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Natural convection heat transfer along vertical cylinder heat sinks with longitudinal fins

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Heat sinks appearing as one vertical cylinder with longitudinal fins are widely used in heat dissipation of LED bulbs and other electric devices. Numerical models of natural convection heat transfer are built based on CFD method. Thermal boundary layers are discussed along with the air flow patterns. Numerical results disclose that the edge of thermal boundary layer can be fitted into a sinusoidal function in 2D polar coordinate system. The local *Nu* distributions on both cylinder and fin surfaces are presented and analyzed. It is found that average Nusselt number along the fin length is approximately proportional to an exponential function of *z*. A new correlation is proposed by introducing finned ratio, achieving a deviation within ±10%.

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1. Introduction

The incandescent light is fading away while the LED (Light Emitting Diode [1]) light is taking its place in the illumination market. However, due to the junction temperature limit, the LED bulb is facing very severe challenges in heat dissipation, especially for high power LED bulbs. It can hardly dissipate the generated heat only via thermal radiation. Therefore, there is now a considerable concern for the performance of the heat sink utilizing natural convection heat transfer. Heat sinks appearing as a cylinder with longitudinal fins are frequently applied in LED bulbs. Such kind of heat sink can dissipate heat by both natural convection and radiation while keeping the appearance as a bulb component.

Heat sinks that work by natural convection require no additional power supply. Additionally, they are almost maintenance-free, which is suitable for long lifetime devices, such as LED devices. Natural convection heat sinks usually have extended fins on either a flat base or a cylinder base. Plate fin, which is the basic fin type on a flat base, can be further manufactured to strip fin [2] or even pin fin [3]. Some advanced fins also derive from these basic fins, such as variable width plate-fin [4], inclined fin [5], elliptical pin fin [6], and hollow perforated pin fin [7]. Natural convection on plate fin heat sinks had been well studied. Parametric optimization had been conducted by Vollaro et al. [8], Bar-Cohen et al. [9], Zhang and Liu [10] and Kim et al. [11]. Meanwhile, orientation effects on the natural convection heat dissipation of plate fin heat sink recently has drawn lots of attention [12–14], which was very helpful to ensure adequate cooling for LED devices at arbitrary installation angle. On the other hand, fins on a cylinder can be either annular or longitudinal. Annular fins are more suitable for horizontal cylinders in natural convection [15,16], while longitudinal fins for vertical cylinders. Longitudinal fins on horizontal cylinders were studied by Haldar et al. who recommended more fins with low fin height for a given area [17,18]. For longitudinal fins on vertical cylinders, there are some similar cases for reference. Analysis on internally finned vertical tubes had been made by Prakash and Patankar [19]. Natural convection within a vertical annulus with longitudinal fins was studied by Kumar [20]. Yu and his coworkers studied natural convection associated with a type of radial heat sink where longitudinal fins were attached on a horizontal disk acting as a base [21], and further improvements were made, such as combining long and short fins [22] and optimizing fin height profile [23]. Experimental exploration on natural convection heat transfer of vertical cylinder with longitudinal fins was recently conducted by An et al. [24], who proposed a correlation for such heat sinks with error within ±20%.

The above literature survey indicates that only experimental work has been done on vertical cylinder heat sinks with

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Nomenclature

A, B	mean radial coordinate and amplitude defined in Eq. (7)
$A_{overall}$	overall surface area, m^2
a	coefficient in Eq. (10)
c_p	heat capacity, J kg^{-1} K^{-1}
D	diameter of the cylinder, mm
g	gravitational acceleration, m s^{-2}
Gr	Grashof number, $Gr = g\beta\Delta TL^3/\nu^2$
H	fin height, mm
h	heat transfer coefficient, W m^{-2} K^{-1}
$\overline{h}$	overall heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity of solid, W m^{-1} K^{-1}
L	fin length, mm
m	exponent in Eq. (10)
N	number of fins
Nu	Nusselt number, $Nu = hL/\lambda$
$\overline{Nu}$	overall Nusselt number, $\overline{Nu} = \overline{h}L/\lambda$
p	pressure, Pa
Pr	Prandtl number, $Pr = \nu/\alpha$
Q	total heat transfer rate, W
q	total heat flux, W/m^2
Ra	Rayleigh number, $Ra = g\beta\Delta TL^3/\nu\alpha$
r	radial coordinate in cylindrical coordinate system
T	temperature, K
ΔT	temperature difference, $\Delta T = T_b - T_a$
t	fin thickness, mm
u, v, w	velocity components in the x, y, z directions, respectively
$\vec{V}$	velocity vector
x, y, z	Cartesian coordinates

Greek symbols

α	thermal diffusivity of air, m^2 s^{-1}
β	coefficient of thermal expansion of air, K^{-1}
ε	emissivity, dimensionless
θ	angle between two adjacent fins
λ	thermal conductivity of air, W m^{-1} K^{-1}
μ	dynamic viscosity, Pa s
ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}
φ	angle in cylindrical coordinate system
ψ	finned ratio, $\psi = 1 + \frac{2NH}{\pi D}$

Subscripts

a	ambient
b	cylinder base
exp	experiment
fit	correlation fit
z	Cartesian coordinate in vertical direction

longitudinal fins. CFD simulation and related analysis are still lacking. In the present study, CFD simulation and analysis are conducted to obtain a better understanding of the heat transfer process. The edges of thermal boundary layer and the local Nusselt number distributions are obtained, through which contributions of geometric parameters can be assessed. Particularly, with the deeper understanding on parametric effects, a more accurate correlation of Nusselt number is proposed by introducing finned ratio.

2. Physical model and numerical analysis

Based on the yielded experimental results by Ref [24], CFD models are built under similar conditions in Ref [24], including the alloys chosen for the base and fins. Since the Φ60 cylinder is heated by a Φ20 heating bar, the aluminum cylinder is considered adequately thick to form a constant temperature surface, and heat conduction is calculated along fins in the present simulation. The heat sink is surrounded by ambient air at 300 K and is cooled by the natural convection induced by thermal buoyancy force and thermal radiation. Six heat sinks are modeled and the geometric parameters are listed in Table 1, while the schematic diagram is shown in Fig. 1. For convenience, each heat sink is named by the two key parameters N and H, e.g. *N9H10*. The material of each component is listed in Table 2, with thermal properties provided. The cylinder is not included in the domain and just act as a boundary surface of constant temperature. The flow boundaries are considered as pressure boundaries at 1 atm, and $\varphi = 0$ and $\varphi = \theta$ are two symmetric planes. The pressure boundaries are treated as 300 K blackbody in the radiation computation.

Table 1
Parameters of vertical cylinder heat sinks with longitudinal fins.

N	θ/rad	H/mm	D/mm	L/mm	t/mm
9	$2\pi/9$	10, 20, 30	60	50	1
18	$\pi/9$	10, 20, 30	60	50	1

*Each heat sink is named by N and H, e.g. *N9H10*.

2.1. Governing equations

Using the Boussinesq approximation for the buoyancy term, the governing equations of the present flow, which is assumed incompressible, steady, and laminar, can be written as follows.

Continuity equation

$$\rho\nabla\cdot\left(\vec{V}\right) = 0. \tag{1}$$

Momentum equations

$$\rho\nabla\cdot\left(u\vec{V}\right) = -\frac{\partial p}{\partial x} + \mu\nabla^2 u, \tag{2}$$

$$\rho\nabla\cdot\left(v\vec{V}\right) = -\frac{\partial p}{\partial y} + \mu\nabla^2 v, \tag{3}$$

$$\rho\nabla\cdot\left(w\vec{V}\right) = -\frac{\partial p}{\partial z} + \mu\nabla^2 w + (\rho - \rho_a)g. \tag{4}$$

Energy equation of fluid

$$\rho\nabla\cdot\left(\vec{V}T\right) = \frac{\lambda}{c_p}\nabla^2 T. \tag{5}$$

Energy equation of solid

$$\nabla^2 T = 0. \tag{6}$$

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