EXPERIMENTAL STUDY ON COLLABORATIVE ENHANCEMENT OF LED HEAT DISSIPATION CHARACTERISTICS BY PULSATING HEAT PIPE AND HEAT PIPE

by

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The objective of this research is to experimentally evaluate the specific impact of a collaborative heat sink composed of gravity heat pipes (GHP) and pulsating heat pipes (PHP) on the thermal efficiency of LED light sources. The heat sink developed in this experiment is designed to improve the thermal management system, ensuring that LED operate within a safe temperature range, which is crucial as the performance of LED is directly affected by their junction temperature. An HP-PHP collaborative heat sink was employed in the experiment, where PHP served as heat dissipating fins to enhance its thermal performance, while HP handles the majority of the heat transfer tasks. The results showed that under forced convection conditions, the HP-PHP collaborative heat sink can increase the maximum thermal power capacity of LED to 192 W. The HP-PHP collaborative heat sink can reduce the substrate's temperature to below 70.5 °C in passive mode when the LED input power does not exceed 96 W. Additional experimental results show that the minimum thermal resistance of the collaborative heat sink is 0.19 K/W under natural-convection conditions, under forced convection conditions, this value drops to 0.15 K/W, which still lower than the non-collaborative heat sink. These results demonstrate that the contact thermal resistance between HP and PHP significantly enhances the thermal performance of the collaborative heat sink. Therefore, this collaborative type of heat sink is an effective method for cooling high power LED.

Key words*: PHP, HP-PHP collaborative heat sink, LED, GHP*

Introduction

The LED, as an energy-saving lighting source, offer numerous advantages over traditional lighting technologies, such as longer lifespan, high reliability, low energy consumption, quick response time, and environmental friendliness [1, 2]. Currently, the optical energy conversion efficiency of high power LED is approximately 15-20% [3], with the remaining 80% or more of the energy being converted into heat. During this process, the substantial heat generated causes an increase in chip temperature, and it is crucial to maintain the chip junction temperature within an appropriate range (usually below 120 °C) to ensure performance and

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lifespan. Various studies have indicated [4] that excessively high temperatures can lead to the yellowing of the epoxy resin (encapsulant material), a reduction in optical efficiency, a shift in emission wavelength, and ultimately a shortening of the device's lifespan, resulting in device failure. Therefore, it is particularly important to design a well-functioning thermal management system to control the temperature of high power LED chips.

In the field of research on cooling high heat flux devices, phase change cooling technology has gradually become a focus in recent years [5-7]. Heat pipe technology [8-13] as a part of phase change cooling strategies, is considered an efficient heat transfer component due to its outstanding heat transfer performance. This technology has been widely applied in the structures of various LED cooling systems. The current trend in LED cooling technology development primarily involves the use of passive cooling methods, such as integrated components combining heat pipes and metal fins [14].

Traditional coolers, on the other hand, have certain drawbacks. The most fundamental difficulty is the typical metal fins on the surface of the heat pipes comparatively low thermal conductivity. Due to this flaw, there will be a substantial temperature difference over the fin surfaces, increasing the internal thermal resistance of the cooler. Furthermore, to achieve appropriate cooling effects, additional metal fins are needed to enhance the cooling surface area, which increase the overall weight of the cooler and can potentially complicate installation. The combined impact of these variables may be highly detrimental to the overall cooling effectiveness of the cooler. Consequently, these flaws restrict the functionality of conventional coolers in high power, high thermal flux density applications and provide a chance to develop new, more effective cooling solutions.

This work proposes and builds a lantern-shaped heat pipe cooling structure through design and research. It is distinguished by combining two different types of heat pipes transfer components. Firstly, the primary heat transfer elements of this cooling structure is chosen to a GHP. Secondly, to enhance their cooling capacity, PHP are employed as cooling fins. The evaporative end of the GHP is deftly integrated into a flat-surfaced copper substrate in this design. The goal of this kind of design is to provide a reliable and effective cooling platform for LED light sources. Moreover, the complete system is integrated into a unified cooler, known as the HP-PHP collaborative heat sink, which includes the LED, copper substrate, PHP, and GHP.

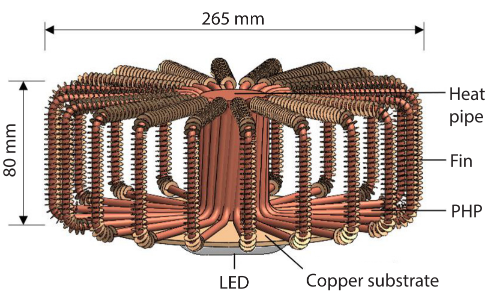

Figure 1. The HP-PHP collaborative heat sink structure

Experimental design and data analysis

Radiator design

Figure 1 illustrates the construction of the HP-PHP collaborative heat sink, whose components include an LED light source, a copper substrate, a cylindrical GHP, PHP, and spiral fins. In this work, a copper substrate is selected as the substrate base for the LED. The construction feature of the PHP is a copper tube bent into a 3-D spiral shape with both ends connected, along with an array of spiral fins located at the condensing end of the PHP. The purpose is to expand the cooling surface area and enhance cooling efficiency. The specific geometric parameters of the PHP are detailed in tab. 1.

The heat pipe is placed in a central position, serving as the core structure of the cooler, while also providing a stable mounting point for the PHP. The PHP, which is wrapped around the side surface of the heat pipe, is fixed in place with a throat hoop. Subsequently, this assembly is affixed to the LED substrate base using an aluminum clamping plate, forming a complete cooling structure. To optimize thermal conductivity and reduce heat loss, thermal grease is applied between the heat pipe, the PHP, and the copper substrate.

Table 1. Parameters of PHP

Parameter	Value	Parameter	Value
Evaporating end length	5200 mm	Material	Red copper
Condensing end length	2800 mm	Working fluid	Deionized water
Inner diameter	4 mm	Liquid filling rate	50%
Outer diameter	6 mm	Internal absolute pressure	0.022 Pa

Experimental set-up and measurements

The test system is shown in fig. 2, comprising a heating system, fan, data acquisition system, and the HP-PHP collaborative heat sink.

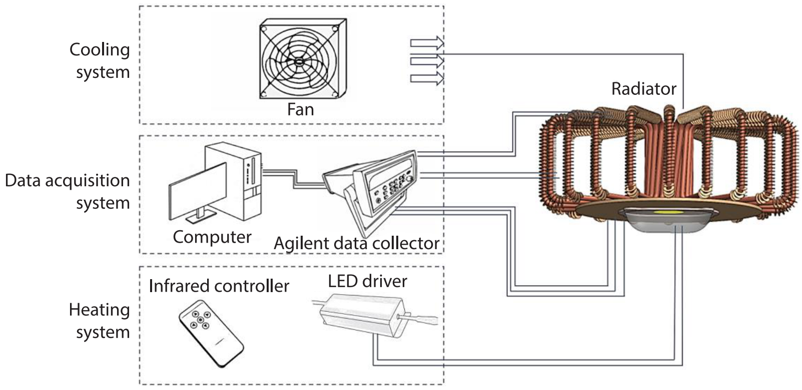

Figure 2. Experimental system

The output power of the LED light source is controlled by an adjustable driver operating in constant current mode, with a rated output power of up to 240 W. During each experimental cycle, the power of the LED driver is adjusted using an infrared controller, thereby precisely controlling the light output and thermal flux density of the LED chip. To assess the temperature response characteristics of the HP-PHP collaborative heat sink, multiple *K*-type thermocouples are directly connected to predetermined temperature measurement points to obtain relevant temperature data.

As shown in fig. 3, the lay-out of the thermocouples encompasses three key temperature measurement areas, which are marked as the LED substrate area (sequentially numbered as T1, T2, T3), the heat sink main body area (heat pipe evaporator end T4; heat pipe condenser end T5), and the heat sink fin area (PHP evaporator end T6, T7; PHP condenser end T8, T9). To process the electrical signals obtained from the thermocouples, an Agilent 34970A data acquisition system was selected (with an accuracy of 0.4%), and all measurement data were stored on a computer.

The temperature data of the LED at various input powers were obtained by adjusting the power of the LED driver. To further enhance the convective cooling efficiency of the heat sink, a fan cooling system with a power of 17 W was designed, and the air-flow velocity of the cooling fan was measured using a hot-wire anemometer. In the experimental process, the input power of the LED was gradually increased from 48 W to 192 W, with each increase of 24 W. When the heating power reached 120 W, to break through the upper limit of LED power consumption, an active cooling strategy with constant wind speed was adopted. To ensure the accuracy and reliability of the experimental data, multiple independent tests were conducted at each power level.

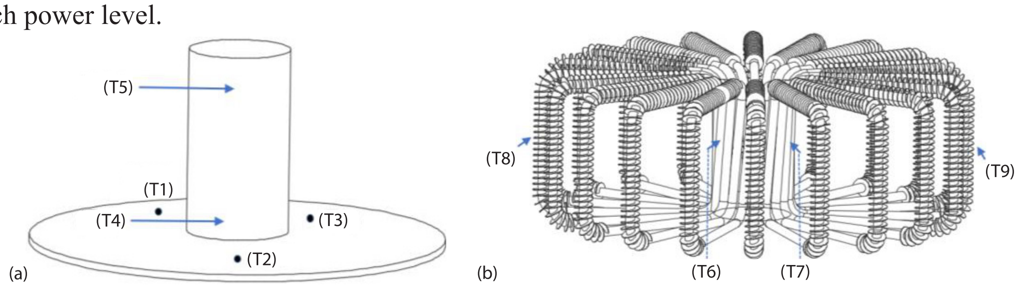

Figure 3. Location of the thermocouple; (a) substrate and heat pipie thermocouple mounting points and (b) PHP thermocouple mounting points

Experimental data processing

Within the domain of heat dissipation technology, thermal resistance is commonly regarded as a crucial measure for assessing the heat transfer efficiency of radiators. Theoretically, a lower thermal resistance indicates a stronger thermal conduction capability of the heat sink. In most applications, the thermal transfer process for assembling high power LED heat sinks can be distinctly divided into two main parts: the internal thermal resistance of the LED chip, as shown in fig. 4, and the thermal resistance of the heat sink itself. Based on this division, the thermal resistance analysis of the complete thermal transfer process is accordingly decomposed into two major parts: the thermal resistance of the LED and the thermal resistance of the heat sink. Considering that 80% of the electrical energy of a high power LED chip not converted into light accounts for the total input energy, which is then transferred in the form of heat to the heat sink substrate. Taking into account this specific thermal transfer path, the thermal resistance from the LED junction the heat sink substrate, R_1, is expressed:

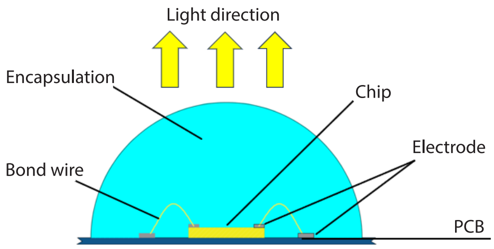

Figure 4. The COB LED internal structure

$$R_1 = \frac{t_d - \overline{t}_{pl}}{0.8Q} \tag{1}$$

where Q is the heating power input to the LED, t_d – the LED junction temperature, and $\overline{t}_{pl}$ the average temperature of the heat sink substrate, with the calculation formula:

$$\overline{t}_{pl} = \frac{t_1 + t_2 + t_3}{3} \tag{2}$$

In this work, the HP-PHP collaborative heat sink thermal transfer performance is the main focus. Therefore, this paper primarily calculates and analyzes the thermal resistance, R_2, which represents the thermal performance of the collaborative radiator itself, while temporarily disregarding R_1, which represents the internal thermal resistance of the LED. The formula for calculating R_2:

$$R_2 = \frac{\overline{t_{pl}} - t_a}{0.8Q} \tag{3}$$

where t_a represents the ambient temperature. It is a common phenomenon to encounter errors in experimental research. To account for measurement uncertainties in power and temperature, the thermal resistance estimate uncertainty is calculated:

$$\frac{\delta R}{R} = \sqrt{\left(\frac{\delta t_{pl}}{\overline{t_{pl}} - t_\alpha}\right)^2 + \left(\frac{\delta t_a}{\overline{t_{pl}} - t_a}\right)^2 + \left(\frac{\delta Q}{Q}\right)^2} \tag{4}$$

The measurement accuracy are listed in tab. 2.

Table 2. Accuracy of temperature and power

Physical quantities	Measurement methods	Range	Precision
Temperature	*K*-type thermocouple	Calibration range: 0~1300 °C	±0.1 °C
Power	Wattmeter	0~240 W	±1.25 W

Analysis of results

Temperature and thermal load measurement

The experiments were conducted in a constant temperature of 21-22 °C. Initially, the experiment set an input power of 48 W. To further explore the performance of the HP-PHP collaborative heat sink, the experimental design included two cooling modes: natural-convection cooling at low thermal loads and forced convection cooling at high thermal loads.

Figures 5 and 6 present detailed measurement data regarding temperature and thermal load. From these data, it can be observed that under forced convection conditions, there are significant fluctuations in the wall temperatures of both heat pipe and PHP. The primary cause of these oscillations is the high thermal load, which induces turbulence or unstable flow characteristics in the working fluid inside the heat pipes. Such instability of the fluid can trigger an uneven distribution of heat, thereby causing abrupt temperature fluctuations.

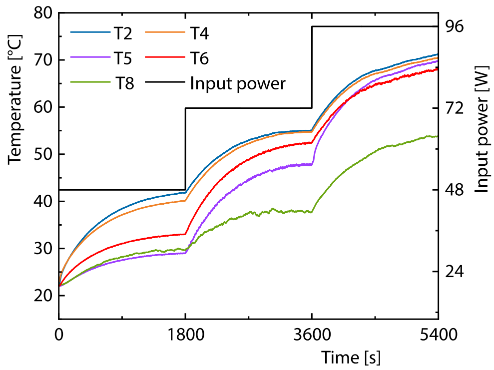

Figure 5. Temperature and heat load measurement under natural-convection conditions

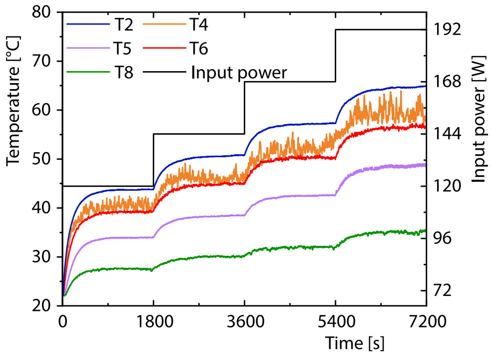

Figure 6. Temperature and heat load measurement under forced convection conditions

In a natural-convection environment, there is a certain correlation between the thermal power applied to the LED chip and the temperature of its radiator substrate. When the heating power is increased to 96 W, the substrate temperature, T2, of the LED radiator rises to

70.5 °C. If the thermal power continues to increase, the substrate temperature will approach the critical temperature for the safe operation of the LED (80 °C). However, under forced convection conditions, the maximum thermal power that the LED radiator can tolerate is significantly increased to 192 W, thereby effectively expanding the upper limit of LED power consumption. Compared to the results under natural-convection conditions, the power has doubled.

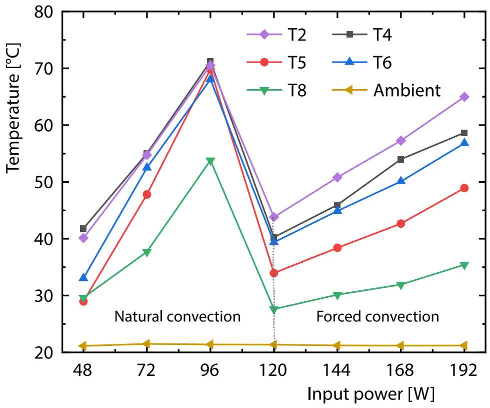

Figure 7. Steady-state temperature *vs.* heat load

The relationship between steady-state temperature and thermal load

The steady-state temperature is determined based on the average of the temperature data from the last ten minutes of the steady-state period. Figure 7 depicts the steady-state temperature curves for the LED substrate, and the evaporator and condenser ends of the heat pipe and PHP, in relation changes in thermal load under both natural and forced convection conditions. As the thermal load gradually increases, the wall temperature of the HP-PHP collaborative radiator also shows an upward trend. This is consistent with the basic principles of thermodynamics: as the thermal load increases, the temperature difference between the HP-PHP collaborative radiator surface and the environment must also increase to ensure that heat can be effectively dissipated into the surroundings.

The temperature curves of the three measurement points T2, T4, and T6 show very similar trends under the two cooling conditions, regardless of cooling air-flow intervention. This indicates a significant thermal conduction synergy between the LED substrate, heat pipe, and the evaporator end of PHP. With the increase of the thermal load, the wall temperature of the LED substrate, T2, shows an upward trend under both cooling conditions. Detailed data indicate that under natural-convection with a thermal load of 96 W, the wall temperature of the LED substrate is 70.5 °C. Under forced convection conditions corresponding to a thermal load of 192 W, the wall temperature of the LED substrate is 64.9 °C. According to conventional operating standards, the junction temperature of the LED should avoid exceeding 120 °C. Therefore, the temperature of the LED substrate should be maintained below 80 °C. This study demonstrates that by combining the synergistic action of GHP and PHP, it is indeed possible to ensure the stable and safe operation of high power LED lighting in high thermal load environments.

Figure 8 displays the steady-state temperature difference changes between the evaporator and condenser ends of the heat pipe and PHP. Under natural-convection conditions, as the heating power increases, the temperature difference $\Delta T_{4\text{-}5}$ between the evaporator and condenser ends of the heat pipe shows a gradually decreasing trend. When the thermal load reaches 96 W, this temperature difference is minimized to 1.4 °C, which reveals the superior performance of the GHP in terms of temperature uniformity. In contrast, the temperature difference $\Delta T_{6\text{-}8}$ between the evaporator and condenser ends of the PHP under this thermal load is 14.2 °C. The relatively large temperature difference of the PHP is mainly due to the increased temperature at the PHP evaporator end caused by the heat transferred from the GHP with good temperature uniformity. Under forced convection conditions, the temperature difference between the evaporator and condenser ends of the heat pipe and PHP rises with the increase in thermal load. This phenomenon may originate from local changes in the heat transfer coefficient on the heat pipe surface caused by the cooling wind, which leads to a decrease in the temperature at the condensing end and thus a gradual increase in the temperature difference. When the thermal load is 192 W, there is a notable decrease in the temperature difference between the evaporator and condenser ends of the heat pipe, which might be due to the heat pipe approaching its heat transfer limit. The velocity of the cooling wind can no longer further reduce the temperature at the condenser end of the GHP, resulting in a reduced temperature difference.

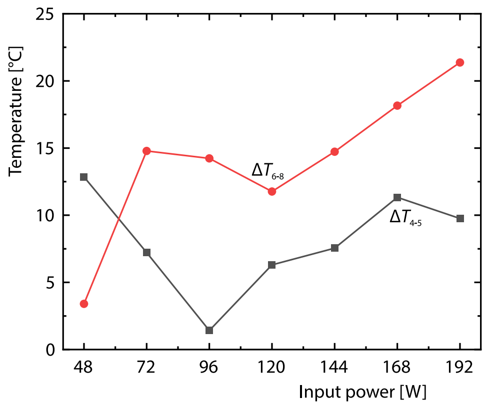

Figure 8. Steady-state temperature difference variation of HP and HPH

Xu *et al.* [15] investigated a unique design of a thermosiphon radiator, which includes rectangular radial fins. Specifically, the thermosiphon used was made of copper, consisting of a 60 mm long evaporator and a 180 mm long condenser. Fixed to the outside of the condenser were 20 rectangular radial fins, with geometric parameters: 2 mm thickness, 180 mm height, and 30 mm length. In addition, the height and diameter dimensions of the radiator base were 2 mm and 85.4 mm, respectively. In the experiments, with the ambient temperature maintained at 25 °C and the heat input gradually increasing from 10-50 W, the highest temperature measured at the bottom of the thermosiphon radiator increased from 43.1-73.4 °C.

In contrast, studies based on GHP cooling technology have found that using PHP as cooling fins attached to the outer surface of an heat pipe shows significant advantages in heat transfer efficiency. Especially under natural-convection conditions, when the continuous thermal load gradually increased from 48-96 W and the ambient temperature was stable at 22 °C, the measured temperature range of the HP-PHP collaborative radiator substrate was from 40-70.5 °C. Synthesizing the aforementioned data, it can be concluded that the HP-PHP collaborative radiator is more suitable for dealing with higher thermal load cooling requirements compared to thermosiphon radiators equipped with rectangular radial fins.

Thermal resistance

Figures 9 and 10 systematically reveal the detailed schematic structures of the PHP and heat pipe side surfaces in both connected and non-connected states. Because the PHP, serving as fins, is tightly connected to the side surface of the heat pipe even with the use of thermally conductive silicone grease-the inherent cylindrical design of the heat pipes still results in a limited actual contact area between the two heat pipes. This structural feature may introduce a higher contact thermal resistance. Based on this, in order to delve into the potential impact of this contact thermal resistance on the performance of the collaborative heat sink, this

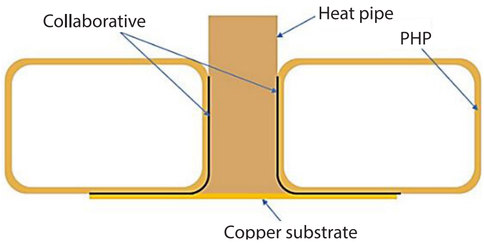

Figure 9. The HP-PHP collaborative heat sink

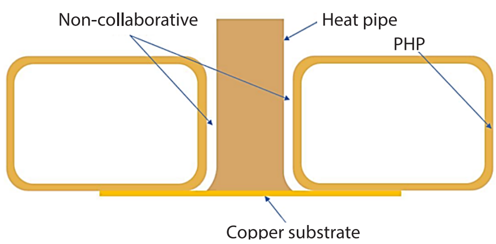

Figure 10. The HP-PHP non-collaborative heat sink

paper further studies a non-collaborative HP-PHP heat sink structure, as shown in fig. 10. This non-collaborative design not only eliminates the contact thermal resistance between the GHP and the PHP but also optimizes the heat dissipation area between the GHP, the PHP, and the ambient air. In the experimental comparative analysis, the cooling performance of these two different structures was specifically evaluated.

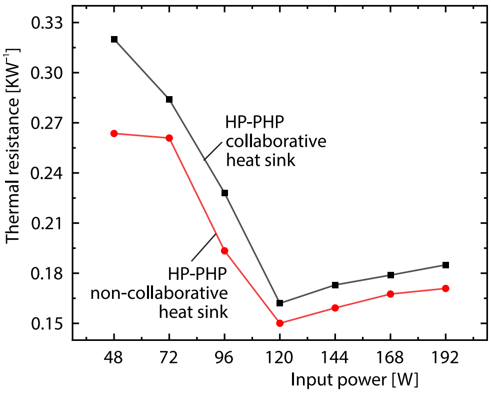

Figure 11. Thermal resistance between LED substrate and ambient air

The data analysis from fig. 11 can clearly identify that the thermal resistance of the collaborative heat sink is significantly lower than that of the non-collaborative heat sink. This observation strongly suggests that the contact thermal resistance has a positive effect on the performance of the collaborative heat sink. Furthermore, fig. 11 also reveals the correlation between the thermal resistance of the heat sink and the thermal load under different convective conditions. Specifically, under natural-convection conditions, as the thermal load increases, the thermal resistance of the HP-PHP collaborative heat sink shows a downward trend. The root of this phenomenon is that as the heating power gradually increases, the amount of heat released by the LED chip located on the collaborative heat sink correspondingly increases, causing the surface temperature of the heat sink to rise. Consequently, the temperature of the air near the heat sink increases, and its density decreases accordingly, which then promotes the buoyant hot air to engage in intense convection with the surrounding cooler air. This further enhances the natural-convection heat transfer coefficient, which is related to the Grashof number:

$$\mathrm{Gr} = \frac{g a_v \Delta t l^3}{\nu^2} \tag{5}$$

Under forced convection conditions, the thermal resistance tends to increase with the increase of heating power. The key reason behind this phenomenon lies in the fact that, in forced convection, the air-flow speed generated by the fan is relatively constant, and when the heating power is increased to a certain extent, this constant flow speed may not be sufficient to meet the increased heat transfer demand, resulting in a relative increase in thermal resistance. To provide a specific quantitative description, the lowest thermal resistance of the HP-PHP collaborative heat sink under natural-convection conditions is 0.19 K/W, while under forced convection conditions, its lowest thermal resistance further decreases to 0.15 K/W.

Conclusions

This experimental work showcased and verified a unique HP-PHP collaborative heat sink for cooling high power LED, importantly monitoring the temperatures of several important heat sink components were monitored at various LED input powers in order to better investigate its thermal performance. The following key conclusions were drawn.

- Under forced convection conditions, the HP-PHP collaborative heat sink allowed the maximum permissible thermal power to be increased to 192 W, thereby effectively expanding the upper limit of LED power consumption. Compared with the results under natural-convection conditions, the power was doubled.

- Under conditions where the LED input power is less than 96 W, the HP-PHP collaborative heat sink can maintain the temperature of the LED substrate below 70.5 °C in passive cooling mode. Compared to thermosiphon radiators with traditional rectangular radial fin designs, this HP-PHP collaborative heat sink exhibited superior cooling performance for relatively higher heat loads.
- As the LED power consumption gradually increased, the thermal resistance of the LED substrate relative to the environment gradually decreased under natural-convection conditions. The lowest thermal resistance measured for this collaborative heat sink under natural-convection conditions was 0.19 K/W. Under forced convection conditions, the thermal resistance further decreased to 0.15 K/W.
- Compared with the non-collaborative heat sink, the collaborative heat sink had significantly lower thermal resistance. This experimental data confirmed that the contact thermal resistance between the GHP and the PHP played a positive role in enhancing the thermal performance of the HP-PHP collaborative heat sink.

Acknowledgment

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Nomenclature

a_v – thermal expansion coefficient, [K^{-1}]
Gr – Grashof number ($=g\alpha_v \Delta t l^3/\nu^2$), [–]
g – gravity acceleration, [ms^{-2}]
Q – heat load, [W]
R – thermal resistance, [°CW^{-1}]
t – temperature, [°C]
t_a – ambient temperature, [°C]
t_d – LED junction temperature, [°C]
t_{pl} – average substrate temperature, [°C]

Greek symbols

δ – uncertainty
ν – kinematic viscosity, [m^{2}s^{-1}]

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