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Correlation of cross-cut cylindrical heat sink to improve the orientation effect of LED light bulbs

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ABSTRACT

We investigate cross-cut cylindrical heat sink to improve cooling of a light-emitting diode (LED) bulb compared with conventional plate-fin cylindrical heat sinks using various installation angles. In contrast to plate-fin heat sinks, whereby the fins in the upper heat sink do not provide effective cooling at large installation angles, flow formed between the cross-cut fins and the air reached the upper region of the heat sink. As the installation angle increased, the length of the flow path increased. Accordingly, the flow efficiency of the cooling air improved, and the dependence of the cooling on orientation was reduced. The improvement of the cross-cut heat sink was more noticeable in the shorter cross-cut length. We suggest a correlation to predict the degree of improvement in the thermal resistance compared with a plate-fin heat sink as a function of design parameters and installation angle, and develop a contour map describing the optimum heat sink shape for various installation angles.

1. Introduction

Light-emitting diodes (LED) are rapidly replacing conventional light bulbs in light applications because of their increased life span and energy efficiency. LED light bulb is used in various places, and the installation angle can vary based on the application and the environment, which significantly affects the cooling of the LED. If the heat dissipation performance of the LED light bulb is insufficient, both the life of the LED and its luminous efficiency decrease. Cooling of LED light bulbs is typically achieved by using a cylindrical heat sink. Thus, there is a need to study the effect of installation angle on the cooling performance of cylindrical heat sinks.

Several recent studies have focused on cylindrical heat sinks for LED lighting applications. Yu et al. [1–3] and Jang et al. [4,5] studied natural convection heat sinks, where the fins were attached perpendicular to a horizontal circular plate. However, these heat sinks were designed for down-lights, which are installed in the ceiling so that the fins are oriented vertically. These studies did not consider the changes in heat dissipation in terms of the variation in the installation angle, so their findings have limitations to be applied to the cylindrical heat sinks used in LED light bulbs. Jeng [6] investigated the combined convection around the cylindrical heat sink with motor fan for LED light bulb. He tested various assemblies of heat sinks and fans and analyzed the effect of fans. Finally, he suggested the correlation to predict the Nusselt number of combined convection. Although this study obtained valuable results with regard to the cooling of LED light bulb, the orientation effect was not discussed and the cooling mechanism was combined convection.

With regard to the natural convection effects in terms of the orientation of heat sinks, Huang et al. [7] experimentally studied the effects of orientation on a square pin–fin heat sink, compared seven types of heat sinks, and proposed optimal designs for each orientation. Shen et al. [8] studied the effects of rectangular plate-fin heat sink orientation in LED lighting applications. They found that a smaller fin pitch resulted in a greater dependence of the cooling performance on the orientation, and proposed a Nusselt number correlation for various angles. These studies investigated the effects of orientation for rectangular heat sinks, and observed the same orientation effect for each fin because the fins were parallel. However, because the fins are arranged radially in a cylindrical heat sink, flow characteristics will vary for each fin. For this reason, it is difficult to apply the results of studies on rectangular heat sinks to LED light bulbs. Recently, Jang et al. [9] investigated the effects of the orientation of a cylindrical heat sink for applications in LED light bulbs, and proposed a Nusselt number correlation. However, the study only analyzed the mechanism of orientation effect, and did not propose a method for reducing its dependence on the installation angle.

The purpose of this study is to analyze cross-cut cylindrical heat sinks to improve the orientation effect of a conventional plate-fin

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Nomenclature

C	specific heat capacity [J/(kg K)]
D	diameter of heat sink [mm]
$\mathbf{F}$	body force vector per unit volume
Gr	Grashof number
g	acceleration due to gravity [m/s^2]
H	fin height [mm] [W/m^2 K^4]
h	heat transfer coefficient [W/(m^2 K)]
k	thermal conductivity [W/(m K)]
L	fin length [mm]
l	length [mm]
N	number of fins
Nu	Nusselt number
n	normal direction vector
P	pressure [Pa]
$\dot{Q}$	heat transfer rate [W]
$\dot{q}$	heat flux [W/m^2]
R_{TH}	thermal resistance [°C/W]
s	Surface
T	temperature [K]
t	thickness of fin [mm]
u	velocity [m/s]
$\mathbf{v}$	velocity vector [m/s]

Greek symbols

ξ	augmentation factor, $\frac{R_{\text{cross-cut}}}{R_{\text{plate}}}$
β	coefficient of volume expansion [K^{-1}]
σ	Stefan–Boltzmann constant
ρ	density [kg/m^3]
ε	emissivity
θ	angle of inclination [°]

Subscripts

avg	average
c	cross-cut
f	unit fin
film	film temperature
L	average over the heat sink length
i	inner
in	in
o	outer
out	out
∞	ambient

cylindrical heat sink for LED light bulbs. We compare the changes in the flow characteristics and thermal resistance between plate-fin and cross-cut heat sinks with various installation angles. In addition, we suggest a correlation to predict the degree of improvement in the thermal resistance compared with a conventional plate-fin heat sink as a function of the heat sink design parameters, the installation angle, and the Rayleigh number. Finally, we develop a contour map showing the optimum heat sink shape as a function of the installation angle.

2. Mathematical modeling

2.1. Numerical model

Fig. 1(a) shows the subject of this study, a cross-cut cylindrical heat sink. The heat sink is composed of a cylindrical base and vertical cross-cut fins. The fins are placed radially with a uniform angle. To restrict the excessive degrees of freedom in terms of the heat sink shape, the cross-cut length l_c was assumed to be the same as the unit fin length l_f. The computational domain used for the analysis of the heat sink is shown in Fig. 1(b). To generate a control volume that is independent of the installation angle, a sphere with a heat sink at the center was selected as the computational domain. Additionally, because the heat sink is symmetrical, only the heat sink and the air region contained in one hemisphere were selected for the control volume. The following assumptions were adopted for numerical analysis:

(1) The airflow is three-dimensional, steady-state and laminar.
(2) The air density is calculated by assuming that the fluid is an ideal gas.
(3) The surface of the heat sink is gray and diffuse.

The governing equations are as follows:
Continuity equation:

$$\nabla \cdot (\rho \mathbf{v}) = 0 \tag{1}$$

Momentum equation:

$$\rho \frac{D\mathbf{v}}{Dt} = -\nabla P + \mu \nabla^2 \mathbf{v} + \mathbf{F} \quad (\text{for } z\text{-direction } \mathbf{F} = -\rho g) \tag{2}$$

Energy equation

$$\rho C \frac{DT}{Dt} = \nabla \cdot (k \nabla T) + \frac{DP}{Dt} \tag{3}$$

The boundary conditions of Ref. [9] were applied in this study. Radiation heat transfer was calculated using the discrete transfer radiation model (DTRM) [10,11], which is applicable to symmetric conditions.

2.2. Numerical methods

The SIMPLE algorithm was used to solve for the flow field by combining pressure and velocity. To improve accuracy, the convective terms of the governing equation and the energy equation were discretized using a second-order upwind scheme. Convergence of the dependent variables was considered to have occurred when the maximum value of the relative error was less than 10^{-5}. The radius of the computational domain was varied from 2 to 4 times the radius of the heat sink $(H + D_o/2)$ based on the temperature convergence and computation time. As a result, we selected 3 times the radius of the heat sink because the temperature change was less than 0.5%. The number of grid points was varied in the range 350,000–1,300,000, and we found that 784,244 nodes provided a temperature change of less than 0.5%, and so that grid was used as the reference grid.

3. Experimental validation

Experiments were carried out to validate the numerical model. The design parameters of the heat sink used in the experiment were $N = 12$, $L = 50$ mm, $H = 30$ mm, $l_c = 10$ mm, $D_i = 20$ mm, $D_o = 60$ mm, and $t = 2$ mm. The heat sink was formed of 6061 aluminum alloy, and was anodized black with an emissivity of 0.9. The test section of the experimental apparatus is shown in Fig. 2(a). To decrease thermal contact resistance, thermal grease was applied to the contact surface between the heat sink and the cartridge heater. To calculate heat loss through the top and bottom sides of the heat sink, polystyrene (k = 0.2 W/m °C) was placed in these areas. The thermal resistance was used as the performance index, and can be defined as follows: