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Experimental and numerical investigations of heat transfer and flow characteristics of cross-cut heat sinks

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ABSTRACT

This study aims to experimentally and numerically investigate the heat transfer and flow characteristics of plate-fin and cross-cut heat sinks. The accuracy of the measured data is verified by comparing them with the exiting correlations. In the case of the cross-cut heat sink, the effects of cross-cut lengths (L_c) and number of cross-cuts (N_c) are presented. The experimental results show that the cross-cut heat sink with L_c = 1.5 mm gives the lowest thermal resistance. In the case of a numerical study, the numerical results are validated by comparing them with the measured data. The comparison results show that the present model gives reasonable agreement with the measured data. Consequently, the present model is used to find the optimal length and number of cross-cuts. From the numerical results, it is found that the thermal resistance of a cross-cut heat sink with L_c = 1.5 mm and N_c = 6 is 16.2% lower than that of a plate-fin heat sink at the same pumping power.

1. Introduction

Over the past several years, the development-related tendency of electronic devices has been to become more compact. Meanwhile, the performance of electronic devices still remains the same or higher than that of older electronic devices. Therefore, the thermal performance improvement of electronic cooling systems is very important. The heat sink is a major part of an electronic cooling system and leads to an increase in the heat transfer area. It is commonly made from aluminum or copper. It can be divided into two main types of fin shapes: the plate fin and pin fin. Many researchers have focused on the optimum design of a plate-fin heat sink based on various fin heights and levels of thickness [1–3]. Li and Chao [1] experimentally investigated the effects of fin height, fin thickness, and Reynolds number on the thermal performance of plate-fin heat sinks. For a given fin thickness, the results showed that the thermal resistance of the plate-fin heat sink decreased with an increase in the fin height. This was due to the fact that the heat transfer area of the higher fin height was larger than that of the lower fin height. For a given fin height, the optimal levels of fin thickness that provided the lowest thermal resistance were obtained at different Reynolds numbers. Li et al. [2] presented the effects of fin height, fin thickness, and number of fins on the thermal resistance of plate-fin vapor chamber heat sinks. The experimental study was conducted under various Reynolds numbers from 10,000 to 60,000. Moreover, the surface temperature distribution of heat sinks was also presented by using infrared thermography. For a given fin height and number of fins, the effect of fin thickness on the overall thermal resistance decreased as the Reynolds number increased. In the case of various fin heights, at a given fin thickness and number of fins, the overall thermal resistance decreased with an increase in the fin height because the heat transfer area of the heat sink increased. Notably, the effect of fin height on the decrease of thermal resistance became less significant as the fin height exceeded 20 mm. Wu et al. [3] presented the theoretical model to predict the thermal performance of a plate-fin heat sink. It was developed for a wide range of Reynolds numbers. Finally, they proposed the optimized geometries of the plate-fin heat sink by using their present model.

However, some research has been done on heat transfer enhancement through increasing the turbulent flow of the working fluid at the inlet and fin tip clearance of the plate-fin heat sink [4–9]. Recently, Chingulpitak and Wongwises [4] reviewed the literature on the effect of flow directions and behaviors on the thermal performance of heat sinks. They found that only some

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Nomenclature

$c_{\varepsilon 1}$, $c_{\varepsilon 1}$, c_{μ}	turbulent model constant
c_p	specific heat (kJ/kg °C)
D_h	hydraulic diameter (m), $D_h = 2H_f W_c/(H_f + W_c)$
H	height (m)
k	turbulent kinetic energy (m^2/s^2)
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (Pa)
ΔP	pressure drop (Pa)
P_p	pumping power (W), $P_p = \dot{V}\Delta P$
Q	heat transfer rate (W)
Re	Reynolds number
R_{th}	thermal resistance (°C/W)
t	time (s)
t_f	fin thickness (m)
T	temperature (°C)
V	velocity (m/s)
$\dot{V}$	volume flow rate (m^3/s)
U	velocity vector
u, v, w	velocity components (m/s)
W	width (m)
x, y, z	Cartesian coordinates

Greek letters

Δ	differential
ε	dissipation kinetic energy (m^2/s^3)
μ	viscosity (kg/m·s)
μ_t	turbulent viscosity (kg/m·s)
Φ	dissipation function
ρ	density (kg/m^3)
σ_k	diffusion Prandtl number for k
σ_ε	diffusion Prandtl number for ε
λ	thermal conductivity (W/m^2·°C)

Subscripts

a	air
avg	average value
b	base of heat sink
c	cross-cut
ch	channel
f	fin
in	inlet
out	outlet
plate	plate-fin heat sink

papers discussed the behavior of a fluid flowing into a heat sink. Li et al. [5] studied the effects of a triangular vortex generator, which was installed in front of a test section, on the thermal resistance and pressure drop of a plain-fin heat sink. They conducted a study on the effect of the attack angles of vortex generators, the distance between both of the trailing edges, the distance between a vortex generator and heat sink, and Reynolds numbers. They concluded that the optimal position of a vortex generator occurs when the distance between the trailing edge of the vortex generator and the heat sink is zero. The proper position of the distance between the trailing edges of the vortex generator equals the width of the heat sink. Moreover, the optimized thermal performance of the plain-fin heat sink was achieved at an attack angle of 30°. At a Reynolds number of 10,000, the result indicated that the thermal resistance of the heat sink with the vortex generator was lower than that of the heat sink without vortex generator by about 27%. Zhang et al. [6] studied the effects of a shield on the heat transfer performance of a plate-fin heat sink. The shield was attached to the inner-top surface of the wind duct in order to increase the turbulent flow on the fin tip. The experimental results indicated that the optimized height and position of the shield were equal to half of the fin height and zero, respectively. Suzana et al. [7] numerically obtained the effects of the bypass flow on the thermal performance of a heat sink by using CFD code. They varied the fin density, air flow rate, and clearance area ratio to study the effects on the bypass flow over the heat sink. The results showed that the bypass flow increased with increasing fin density and clearance distance. Prstic and Bar-Cohen [8] experimentally and numerically investigated the effects of fully shrouded, partially shrouded and shielded heat sinks on the thermal performance of a plate-fin heat sink. A shield for the plate-fin heat sink was used to reduce the effect of the bypass flow. The results show that the pressure drop of the shielded heat sink was found to be much lower than that of the fully shrouded heat sink. Furthermore, the thermal resistance of the shielded heat sink was similar to that of the fully shrouded heat sink. Elshafei [9] presented the bypass flow effect of a plate-fin heat sink with various air flow rates, number of fins, and tip-to-shroud distances. The experimental results at the high number of fins showed that the air flows more through the clearance distance between the fin tips and shroud than into the fin passages. This is because the air flow was obstructed due to the increase of fin density. Finally, the proper size of the tip-to-shroud distance was obtained at a Reynolds number above 7000.

Furthermore, several research studies have been done on heat transfer improvement with an increase in the turbulent flow of a working fluid in the flow passage of a plate-fin heat sink [10,11]. Yang et al. [10] experimentally studied the pressure drop and heat transfer coefficient of plate-fin heat sinks with eight different fin patterns. The fin patterns of the tested heat sinks were: (1) plain fin; (2) delta vortex generators (delta VG) fin; (3) semi-circular VG fin; (4) delta VG and plain fin; (5) triangular VG fin; (6) triangular-attack VG fin; (7) dimple VG fin; and (8) two-group dimple VG fin. The experimental results show that the triangular-attack VG fin was considered the optimum design of fin patterns. The surface area of the triangular-attack VG fin was 12–15% lower than that of the plain fin at a frontal velocity of about 3–5 m/s and the same heat transfer capacity and pumping power. The experimental results showed that all of the vortex generators were more beneficial than the plain fin pattern was. Kim and Kim [11] experimentally studied the effects of cross-cuts on the thermal performance of heat sinks with various cross-cut lengths, positions, and number of cross-cuts. Among the variable parameters, the results showed that the cross-cut length gave the most significant effect on the thermal performance of a heat sink. The thermal resistance of a cross-cut heat sink was lower than that of plate-fin and square pin-fin heat sinks—about 5–18% and 14–16%, respectively. Moreover, the comparison results were presented in a contour map for selecting the type of heat sink, which was plotted as a function of the dimensionless pumping power and the length of the heat sink.

As mentioned above, most of the previous works have studied the effect of the geometry parameters of a plate-fin heat sink, such as thickness and fin height on thermal performance. However, there is still room for improvement in the fin shape of a plate-fin heat sink to increase the thermal performance, which is especially based on reliable manufacturing processes. Thus, this research