined by:

$$R_t = \frac{T_{ev} - T_c}{q} \tag{2}$$

where q is the heat transfer rate, T_{ev} is the evaporator temperature, given by thermocouple T_1 measurement, and T_c is the condenser average temperature, given by the average of thermocouples T_3, T_4 and T_5 readings for the tests with forced water convection, and by T_4 for natural air convection. The thermal resistance uncertainty was evaluated by the error propagation method explained by Holman (2011). It is associated with the error given by: thermocouples, data acquisition system, and the power supply unit. All thermocouples were calibrated using the same experimental apparatus. The thermal resistance uncertainty, δR, can be estimated as:

$$\delta(R) = \left(\left[\frac{\partial R}{\partial T_{ev}} \delta T_{ev} \right]^2 + \left[\frac{\partial R}{\partial T_c} \delta T_c \right]^2 + \left[\frac{\partial R}{\partial q} \delta q \right]^2 \right)^{1/2} \tag{3}$$

where δT_{ev} and δT_c are the temperature uncertainty of evaporator and condenser, respectively. The heat load was the heat transfer rate applied on the outer surface of the evaporator, defined as:

$$q = U \cdot I \tag{4}$$

where U is the voltage output and I is the current output of the power supply. The standard uncertainty of voltage and current was provided by the power supply manufacturer, taking into account the maximum value as 0.15 V and 34 mA.

4. RESULTS AND DISCUSSION

The startup might be considered as one of the most significant aspects of the thermal performance evaluation of LHPs. Evaporation of the working fluid will occur when the evaporator wick structure is saturated with liquid and heat is directly applied to the outer surface of the evaporator. The vapor circulates to the condenser section direction, where heat is removed, condensing the working fluid. The liquid returns to the evaporator by the generated capillary pressure in the porous media, closing the loop. In order to LHPs startup, a necessary thermodynamic condition must exist, i.e., a minimum temperature difference between the liquid-vapor interface inside the evaporator wick structure and the evaporator inlet region (Chernysheva et al., 2007). Usually, this thermodynamic condition is easily achieved when high power inputs are applied, favoring the LHP startup. Therefore, operating small-scale LHPs can be difficult when compared to larger ones, as the conduction heat transfer within the metallic components of the device tends to dump temperature differences. In the present research, the LHP is considered to start operating when vapor reached the condenser, observed by its temperature increase.

Figure 3 shows the transient temperature behavior for each applied power input level of the LHP in the horizontal orientation and cooled by natural air convection. The data did not indicate any evaporator temperature overshoot. Below 2 W, conduction rules the heat transfer rate from the evaporator to the condenser. With 2 W and at approximately 3200 s, the liquid started to change of phase and vapor was able to reach the condenser region, increasing the temperatures T_3 and T_4. At the same time, sub-cooled liquid enters the evaporator, decreasing its temperature. This unbalanced temperature between the evaporator (T_1) and evaporator inlet region (T_6) was of 2.92 °C. Above 3 W, thermocouples T_1, T_2, T_3, and T_4 (evaporator and vapor line) are nearly equivalent, presumably because most of the condenser length is fulfilled with vapor. The LHP operated with heat input power up to 8 W, almost reaching the limit temperature of 100 ºC. Some temperature fluctuations were observed in the data, which might be related to the two-phase phenomena or caused by variations in the natural convection.

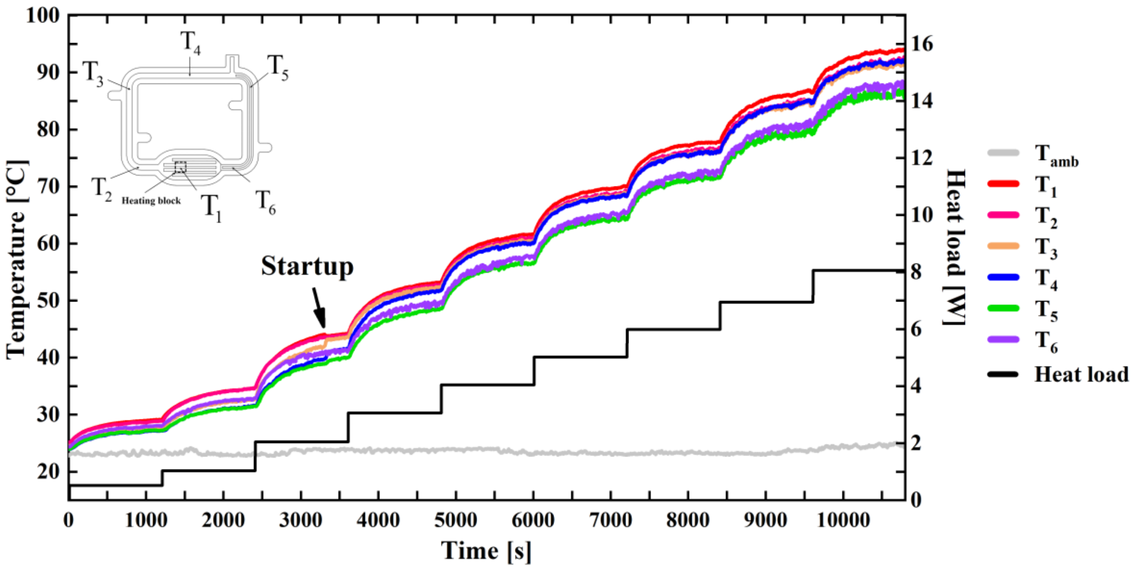


Figure 3. Transient temperature of the LHP in the horizontal orientation and cooled by natural air convection.

Figure 4 illustrates the transient temperature behavior for each applied power input level of the LHP in the horizontal orientation and cooled by forced water convection (prescribed temperature of 20°C). As in the previous thermal test (Figure 3), the LHP started up at 2 W when the inlet condenser temperature increased and sub-cooled liquid entered the evaporator, decreasing its temperature. The startup unbalance temperature between the evaporator (T_1) and evaporator inlet region (T_6) was of 2.37 °C. It is clear that, when vapor reaches the condenser inlet, the heat sinks instantly condensates the working fluid in all power input levels above 2 W. For this reason, the LHP requires more ethanol to circulate inside the closed loop (FR of 49%) compared to the previous test (FR of 39%), with the LHP cooled by natural air convection. Furthermore, above 4 W, increasing the applied heat load, the evaporator and condenser temperature readings slowly sort out at each power input. When high heat fluxes are removed from the condenser, a small area of the entry region of the condenser is enough for condensing the vapor. The resulting condensate is able to return to the evaporator by the liquid line, as the small dimensions of channels provide the capillary effect needed for the liquid return. This means that only the evaporator, vapor line and the entrance of the condenser are active and the LHP works as a conventional heat pipe. This "heat pipe", although with lower thermal performance, is able to transport higher heat powers, of up to 16 W, before reaching the limit temperature of 100 ºC.

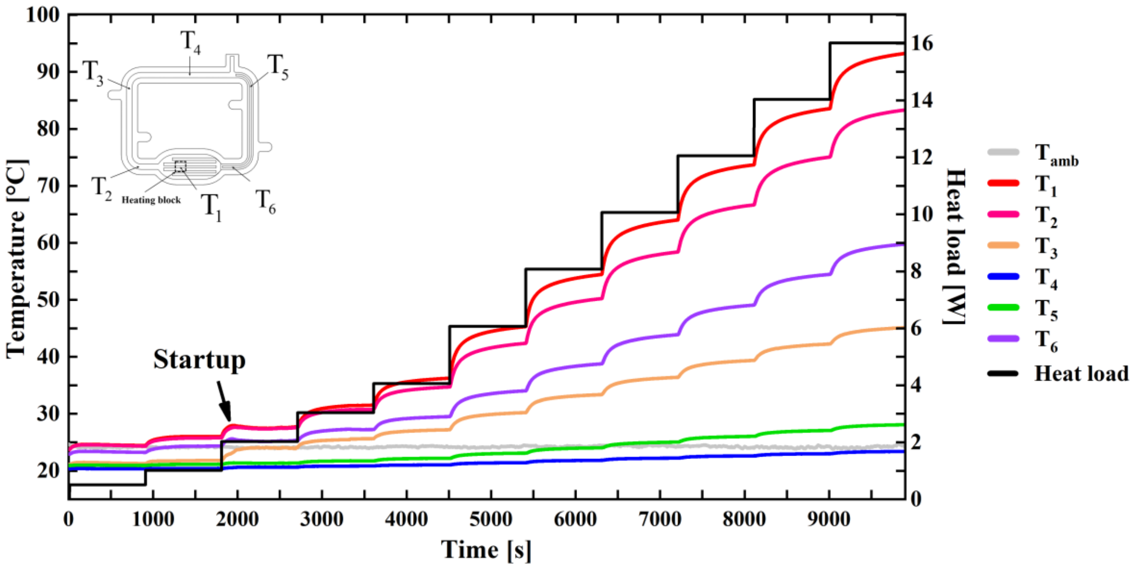


Figure 4. Transient temperature of the LHP in the horizontal orientation and cooled by forced water convection (20 °C).

Figure 5 presents the transient temperature behavior for each applied power input level of the LHP in the horizontal orientation and cooled by forced water convection (prescribed temperature of 36°C), i.e., the same testing conditions of Figure 4, but with higher inlet temperature levels (and so, lower cooling capacity).

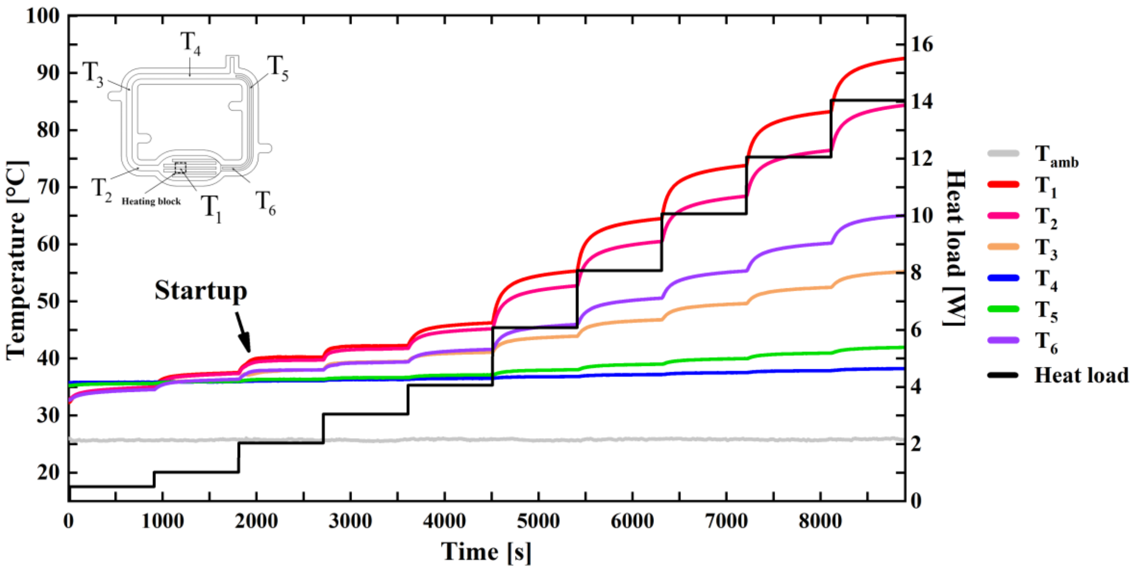


Figure 5. Transient temperature of the LHP in the horizontal orientation and cooled by forced water convection (36 °C).

As in the previous thermal tests (Figure 3 and Figure 4), the LHP started up at 2 W (See Figure 5), the produced vapor is condensed only in the condenser entry region. The necessary startup unbalance temperature between the evaporator (T_1) and evaporator inlet region (T_6) was of 2.25 °C. At 0.5 W, the LHP evaporator temperature is lower than the condenser one due to the constant heat sink temperature of 36 °C. Note that the filling ratio, considered as the best one, remained the same. However, in this case, the evaporator and condenser temperatures are closer than in the former test, presumably because the heat sink has a higher temperature. In this condition, the device worked as a conventional heat pipe and, although with lower thermal performance when compared to the LHP operation, was able to transfer higher heat loads, of up to 14 W, before reaching the limit temperature of 100 ºC.

Figure 6 shows the thermal resistance of the LHP as a function of the applied heat loads for the three experimental cases and two orientations. In all cases, for low heat inputs, the mass flow rate of the working fluid is small. Consequently, only a fraction of the condenser length is used for condensation. As the applied heat load increases, the mass flow rate raises, and the vapor can occupy a larger portion of the condenser, achieving the best thermal performance (lower thermal resistance). This explain why the thermal resistances decreases as the power input increases at low levels.

It should be noted that the thermal resistances of the empty LHP, i.e., without any working fluid inside, only evacuated, were 3.00 °C/W and 4.26 °C/W for the LHP cooled by natural air and forced water convection. These resistances are lower than those observed for low power input levels (less than 2 W) in all conditions tested when the device is not operating in phase change mode. This happens because, due to the presence of the porous media, the liquid is accumulated in the evaporator and can store heat, by increasing the evaporator region temperature, until the phase change conditions are reached and the device starts operating as a two-phase device. Besides that, at low heat loads and with forced water at 36 °C, the thermal resistance was small because the applied heat input was not able to sufficiently increase the evaporator temperature (See Figure 5), which is usually higher than the condenser temperature.

For both horizontal and gravity-assisted orientation, the LHP worked satisfactorily, independently of the gravity action. The condenser cooled by natural air convection yielded the best thermal performance for all cases, achieving a minimum value of 0.24 ºC/W in both tested orientations. For the condenser cooled with forced water at 20 ºC, the minimum thermal resistance was of 2.80 °C/W in the horizontal position and of 2.37 °C/W in the gravity-assisted orientation. As for the forced water at 36 °C, the thermal resistances reduced to 1.56 °C/W and 1.43 °C/W in the horizontal and gravity-assisted position, respectively. Although the natural air case presented the best thermal performance, the LHP could operate only until 8 W/cm^2, half power input when compared to the forced water convection condition, of 16 W/cm^2.

Figure 6 demonstrates that, as already observed, if the LHP condenser is cooled by forced water convection, the heat can be removed from the condenser faster than the working fluid is able to transport heat. In this case, the vapor is condensed only in the condenser inlet region. The porous media within the liquid line is able to keep the condenser outlet region filled with working fluid. Since the vapor could not fulfill the total condenser length, the effectiveness of the LHP was affected, leading to an increase in the evaporator and condenser temperature difference and, consequently, a higher thermal resistance. Furthermore, Figure 4 and Figure 5 enlighten that, as phase change is not happening in most of the condenser region, the outlet condenser temperature is increased by conduction heat transfer through liquid line (case material and liquid line wick structure of the liquid line), once the outlet condenser temperature (T_5) is higher compared to its center region temperature (T_4), which means that the heat transfer device was not working as an LHP but that the evaporator, vapor line and condenser inlet region are working as a conventional heat pipe.

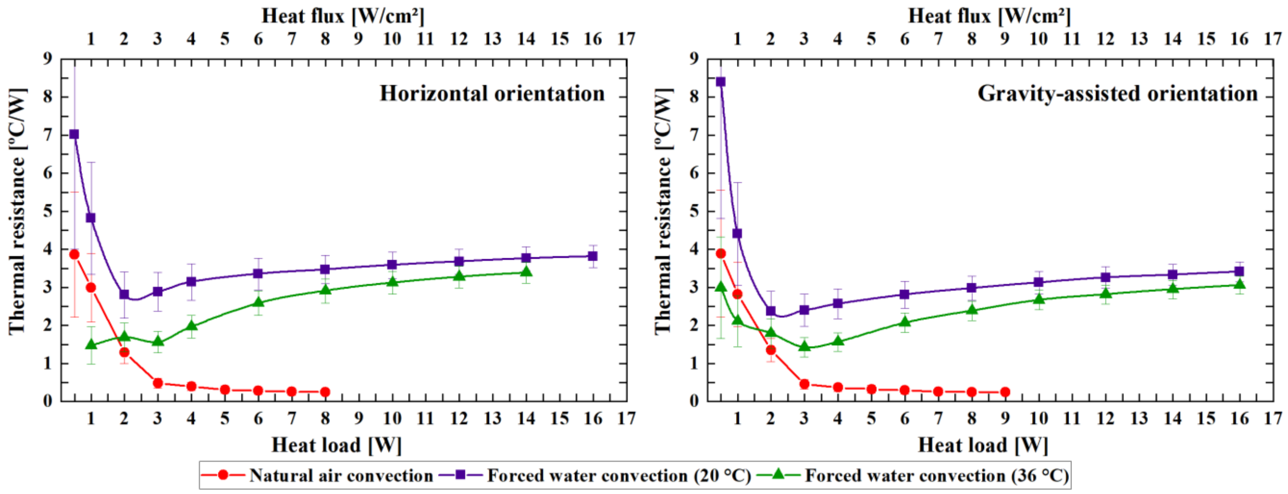


Figure 6. Overall thermal resistance of the LHP in the horizontal and gravity-assisted orientation.

Figure 5 and Figure 6 also show that, as expected, the LHP works better (lower thermal resistances) at higher temperature levels, although the lower condenser temperatures allow the system to transfer more heat.

Therefore, in addition to the importance of the correct LHP evaporator´s design (as well recognized in the literature), this paper shows that the heat transfer capacity of the LHP condenser must also be properly considered (neither under nor over dimensioned).

5. CONCLUSION

The present work developed a mini loop heat pipe with 1.6 mm of thickness, fabricated by sintering processing and diffusion bonding, for cooling electronic components. Two workbenches were designed to evaluate the heat transfer performance under different cooling systems, natural air and forced water convection. The LHP was tested in the horizontal and gravity-assisted orientation. The heating system delivered heat in an area of 1 cm², to simulate a typical electronic application. The conclusions are summarized as follows:

i. Wick structures in the evaporator and in the liquid line generated the necessary capillary force to pump the working fluid inside the loop heat pipe.
ii. For both cooling systems (natural air and forced water convection), the heat transfer device started the operation with 2 W/cm^2, resulting in a temperature difference between the evaporator and condenser of 2.92 °C and 2.37 °C for natural air and forced water convection, respectively.
iii. Under natural air convection, the loop heat pipe operated at a low heat flux of 2 W/cm² up to 8 W/cm² in the horizontal orientation and up to 9 W/cm^2 in the gravity-assisted orientation. Besides, a nearly to isothermal temperature distribution along the device was observed (less than 2 °C for all power input levels tested), as a result of the low thermal resistance of the device, of 0.24 ºC/W.
iv. Under the effect of a forced water convection cooler, with constant temperature of 20 °C, the loop heat pipe was able to transport up to 16 W/cm^2 for both horizontal and gravity assisted positions before achieving the limit temperature of 100 °C. The minimum thermal resistances of 2.80 °C/W and 2.37 °C/W were obtained in the horizontal and gravity-assisted orientation.
v. Under forced water convection with a prescribed temperature of 36 °C, the loop heat pipe worked transporting up to 14 W/cm^2 and 16 W/cm^2 with a minimum thermal resistance of 1.56 °C/W and 1.43 °C/W in the horizontal and gravity-assisted orientation, respectively.
vi. Although working under forced water convection, the LHP could not operate successfully as a loop heat pipe, but as a heat pipe, once the vapor could not fulfill the entire condenser.
vii. The suggested mini loop heat pipe contributes as a promising technology for the thermal control of electronics components.

The present study helped to understand the effect of different cooling systems to be used to remove heat from loop heat pipes. The main contribution of this research is to highlight that the heat removal conditions must be seriously considered in the design of LHPs.

As a suggestion for future work, further experimental research using the same LHP, but with different heat removal conditions, such as forced air convection, must be performed and the results compared to previous tests.

6. ACKNOWLEDGEMENTS

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