International Journal of Heat and Mass Transfer 124 (2018) 752–760

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Thermal performance improvement based on the partial heating position of a heat sink

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ARTICLE INFO

Article history:
Received 10 January 2018
Received in revised form 22 March 2018
Accepted 23 March 2018

Keywords:
Electronic cooling
Heat sink
Forced convection
Partial heating

ABSTRACT

The thermal performance of a heat sink is analyzed according to the position of the partially heated surface. Numerical models for simulating forced convection were used to analyze the heat transfer between the heat sink and ambient air. The optimal partial heating position was discussed in terms of the effects of total heat transfer rate, air velocity, the ratio of total heat sink length to partially heated surface width, the thermal conductivity of the heat sink, and the thickness of the heat sink base. Finally, a correlation was suggested to determine the partial heating position that maximizes thermal performance by using the experimental design method. It was thus possible to reduce the thermal resistance of the heat sink by up to approximately 30% by finding the optimal partial heating position.

1. Introduction

As the performance of electronic devices improves, the amount of heat generation increases, which negatively impacts their performances. For the effective cooling of electronic devices, Joo et al. [1] optimized the fin geometry and compared the thermal performance, and Li et al. [2] promoted natural convective air flow by using a perforated heat sink. Many researchers have also improved thermal performance by changing the geometry of the heat sink [3–7] and by using surrounding structures [8–11]. The heating conditions of the studies described above are the case where heat is applied to the bottom parts of the heat sink. However, in the case of power electronic devices, the size of the heat sink is typically larger than the size of the heating element due to high heat generation. An insulated gate bipolar mode transistor (IGBT) is a typical example, as shown in Fig. 1. In the case of this partial heating condition, the above-mentioned geometric shape change or the use of surrounding structures cannot be directly applied because the temperature distribution and heat transfer characteristics of the heat sink are different.

The effective cooling method of partial heating conditions has been studied by several researchers, mainly using microchannel heat sinks. In microchannel heat sinks, since the heat exchanging area is relatively small, nanofluids are mainly used as a working fluid to ensure heat transfer performance [12]. Toh et al. [13] studied the position of the maximum temperature in the microchannel heat sink under partial heating conditions. Sudhakar et al. [14] studied the transient temperature response of the microchannel heat sink under partial heating conditions, and Yoon et al. [15] found that the thermal performance was better in pin-fin than in strip-fin of the microchannel heat sink for the partial heating case. However, none of them investigated how the thermal performance can vary with the partial heating position in spite of their importance in improving the cooling performance.

According to Cho et al. [16], a microchannel heat sink with a trapezoidal header has the highest temperature when the heater is located upstream. However, this is due to the flow distribution difference according to the header shape. Therefore, it cannot be generalized that the thermal performance is always poor when the heater is located at the upstream of the heat sink. In the studies by Anbumeenakshi et al. [17] and Lelea et al. [18], the thermal resistance had the smallest value when the heater was located at the upstream of the microchannel heat sink. However, the base thickness used in these studies was on the microscale, and the effects of axial heat conduction of the heat sink base are different in macroscale heat sinks. Therefore, in a macroscale heat sink, the thermal resistance is not always low when the heater is located at the upstream. In addition, all of the studies mentioned above utilized water or various nanofluids as the working fluid. Therefore, it is necessary to study the cooling system of a macroscale heat sink that uses air as a working fluid in the partial heating condition.

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https://doi.org/10.1016/j.ijheatmasstransfer.2018.03.080
0017-9310/

Nomenclature

A	surface area [mm^2]
D_h	hydraulic diameter [mm]
H	height [mm]
h	convective heat transfer coefficient [W/m^2 K]
k	kinetic energy of turbulence [m^2/s^2]/thermal conductivity [W/m K]
L	length [mm]
N	number of fin arrays
p	pressure [Pa]
$\dot{q}$	heat flux [W/m^2]
$\dot{Q}$	heat transfer rate [W]
R_{th}	thermal resistance [°C/W]
s	space [mm]
T	temperature [°C]
t	thickness [mm]
u	velocity [m/s]

Greek symbols

η	normal direction vector
ρ	density [kg/m^3]
μ	dynamic viscosity [N s/m^2]
ω	specific dissipation rate [s^{-1}]

Subscripts

a	air
avg	average
b	base
bot	bottom
ch	channel
eff	effective
f	fin
h	heat sink
in	in
p	polyethylene
t	turbulent
top	top
u	upstream
∞	ambient

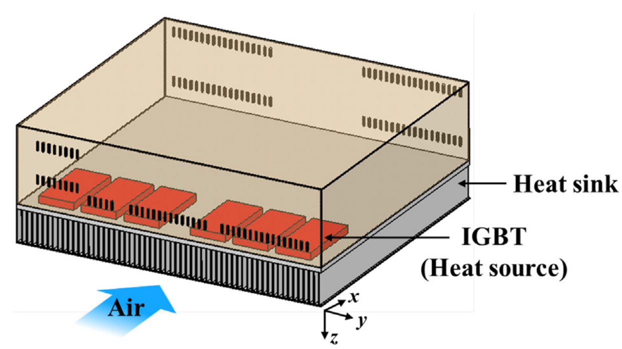

Fig. 1. IGBT cooling system in an industrial inverter.

Emekwuru et al. [19] performed a numerical analysis by dividing the position of the partially heated surface into three positions (upstream, center, and downstream) for an air-cooled heat sink. The study found that the thermal resistance of the heat sink was the lowest when the partially heated surface was located at the center. However, since the partial heating position was limited to only three positions, the central position cannot be the optimum position. In addition, the average temperature of the entire heat sink was used as a measure of thermal performance. However, this limits the accurate evaluation of the optimal cooling of a macro-scale heat sink for the partial heating condition because the heat sink typically has a large temperature distribution.

In this study, the thermal performance of an air-cooled macro-scale heat sink was numerically investigated in terms of the position of partial heating in the heat sink after verifying the numerical model with experiments. Here, the temperature used to calculate the thermal resistance of the heat sink is not the average temperature of the heat sink but the average temperature of the partially heated surface, which is suitable for power electronic device cooling in industrial fields. The effects of various factors, such as the total heat transfer rate, the velocity of the air, the ratio of total heat sink length to partially heated surface width, the thermal conductivity of the heat sink, and the thickness of the heat sink base, on the optimal partial heating position were investigated. Finally, a correlation was suggested to predict the optimal partial heating position to minimize the thermal resistance of the heat sink under various operating conditions by using the design of experiments method.

2. Mathematical modeling

2.1. Numerical model

Fig. 2 illustrates the heat sink and the heater used in this study.

The heat sink consisted of a fin and a base, with the top, left, and right sides making a physically closed system.

The following assumptions were applied for the numerical analysis.

(1) The flow is three-dimensional and steady state.
(2) The air properties do not change with temperature.
(3) Radiative heat transfer of the heat sink can be neglected.
(4) Heat transfer occurs only in the air and the heat sinks, and there is no heat transfer on the top and the sides surrounding the heat sink.

The governing equations used in the numerical analysis were as follows, and the shear stress transport (SST) $k-\omega$ model was used for turbulent flow. Validation of the numerical analysis and selection of the turbulence model are described in Section 2.3.

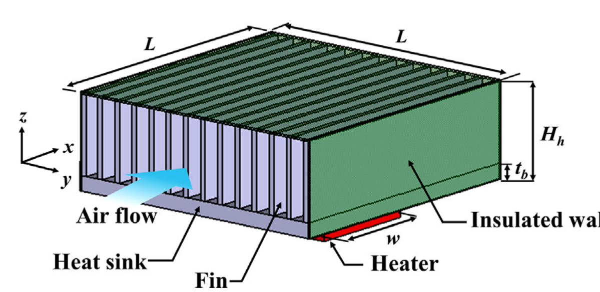

Fig. 2. Schematic of the heat sink.