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Orientation effect on heat transfer of a shrouded LED backlight panel with a plate-fin array ☆

Jin-Cherng Shyu [a], Keng-Wei Hsu [a], Kai-Shing Yang [b], Chi-Chuan Wang [c,*]

[a] Department of Mechanical Engineering, National Kaohsiung University of Applied Sciences, Kaohsiung 80778, Taiwan
[b] Green Energy and Environment Research Laboratories, Industrial Technology Research Institute, Hsinchu 31040, Taiwan
[c] Department of Mechanical Engineering, National Chiao Tung University, Hsinchu 30010, Taiwan

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ABSTRACT

This study reports thermal performance of a shrouded 348 mm × 558 mm aluminum plate-fin heat sink subject to various input powers and orientations. Effects of clearance (C) and the orientation on the heat transfer of the heat sink were investigated. Results show that the clearance effect is detectable only in a "window region" between 5 mm and 10 mm where an appreciable rise of heat transfer coefficient is encountered. As the tilted angle (θ) of the LED panel is increased, the heat transfer coefficient is reduced and the clearance effect on heat transfer becomes more pronounced. The heat transfer coefficients are similar between two cases in which the tilted angles of the LED panel are supplementary irrespective of clearance and input power. Except the cases of a horizontal heat sink, heat transfer coefficient of the shrouded heat sink having a fin array facing downward is usually slightly higher than that having supplementary tilted angle.

1. Introduction

Among numerous effective ways to remove heat from LEDs, passive means employing heat sink is still the most preferable method in the mainstream applications due to its noiseless, reliable, and cost-effective operation. Considerable researches concerning natural convective heat transfer of a plate-fin heat sink with various geometric parameters were carried out [e.g., 1–4]. Some studies concerning the shroud effect on the thermal performance of a fin array [5–7] had also been performed. Fujii and Imura [8] proposed correlations for the prediction of Nusselt number of two flat plates based on Reynolds numbers and the Rayleigh numbers at various orientations. Starner and McManus [9] showed that vertical heat sink showed the best thermal performance among all orientations. Because the interaction between thermal boundary layers develop over the plate-fin array, a vertically-orientated heat sink having larger fin spacing yielded better thermal performance. An investigation [10] on the natural convection heat transfer of a plate-fin heat sink facing downward with a tilted angle ranging from 0° to 30° showed that heat transfer was enhanced with a tilted heat sink due to the removal of stagnation point of the induced air flow passing over the plate-fin array.

The foregoing studies were conducted for uniform heat sources subject to natural convection. However, LED array used in large-size LED TVs is normally enclosed in a slim housing, and the installation of a LED panel may be subject to inclination due to space constraint or application needs. Therefore, this study aims to experimentally examine both effects of orientation and the clearance between the shroud and the fin tip of the heat sink on the heat transfer of a plate-fin heat sink attached to a LED array.

2. Experimental apparatus and data reduction

The experimental setup consists of a 558 mm × 348 mm MCPCB having 270 evenly distributed 1-W LEDs enclosed by an acrylic housing, a power supply and a power meter to light up LEDs, an aluminum plate-fin array screwed to the backside of the LED panel and an environmental chamber, as well as a data acquisition unit to record temperatures transmitted from thermocouples. More detailed contents concerning the test facility is referred in a previous study [11]. The clearance (C) and the orientation (θ) is depicted in Fig. 1. Since the overall surface efficiency of the present heat sink was about unity, the average heat transfer coefficient, $\overline{h}$, was estimated by

$$q = \overline{h}A(T_s - T_a) \tag{1}$$

where A is the total surface area of the heat sink, and T_s and T_a are the average temperature of the seven longitudinal temperatures in the panel and the ambient temperature in the chamber, respectively. The heat generated by those LEDs, q, is estimated as the 75% of the total power input [12], Q, and ranges from 140 W to 230 W. The uncertainty of the present heat transfer coefficients [13] ranges from 1.88% to 2.88%. The experimental conditions of the present study are shown in Table 1.

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* Corresponding author at: Department of Mechanical Engineering, National Chiao Tung University, No. 1001, Ta Hsueh Road, Hsinchu 30010, Taiwan.
E-mail address: ccwang@mail.nctu.edu.tw (C.-C. Wang).

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Nomenclature

A	Heat transfer surface area (m^2)
C	Distance between fin tip and acrylic housing (m)
H	Fin length (m)
$\overline{h}$	Average convective heat transfer coefficient (W/m^2K)
Q	Total input power to LED panel (W)
q	Heat transfer rate from the LED array panel (W)
T_a	Ambient temperature of the environment (K)
T_s	Average vertical temperatures of the heat sink (K)

Greek symbol

θ	Orientation of the LED backlight panel (°)

3. Results and discussion

Fig. 2 shows the variations of heat transfer coefficient of a 120°-tilted heat sink subject to clearance, C, at various input powers, Q. It is seen that the clearance effect is negligible as C is larger than 10 mm. When the shrouded heat sink is tilted with an angle, the shroud imposes frictional resistance on the rising air flow. At the same time, the buoyancy force for the rising air has to be multiplied by a factor of cos(θ-90°) due to the tilted angle. As a result, both the abovementioned effects lead to a smaller heat transfer coefficient than the case of $\theta=90°$ for the shrouded heat sink as shown in Fig. 2. As the LED panel is further tilted with a larger angle, the heat transfer coefficient of the shrouded heat sink with an orientation of 30° for various clearances and input powers is shown in Fig. 3. It shows that the heat transfer coefficient is decreased further and the clearance effect on the heat transfer coefficient becomes more pronounced than the results shown in Fig. 2. With the increase of the clearance, the heat transfer coefficient gradually increases, except the particular condition with $C=0$ in Fig. 3. In addition, the heat transfer

Table 1
Present experimental conditions.

LED backlight panel size (mm): 558×348	
Aluminum plate-fin heat sink	
Fin length: 558 mm	Fin spacing: 9.33 mm
Fin thickness: 1 mm	Fin height: 10 mm
Tested conditions	
Ambient temperature, T_a (°C)	30
LED Power, Q (W)	140, 170, 200, 230
Tilted angle, θ (°)	0, 30, 60, 90 120, 150, 180
Clearance, C (mm)	0, 5, 10, 15, 20, ∞

coefficient for heat sink with clearance of 10 mm in Fig. 3 seems approximately the same as the input power is increased from 170 W to 200 W. For the 30°-tilted shrouded heat sink, most of the induced hot air moves towards the acrylic shroud along the clearance between fin tip and shroud rather than the interfin region. This is because the plate-fin array is basically facing upward. Therefore, for a shrouded heat sink with a clearance of 10 mm, the flow passage is so small that the heat transfer enhancement due to input power is not so evident. However, the heat transfer coefficients measured at $C=0$ mm, denoted as 0 mm in Figs. 2 and 3, are always higher than those measured at $C=5$ mm at various input powers.

Fig. 4 shows the effect of tilted angle, θ, on heat transfer coefficient subject to clearance with different input powers. For a shrouded heat sink with an orientation between 60° and 120°, the heat transfer coefficient remains virtually unchanged or very slightly decreased when the shroud clearance, C, is less than 5 mm. A further increase of clearance from 5 mm leads to a gradual increase of the heat transfer coefficient and it peaks at $C=15$ mm. Above $C=15$ mm, the heat transfer coefficient of the heat sink remains almost unchanged. The results suggest that the existence of a shroud above a heat sink with a slight tilted angle ($60°\le\theta\le120°$) may improve or impair the heat transfer depending on the shroud clearance. With a larger tilted angle of 30° or 150°, the heat transfer coefficients are inferior to those measured with

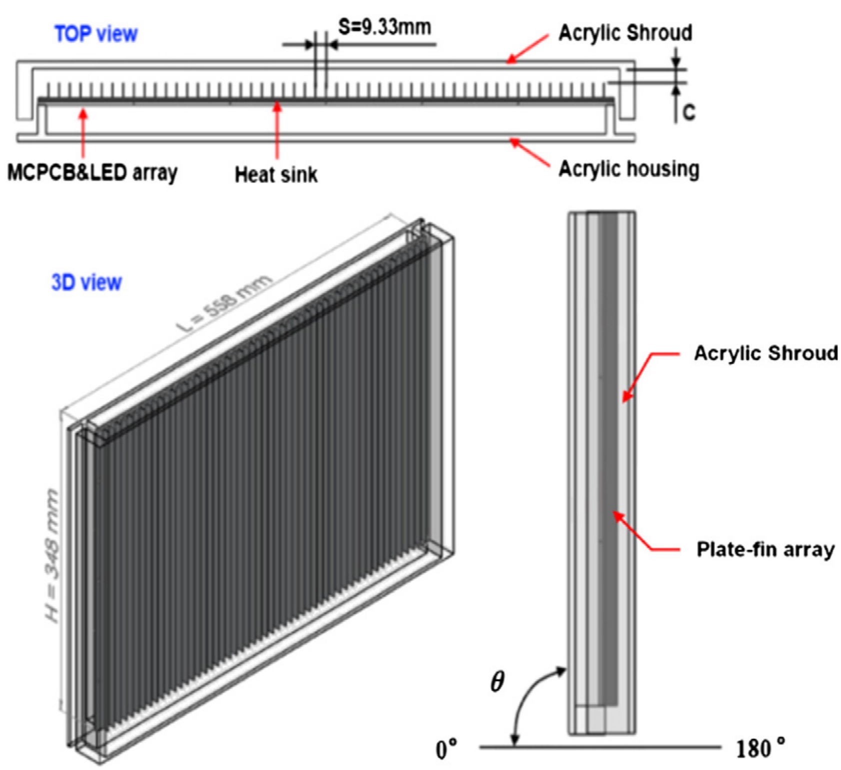

Fig. 1. Schematic diagram of the present shrouded plate fin array on an LED panel.

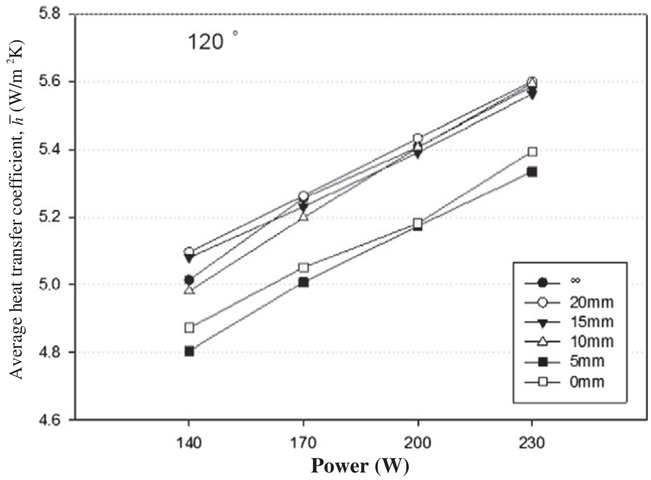

Fig. 2. Effects of clearances (C) on heat transfer coefficient of the present heat sink with an orientation of 120° at various input powers.

$60° \leq \theta \leq 120°$ irrespective of the clearance. Another feature is that the heat transfer coefficient is continuously enhanced as the clearance increases from 5 mm to the condition without a shroud. Besides, Fig. 4 also shows that the heat transfer coefficient is gradually reduced as the shrouded heat sink becomes horizontal. The lowest heat transfer coefficients among all tilted angles in Fig. 4 occur when the shroud heat sink is either 0° or 180°. With $C = 0$ mm, the difference in heat transfer coefficient between $\theta = 0°$ and $\theta = 180°$ is insignificant. Under such conditions, the air induced by the buoyancy force is confined within the acrylic shroud and is not able to diffuse freely into the ambient air. In essence, the heat transfer from the LED backlight panel to ambient air mainly depends on the induced airflow passing over the heated acrylic shroud. Once the clearance is increased, the heat transfer coefficient is initially gradually reduced and followed by a monotonous increase with a minimal heat transfer coefficient occurring at $C = 5$ mm and at $C = 15$ mm for $\theta = 0°$ and $\theta = 180°$, respectively as shown in Fig. 4. In particular, the heat transfer coefficient for both cases has the greatest difference occurring without a shroud. This is because the rising air induced by the buoyancy force over a downward finned surface is subject to a much larger frictional force than that passing over an upward finned surface, as well as the stagnation point of the induced air flow located at the downward facing finned surface [9], hence leading to a notable difference in heat transfer coefficient.

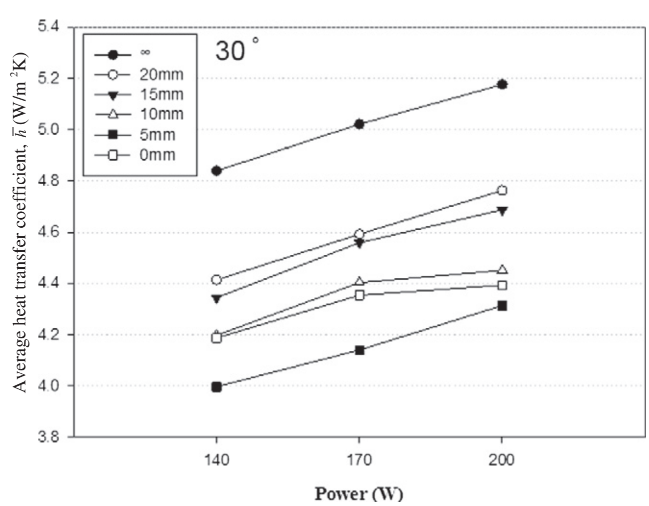

Fig. 3. Effects of clearances (C) on heat transfer coefficient of the present heat sink with an orientation of 30° at various input powers.

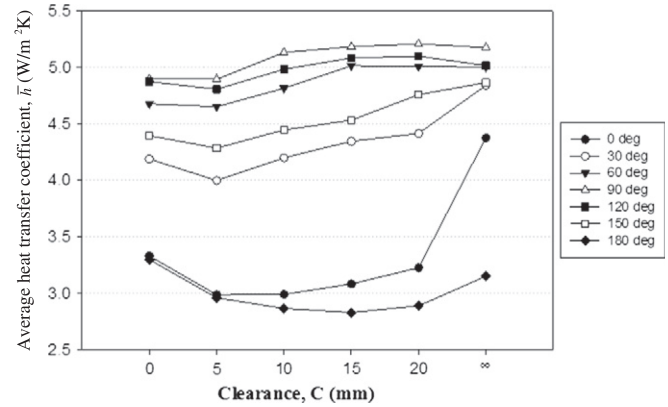

Fig. 4. Effect of tilted angle, θ, on heat transfer coefficient subject to clearance with input power of 140 W.

4. Conclusions

An experimental study investigating the effects of clearance between the shroud and plate-fin tip and the orientation on the thermal performance of a shrouded heat sink attached to LED backlight panel was performed. Results show that a "window region" exists concerning the effect of shroud. There is virtually negligible influence of shroud clearance on the heat transfer coefficient when C is above 15 mm or below 5 mm but an appreciable rise of heat transfer coefficient is encountered when C is increased from 5 mm to 15 mm. Besides, considering that the tilted angle of the LED panel increases, the heat transfer coefficient is reduced and the clearance effect on heat transfer becomes more pronounced with the input powers. In general, irrespective of clearance or input power, the heat transfer coefficients would always be similar between two cases in which the tilted angles of the LED panel are supplementary to each other. The heat transfer coefficient of the shrouded heat sink having a fin array facing downward is usually slightly higher than that in the other case having supplementary tilted angle. However, a reverse trend occurs when the LED panel is placed horizontally.

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