



## Thermal characterization of shrouded plate fin array on an LED backlight panel

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### ABSTRACT

This study experimentally investigates a 348 mm × 558 mm LED backlight panel consisting of 270 1-W LEDs and a plate-fin heat sink in an acrylic housing. Effects of shroud clearance, obstructions at entrance or exit on the overall performance are also examined. The results show that the heat transfer coefficient is very slightly reduced with the rise of shroud clearance from 0 mm to 5 mm, followed by a notable rise but it peaks at a shroud clearance between 10 and 20 mm. A further increase of shroud clearance leads a marginal decrease of heat transfer coefficient. It is found that the maximum temperatures within the LED panel are not necessarily located at the center region due to the edge effect. On the other hand, the temperature variation along the longitudinal direction indicates a local minimum Nusselt number occurring near the exit of LED panel. This unique phenomenon is associated with the development of velocity and temperature field, and is consistent with previous numerical examinations. On the other hand, the obstruction placed at the exit are often severe than at the entrance.

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### 1. Introduction

In recent years, applications of light emitting diodes (LEDs) in illumination have become more and more popular for its superior advantages such as high efficiency, good reliability, long life, and low power consumption over traditional light sources. To date, high-power, high-brightness LEDs have penetrated into almost every aspect of lighting applications [1]. For instance, using LEDs array as backlight of TFT-LCD instead of the conventional cold cathode fluorescent lamps (CCFL) has become one of the main-streams for large-size and energy-saving LCD TVs.

However, a greater portion of the input power for LEDs turns into heat which raises severe problems to maintain a low LED die temperature. It is well known that keeping LEDs in a lower junction temperature is a key for higher luminous efficacy, longer lifetime, and stable emission wavelength of the light output [2–5]. Normally heat dissipation for LED via radiation heat transfer is rather small due to their relatively low die temperature ( $\leq 120$  °C) when compared with that of an incandescent lamp [6]. Thus, thermal management of LEDs solely depends on both conduction heat transfer and convection heat transfer in package level and system level, respectively.

Consequently, an effective thermal management of LEDs in system level is of vital importance. Placing the LED chip onto

a metallic board attached to a heat sink is recognized as a better LED cooling design in package level. Heat is thus dissipated to ambient from the attached heat sink via either natural convection or forced convection [7]. It is no doubt that cooling of LED array in a confined enclosure by passive methods would be preferred to active means, since either noise or bulky size, or higher cost accompanied the employment of those active cooling technologies are usually inevitable.

Elanbaas [8] conducted the pioneer work of natural convection of a heat sink. Geometric influences on the associated heat sinks including the distance between plates and plate height, on the heat transfer of parallel rectangular plates were investigated. He found that the distance between plates played an important role on the thermal performance of the parallel plates due to its influence on the boundary layer development. They proposed an empirical correlation to calculate the optimal fin thickness. Bar-Cohen [9] and Bar-Cohen and Rohsenow [10] developed empirical correlations based on Nusselt number for estimation of the optimal fin thickness and fin spacing. Bar-Cohen et al. [11] also performed a natural convection cooling experiment for a vertical plate-fin heat sink. Effects of fin parameters including thickness, spacing and height, as well as material on the heat transfer were investigated.

Arquis and Rady [12] performed a two-dimensional numerical simulation to investigate the natural convection heat transfer and fluid flow characteristics from a horizontal fluid layer with a finned bottom surface. The effects, such as fin height ( $L$ ) and fin spacing ( $S$ ),

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and layer height ( $H$ ), on both natural convection heat transfer and finned surface effectiveness were investigated for a sufficiently wide range of Rayleigh number. They found that the maximum heat transfer rate from a single fin spacing occurs at about  $H/S = 0.5–0.75$ . For the range of Rayleigh number in the study ( $Ra = 2000–30000$ ), the effectiveness of the finned surface is a strong function of fin spacing and fin height.

Cengel and Nagi [13] performed an experiment to test four different types of heat sink. They found that the shrouded heat sink having higher ratio of distance between the shroud and fin tip to fin height caused a less maximum temperature over the heat sink. Moreover, they also reported that the presence of a shroud not necessarily degraded the cooling performance under natural convection condition due to the interactions amid several factors such as the friction force and buoyant force exerted on the induced air flow, as well as the radiation heat transfer contributed by shroud surface.

Yu and Joshi [14] investigated the heat transfer for a pin-fin heat sink placed in an enclosure. They found that the thermal performance of a pin-fin heat sink in an enclosure was better than that without pin-fin array in free space. Besides, experimental results showed that the presence of an additional opening for horizontal enclosures is much effective than for vertical enclosures by almost a factor of 2. In addition, vertical orientation gives a lower thermal resistance for complete enclosures, the enhancement of heat transfer from the discrete heat source using pin-fin arrays was found to be more significant than that for horizontal enclosures. For horizontal fin arrangement having top opening shows a better heat transfer performance than that of side opening with vertical fins.

Nada [15] also explored the natural convection heat transfer and fluid flow characteristics in horizontal and vertical narrow enclosures with rectangular heated finned base plate. The results

showed that increasing fin length also increases  $Nu$  and finned surface effectiveness. Besides, increasing  $Ra$  increases  $Nu$  for any fin-array geometries. In addition, for any fin-array geometry and at  $Ra > 10,000$ , increasing  $Ra$  decreases finned surface effectiveness while for fin-array geometries of large fin spacing and at  $Ra < 10,000$ , the finned surface effectiveness rises with  $Ra$ . Regardless the orientation (horizontal or vertical) both  $Nu$  and fin effectiveness attain optimal values when the ratio of fin spacing to the clearance is equal to 1. The foregoing studies are conducted for uniform heat sources subject to natural convection, it is expected that the LED may reveal an even worse situation due to its concentrated heat source behavior. Moreover, the temperature stratification and enclosure confinement may bring about serious catastrophe upon heat dissipation with a large surface area (e.g. LED TV backlight), yet practical applications may involve some blockage either at the entrance or exit of the heat sink that may accentuate the difficulty of heat dissipation. In this sense, the aim of this study is to explore the thermal performance of a large LED array panel under natural convection in a confined space. The effects of shroud located at the entrance, exit, and above the heat sink are examined to clarify the influence of the space between the enclosure of the LED array panel and the heat sink on thermal performance of the LED array panel attached to an aluminum plate-fin heat sink.

## 2. Experimental apparatus and data reduction

The experimental setup consisting of an environmental chamber, an LED backlight panel attached with an aluminum heat sink, and a power supply system, as well as a data acquisition system, is schematically shown in Fig. 1. In order to maintain a constant and uniform ambient temperature throughout the chamber without any fan during the experiment, an environmental

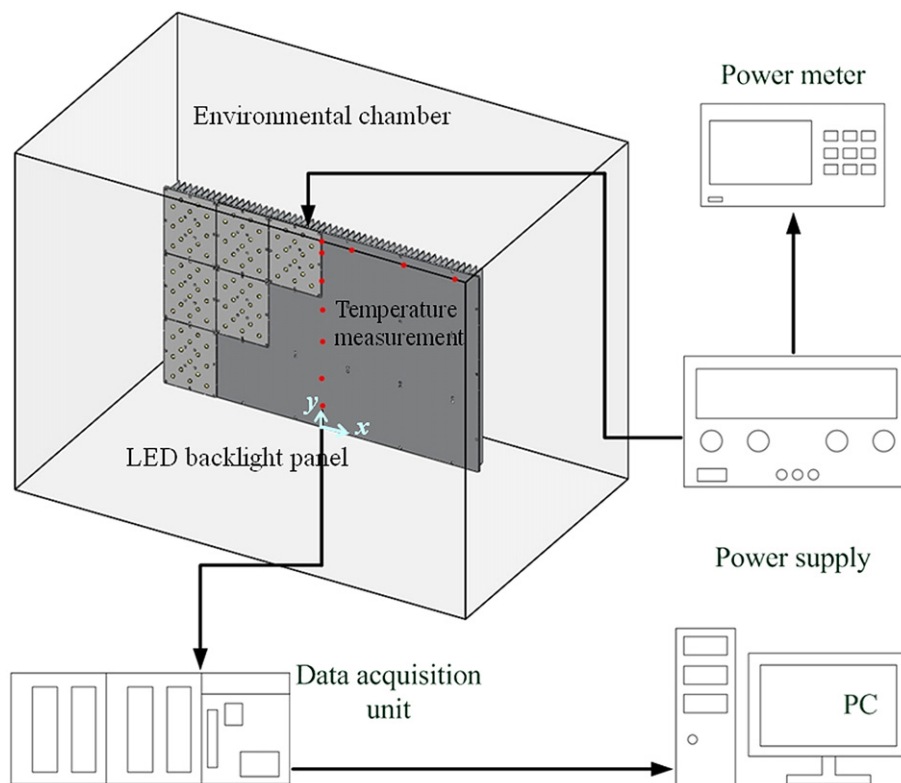


Fig. 1. Schematic diagram of the present experimental setup.

chamber having a volume of 0.86 m (*L*) × 0.86 m (*W*) × 1.16 m (*H*) was employed to carry out the experimental tests. The ambient temperature in the chamber was set to be 30 °C with a controlled resolution of 0.2 °C during the experiment.

Besides, a 558 mm × 348 mm metal core printed circuit board (MCPCB) with 270 evenly distributed 1-W white light LEDs enclosed between two acrylic housings was operated in the chamber to simulate a LED backlight panel in an enclosure. The effects of the geometric parameters on the natural convection are depicted in Fig. 2. The parameters are the clearance between the fin tip and the acrylic housing, denoted as *C*, and the clearance between the lower edge of the plate fins and acrylic housing, denoted as *C<sub>i</sub>*, as well as the clearance between the higher edge of the plate fins and acrylic housing, denoted as *C<sub>o</sub>*. For comparison, besides the baseline case without shroud, the clearances *C<sub>i</sub>* and *C<sub>o</sub>* were from 5 mm to 30 mm in an increment of 10 mm, while the clearance *C* was from 0 mm to 20 mm in an increment of 5 mm. An aluminum heat sink composed of 54 10-mm-high and 1-mm-thick rectangular plate fins parallel to gravitational direction and 3-mm-thick base plate was screwed to the backside of the LED array panel. The spacing between plate fins was 9.33 mm with a length of 348 mm.

A power supply (GW 7550D) is to power the LED panel and a power meter (Yokogawa WT230) is to measure the consumed power for those LEDs of the present experiment. Note that the corresponding efficiency indicated the ratio of net heat dissipation to total electric power input is 0.75 according to the luminous efficacy of the present LEDs provided by the manufacturer and the energy balance diagram for high-power white LEDs proposed by Krames et al. [16]. Besides ambient temperature measurement, additional ten copper-constantan thermocouples (Omega), with their locations being represented by the red dots in

**Table 1**  
The present experimental conditions.

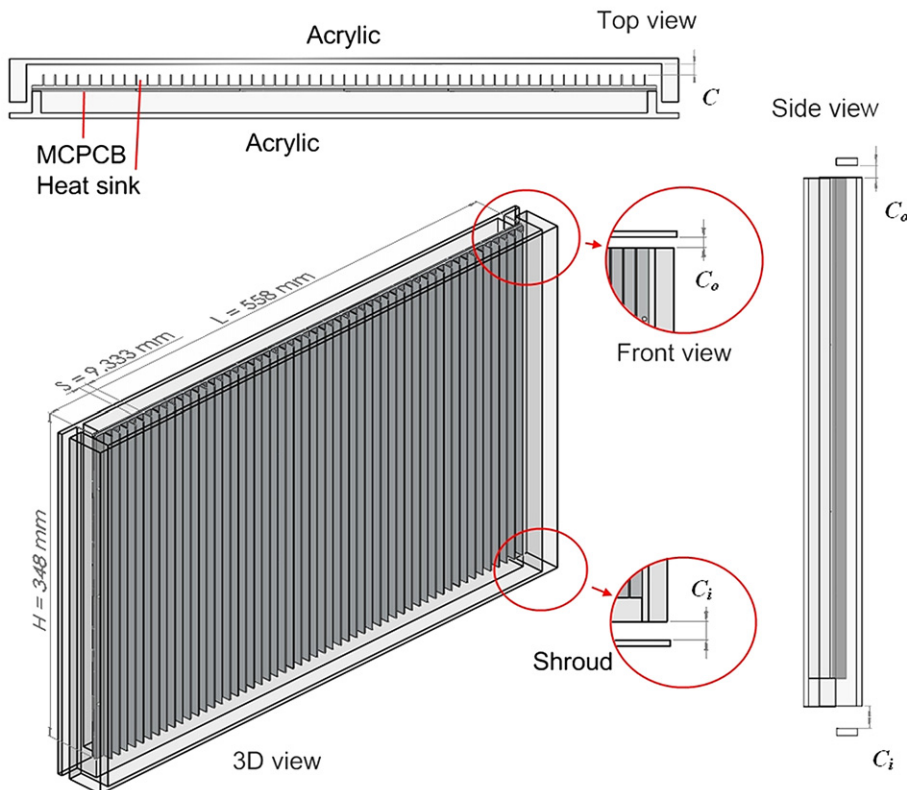
Ambient temperature, <i>T<sub>a</sub></i> (°C)	30
Power, <i>Q</i> (W)	140, 170, 200, 230
Clearance, <i>C<sub>i</sub></i> (mm)	5, 10, 20, 30, ∞
Clearance, <i>C<sub>o</sub></i> (mm)	5, 10, 20, 30, ∞
Clearance, <i>C</i> (mm)	0, 5, 10, 15, 20, ∞
LEDs array size (mm)	558 × 348
Aluminum heat sink	
Fin type	Rectangular plate fin
Fin length (mm)	558
Fin spacing (mm)	9.33
Fin thickness (mm)	1
Fin height (mm)	10

Fig. 1, were used to measure the temperature distribution of the LED panel. All the measured temperatures were recorded by a data acquisition unit (Yokogawa MX100) for further heat transfer analysis.

The average heat transfer coefficient,  $\bar{h}$ , was estimated from the Newton's cooling law as follows,

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

Where *A* is the total surface area of the heat sink, *T<sub>s</sub>* and *T<sub>a</sub>* are the average temperature of the seven vertical 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, *Q*, ranging from 140 W to 230 W. The uncertainty of the heat transfer coefficient estimation was ranged from 1.88% to 2.88%. The experimental conditions of the present study are shown in Table 1.



**Fig. 2.** Schematic diagram of the present shrouded plate fin array on an LED panel.

3. Results and discussion

In this study, experiments were carried out two parametric effects,  $C_i$  and  $C_o$ , on the heat transfer performance of a LEDs backlight panel in a confined space by both free convection and radiation. The  $C_i$  and  $C_o$  effects, ranging from 5 mm to 30 mm with  $C = 10$  mm subject to various power inputs along with the  $C$  effects, ranging from 0 mm to 20 mm on the thermal performance of the plate fin heat sink subject to the obstruction at air inlet and outlet are examined in this study.

Prior to the heat transfer coefficient investigation at various conditions, heat transfer coefficients estimated by Eq. (2) proposed by Van de Pol and Tierney [17] at various Rayleigh numbers were compared with the present test results as shown in Fig. 3. The empirical correlation for flat plate heat sink with fin spacing and fin length of  $S$  and  $H$ , respectively, is as follows,

$$Nu_r = \frac{\bar{h}r}{k} = \frac{Ra^*}{\psi} \left[ 1 - e^{-\psi(0.50/Ra^*)^{3/4}} \right] \quad (2)$$

Where  $\psi$  and  $Ra^*$  are as follows,

$$\psi = \frac{(24)(1 - 0.483e^{-0.17/a})}{\left\{ (1 + a/2) \left[ 1 + (1 - e^{-0.83a}) (9.14a^{1/2}e^{-0.4646S} - 0.61) \right] \right\}^3} \quad (3)$$

$$Ra^* = Ra_r \frac{r}{L} \quad (4)$$

and  $a, r, L, Ra_r$  and  $Ra^*$  are heat sink aspect ratio defined as  $S/H$ , characteristic length defined as  $2SH/(2H + S)$ , Rayleigh number based on the characteristic length of  $r$ , and modified Rayleigh number, respectively.

It can be found that both heat transfer coefficients obtained by calculation and experimental are of consistent trend. However, the measured heat transfer coefficient shown in Fig. 3 was underestimated by approximately 10%.

Fig. 4 shows the variations of heat transfer coefficient subject to  $C_i$  and  $C_o$ , respectively having different input power,  $Q$ . It can be found in Fig. 4 that the heat transfer coefficients indicated by dashed curves were increased by approximately  $0.2 \text{ W m}^{-2} \text{ K}^{-1}$  as the input power is increased from 140 W to 230 W with an increment of 30 W at a given  $C_i$  value. Similarly, heat transfer coefficients

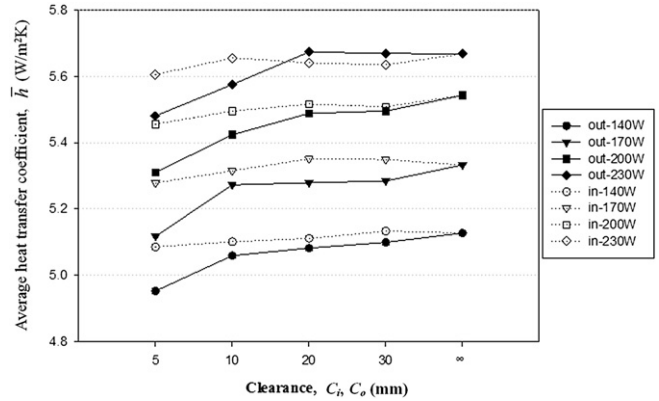


Fig. 4. Effects of clearances at air inlet ( $C_i$ ) and at air outlet ( $C_o$ ) on heat transfer coefficient at various input powers at  $C = 10$  mm.

indicated by solid curves were increased as the input power is increased from 140 W to 230 W with an increment of 30 W at a given  $C_o$  value. Based on Eq. (2), it is reasonable that Rayleigh number would be increased due to higher input power, and thus results in a higher heat transfer coefficient of a fixed heat sink with a given clearance,  $C_i$  or  $C_o$ .

For the same blockage distance, normally the heat transfer coefficient for the inlet blockage is slightly higher than those of outlet blockage, yet the difference is reduced with the increase of the clearance. The reasons for this phenomenon are explained as follows. Firstly, the density of the rising hot air is smaller at the exit than at the entrance, thereby leading to a higher velocity at the exit. In this regard, blockage at the exit offers larger resistance than at the entrance. Secondly, the results are somehow conceivable that the blockage at the downstream places resistance to the rising air while the blockage at the entrance still offers some tiny gap where cold air can easily pass through. The results are further made clear from the fact that varying  $C_i$  from 5 mm to an extreme without shroud at the air inlet, denoted by  $\infty$ , casts a negligible influence on the heat transfer coefficient at a given input power as shown in Fig. 4.

Fig. 5 shows both effects of input power and clearance,  $C$ , on the heat transfer coefficient without any obstruction at the air inlet or outlet ( $C_i = C_o = \infty$ ). As shown in the figure, with increasing shroud distance, the heat transfer coefficient remain virtually unchanged

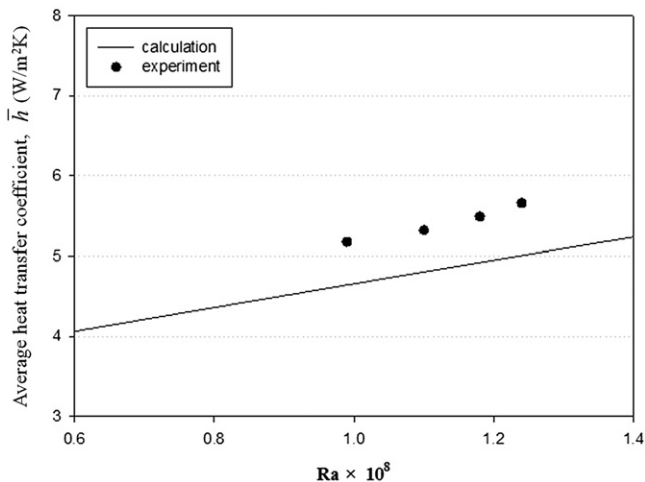


Fig. 3. Comparison of heat transfer coefficient between estimated values and measured values at various Rayleigh numbers.

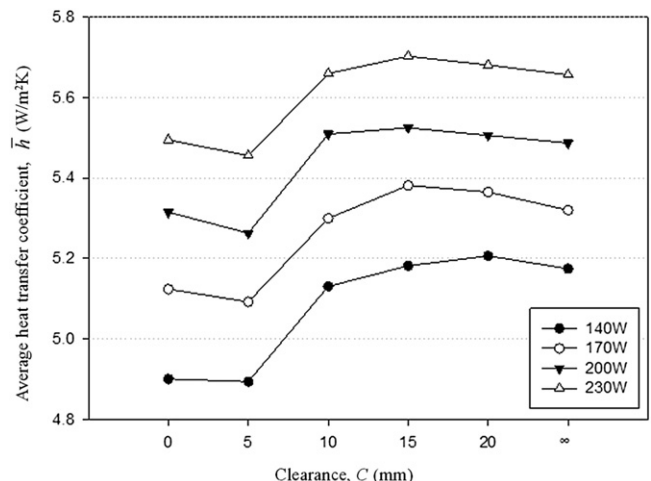


Fig. 5. Variation of heat transfer coefficient at different clearance.

(or very slightly decreased) when the shroud clearance,  $C$ , is below 5 mm. Above the threshold value,  $C = 5$  mm, the heat transfer coefficient is gradually increased to a plateau, and is slightly decreased thereafter. Roughly speaking, the heat transfer coefficient can be divided into two regions, the region for  $C < 5$  mm and the region above it. The results suggest that the presence of shroud may improve or impair the heat transfer coefficient depending on the shroud clearance. Notice that the heat transfer coefficients measured at  $C = 0$  mm, denoted as 0 mm in Fig. 5, were always higher than that measured at  $C = 5$  mm at various input powers. The results are associated with the complex interactions between buoyancy force, viscous force, and shroud surface. Note that shroud adjacent to the heat sink cast two effects upon the fluid flow. Firstly, a friction forces develops at the contact surface opposed to the direction of air flow, yet this effect normally constrain the induced flow rate, and deteriorate the heat transfer performance when shroud surface is added. On the other hand, a shroud placed above the heat sink introduces extra buoyancy especially for the case of  $C = 0$  mm. This is because the shroud temperature is higher than the ambient air due to the direct contact of the shroud and the aluminum heat sink. With a smaller shroud clearance, the viscous friction is comparatively large, whereas larger shroud clearance may offer passage for buoyant airflow; these two mechanisms counteract with each other and show an indiscriminate difference in heat transfer coefficient. On the other hand, with a further increase of shroud clearance, more induced buoyant airflow enters into the heat sink, resulting in a rise of heat transfer coefficient. However, with a further rise of shroud clearance, part of the induced buoyant airflow may bypass the heat sink and does not contribute to the increase of heat transfer coefficient directly. In this regard, one can see that the heat transfer coefficient shows a plateau phenomenon with the shroud clearance. For the present study, the maximum heat transfer coefficients at various input powers were obtained at a threshold clearance ranging from  $C = 10$  mm to  $C = 20$  mm for the input power between 140 W and 230 W as shown in Fig. 5. Test results shown in Fig. 5 are in line with the test results by Gugliemini et al. [18] who tested four staggered pin-fin heat sinks subject to the influence of shrouding. Some of their samples, e.g. the sample with lower fin height, revealed a decrease of heat transfer performance when a smaller shroud clearance ( $C = 2.5$  mm) is used, and the trend is reversed with a further increase of shroud clearance ( $C = 7.5$  mm).

Fig. 6 shows the variation of the ratio of heat transfer coefficient without a shroud to that with a shroud,  $\bar{h}_{ns}/\bar{h}$ , at different ratios of

clearance to fin height,  $C/H$ . It can be found that the heat transfer was noticeably augmented when the  $C/H$  value was increased from 0.5 to 1.0. Once the  $C/H$  value was greater than 1.5, the heat transfer coefficient measured with the presence of a shroud surpassed that of no shroud, namely  $\bar{h}_{ns}/\bar{h} < 1$ , for all tested powers. The results reported that the presence of a shroud above the heat sink is able to enhance the heat transfer performance although such enhancement is quite minor. Besides, the optimal  $C/H$  value for the highest heat transfer coefficient depends on the input power in the present experiment. Similar results showing more enhanced heat transfer of a heat sink with a shroud than the no-shroud case as  $C/H$  value increased at a given input power were also reported by Cengel and Ngai [13]. They explained that the presence of a shroud adjacent to a heat sink introduces both effects on heat transfer: including extra buoyancy as a result of the elevated temperature of the shroud surface, and slowing down the fluid by acting as an added obstacle on the flow path. Therefore, shrouding a heat sink can either enhance or reduce the heat transfer, depending on which effect is dominant.

Fig. 7 shows that the horizontal temperature variation at 140 W and different clearances without any obstructions either at the inlet or at the exit. The schematic of the temperature measurement positions are termed as 4 horizontal red dots in Fig. 1. The variation of temperature along  $x$ -axis initially remains roughly the same but is slightly reduced when  $x$  is increased to approximately 150 mm (note that  $x = 0$  represents the center line of the panel as shown in Fig. 1), followed by a slight increase of temperature with a further increase of  $x$ . The results are quite interesting for one normally think that the temperatures in the center region exceed other locations. The reason is due to the friction force exerted on the upward air flow at the edge of the acrylic shroud which causes less air flow on the edge of the LED array panel and thus result in a slightly higher temperature at  $x = 234$  mm. The lesser air flow at the edge region implies a higher air flow next to the edge, the higher air flow results in a lower temperature accordingly. Therefore, the temperature at  $x = 150$  mm was usually the lowest among the measured four temperatures along  $x$ -axis for all clearances, except the condition at which the clearance was prescribed at 20 mm. Even though there was variation in temperatures from  $x = 0$  mm to  $x = 234$  mm in Fig. 7, it can be observed that the maximum temperature difference along  $x$ -axis was less than  $1^\circ\text{C}$  no matter how far the shroud was presented.

The longitudinal temperature variation from  $y = 30$  mm to  $y = 330.5$  mm for an input power of 140 W subject to influence of

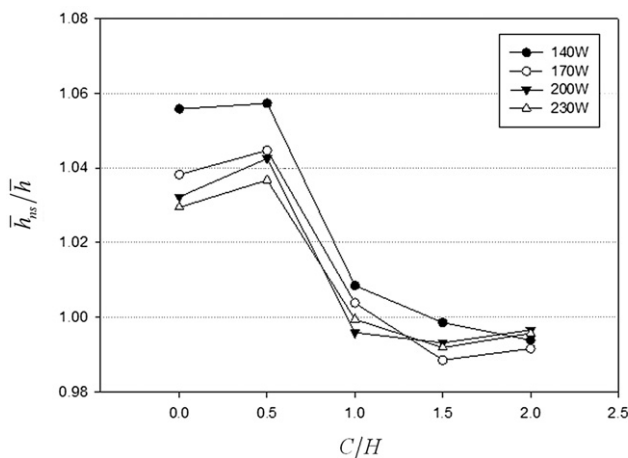


Fig. 6. Variation of the ratio of heat transfer coefficient without shroud to that with a shroud,  $\bar{h}_{ns}/\bar{h}$ , at different ratio of clearance to fin height,  $C/H$

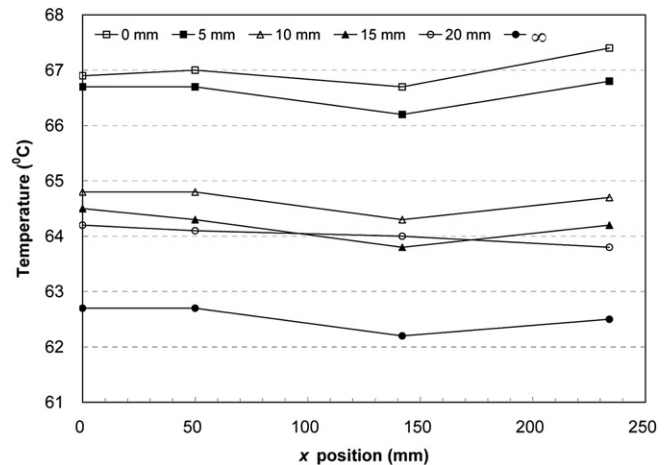


Fig. 7. Temperature variation at different horizontal positions at 140 W and different clearances.



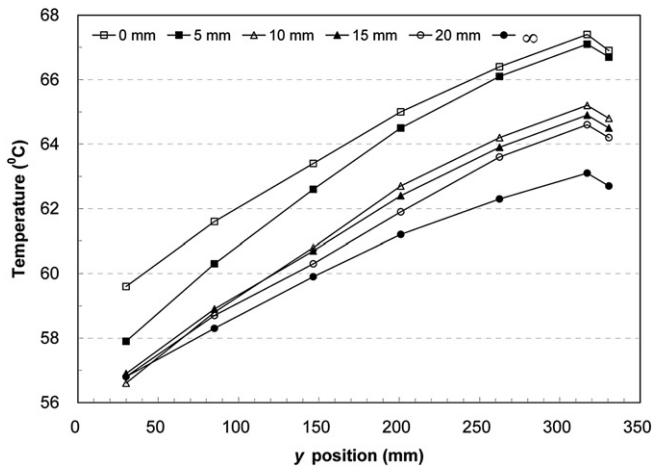


Fig. 8. Temperature variation at different vertical positions at 140 W and different clearances.

shroud clearance is shown in Fig. 8. Converse to the temperature variation along the lateral position shown in Fig. 7 with approximately only 1 °C, the variation of temperature along the longitudinal direction ranged from 5 °C to nearly 10 °C. It is interesting to know that the temperature continuously rises from entrance to  $y = 317$  mm. However, a sudden drop in temperature from  $y = 317$  mm to  $y = 330.5$  mm was observed for all tested clearances. The results indicate a local minimum of Nusselt number occurring within the LED panel. This phenomenon is quite unique and was numerically demonstrated by Karki and Patankar [19] who showed that the local Nusselt number revealed a monotonic decrease from the inlet to exit of the duct but the local Nusselt number shows a minimum within ducts if the fin is closely spaced. In fact, their simulations showed that the minimum is very close to the exit like the present study. As explained by Karki and Patankar [19], this phenomenon is from the development of velocity and temperature field. The fluid in the interfin region migrates toward the clearance region and its temperature approaches the wall temperature, and eventually a state is reached where the fins are in contact with fluid of nearly the same temperature, and no additional heat transfer can take place. After the stage, the cold fluid in the clearance region moves into the interfin region due to buoyancy, and thus results in a slight increase of heat transfer at the exit. This phenomenon occurs for a plate-fin heat sink having a very narrow fin spacing or a very long fin length like the present study.

#### 4. Conclusions and suggestions

The present study performs an experimental study concerning thermal performance of shrouded rectangular plate fin array on an LED backlight panel. A total of 270 1-W LEDs having an efficiency of 75% are used to simulate a large LED panel. The size of the cooling heat sink is 348 mm × 558 mm having plate fins parallel to gravity with a fin spacing of 9.33 mm. The effects, including the clearance between shroud and heat sink and the obstruction at the entrance or at the exit, on the thermal performance of the heat sink are investigated. For the effect of shroud distance on the heat transfer coefficient, it is found that the heat transfer coefficient is very slightly reduced with the rise of shroud clearance from 0 mm to 5 mm. A further increase of the clearance leads to a notable raise of heat transfer coefficient, and depending on the input power, a plateau of heat transfer coefficient is observed at the clearance of 10–20 mm. A further increase of the shroud clearance leads to a marginal decrease of heat transfer coefficient. The peak

phenomenon is associated with the complex interactions amid the friction force, buoyancy force, and shroud surface. It is found that the maximum temperatures within the LED panel are not necessarily located at the center region due to the edge effect. On the other hand, the temperature variation along the longitudinal direction indicates a local minimum Nusselt number occurring near the exit of LED panel. This unique phenomenon is associated with the development of velocity and temperature field, and is consistent with previous numerical examinations. On the other hand, the obstruction placed at the exit are often severe than at the entrance.

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#### Nomenclature

$A$	heat transfer surface area ( $m^2$ )
$a$	heat sink aspect ratio
$C$	distance between fin tip and acrylic housing (m)
$C_i$	entrance blockage distance (m)
$C_o$	exit blockage distance (m)
$H$	fin length (m)
$\bar{h}$	average convective heat transfer coefficient ( $W/m^2K$ )
$\bar{h}_{ns}$	average convective heat transfer coefficient without shroud ( $W/m^2K$ )
$Nu$	nusselt number
$Q$	total input power to LED panel (W)
$q$	heat transfer rate from the LED array panel (W)
$Ra$	Rayleigh number
$Ra_r$	Rayleigh number based on the characteristic length of $r$
$Ra^*$	modified Rayleigh number
$r$	characteristic length
$S$	spacing between two plate-fins (m)
$T_a$	ambient temperature of the environment (K)
$T_s$	average temperature of the heat sink (K)
<i>Greek</i>	
$\psi$	heat sink configuration factor used in Eq. (2)

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