International Communications in Heat and Mass Transfer 42 (2013) 51–54

Contents lists available at SciVerse ScienceDirect

International Communications in Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ichmt

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

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ARTICLE INFO

Available online 31 December 2012

*Keywords:*
Orientation
Clearance
Plate-fin
Natural convection

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 ($\theta$) 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 ($\theta$) 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.

☆ Communicated by W.J. Minkowycz.

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0735-1933/$ – see front matter 
http://dx.doi.org/10.1016/j.icheatmasstransfer.2012.12.006

**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*

| | |
|---|---|
| $\theta$ | Orientation of the LED backlight panel (°) |

**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, $\theta$ (°) | 0, 30, 60, 90 120, 150, 180 |
| Clearance, $C$ (mm) | 0, 5, 10, 15, 20, ∞ |

## 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($\theta$-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 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, $\theta$, 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 ($60° \leq \theta \leq 120°$) 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

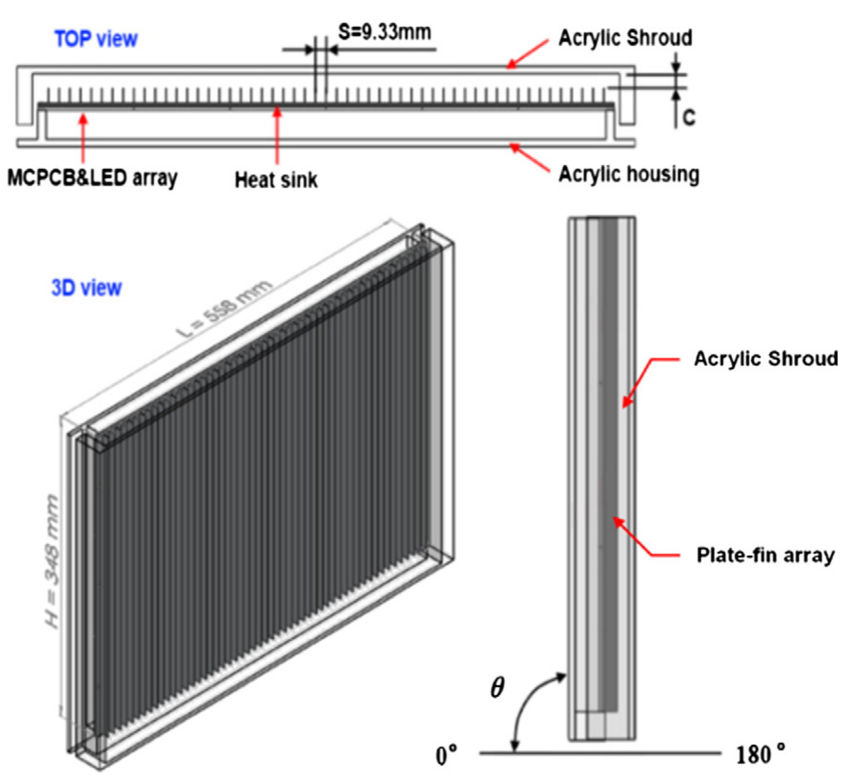


**Fig. 1.** Schematic diagram of the present shrouded plate fin array on an LED panel.