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Design of vertical fin arrays with heat pipes used for high-power light-emitting diodes

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ABSTRACT

As Light-Emitting Diodes (LEDs) are negatively affected by high temperature, the thermal design for them is critical for better light quality, reliability and lifetime. In this work, a thermal design of vertical fin arrays with heat pipes as passive cooling was applied. The heat pipes can supply high thermal conductivity with much less weight and volume compared to copper or aluminum base and consequently less obstruction to air flow with enhanced natural convection. As the natural convection and radiation dominate heat transfer in this case, the optimum vertical fin spacing was calculated by the most used empirical correlations. Then, the design was numerical investigated by Computational Fluid Dynamics (CFD) to obtain best thermal performance. As the fin spacing was both optimized by correlations and modelling, the optimum thermal design achieved. Finally, we manufactured and tested the design experimentally which consistently approved the thermal design compared to correlations and simulation.

1. Introduction

Light-Emitting Diodes (LEDs) are the second revolution in the history of lighting with its high efficiency, tunable chromatic lights and long life time [1,2]. However, LEDs require relatively low temperature and much better thermal structure compared to traditional light sources which are designed to obtain a very high temperature. Then, a new challenge emerged to control temperature without hindering lights. In order to maintain the characteristic of high efficiency and reliability to LED system, passive cooling solutions are preferred [3]. Although high heat fluxes in LED systems exceeded 100 W/cm^2 in recent years [4,5], the generated heat in normal applications was still lower than compact and micro-sized electronic components. Therefore, thermal management under natural convection with fin arrays is attractive as they are relative more reliable, noise free and economical.

Former researchers mainly focused on the traditional heat sink with vertical or horizontal fins on vertical or horizontal base. Compared to a bare plate, fins dramatically increased the heat transfer interface between heat sink and environment and consequently achieved higher heat dissipation to volume ratio. However, the performance of heat sinks with natural convection is highly dependent on the geometry parameters. In limited volume, the vertical fin length, fin inclination and spacing are critical to optimize the performance of heat sinks. Elenbaas [6] firstly measured the heat dissipation of two plates of $h \times h$ cm^2 as function of distance. They found that the natural convection coefficient between two plates rose until a certain distance around 1 cm. Bar-Cohen and Rohsenow [7] gave theoretical analysis aimed at establishing an analytical solution for optimization of vertical arrays with natural convection. Starner and McManus [8] presented natural convection heat transfer performance data for four large rectangular fin arrays oriented vertically, 45 °C and horizontally. Form their conclusion, the vertically based orientation is the most favorable system for application of the base and fin type heat sink. Welling and Woolbridge [9] reported there existed an optimum values of the ratio of fin height to spacing for rectangular vertical fins. From then on, many researchers experimentally studied vertical and horizontal fin arrays with natural convection and consequently some simplified correlation were reported [10–13]. Recently, Yazicioğlu and Yüncü [14] reported the convective heat transfer rate from fin arrays obtained a maximum value as a function of fin spacing and height for a given base-to-ambient temperature difference. Their experiments were conducted by fin length of 250 and 300 mm, aluminum fin thickness of 3 mm, fin height varied from 5 to 25 mm and fin spacing differed from 5.75 to 85.5 mm. Besides,

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Nomenclature

c_p	specific heat (J/kgK)
F	correlations for view factor
$\hat{F}$	grey body view factor of the channel formed between adjacent fins
F_c	channel view factor
Gr'	modified Grashof number
g	acceleration due to gravity (m/s^2)
H	Fin horizontal height (mm)
h	average heat transfer coefficient (W/m^{2}K)
k	thermal conductivity (W/mK)
L	Fin vertical length (mm)
Nu	nusselt number
n	Fin number
P	pressure (Pa)
Pr	prandtle number
Q	heat transfer rate (W)
s	Fin spacing (mm)
T	temperature (°C)
t	Fin thickness (mm)
u	x component of velocity (m/s)
v	y component of velocity (m/s)
W	total fin width (mm)
w	z component of velocity (m/s)
x	x Cartesian coordinate
y	y Cartesian coordinate
z	z Cartesian coordinate

Greek symbols

α	thermal diffusivity (m^2/s)
β	volumetric thermal expansion coefficient (1/K)
ε	emissivity
μ	dynamic viscosity (Ns/m^2)
ρ	density (kg/m^3)
σ	Stefan-Boltzmann constant (W/m^{2}K^4)
υ	kinematic viscosity (m^2/s)
ΔT	temperature difference between fins and air T_f-T_a (K)

Subscript

0	initial condition
a	air
c	natural convection
f	Fin
r	radiation

three-dimensional numerical studied on this subject were carried out which obtained good agreement with experimental results [15,16]. And the numerical method can predict Nusselt numbers with correlations based on the effects of various fin geometries and spacing [17,18]. More recently, Tari and Mehrtash [19] generated a set of dimensionless correlations for the convective heat transfer rate from steady-state simulation. Their results have a practical use in electronics cooling applications due to the investigated ranges of parameters are suitable for electronics device cooling. In addition, thermal radiation which typically accounts for 25% of the total heat dissipated by natural convection finned heat sink cannot be simply neglected [20]. Rammohan Rao and Venkateshan [21] calculated natural convection and radiation contributions independently and then adding them to obtain the total heat loss from the fin arrays. Khor et al. [22] concluded that the errors induced are more than 30% when thermal radiation is neglected, and exceed 60% when thermal radiation is considered without including view factor.

In this work, we designed a heat sink with heat pipes and parallel vertical fins to control the case temperature of a 80 W LED under 70 °C, as about 75% of electric power was estimated as heat [23,24]. The used heat pipes can replace the base which significantly reduced the total weight. Beside, as the base especially on the top position will dramatically decrease the fin efficiency [19,25], the heat pipes can achieve sufficient conduction with less volume and consequently less obstruction for vertical air flow of natural convection. Compared to previous work, both empirical correlations and numerical simulations were conducted for designing the heat sink with consideration of natural convection and thermal radiation. As shown in Fig. 1, the high power LED package was positioned on the top of aluminum heat spreader with three heat pipes underneath. The designed aluminum fins connected to the three heat pipes at right-hand and left-hand side. Due to the application, there was limitation of 85 mm total width for fins at each side. As the fins dominate the convective and radiative heat transfer from heat sink to environment, the analytical solutions were conducted firstly to decide the fin spacing. Then, a set of testing with steady-state natural convection and radiation is numerically investigated according to the parameters of analytical solutions in order to obtain accurate prediction of thermal performance and achieve the optimum thermal design. Finally, the experiment tests were applied to validate the design.

2. Numerical modeling

The computational domain can be reduced to a quarter model which significantly decreased the calculation time. As shown in Fig. 2, a simplified LED package with a short cylinder heat source and a thin rectangle substrate was positioned on the aluminum heat spreader, and aluminum fins thermally connected to three heat pipes. Then heat can be conducted from heat source to fins through the heat spreader and the heat pipes. All the solid components were simulated in the air box where heat finally should be transferred to air by natural convection and thermal radiation. The dimension of aluminum heat spreader was designed as $72 \times 68 \times 10$ mm^3 and aluminum fin dimension was designed as $1 \times 68 \times 62$ mm^3 on xyz directions.

2.1. Governing equations for air

The physical phenomenon can be expressed by a set of partial differential equations which represent the conservation of mass, momentum and energy and these equations were solved using a commercial solver with adequate boundary conditions.

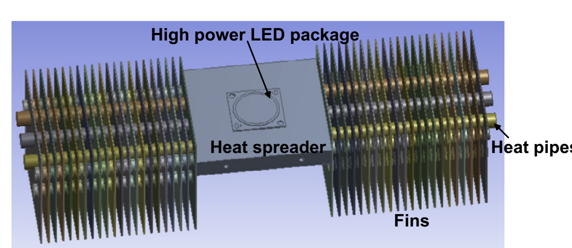

Fig. 1. Designed structure with heat pipes for air cooling of a high power LED package.