The 2013 World Congress on
Advances in Nano, Biomechanics, Robotics, and Energy Research (ANBRE13)
Seoul, Korea, August 25-28, 2013

Chimney Effect on the Fluid Flow and Heat Transfer Characteristics of Finned Heat Sink for LED lamp

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ABSTRACT

Using heat sink to increase the cooling area is the most familiar and fundamental method in LED heat management technology. We develop a new configuration of finned heat sink for LED lamp. It is a circular cylinder with many plate fins radially arranged on the external surface. There are many straight ducts inside the peripheral wall of the cylinder. These ducts form the chimney effect to enhance the heat transfer. This work experimentally investigated the chimney effect on the fluid flow and heat transfer characteristics of the finned heat sink for the LED lamp under the free convection state. There are four kinds of test specimens made of aluminum alloy to be employed. The results show that the chimney holes seemed to strengthen the cooling performance at the smaller temperature difference. Besides, the heat transfer enhancement would be reduced when the height of the heat sink was decreased.
Keywords: chimney effect, free convection, finned heat sink, experiment

1. INTRODUCTION

Due to the increase of the dissipated heat and the decrease of the dimension, the average output heat of the chip has expanded to be 250W/cm2 at 2011. It demonstrates the light, thin, short, small and multi-function trend of the electric products. Some engineers even predict that, till 2015, the operating temperature of the chip will reach the surface temperature of the sun. Therefore, the demand of the thermal control for the

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electric products is becoming more and more serious.

For the high-power LED lamp, the challenge of the cooling demand is not less than that of the chip. Only about 15 percent of the total input electric power is transformed to be light energy; the remaining 85 percent of the input electric power will be dissipated to the ambient. If a poor cooling design is used, the LED will be destroyed due to the heat accumulation. In order to conduct the heat rapidly, the present commercial LED lamps usually employ the circular-cylinder alloy heat sink with many plate fins radially arranged on the external surface. Some new design should be provided to promote the present finned heat sink to satisfy the serious cooling demand.

There are many studies concerning about the free convection heat transfer of the finned heat sink. Welling and Woolbridge [1] experimentally investigated the heat transfer of fixed-length plate fins installed vertically on the LED base plate. They found that the optimal fin heights for various spaces between fins. Mendez and Trevino [2] numerically explored the steady-state free convection heat transfer of a vertical plate. Maranzana et al. [3] experimentally studied the transient free convection of the vertical plate. Polidori and Padet [4] experimentally investigated the transient free convection of the vertical plate with transverse ribs. They found that the vortices reduced the local heat transfer at the ribbed regions. Gianfrini et al. [5] numerically investigated the free convection heat transfer between two heated horizontal circular cylinders. They indicated that the heat transfer performance of the bottom cylinder fell when cylinders were quite close. However, the heat transfer performance of the upper cylinder would be strengthened when the distance between both cylinders was reduced further. Mobedi and Sunden [6] numerically simulated the coupled steady-state heat transfer behavior of the thermal conduction and the free convection for the vertical plate fin with local heat source. They demonstrated that there were optimal positions of local heat source to obtain the maximum total heat transfer for various Prantle and Biot numbers. Kasayapanand and Kiatsiriroat [7] numerically studied the electrophoresis effect on the free convection between two parallel and vertical plates. They indicated that the electrophoresis effect was more obvious at smaller Reynolds number. Yilmaz and Fraser [8] explored the turbulent characteristics between both parallel and vertical plates with asymmetrically heating. Haldar [9] numerically investigated the free convection heat transfer characteristics of the pin fins installed vertically on the horizontal base plate. They found that the fluid flowed horizontally toward the pin fins from the distant place firstly, then flowed up vertically along the pin fins, and finally left away.

This work experimentally investigated the chimney-hole effect on the free convection heat transfer of the finned heat sink. The test specimens were made of aluminum alloy. There were four kinds of test-specimen configurations. The major objective is to explore the chimney-hole effect on the free convection of the finned heat sinks under various heating temperatures.

2. Experimental Method

2.1 Experimental setup

The experimental setup was shown as Fig. 1, including (1) the test specimens, (2) the test section and heating system, (3) the data-acquiring system, (4) the smoke generator, (5) the laser-light generator, (6) the infrared thermal-image camera, and (7) the personal computer. The test specimens were four kinds of finned heat sinks made of aluminum alloy. They were: (1) the vertical and parallel plate-fin heat sinks without/with a chimney hole (Model A / Model B), (2) the low, vertical and radial plate-fin heat sinks without/with chimney holes (Model C / Model D), and (3) the high, vertical and radial plate-fin heat sinks without/with chimney holes (Model E / Model F). Figure 2 shows the configurations and dimensions of the heat sinks with chimney holes (i.e. Models B, D and F). Besides, two kinds of test sections made of Bakelite were employed. One was used for the heat sinks of Models A and B, and the other was used for those of Models C, D, E and F. Film heaters, which were heated by the DC power supply, were adhered to the surface of the test sections. The finned heat sink was adhered to the film heater using the highly-conductive thermal grease (OMEGABOND 200, k=1.385W/m°C). The Bakelite test section had a very low conductivity and could reduce heat loss from the film heater. Many thermocouples are uniformly embedded in the test sections to measure the temperatures of the heated walls of the finned heat sinks. The dimensions of test sections and positions of thermocouples were shown in Fig. 3. All thermocouples connected to a YOKOGAWA MX-100 data recorder. Another thermocouple was used to measure the ambient temperature. In order to avoid bothering by the ambient, all tests were performed in a 400mm×400mm×300mm acrylic box. The system was assumed to be at the steady state condition when the temperature variation was within 0.2 °C in 30 minutes.

2.2 Data reduction and uncertainty analysis

The present study measured the thermal resistances of the finned heat sinks without/with chimney holes under free convection state. The definition of the thermal resistance was expressed as follows.

$$\mathrm{R} = \frac{T_w - T_0}{Q_t - Q_{Loss}}, \tag{1}$$

where Tw was the weighted average temperature at the bottom of the finned heat sink; T0 was the ambient temperature in the acrylic box; Qc represents the total input heat (Qt=I·V, I was the electric current and V was the electric voltage); and QLoss was the heat loss. The heat loss (QLoss) can be measured at the plate system without any heat sink. For such condition, the total input heat (Qt) is divided into two parts: (1) the convective heat (Qplate) directly from the heated surface to the above air, and (2) the heat loss (QLoss) from the heated surface to the ambience through the Bakelite test section.

$$\mathrm{Q_{Loss} = Q_t - Q_{plate} = V \cdot I - h_{plate} \cdot A(T_w - T_0) = h_{Loss} \cdot A(T_w - T_0)}, \tag{2}$$

The hplate is the heat transfer coefficient of the nature convection from the heated plate surface to the above air. The empirical formula of hplate is obtained from Ellison [10].

$$h_{plate} = 1.361 \cdot \left(\frac{T_w - T_\infty}{D/4}\right)^{0.25}, \quad (3)$$

In the heat-loss tests, different values of Qt result in various (Tw-T0). Using the results of the heat-loss tests and Eqs. (2)-(3), the heat loss (QLoss) as a function of (Tw-T0) can be obtained. Also, the thermal resistance (R) in Eq. (1) can be derived.

The standard single-sample uncertainty analysis, as recommended by Moffat [11], was performed. Data supplied by the manufacturer of the instrumentation states that the measurement of flow velocity had a 1% error. The uncertainty in the measured temperature was ±0.2°C. The experimental data herein reveal that the uncertainty in the thermal resistance (R) was 6.8%.

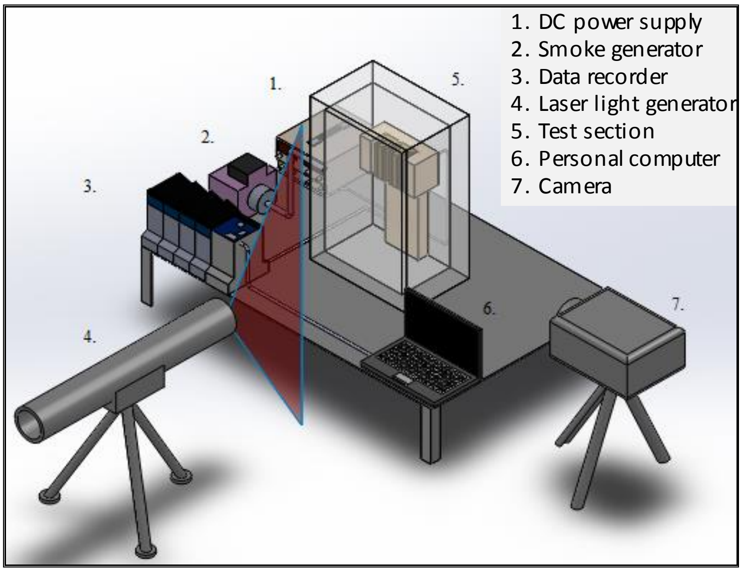
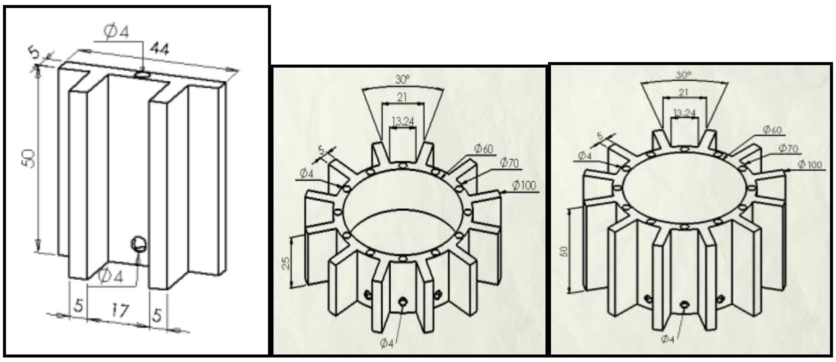
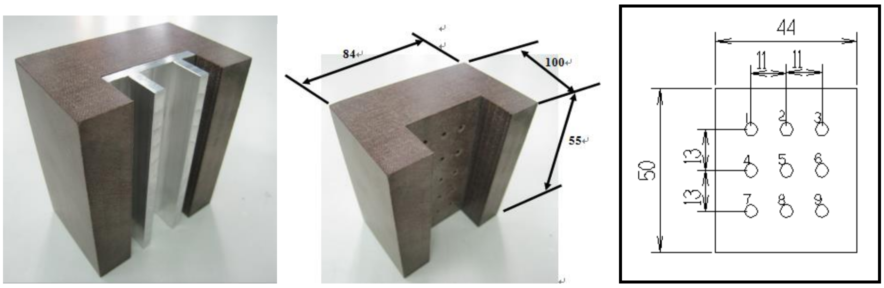


Fig. 1 Experimental setup

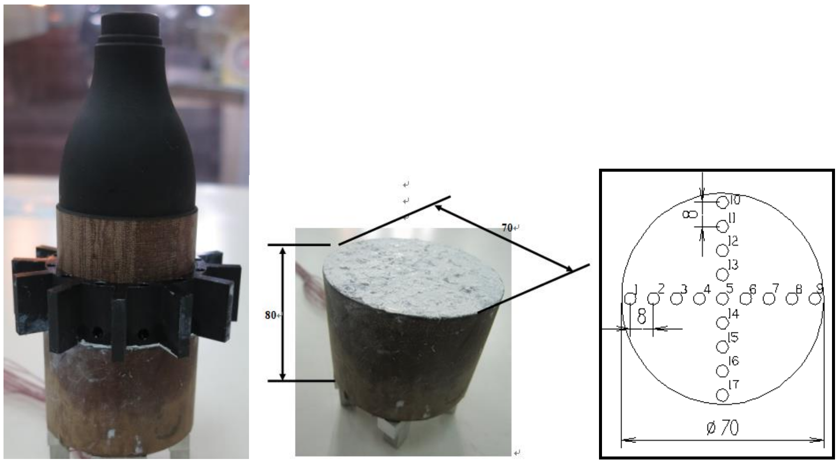


(a) Model B (b) Model D (c) Model F

Fig. 2 Configurations and dimensions of heat sinks with chimney holes



(a) Models A and B



(b) Models C, D, E and F

Fig. 3 Dimensions of test sections and positions of thermocouples

3.Results and Discussion

Figure 4 displays the relationship between the thermal resistance (R) and the temperature difference (ΔT=Tw-T0). The results indicate that the thermal resistance of the Model A decreased when the temperature difference increased. It meets the heat transfer characteristics of the free convection. When the temperature difference is bigger, the effect of fins heating the air within the thermal boundary is more obvious. Therefore, it results in better buoyancy effect and smaller thermal resistance. However, the thermal resistance of Model B was only 20℃/W at ΔT=22℃. It was much lower than 48℃/W of Model A. When the temperature difference increased till 30℃, the thermal resistance of Model B became 64℃/W sharply. When the temperature difference increased to be 37℃, the thermal resistance of Model B dropped to be 23℃/W. These results were verified by repeated tests. One can find that, at smaller temperature difference, the Model B with a chimney hole had a much better heat-transfer performance than the Model A did. But at the medium temperature difference, the heat transfer performance of Model B would become much worse. At the bigger temperature difference, Model B and Model A had similar heat transfer performance. The results seem indicate that, for the vertical and parallel plate-fin heat sinks, the chimney holes strengthened the cooling performance at the smaller temperature difference. It should be owing to the effect of two heat-transfer paths in the vertical and parallel plate-fin heat sink with a chimney hole. Figures 5 and 6 are the Infrared photos of Model A and Model B heat sinks, respectively. For various temperature differences, all the infrared photos indicate that the surface temperatures of the vertical fins were lower than those of the spreader.

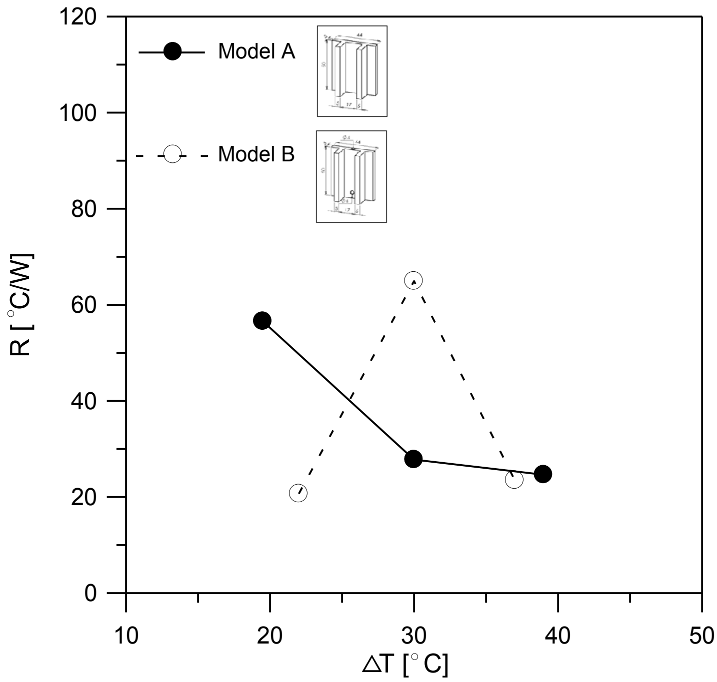
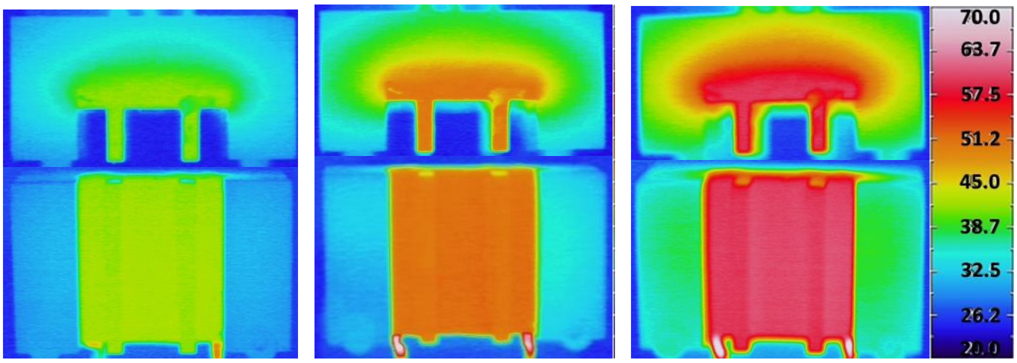
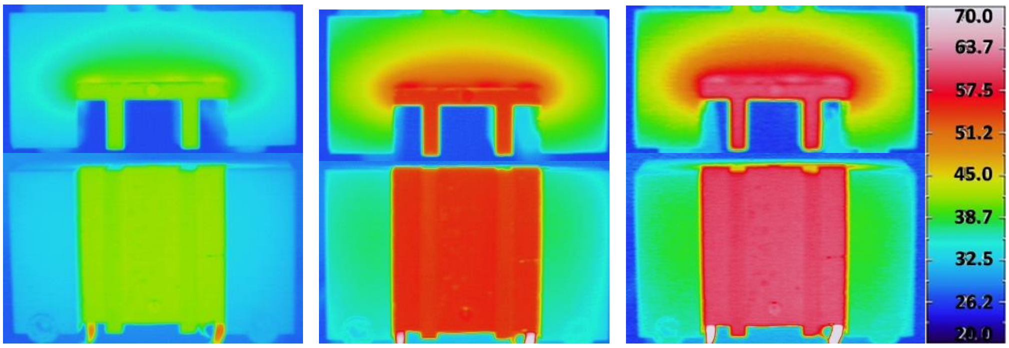


Fig. 4 Thermal resistance as a function of temperature difference for Models A and B



(a) ΔT=20℃ (b) ΔT=30℃ (c) ΔT=39℃

Fig. 5 Infrared photos of Model A heat sink



(a) ΔT=22℃ (b) ΔT=30℃ (c) ΔT=37℃

Fig. 6 Infrared photos of Model B heat sink

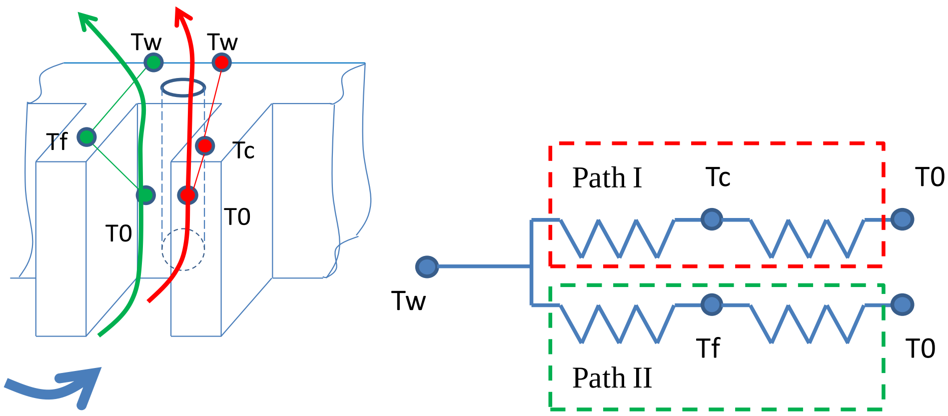


Fig. 7 Thermal network paths for Model B

The thermal network paths for Model B were shown in Fig. 7. The Model B had both heat-transfer paths (i.e. path I and II), but the Model A had only one heat-transfer path (i.e. path II). At ΔT=22℃, the conduction path of the heat-transfer path II is too long to build a sufficient buoyancy between fins to drive air upward. It made that Model A had poor heat transfer. For Model B, the heat-transfer path I could build a good free convection in the chimney duct since that it was very near the heated wall. Therefore, the heat-transfer performance of Model B was better than that of Model A at ΔT=22℃. When the temperature difference was raised to be 30℃, both heat-transfer paths of Model B had the similar buoyancy to drive air upward. The total convective air was divided into two parts to flow into the chimney duct and the passage between fins. It made the convective air in both heat-transfer paths too small to carry sufficient amount of heat away. When the temperature difference was large (i.e. ΔT=37℃), the buoyancy in both heat-transfer paths would be large enough. It would make sufficient amount of convective air and then reduce the thermal resistance again.

Figures 8 and 9 depict the thermal resistance of the vertical and radial plate-fin heat sinks without/with chimney holes as a function of the temperature difference. The results indicate that the total free convection heat transfer performances of low heat sinks (Models C and D) were smaller than those of high heat sinks (Models E and F), especially for the test cases with high temperature differences. It is because that the high heat sinks had much more total extended surface area. The other reason should be owing to the longer chimney duct of high heat sinks, leading to stronger free convection. Besides, the chimney holes did enhance the total free convection for the high, vertical and radial plate-fin heat sink at the operating state of medium temperature difference (ΔT=25-30℃). Figure 10 shows the Infrared photos of the vertical and radial plate-fin heat sinks at ΔT=25-35℃. The higher surface temperatures of the heat sinks demonstrate the higher thermal resistances since the amounts of the input electric heat powers were similar.

4. Conclusions

This work experimentally investigated the chimney effect on the fluid flow and heat transfer characteristics of the finned heat sink for the LED lamp under the free convection state. There are four kinds of test specimens made of aluminum alloy to be employed. They were: (1) the vertical and parallel plate-fin heat sinks without/with a chimney hole (Model A / Model B), (2) the low, vertical and radial plate-fin heat sinks without/with chimney holes (Model C / Model D), and (3) the high, vertical and radial plate-fin heat sinks without/with chimney holes (Model E / Model F). The results show that the chimney holes seemed to strengthen the cooling performance at the smaller temperature difference. Besides, the heat transfer enhancement would be reduced when the height of the heat sink was decreased.

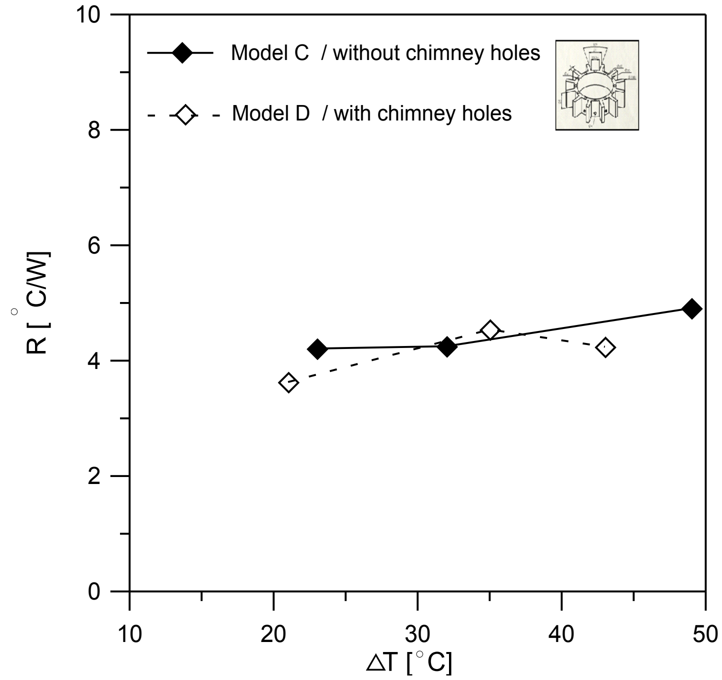


Fig. 8 Thermal resistance as a function of temperature difference for Models C and D

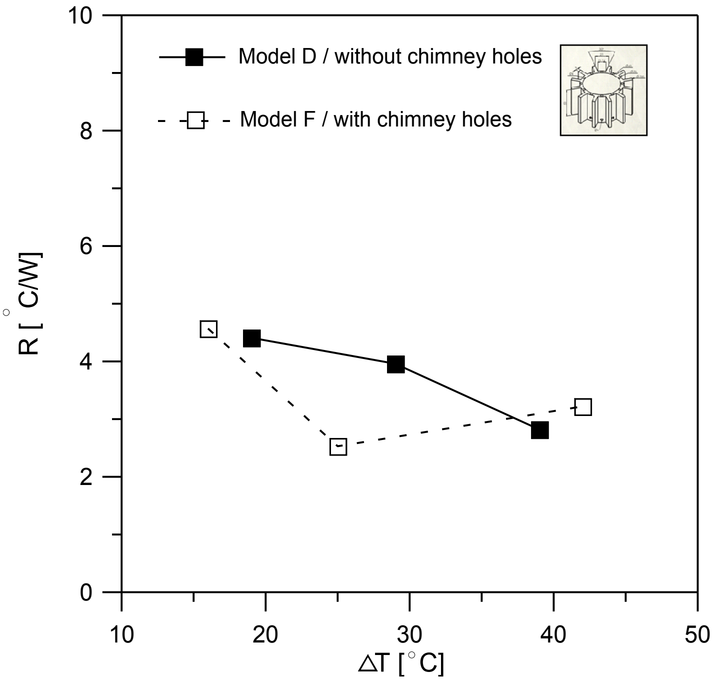
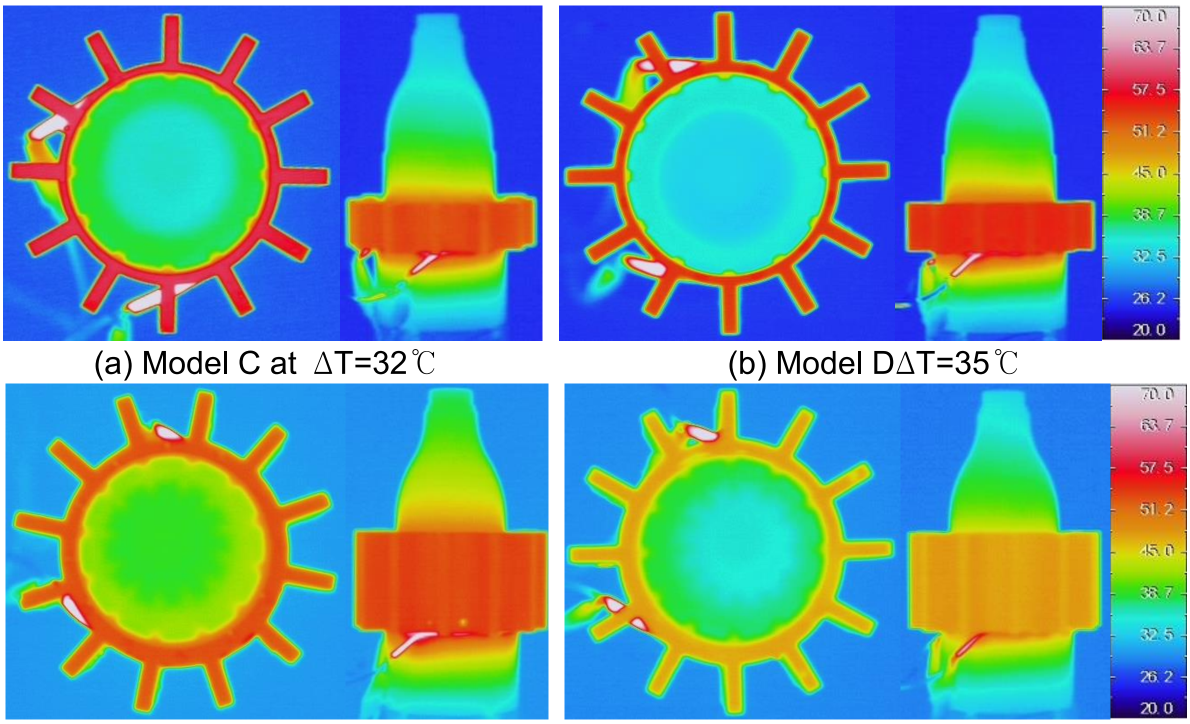


Fig. 9 Thermal resistance as a function of temperature difference for Models E and F



(a) Model C at ΔT=32℃ (b) Model DΔT=35℃

(c) Model E at ΔT=29℃ (d) Model FΔT=25℃

Fig. 10 Infrared photos of the vertical and radial plate-fin heat sinks

Acknowledgements

The authors would like to thank the National Science Council of the Republic of China for financially supporting this research under Contract No. NSC-100-2632-E-270-001-MY3.

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