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Experimental investigation of loop heat pipe with a large squared evaporator for multi-heat sources cooling

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ABSTRACT

A heating area of 190 mm × 90 mm large flat-plate loop heat pipe was designed for the heat dissipation problem of multi-heat sources. The design process was also briefly introduced. The evaporator was made of aluminum alloy, and heat dissipation fins were arranged on the back side of the compensation chamber to enhance the heat transfer between the compensation and the ambient. The stainless steel wire mesh worked as the porous wick, and the acetone was chosen as the working fluid. Six ceramics heating blocks were used as the heat sources. The results showed that the system could start up and work normally between 20 W–140 W, and maintained the heating surface temperature below 90 °C. The system behaved as a zigzag start below 20 W, and the condenser inlet temperature oscillated periodically, and the system could start up stably between 25 W and 140 W. When the heat load was increased, there occurred periodic temperature fluctuation in condenser outlet. The system could establish a new balance quickly during variable heat loads operation, which reflected the good reliability of the LHP. The experiment of changing the heat dissipation condition on the condenser side and the evaporator side was carried out. When the heat load was 120 W and the ambient temperature was constant, the system equilibrium temperature difference caused by the air ventilation of the condenser was changed, which was less than the heat dissipation of the evaporator under the same conditions. The evaporator thermal resistance decreased with the increase in heat load, and the minimum thermal resistance of 0.032 °C/W was achieved at the heat load of 120 W. The total thermal resistance of the LHP was distributed between 0.312 °C/W and 0.212 °C/W. It was also pointed out that it was very important to improve the thermal uniformity of the heated surface of a large-plane loop heat pipe system with multiple heat sources.

1. Introduction

In supercomputers, chips, and inverters, the thermal load generated by high-power electronic devices are generally large. The heat dissipation problem has become the focus on ensuring the normal operation of electronic devices. With the increase of electronic equipment components, the multi-point distribution of heat sources caused by the spatial arrangement of high-power components on the modules will become another problem to be solved in the heat dissipation problem of high heat flux density. The goal of a cooling system is to keep the conditioner frequency converter pack within its operating temperature range. IGBT modules, IPM modules and other components of the converter have relatively large heat generation. The natural heat dissipation of the module itself cannot meet the requirements of semiconductor operation.

In most cases, air cooling can achieve the cooling requirements of air conditioner inverters, except in the case of high ambient air temperature or high load operation of the inverter. There are few research reports on the new heat dissipation methods of frequency converters under bad working conditions. As highly efficient heat transfer devices, loop heat pipe (LHP) can solve this problem very well due to the characteristics of lower thermal resistance and a simple structure. In recent years, it has been studied by many scholars [1,2].

Loop Heat Pipe is a new type of thermal control technology based on heat pipe [3]. It uses phase change heat transfer of

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Nomenclature

d	Pipe diameter (m)
d_w	Mesh diameter (m)
h	Heat transfer coefficient (W/(m^2• K))
K	Permeability (m^2)
P	Pressure (Pa)
Q	Thermal power (W)
r	Capillary radius (m)
R	Thermal resistance (K/W)
Re	Reynolds number
u	Velocity (m/s)

Greek symbols

ε	Porosity
ρ	Density (kg/m^3)
μ	Dynamic viscosity (Pa• s)
σ	Surface tension (N/m)
θ	Contact angle (°)
φ	Welding coefficient

Subscripts

cap	Capillary wick
cond	Condenser
evap	Evaporator
g	Gravity
l	Liquid
ll	Liquid line
v	Vapor
vl	Vapor line

Abbreviations

DC	Direct current
LHP	Loop heat pipe

working fluid. The working fluid absorbs heat in the evaporator, cools it into liquid in the condenser, and relies on capillary structure. The capillary force provided is returned to the evaporation side. It has the advantages of high heat transfer capability, low thermal resistance, high efficiency and long transmission distance. Based on these advantages, it has great application potential in electronics cooling.

LHP was first proposed in the 1970s, when the cylindrical LHP was designed for thermal control of aerospace and optical instruments [4,5]. As the demand for heat dissipation from electronic devices has become more prominent, LHP began to be applied to new scenes and was designed into various shapes, such as cylindrical, rectangular, flat-oval and disk-shaped [6]. The flexibility of the positional arrangement of the flat plate evaporator LHP and the contact area with the heat source can be increased compared with the cylindrical LHP of the same size, so that it can be applied to different conditions and convenient adjustment of gas flow direction to achieve synergy between temperature field and velocity field.

Researchers have studied LHP's start-up performance, working fluids, filling rate, wick structure and evaporator structure [6–9]. There are also analyses on non-condensable gas, pump assist, and loop pressure drops. In the experimental aspect, Wang et al. [1] designed a copper-methanol system with a diameter of 40 mm and a height of 19 mm. The startup under different heating load were tested. The results show that the system can operate stably between 20 W and 240 W. The sealing method plays an important role in improving the flat-plate type LHP. Anand [10] designed a flat-panel LHP with a visual condenser and tested with different working fluids for various heat inputs. The results show that n-pentane has the lowest operating temperature in the fluid tested, while methanol could work in the widest range of thermal load. Liu [11] also found that the two-phase flow is closely related to the physical properties of the working fluid. Liu [12] studies have found that acetone has better start-up performance in the LHP system, while the methanol system has a stronger heat transfer capacity. Tsai [13] designed a new type of flat-plate loop heat pipe, using a special semi-closed copper mesh structure, using copper as the material of the evaporator and pipeline, and the condenser is cooled by aluminum sheets and fans. The experimental results show that the system can operate at 50 W, and the operating temperature of the evaporator is lower than 85 °C.

For the problem of heat dissipation area of multiple heat sources, there are generally several methods when using a loop heat pipe as a heat sink. The first one is to design a multi-evaporator heat pipe system to increase the contact area with the heat source, but the multi-evaporator system has the problems of weight increase and unstable start-up. Many scholars have carried out demonstration and experimental research on this aspect [14–17]. Qu [18] also write a review about thermal performance in multiple evaporators loop heat pipe. Another method is to cover the heat source with a large surface evaporator to achieve heat transfer of the various components of the electronic device. There is not much research on large-scale loop heat pipes. One of the reasons is that researchers pay more attention to the heat dissipation of local heat sources in highly integrated electronic products, and the heat treatment of multiple heat sources has not given sufficient attention. In fact, for high-power electronic components, increasing the heat dissipation area can effectively reduce the heat flux density and thus protect it. For this reason, an aluminum-acetone loop heat pipe with a large squared evaporator have been designed, fabricated and tested for the first time for heat dissipation with multiple heat sources. The start-up characteristics and variable power operation of the system under different loads are studied, and its operation and system performance were analyzed. It is of great significance to design a more reasonable large-scale LHP in the future.

1.1. The design of LHP

Many factors should be considered during the design of a LHP for specific function. Usually, the following points need attention: working fluid; Structural material; porous wick; and working limit temperature.

1.2. Design of evaporator

Working limit temperature was firstly considered in many study, since the goal of LHP is to ensure the temperature of the electronics within acceptable limits, while most electronic components should have a temperature below 100 °C. For this reason, the working fluid could be ethanol, methanol, and acetone. Structural material are depend on the compatibility with liquids as well as the economic requirements. But high pressure should not be ignored under high power operation, especially when the working medium is ammonia. In fact, the design of the shell is generally considered from strength.

The design thickness calculation formula of the shell can be

derived as follows:

$$\delta = \frac{pD}{2[\sigma]^t\varphi} + C_2$$

where p is the safe withstand pressure, D is the calculated diameter, $[\sigma]^t$ is the allowable stress at design temperature, φ is the welding coefficient. C_2 is the corrosion allowance.

$$\delta_B = D_c\sqrt{\frac{Kp}{[\sigma]^t\varphi}}$$

where K is the structural coefficient, which could be found in the relevant table.

Under the premise of ensuring the strength and stiffness of the evaporator shell, reducing the wall thickness can effectively reduce the adverse effect of the lateral wall heat conduction, preventing the heat from entering into the compensation cavity through the outer wall, and reducing the wall thickness can reduce the metal consumables, thus saving the economic cost.

1.3. Design of capillary wick

Another design principles for the LHP system is the capillary limit. The capillary pumping force created in the wick was considered based on the total pressure loss during loop operation as follows:

$$\Delta p_{cap} \geq \Delta p_{total} \tag{1}$$

And the capillary pressure is calculated by:

$$\Delta P_{cap} = \frac{2\sigma\cos\theta}{r} \tag{2}$$

where θ is the angle of contact, it may be taken to be equal to zero for well-wetting liquid. The total losses of pressure in the LHP are composed of the following parts:

$$\Delta p_{total} = \Delta p_{groove} + \Delta p_{vl} + \Delta p_g + \Delta p_{cond} + \Delta p_{ll} + \Delta p_{wick} + \Delta p_{bend} \tag{3}$$

where Δp_{groove} is the pressure drop in the vapor groove; Δp_{vl} is the pressure drop in the vapor line; Δp_g represents the pressure drop due to gravity; Δp_{cond} means pressure drop in the condenser; Δp_{ll} refers to the pressure drop in the liquid line; Δp_{wick} is the pressure drop through the wick; σ is the surface tension of the working fluid, r is the effective pore radius of the capillary wick. To determine the total pressure drop in an LHP, we need to calculate all terms of Eq. (3).

According to fluid mechanics analysis, either in the vapor line, liquid line or vapor groove, we have:

$$\Delta p_i = \zeta_i \frac{L_i}{d_i} \frac{\rho_i u_i^2}{2} (i = groove, vl, ll) \tag{4}$$

$$u_i = \frac{m_0}{\rho_i A_i} = \frac{Q}{h\rho_i A_i} \tag{5}$$

where u is the average velocity of the working fluid in local, L is the length of line, d is equivalent diameter, Q is the thermal Power, h is the latent heat of the working fluid, ζ is a function of Re,

$$\zeta = \begin{cases} 64/Re & 0 \leq Re \leq 2300 \\ 0.00063\sqrt{Re} & 2300 \leq Re \leq 4000 \\ 0.3164/Re^{0.25} & 4000 \leq Re \leq 10^5 \end{cases} \tag{6}$$

In the liquid line, the Re is usually lower than 2300, we have:

$$\Delta p_{ll} = \frac{32\mu u_1 L_{ll}}{d^2} \tag{7}$$

Pressure drop in the condenser could be calculated by this equation:

$$\Delta p_{cond} = \sum \zeta_{cond}^i \frac{L_{cond}^i}{d_{cond}^i} \frac{\rho_i u_i^2}{2} (i = v, l) \tag{8}$$

$$\sum L_{cond}^i = L_{cond} (i = v, l) \tag{9}$$

For the pressure drop at the bend, we have

$$\Delta p_{bend} = N\zeta_{bend}\frac{1}{2}\rho u^2 \tag{10}$$

where N is the number of bend, ζ_{bend} is the local loss coefficient. In this study, all the bend angle equal to 90°, then ζ_{bend} is 0.83.

$$\Delta p_g = \begin{cases} \rho_v g\Delta H - \rho_1 g\Delta H & (1) \\ \rho_1 g\Delta H - \rho_v g\Delta H & (2) \end{cases} \tag{11}$$

where (1) evaporator is located above the condenser, (2) evaporator is located below the condenser. ΔH is the height difference between wick surface and condenser.

$$\Delta p_{wick} = \frac{\mu}{K} u_w L_{wick} + R_f \rho u_w^2 L_{wick} \tag{12}$$

where K is the permeability coefficient, μ is the dynamic viscosity, u_w is the apparent velocity of single-phase flow, ΔL_{wick} is the thickness of the wick. The permeability of the capillary core made of screen is calculated by the following empirical formulas

$$K = \frac{\varepsilon^3 d_w^2}{122(1-\varepsilon)^2} \tag{13}$$

d_w is the diameter of the mesh; ε(porosity) can generally be measured by experiments.

Through the above formula, the main pressure drop of the system in this paper was calculated at a heat load of 140 W and a vapor temperature of 80 °C for acetone, and the result was shown in Fig. 1. It can be seen that the maximum pressure drop exists in the vapor line flow, mainly because of its high flow rate. In this system, hydrostatic pressure drop also occupies an important part (see the following section for specific system structure parameters), which means that more vapor pressure is needed during startup.

The LHP consists of three parts: evaporator, condenser, liquid line and vapor line. The evaporator is the core component of the LHP, which directly affects the startup, operation and reliability of the system. For flat LHPs, the evaporator contains a compensation chamber and a capillary wick. The main function of the compensation chamber is to store the excess liquid working fluid in the operating process of the LHP system, ensuring that the capillary wick was wetted to provide the necessary working fluid for the phase change.

The large evaporation area of the multi-point heat source means

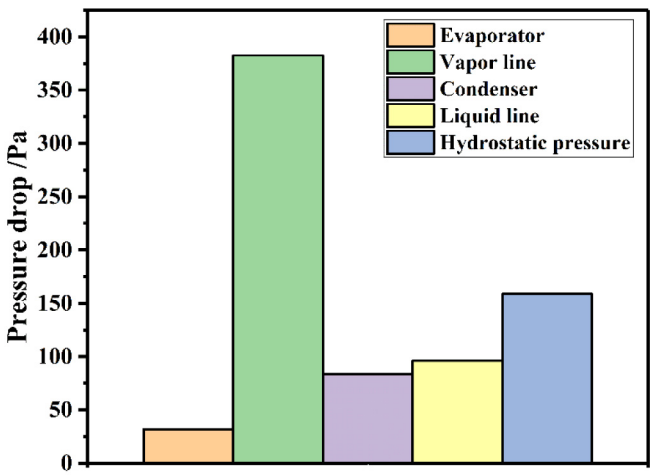


Fig 1. LHP experimental system.

that the capillary wick and its compensation chamber will be also increased accordingly. Due to its large area and good strength, porous wick of the evaporator is made of composite stainless steel meshes which has a thickness of 3.5 mm and is made of 200 mesh and 400 mesh stainless steel wire mesh. In our LHP system, a air-cooled finned tube heat exchanger is used as a condenser. Fig. 2 shows the evaporator and condenser of the LHP.

Before the working fluid was charged, the system was evacuated to 1.5×10^{-4} Pa by a vacuum pump. Acetone was selected as the working fluid considering the compatibility with the aluminum and the experimental temperature range–the vaporization temperature of acetone is 56.5 °C. The charge ratio of working fluid also plays an important role in the operational performance of LHP. In order to adapt to the gravity characteristics and avoid the lack of working medium, the charge ratio was selected as 75% based on the experimental experience. Table 1 shows the geometric material characteristics of the experimental LHP.

The test section and the locations of the thermocouples in this system are shown in Fig. 3. Seven microscale T-type thermocouples were firmly attached onto the evaporator surface near the heat source with aluminum foil tape, to measure the heating surface temperature of the evaporator. Other thermocouples were to measure the following temperatures: the temperature at the external wall of compensation chamber (8), the temperature at the fin (9), the temperature at the outlet of the evaporator (10), the temperature at the middle of vapor line(11), the temperature at the entrance of the condenser (12), the temperature at the exit of the condenser (13), the temperature at the middle of liquid line(14), the temperature at the entrance of the evaporator (15). And the thermal couple No.16 measuring the environment temperature is installed at a bench about 0.5 m nearby the LHP system, where the cooling fans have little impact on the testing of the environment temperature. Deviation of thermocouples is 0.5 °C. Temperature data were acquired every 3 s by the data acquisition unit (Keithley 2700). The atmospheric temperature was controlled by air conditioning at 12 ± 1 °C.

Heat source simulators were ceramic heating blocks and the heat input were controlled by the voltage regulator with an absolute error of 0.1 W (see Fig. 4). The heat source 1, 2, 3, 4 are the same, and connected to the regulator A, and the heat source 5, 6 are the same, and then connected to the regulator B, as shown in Fig. 3. The power of the heater is controlled by adjusting the voltage to simulate different heat loads. The condenser was cooled by three DC cooling fans with input voltage of 12 V and output power of 1.8 W. Fig. 5 is the LHP experimental system.

2. Results and discussion

2.1. Start-up performance

The start-up performance is an important performance indicator for LHP. The startup process of the loop heat pipe generally includes the following three stages: (1) The heat source is loaded on the surface of the evaporator, and the generated heat is mainly transmitted to the evaporator through heat conduction. The surface temperature of the evaporator rises steadily, and the working liquid in the compensation chamber absorbs part of the heat at the same time. (2) After the working fluid in the evaporator absorbs a certain amount of heat, vapor is generated beyond the saturation temperature and reaches a certain degree of superheat. As the vapor reaches the condenser inlet, the condenser inlet temperature also rises. (3) When the heat load is relatively small and the steam generated in the previous period is insufficient to maintain the system to complete the cycle, the condenser inlet temperature will fluctuate and the vapor will accumulate again in the evaporator. When the heat load is large, sufficient steam is generated, the working fluid can continuously work in the LHP, the condenser outlet and the evaporator inlet temperature are lowered, and the temperature change is gradually stable, and the system enters into a stable working state. When the experimental operating temperature change of the system was less than 0.5 °C within 10 min, it can be considered that the heat pipe has reached a stable working state.

From the principle of the LHP, when establishing the loop cycle, the following conditions should be met: (1) There is sufficient

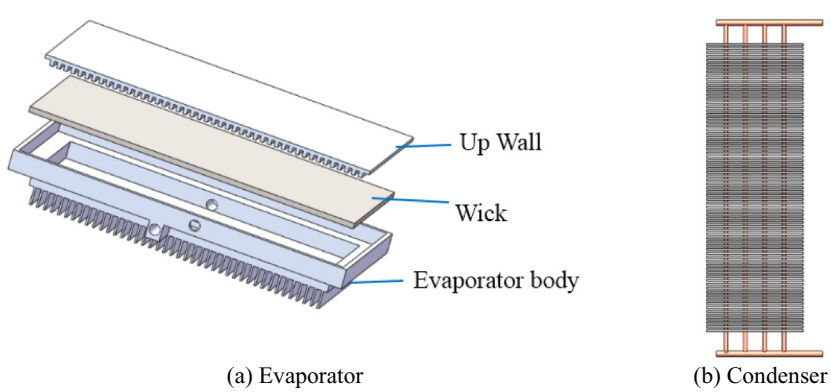


Fig. 2. Evaporator and condenser of the LHP.

Table 1
Geometric material characteristics of the experimental LHP.

Evaporator	L/W/H	190/90/30.5 mm	Groove depth	2 mm
	Material	Aluminum alloy	Groove number	84
	Groove width	2 mm	Working Fluid	Methanol
Compensation chamber	L/W/H	165/58/10 mm		
Wick	Porosity	0.6	Material	SS304
Condenser (Four tubes)	O/I	4/2 mm	Fin thickness	0.5 mm
	Length	160 mm	Fin number	100
Vapor Line	Material	Copper	Length	414 mm
	O/I	6/4 mm		
Liquid Line	Material	Copper	Length	798 mm
	O/I	6/4 mm		

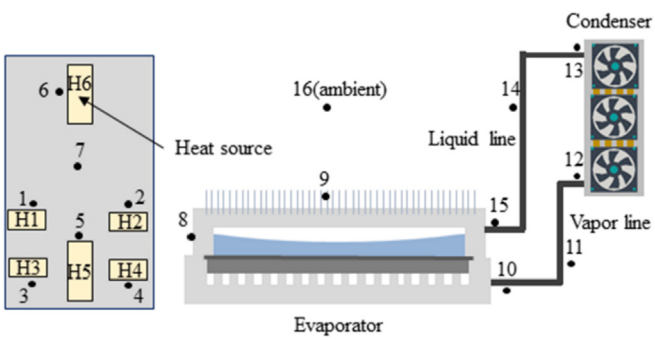
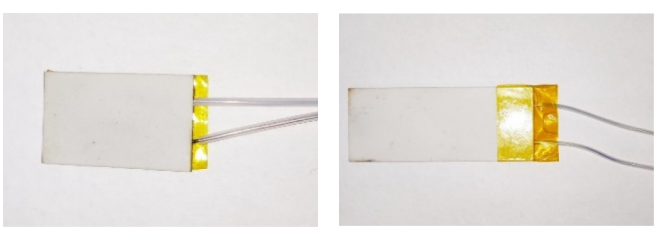


Fig. 3. Schematic of the test section and the locations of the thermocouples.



(a) H1 H2 H3 H4 (b) H5 H6

Fig. 4. Ceramic heating block: (a)220 V 50Ω; (b)220 V 20Ω.

working fluid in the evaporator; (2) Suitable heat load; (3) Sufficient capillary suction.

Fig. 6 shows the LHP the transient temperature oscillations during the startup process at a heat load of 18 W. At the start of the LHP, a temperature jump occurs in the evaporator and the compensation chamber, and the heat transfer in the condenser cannot be stabilized, but at this time there is a phase change because the inlet and outlet temperatures of the condensing are very different. The figure shows that the start-up curve with the

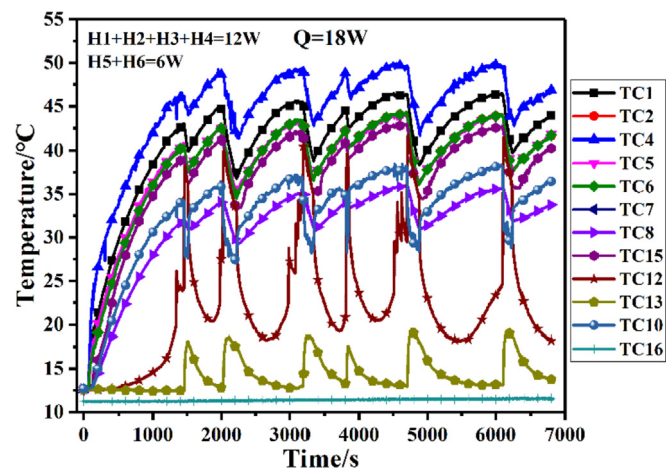


Fig. 6. Start-up process at 18 W.

heat load of 18 W exhibits a zigzag shape, and finally the final system cannot be stabilized. All measured temperature oscillations occur in regular periodic manner - this type of periodic fluctuation has also been reported in other researchers' studies [7,19]. The fluctuation cycle of the evaporator wall temperature is about 1200 s, while the amplitude is approximately 7 °C. It was clear that for the LHP developed in this experiment, the start-up heat load lower than the 18 W was unable to make proper potential for the complete working fluid circulation. The heat input on the evaporator surface is mostly lost from the evaporator surface to the environment. And the condenser is able to cool the working liquid to near ambient temperature.

Fig. 7 shows the start-up process when the heating power is 20 W. When the surface temperature of the evaporator reaches 26.0 °C, the rising speed of the temperature is temporarily stagnated. When the inlet temperature of the condenser reaches

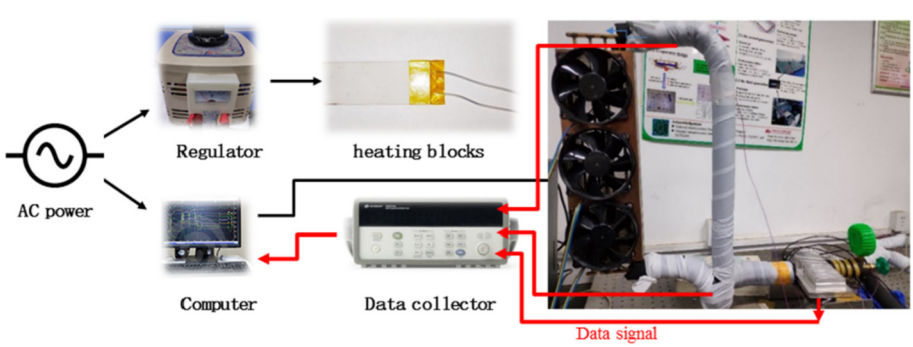


Fig. 5. LHP experimental system.

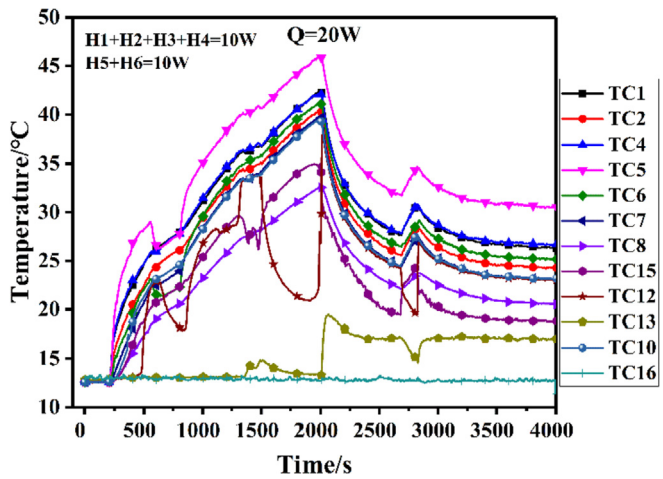


Fig. 7. Start-up process at 20 W.

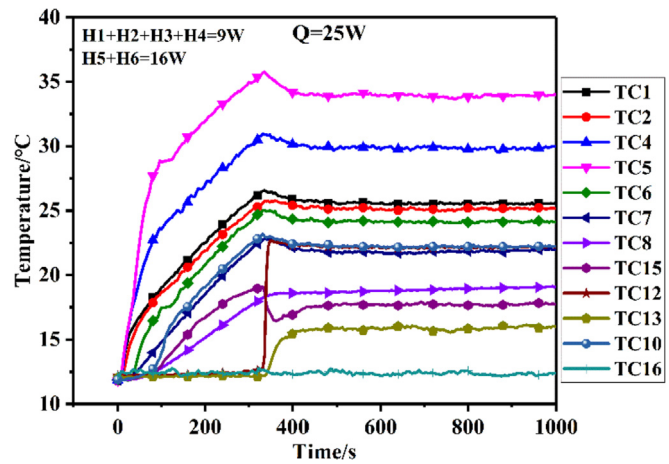


Fig. 8. LHP startup process at a heat load of 25 W.

22.9 °C, the temperature of the condenser drops suddenly and the decreasing range is about 5 °C. It can be seen from the observation that the outlet temperature of the condenser does not change, and the vapor does not flow out from the outlet after condensation. Instead, it returns to the condenser entrance under the action of gravity. When the evaporator temperature reaches 36.9 °C and the temperature rise slows down again, the phase change in the evaporator bns to play a role. The working point of the condenser is that the outlet temperature of the condenser rises and the temperature rises to 14.8 °C. At this time, it did not reach a complete equilibrium because the temperature of the heating surface is still in the rising stage.

After that, two oscillations occurred. At 1800 s, the heating surface reached the highest temperature of 42.9 °C, and began to drop abruptly. After 700 s, the temperature reached 28.5 °C. After a short temperature rise fluctuation, the temperature finally stabilized at 27.3 °C. The last oscillation occurs in the whole temperature drop stage, which is different from the previous oscillation. At this time, the heating surface and the outlet temperature of the evaporator rise, and the outlet temperature of the evaporator may be affected by the side wall heat conduction. The decrease of the inlet temperature of the condenser means that the vapor entering the condenser suddenly decreases, but the power has not changed and the liquid pipeline has flowed. The possible reason is that the working fluid occurs reverse flow. Short-term vapor is insufficient due to insufficient liquids.

When heat load is low, the system starts to experience temperature fluctuations. The phase variable of the working fluid is heavily dependent on the accumulation of heat over time. In the early stage of the start-up, the evaporator wall continues to rise, and steam is generated and enter to the condenser inlet, but the condenser inlet temperature fluctuates, indicating that after the vapor enters the condenser for cooling, it is hard to overcome the resistance to continue to flow, but is pressed back to the condenser inlet due to gravity. As the time increases, the amount of vapor increases, and it re-enters the condenser, and fluctuates a number of times until the evaporation rate of the vapor is greater than the condensation rate of the condenser. And the vapor outlet temperature drops and finally reaches equilibrium.

In order to determine the minimum stable starting power of the system, that is, the thermal load of the heating surface without temperature dip during the starting process, the system increases the thermal load on the basis of 20 W until the system achieves stable starting. The experimental results show that the temperature fluctuation of the system is small when the power continues to increase to 25 W. As shown in Fig. 8, the condenser inlet temperature rises suddenly at 330 s due to sufficient steam, and the temperature reaches about 20 s to reach the maximum temperature of 22.6 °C. During this period, the condenser outlet temperature rises steadily to 15.8 °C, after which the system bns to stabilize. It is apparent that there is a hysteresis in the condenser outlet temperature due to the vertical arrangement of the system condenser and the longer the condenser. Based on these, it was concluded that the start-up heat load of 25 W was sufficient to achieve complete working fluid circulation.

With the heat load increased, enough amount of steam was generated to clear the liquid in the vapor line into the condenser, while the temperature of the condenser outlet took a long time to increase due to the time required for the phase change and the liquid flow rate is less than the gas flow rate. Singh et al. [20] believed the start-up phenomenon of LHP involves satisfying two main conditions. (1)Clearing of liquid from evaporator grooves, vapor line and part of condenser; (2) Setting up required pressure difference across the wick that is necessary to circulate the working fluid around the loop.

One condition that must be met for the LHP to start operation is that there is sufficient temperature difference and pressure drop between the capillary evaporation surface and the compensation chamber. The pressure drop is equal to the sum of the pressure drops of the other parts of the system except the capillary wick.

From the point of view of the startup, the Δp_{cap} should not be used as a criterion because the loop has not yet been established. Steam needs to cross a certain pressure potential to the condenser to start condensing. The sufficiently high vapor pressure to ensure the superheated vapor entrance to the condenser was supposed to be the key factor to circulate the working fluid around the loop, which means:

$$\Delta p_v \geq \Delta p_{groove} + \Delta p_{vl} + \Delta p'_g + \Delta p_{cond} \tag{14}$$

where $\Delta p'_g = \rho_l g \Delta H$, which is different with $\Delta p_g = \rho_v g \Delta H$ that calculated in the process of stable operation.

In this experiment, the length and placement of the condenser determined that gravity pressure drop Δp_g and condensing pressure drop Δp_{cond} accounted for a large proportion of the total pressure drop.

Fig. 9 shows the temperature changes with time at 60 W and 80 W power. When the heat load is 60 W, the starting turning point is 60 s. At this time, the temperature rise rate of the heated surface

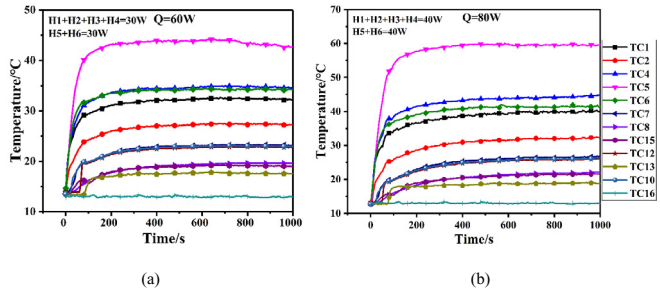


Fig. 9. LHP startup process at a heat load of 60 W (a) and 80 W (b).

decreases, and the condenser inlet temperature rises sharply, which can be considered as the time when the LHP starts to work. It is clearly shown that a very stable operating mode has been achieved under this thermal load and the fluctuations in the temperature profile are negligible. Heating surface temperature stabilized at 34.5 °C.

Fig. 10 shows the temperature dependence of LHP over time at a heat load of 100 W. The results show that under high heat load, LHP can be achieved smoothly, and the startup time is about 400 s. After the system enters a stable state, the heated surface temperature and condenser outlet temperature are stable at 48.9 °C and 26.8 °C. When the thermal load is 120 W. The temperature rise rate of the heated surface is up to 1.7 °C/s. After the system is stable, the condenser inlet temperature is 29.3 °C, and the temperature change at other locations is less than 0.2 °C, but the temperature at the condenser outlet is slightly fluctuating. To verify whether the phenomenon is unique under this heat load, continue to increase the heat load of system.

Fig. 11 shows the system startup temperature change at 140 W. The characteristics of the pre-launch period are similar to the previous power. It is worth noting that the condenser outlet has periodic temperature fluctuations after the startup is successful, and the fluctuation period is irregular. The fluctuation period is between 60 s and 120 s and the maximum amplitude is 3 °C. The comparison found that serrated fluctuations occurred under low load because the insufficient amount of vapor caused the gas-liquid phase transition in the condenser to move. In the experiment with the thermal load of 140 W, there is no serrated fluctuation of the heating surface. Conversely, the outlet temperature changes when the heating surface and the condenser inlet temperature are stable. It can be seen from the figure that the heating surface temperature is near 64 °C, the condenser inlet temperature is around 31.9 °C, but the outlet temperature fluctuates within the range of 24 °C–27 °C. It is considered that this is caused by the phase transition interface movement in the condenser under high load. At this time, the amount of vapor is large, the gas phase in the entire condenser tube occupies a large tube length, and the condensation rate is slightly insufficient relative to the evaporation rate, but it is still within the allowable working range.

Compared with starting at different loads, as the power increases, the starting process tends to be stable, and the temperature of the heated surface increases. The experimental results show that the whole startup process is about 6 min within 100 W, and the startup time is about 9 min after 100 W. The system was able to start-up at input power as low as 20 W, however the start-up time was long at such heat load.

At the same time, for every 20 W increase in heat load, the temperature of the heated surface rises by an average of 5 °C when the system is stable. Because of the direct heating of the heat source on the heated surface, the linearity of this relationship is high, and the condensation temperature can be observed. As the heat load increases, as shown in Fig. 12. The average temperature increment of the condenser is not uniform. This is mainly because the condensing temperature is the average value of the condensing inlet and outlet temperature, and the system does not have a linear relationship in the heat dissipation under various working conditions during the establishment of the equilibrium process. The thermal resistance changes at different powers, which are affected by the heat source, the environment, and the performance of the system itself.

At the low heat loads, the working fluid can be cooled by three fans, but the temperature of the condenser outlet was still higher than the ambient about 7 °C. At the high loads like 140 W, the difference could reach to 16 °C. In another paper [21], the temperature difference was near to 12 °C at the heat loads of 150 W by a standard 120 mm axial DC fan.

2.2. Power cycle tests

The LHP system was tested using different heat flux cycles to verify its operational reliability and transient response to changes in thermal load. Fig. 13 shows the temperature change and

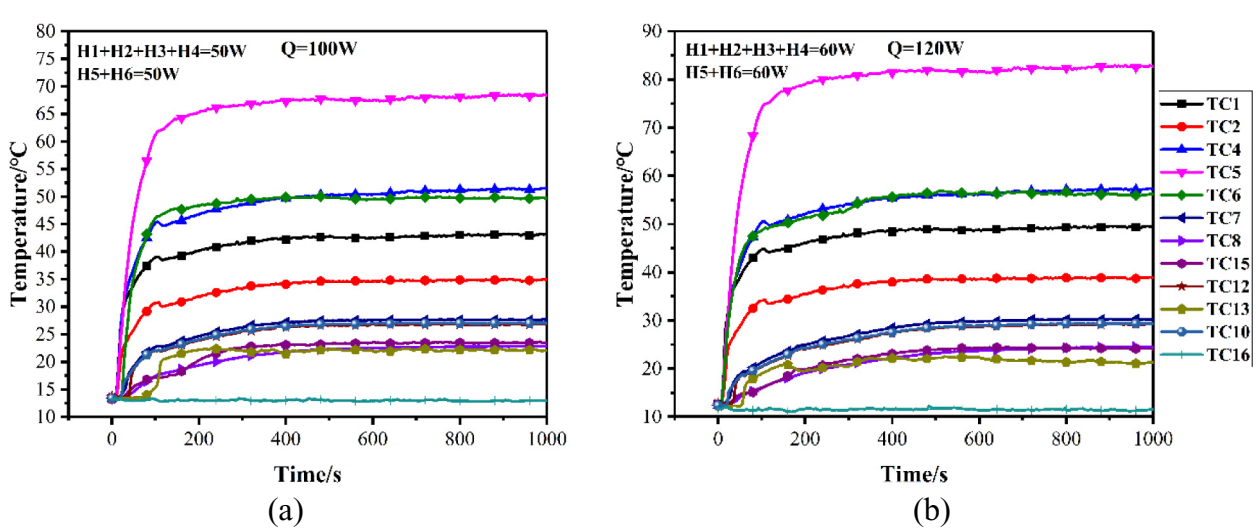


Fig. 10. LHP startup process at a heat load of 100 W (a) and 120 W (b).

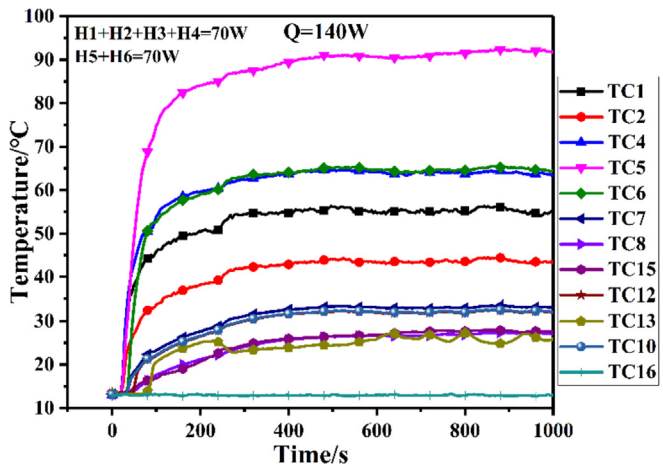


Fig. 11. Start-up process at 140 W.

operating characteristics of the LHP during the increase of the ceramic heating block from 1 W/cm^2 to 4 W/cm^2 in 0.5 W/cm^2 increments. The experimental results show that the system can respond quickly and establish a stable temperature working state under various heat flux, which indicates that the experimental system can run stably under variable load conditions and has good heat transfer performance.

When the heat flux density of the ceramic heating block is 2.5 W/cm^2, the comparison of the constant power start-up experiments shows that under the same high thermal load, the temperature at which the variable heat load operation reaches stability is higher than the temperature directly at the power, and the analysis considers that the system The initial conditions, including the distribution of temperature and working fluid, have a certain influence on the operation of the given heat load of the system. For the start of the fixed heat load, the initial temperature is close to the ambient temperature, and the working medium is all accumulated in the liquid in the area with lower potential energy. When the heat load is running, the flow state and temperature distribution of the working fluid in the previous stable operation are different from those at normal temperature, which causes the temperature difference of the system after establishing a new balance. This also

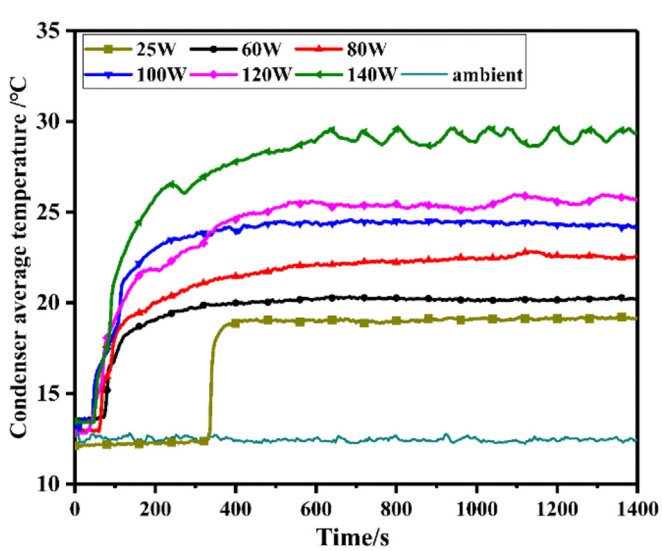


Fig. 12. The condenser average temperature under different heat loads.

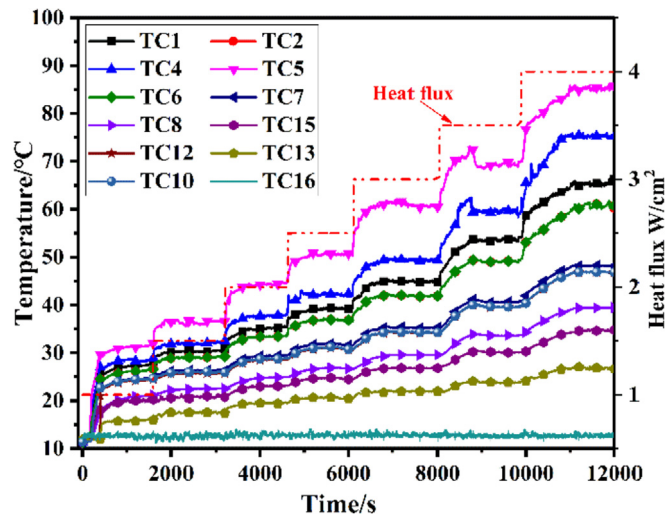


Fig. 13. Operation characteristics of the LHP with increase of heat flux from 1 W/cm^2 to 4 W/cm^2.

means that the stability point of the system is affected by the process. In practical applications, it is instructive to determine the thermal load corresponding to the allowable temperature of the system.

In practical engineering applications, cooling equipment is often encountered operating under different conditions. LHP with backside fins have good sensitivity to variable heat loads [22], and large heated surfaces also mean the system has the potential to cool larger heat load equipment.

2.3. Experiment under variable heat dissipation conditions

When a tube-type heat exchanger was used as condenser, the reliability of the system can be judged by changing the temperature of the cooling water and adding ribs or pins to investigate the influence of the condensation condition on the system [23]. For air forced convection cooling, it is difficult to achieve this, but it is possible to enhance the cooling of the system by changing the air volume of the condenser, for example, adjusting the wind speed and the number of fans. Considering that the fan is controlled by the computer CPU fan line, changing the wind speed is not easy to control. Therefore, the influence of different heat dissipation conditions on the system is explored by reducing the number of fans on the condensing end and adding the number of fans on the back side of the evaporator. As shown in Fig. 14, con-fan represents the amount of fan change on the condenser side, which varies according to the law of 3-2-1-2-3. Eva-fan represents the fan on the evaporator side, as shown in Fig. 14, a fan of the same power is added in about 3600 s.

It can be clearly seen from the experimental results that the heat dissipation on the evaporator side is very significant for the system cooling effect. The interval between each point in the figure is 400 s. The fan added at about 3600 s directly reduces the average temperature of the system by about 14 °C. The maximum temperature of the heated surface decreased from 75.7 °C to about 800 s to 61.3 °C, and the evaporator outlet and condenser inlet temperature changed from 47.9 °C to 33.6 °C, while the condenser outlet temperature changed to 5 °C. The change of the fan on the condenser side is direct and has a great influence on the temperature change of the outlet. It can be known from the formula $Q_{cond} = mc_p(T_{cond,in} - T_{cond,out}) = hA\Delta T$ that the decrease of the heat output of the cold end causes the outlet temperature to rise.

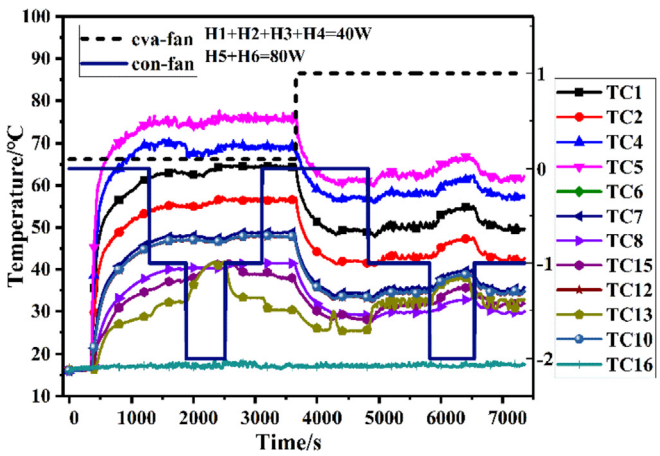


Fig. 14. Operation characteristics of the LHP under different heat dissipation conditions.

Under normal circumstances, reducing one condensing fan causes the outlet temperature to rise by about 5 °C.

It is worth noting that when the two fans are reduced, it will cause more severe Fluctuations, the outlet temperature rose from 32 °C to around 41 °C. Obviously, improving the heat dissipation condition on the evaporator side improves the reliability of the system. Therefore, for the design of the loop heat pipe under high heat load, the heat dissipation near the evaporation side can be strengthened and utilized as much as possible. For example, reducing the thermal resistance of contact between the heat source and the heated surface, and accelerating the forced convection of the heat source or the evaporator casing and air.

2.4. Thermal resistance

Thermal resistance is an indicator of the heat transfer capacity of a loop heat pipe system. The loop thermal resistance (R_{LHP}) is defined as the ratio of the evaporator wall temperature and the difference between the condenser and the heated load, indicating the ability of the heat to be absorbed by the evaporator and discharged to the outside through the condenser. The evaporator thermal resistance (R_{evap}) is the ratio of the difference between the evaporator wall temperature and the steam temperature at the evaporator outlet to the heated load, reflecting the effect of the evaporator's own heat capacity on the heat transfer of the entire system. They are calculated by following equations.

$$R_{LHP} = (T_e - T_c)/Q_a \qquad R_e = (T_e - T_{e,out})/Q_a \tag{15}$$

while T_e represents the average temperature of heating load, $T_{e,out}$ is the evaporator outlet temperature measured by T_{c10}. Q_a is the input power controlled by the voltage regulator. T_c is the average temperature of the condenser inlet and outlet which calculated by $T_c = (T_{c12}+T_{c13})/2$.

It is obvious that the thermal resistance of LHP is related to the heat capacity of the evaporator and the heat dissipation efficiency of the condenser [1,24]. Generally, the thermal resistance of the heat source and the evaporator can be reduced, and the gasification core of the surface of the capillary core can be increased to reduce the thermal resistance of the evaporator. It is also possible to design a more reasonable heat dissipation structure and cooling method on the condenser side for reducing the total thermal resistance of the system.

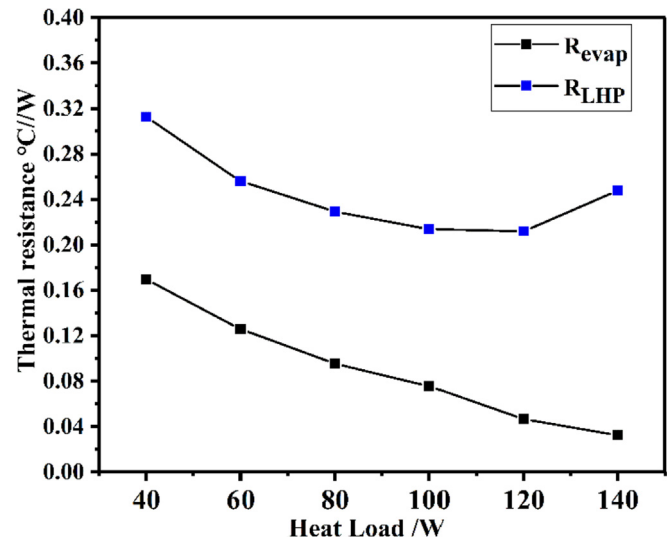


Fig. 15. Analysis of thermal resistance dependence of heat load.

As shown in Fig. 15, the R_{evap} of the LHP decreases with increasing heat load, the total thermal resistance of the system is first reduced and increased after 120 W. In this experiment, the total thermal resistance of the system is between 0.312 °C/W and 0.212 °C/W, and the thermal resistance of the evaporator is between 0.170 °C/W and 0.032 °C W, the minimum value of 0.032 °C/W for R_{evap} is achieved at the heat load of 120 W. At this time, the thermal resistance is already very small, and it is more research significance to improve the low thermal resistance working range of the system compared with the lower thermal resistance of the evaporator.

3. Conclusion

In the present study, experimental investigations were conducted to study the performance of a loop heat pipe with large flat evaporator under multi heat sources. The main conclusions can be itemized as follows:

1. The system can start and work between the heat load of 20 W–140 W and maintain the heating surface temperature below 90 °C. The system has zigzag fluctuations below 20 W, the condenser inlet temperature oscillates periodically due to insufficient steam power, and the system can start stably between 25 W and 140 W. But when the heat load is high, there is a periodic temperature fluctuation at the outlet of the condenser
2. The experimental results show that the system can respond quickly and establish a stable temperature working state under 1–4 W/cm^2 of the heating block.
3. Improve the heat dissipation on the evaporator side with the characteristics of fast response and obvious temperature drop. Some enhanced methods could apply such as, reducing the thermal resistance of contact between the heat source and the heated surface, and accelerating the forced convection of the heat source or the evaporator casing and air.
4. In this experiment, the total thermal resistance of the system is between 0.312 °C/W and 0.212 °C/W, and the thermal resistance of the evaporator range from 0.170 °C/W to 0.032 °C/W, the minimum value of 0.032 °C/W for R_{evap} was achieved at the heat load of 120 W.
5. The large flat LHP system designed to solve the multi-heat source problem has the advantages of simple system and good

coordination stability compared with the multi-evaporator system, but it should also be noted that it is important to improve the thermal uniformity of the heated surface. It is expected to use copper as the evaporator material or design a steam channel matching the heat source distribution for optimization.

Acknowledgment

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