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Impingement heat transfer of a plate fin heat sink with fillet profile

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ABSTRACT

The fluid flow and thermal characteristics of an air-impinged plate fin heat sink was investigated numerically to understand the effect of fillet profiles at the bottom of the plate fin. The numerical model is validated against existing literature. The results of pressure drop and mean Nusselt number were obtained for cases with and without fillet profiles for variable channel widths and inlet widths. It is found that fillet profile enhances the overall thermal performance of a plate fin heat sink, and the U-shaped channel profile is capable of achieving thermal enhancement as high as approximately 13%. The fillet profiles altered the flow pattern such that the flow is smoothened near the bottom of the plate fin, reducing the negative impact induced by the rebounding flow. Furthermore, the addition of fillet profiles at the bottom of the plate fin reduces the constriction resistance and contributes to the overall thermal enhancement.

1. Introduction

Power density in electronic components is steadily increasing due to the advances in modern electronic devices. Thermal management for such devices is becoming more challenging as a result of miniaturization. Heat sinks coupled with peripherals such as jets and fans are continuously developed to meet the requirement of higher heat dissipation. Plate fin heat sink is one of the more common types of heat sink, consisting of plate fins at regular intervals. This research explores the possibility to further enhance the heat transfer efficiency of the plate fin heat sink with the consideration of a fillet profile at the bottom of the plate fin and channel.

Extensive studies have been carried out by researchers on plate fin heat sinks. Kordyban [1] compared the performance of a pin fin and a plate fin heat sink of similar dimensions. The temperature difference between the heat sink base and the ambient air for the pin fin was 50 °C, and for the plate fin it was 44 °C. Although the pin fin had a surface area of 194 cm^2, larger than the plate fin surface area of 58 cm^2, the plate fin showed a merit in heat transfer, recording a lower temperature difference. Forghan et al. [2] published test data of pin fin, plate fin and flared fin heat sinks. They found that for low air velocity between 1 m/s and 4 m/s, the thermal performance for plate fin is at least 20% better as compared to other heat sinks. Some researchers [3–6] developed analytical solutions for the heat transfer of plate fin heat sink in parallel flow. Teertstra et al. [3] developed an analytical model to predict thermal performance for air cooled plate fin heat sinks in parallel flow. The average heat transfer rate predicted was expressed in terms of Nusselt number, which combines the limiting cases of fully-developed and developing flow. The model was found to be in good agreement with the experimental results. Culham and Muzychka [4] developed a procedure that allows design parameters in a plate fin heat sink to be optimized. The model used the apparent friction factor model [5], which was asymptotic between a developing and fully developed flow. The procedure is based on the minimization of entropy generation resulting from viscous fluid effects and heat transfer. Copeland [6] suggested a laminar flow pressure drop model for parallel flow in rectangular channeled (plate fin) heat sinks. The friction factor data for developing laminar flow were taken from Shah and London [7] and fitted to an equation of the Churchill–Usagi form. Optimum dimensions of fin thickness and pitch were calculated for a variety of realistic operating conditions.

The performance of impingement heat transfer of plate fin heat sinks was reported by researchers [8–14]. Kondo and Matsushima [8] completed an experimental study and reported a zonal model of a thermal resistance prediction for impingement cooling of plate fin heat sinks. The impingement flow over the plate fins was divided into six regions. A set of correlations were proposed between the thermal resistance of the heat sink and the geometry of the plate fins. The accuracy of the predicted thermal resistance was found to be within ±25% of the experimental data. They showed that optimized plate-fin heat sinks provide 40% lower thermal resistances compared to optimized pin-fin heat sinks.

Biber [9] carried out a numerical study to determine the thermal performance of a single isothermal channel of plate fin heat sink

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Nomenclature

D_h	hydraulic diameter, m
h_b	thickness of the heat sink base, m
h_c	channel height, m
h_e	extended height at inlet, m
h	heat transfer coefficient, $\mathrm{Wm^{-2}\,K^{-1}}$
$\bar{h}$	average heat transfer coefficient, $\mathrm{Wm^{-2}\,K^{-1}}$
H	Height of heat sink, m
k	thermal conductivity, $\mathrm{Wm^{-1}\,K^{-1}}$
L	half of the heat source length or half of base length of heat sink (Fig. 1), m
L_{in}	half of inlet width, m
L_e	extended length at outlet, m
Nu	local Nusselt number
$\overline{Nu}$	mean Nusselt number along the heat source
P_{in}	inlet pressure, Pa
P_{out}	outlet pressure, Pa
q''	heat flux, $\mathrm{W/m^2}$
Re	Reynolds number
r	fillet radius, m
T_{in}	inlet temperature, K
T_w	bottom wall temperature, K
u, v, w	velocity components along x, y, z-axes, respectively, $\mathrm{m\,s^{-1}}$
w_{in}	inlet velocity, $\mathrm{m\,s^{-1}}$
W	width of heat sink, m
W_c	channel width, m
W_s	section width of heat sink, m
x, y, z	Cartesian coordinates, m

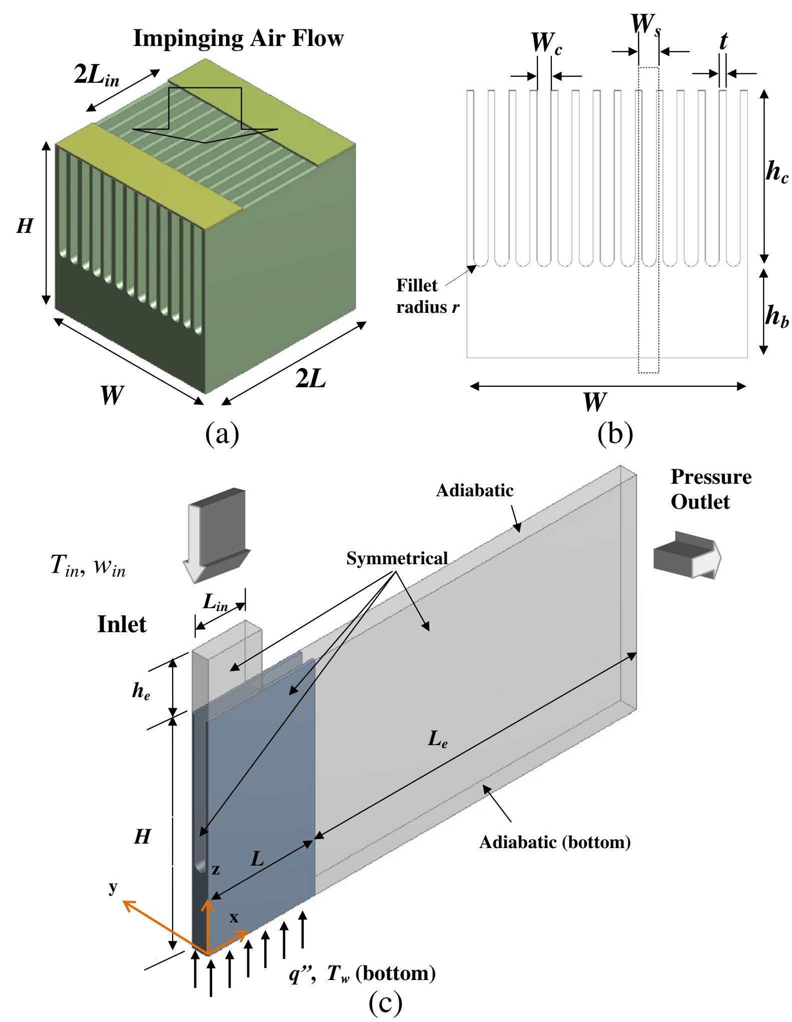

Fig. 1. Schematic of (a) physical model, (b) single section of channel and (c) domain.

with variable widths of impingement flow. Biber [9] studied many different combinations of channel parameters and obtained correlations for channel mean Nusselt number. These correlations cover a wide range of practical plate-fin heat sinks with air supplied by axial fans or other uniform or nonuniform methods. Saini and Webb [10] presented a modified Biber model [9] and validated their model by experiments. The predicted pressure drop is 13–31% lower

Table 1
Comparison of various turbulence models for L_e = 60 mm, Re = 693 and W_c = 2 mm, L_{in} = 20 mm.

Turbulence model	Pressure drop	% Discrepancy as compare to experiment [13]
Spalart–Allmaras	10.525	3.186
Standard k-epsilon	10.788	5.764
Standard k-omega	10.846	6.333