IMPROVEMENT OF THERMAL EFFICIENCY OF A RADIAL HEAT SINK WITH A SURROUNDING STRUCTURE

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ABSTRACT

This study proposed a radial heat sink with a surrounding structure as a cooling system for light-emitting diode (LED) lighting with a circular base. The radial heat sink with long and middle fins was considered. Heat flow and radiative heat transfer were simulated numerically and the numerical model was validated experimentally. Natural convection heat transfer at the heat sink can be improved by controlling the air flow pattern. In this study, the surrounding structure increased the velocity of the inlet air around the heat sink due to the narrow air inlet surface, allowing the cooling air to move to the center of the heat sink. The effects of the material used to make the surrounding structure and the surface treatment of the heat sink were also analyzed. The results showed that the thermal efficiency of the radial heat sink was improved by up to 40% following installation of the chimney-shaped surrounding structure. The material used to make the surrounding structure did not influence the ability of the heat sink to dissipate heat. This research is expected to be helpful in the development of high-power LED lighting with a natural cooling system.

INTRODUCTION

A lighting-Emitting diode (LED) is a light producing semiconductor diode that has a long lifetime and high luminous efficiency. Consequently, conventional light sources have been replaced by LEDs. However, LEDs have a critical weakness: temperature management. If the temperature of a LED exceeds the maximum junction temperature, the lifetime of the LED is shortened dramatically, and its stability deteriorated markedly[1]. Therefore, a sufficient cooling system is essential for LED lighting. Radial heat sinks are passive cooling devices used for circular based LED lights, such as LED downlights (Figure 1). Recently, the amount of heat dissipation necessary increases with the power of the LED lights.

Many studies have examined ways to improve the performance of radial heat sinks[2-6]. A change in the heat sink geometry has little influence due to the the limitations of natural convection and radiative heat transfer. In addition, a complex design increased manufacturing costs. Researchers have tried to enhance convective heat transfer using surrounding structures[7-10]. However, no study has evaluated the effects of a horizontal circular plate and chimney-shaped

NOMENCLATURE

A	[m^2]	Surface area
C	[-]	Turbulent constant
c_p	[J/kg^{o}C]	Specific heat
G	[-]	Generation of turbulent kinetic energy
g	[m/s^2]	Gravity acceleration
h	[m]	Height
I	[-]	Intensity of radiation
k	[W/m^{o}C] [m^2/s^2]	Thermal conductivity, Kinetic energy of turbulence
l	[m]	Length
N	[-]	Number of fin array
$\vec{n}$	[-]	Normal direction vector
p	[Pa]	Pressure
$\dot{Q}$	[W]	Heat release rate of film heater
$\dot{q}$	[W/m^2]	Heat flux
R_{th}	[^{o}C/W]	Thermal resistance of heat sink
r	[m]	Radius
$\vec{s}$	[-]	Ray direction vector
T	[^{o}C]	Temperature
t	[m]	Thickness
u	[m/s]	Velocity component
x	[m]	Distance component
Special characters		
α	[-]	Effective inverse Prandtl number
ε	[m^2/s^3]	Dissipation rate of turbulent kinetic energy, Emissivity
μ	[m2/s]	Dynamic viscosity
ρ	[kg/m^3]	Density
τ	[Pa]	Shear stress
Subscripts		
ave		Average
eff		Effective
h		Heat sink
i		Inner
l		Long fin
m		Middle fin
o		Outer
s		Surrounding structure

surrounding structure.

In this study, the numerical models were used to simulate the natural convection and radiative heat transfer of a radial heat sink with a chimney-shaped surrounding structure. The effects of structure geometry, the materials used to make the chimney-shaped surrounding structure, and the surface

Figure 1 Light-emitting diode (LED) downlight

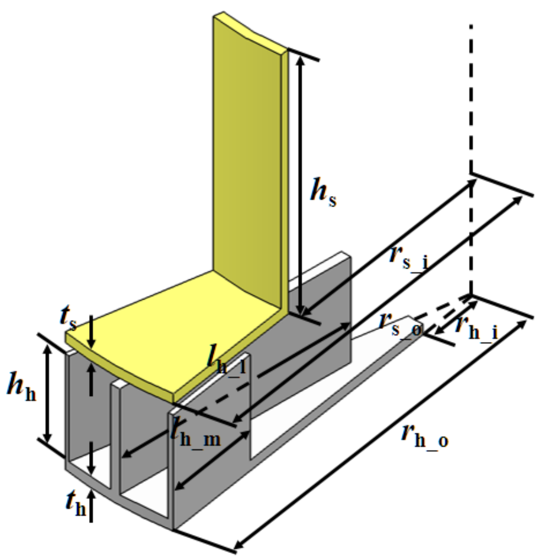


Figure 2 Dimensions of the heat sink and surrounding structure

treatment were analysed.

NUMERICAL MODEL AND PROCEDURE

Figure 2 shows the long and middle fin type radial heat sink and the chimney-shaped surrounding structure, which consists of a ring-shaped horizontal plate and a cylindrical pipe. To minimize the computational effort, only one period domain was computed. To analyze the natural convection and radiative heat transfer around the heat sink and the surrounding structure, a renormalization group (RNG) k-ε turbulence model and discrete transfer radiation model (DTRM) were adopted. The governing equations are as follows:

Continuity

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \tag{1}$$

Momentum

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left((\mu+\mu_t)\frac{\partial u_i}{\partial x_j}\right) + \mathbf{F}$$
$$(\text{for z direction } \mathbf{F} = -\rho_{\text{op}} g) \tag{2}$$

Turbulent kinetic energy

$$\frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j}\left(\alpha_k \mu_{\text{eff}} \frac{\partial k}{\partial x_j}\right) + G_k - \rho\varepsilon \tag{3}$$

Turbulent dissipation rate

$$\frac{\partial(\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j}\left(\alpha_\varepsilon \mu_{\text{eff}} \frac{\partial \varepsilon}{\partial x_j}\right) + C_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}^{*}\rho\frac{\varepsilon^2}{k} \tag{4}$$

Energy

$$\frac{\partial}{\partial x_i}\left(u_i \rho c_p T\right) = \frac{\partial}{\partial x_j}\left(k_{\text{eff}}\frac{\partial T}{\partial x_j} + u_i\left(\tau_{ij}\right)_{\text{eff}}\right) \tag{5}$$

$$\dot{q}_{\text{in}} = \int_{\vec{s}\cdot\vec{n}>0} I_{\text{in}}\vec{s}\cdot\vec{n}\,d\Omega \tag{6}$$

$$\dot{q}_{\text{out}} = (1-\varepsilon_{\text{w}})\dot{q}_{\text{in}} + \varepsilon_{\text{w}}\sigma T_{\text{w}}^4 \tag{7}$$

The values of the parameters used for the reference heat sink and surrounding structure were N = 20, $l_{\text{h\_l}}$ = 40 mm, $l_{\text{h\_m}}$ = 20 mm, $r_{\text{h\_i}}$ = 75 mm, $r_{\text{h\_o}}$ = 75 mm, h_{h} = 20 mm, t_{h} = 2 mm, H_{s} = 200 mm, $r_{\text{s\_i}}$ = 45 mm, $r_{\text{s\_o}}$ = 10 mm, and t_{s} = 2 mm. The height and radius of the analysis air domain were set to 1.5$r_{\text{h_o}}$ and 1.5(h_{h}+h_{s}), respectively. A hexahedral mesh was adopted; the mesh decreased near the surface of the solids. There were 862,746 grid points in the reference model to consider grid dependency. ANSYS FLUENT(ver. 16.1; ANSYS, Canonsburg, PA, USA) was used to analyze the heat flow. The convergence criterion for the dependent variables was set to 10^{-5}.

Figure 3 shows the experimental set-up used to validate the numerical model. The reference heat sink and chimney-shaped surrounding structure were used in the experiment. The heat sink was made of aluminium(Al6061) using a computer-

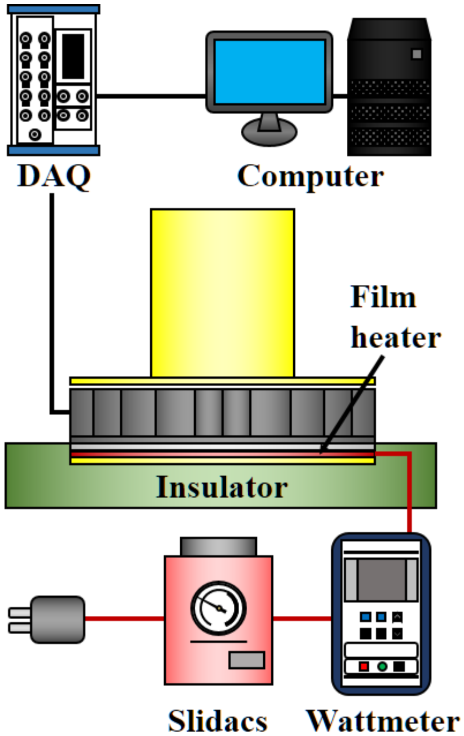


Figure 3 Experimental set-up

controlled machine with no additional surface treatment (ε = 0.2). The surrounding structure was made of acrylic. A circular plate heater was equipped to generate the heat flux on the base surface of the heat sink. The thermal resistance, the thermal performance factor, was obtained from the following equation:

$$R_{th} = \frac{T_{h\_ave} - T_{\infty}}{\dot{q}_{h\_base} \cdot A_{h\_base}} \qquad (8)$$

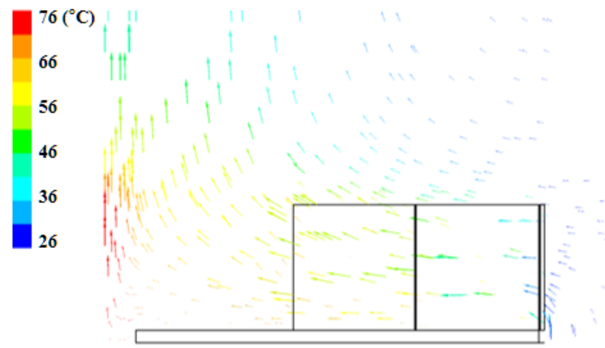
where,

$$\dot{q}_{h\_base} = \frac{\dot{Q}}{A_{h\_base}} \qquad (9)$$

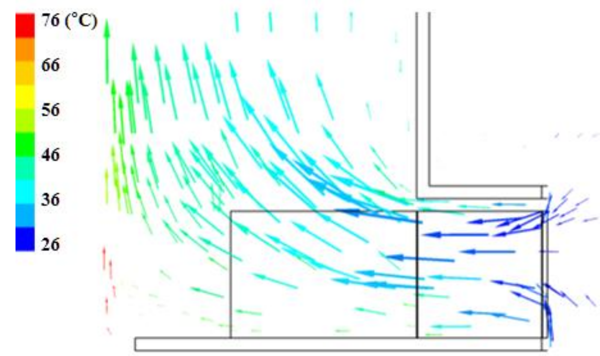
Table 1 compares the numerical and experimental results at T_{∞} = 25^{o}C and $\dot{q}$ = 1000 W/m^2. The difference between the results was 5%. Therefore, the numerical models were capable

Table 1 Experiment and numerical analysis results

-	Without surrounding structure	With surrounding structure
T_{hs_ave} (^{o}C) (Experimental)	65.6	51.2
T_{hs_ave} (^{o}C) (Numerical)	64.7	50.0
R_{th} (^{o}C/W) (Experimental)	2.34	1.51
R_{th} (^{o}C/W) (Numerical)	2.29	1.44



(a)



(b)

Figure 4 Air velocity vector around the heat sink

of simulating natural convection and radiative heat transfer around the heat sink and surrounding structure.

RESULTS

Tables 1 compares the thermal performance of the heat sink with and without the chimney-shaped surrounding structure at T_{∞} = 25^{o}C and $\dot{q}$ = 1000 W/m^2. The surrounding structure improved the performance of the heat sink by 40%. Figure 4 shows the air velocity vector around the heat sink. The surrounding structure increased the velocity of inlet air due to the narrow air inlet. At the air inlet region, the average velocity of the air was 0.055 m/s without the surrounding structure and 0.16 m/s with the surrounding structure. The horizontal plate portion of the chimney-shaped surrounding structure caused the inlet air to penetrate to the center of the radial heat sink.

Surrounding structures made with various materials that can be used in the LED lighting industry were considered. Materials with low to high thermal conductivities were studied. The maximum variation in the thermal resistance was 2%. Therefore, the material of the hollow cylinder does not have a strong effect on the thermal performance of the heat sink.

To determine the effect of the emissivity of the heat sink, no treatment (ε = 0.2), chrome gilding (ε = 0.7), and black anodizing (ε = 0.9) surface treatments were evaluated. Table 2 shows the thermal resistance of the heat sink with various surface treatments. Despite its high emissivity, the black anodizing treatment reduced the thermal performance of the

Table 2 Numerical analysis results with various surface treatments ($\dot{q}$ = 1000 W/m^2)

Surface treatment	ε	R_{th} (^{o}C/W)
No treatment	0.2	1.44
Chrome gilding	0.7	1.48
Black anodizing	0.9	1.49

heat sink by 5% compared with the no-treatment heat sink. This was because the high emissivity heat sink increased the temperature of the surrounding structure, decreasing the amount of inlet air.

CONCLUSIONS

In this study, the installation of the chimney-shaped surrounding structure which enhanced the thermal performance of a radial heat sink was suggested. A numerical model comprising ab RNG k-ε turbulence model and DTRM model was used to analyse the natural convection and radiative heat transfer around the heat sink and surrounding structure. The model was validated experimentally. The numerical analysis results showed found that the installation of a surrounding structure increased the velocity of the inlet air and the heat transfer surface between the heat sink fin and inlet air. The effect of the thermal conductivity of the surrounding structure was negligible. In addition, the surface treatment of the heat sink had no major influence on the thermal performance of the heat sink. The installation of a chimney-shaped surrounding structure improved the thermal performance of the heat sink by up to 40%. This approach should enable the manufacture of high-power LED lighting with sufficient cooling.

ACKNOWLEDGEMENT

This work was supported by the technology upgrade R&D program through the Commercialization Promotion Agency for R&D Outcomes (COMPA) funded by the Ministry of Science, ICT and Future Planning (MSIP) (No. 2014K000244).

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