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Research Paper

Analytical analysis and experimental verification of interleaved parallelogram heat sink

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HIGHLIGHTS

- A novel air-cooled heat sink profile (IPFM) is proposed to compete with the typical design.
- It features two different perimeters with odd fin being rectangular and the rest being parallelogram.
- A new modified dimensionless parameter characterized the flow length in triangular region is proposed.
- The analytical predictions are in line with the experiments for both conventional and IPFM design.
- IPFM design shows a much lower pressure drop and a superior performance especially for dense fins.

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ABSTRACT

In this study, a novel air-cooled heat sink profile is proposed to compete with the conventional design. The new design is termed as IPFM (Interleaved Parallelogram Fin Module) which features two different geometrical perimeter shapes of fins. This new design not only gains the advantage of lower pressure drop for power saving; but also gains a material saving for less fin surface area. An assessment of flow impedance and performance between the conventional and IPFM heat sink is analytically investigated and experimentally verified. A new modified dimensionless friction factor for triangular region is proposed. The analytical predictions agree with experimental measurements for both conventional and IPFM design. In electronic cooling design, especially for cloud server air-cooled heat sink design, the flow pattern is usually laminar with Reynolds number being operated less than 2000. In this regime, the IPFM design shows 8–12% less of surface than conventional design when the flow rate is less than 10 CFM; yet the thermal performance is slightly inferior to the conventional design when the flowrate is raised towards 25 CFM. Yet in the test range of 5–25 CFM, a 10–15% lower flow impedance is observed. The smaller fin spacing, the more conspicuous reduction of flow impedance is observed. The optimization of cutting angle is around 35° for 10 CFM, and it is reduced to 15° at a larger flowrate of 20 CFM.

1. Introduction

In electronic cooling design, air-cooling is still the most widely adapted methods for being reliable and easy implementation. However, due to the poor heat transfer characteristics of air as compared to water, the heat sinks normally need to accommodate numerous fins to lower the thermal resistance [1,2]. Apparently, the fin profile plays a significant role in heat dissipation. Kraus [3] had summarized and compared various kinds of fin profiles. However, for easier manufacturing in practice, the plate fin heat sink as schematically shown in Fig. 1 is still the most widely adopted fin design. There are still many further improvements on plate fin heat sinks existing on the published literatures. For example, Chen and Wang [4] had studied inverse trapezoidal shape of heat sink for performance optimization. In recent years, with the gigantic growth of internet and mobile devices, datacenters are becoming the heart for data storage, information processing, and scientific computations. Hence, cooling of server within computer rack becomes an important issue for thermal solution providers. Unlike PCs, server design not only asks for thermal performance; but also requires lower flow impedance (pressure drop). Due to the limited space in motherboard for effective heat removal, the heat sink is usually designed with small fin spacing to accommodate more fins which may lead to huge pressure drop. In assessing the pressure drop across heat sink in association with estimation of the fan power, White [5] and Holman [6] had proposed the methodology to evaluate the friction factor *f* applicable for internal

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Nomenclature

A_c	flow channel cross sectional area, m^2
A_f	fin surface area, m^2
A_{trig}	L_1 triangular region area in IPFM design, m^2
A_{tot}	total area of $L_{1/2/3}$ 3 region in IPFM design, m^2
A_o	total surface area, including fin and base surface area, m^2
b	fin height, m
C_p	heat capacity, kJ/(kg K)
D_h	hydraulic diameter, m
$D_{h,1/2/3}$	hydraulic diameter of region 1/2/3 of IPFM fin design
F	fin number
f	friction factor, dimensionless
f_{app}	apparent friction factor, dimensionless
$f_{app,1/2}$	apparent friction factor of IPFM design in region 1/2, dimensionless
f_{fd}	fully developed friction factor, dimensionless
$f_{fd,1/2/3}$	fully developed friction factor of IPFM design in region 1/2/3, dimensionless
G_c	mass flow flux, kg/(s m^2)
$G_{c,ip}$	mass flow flux in IPFM design, kg/(s m^2)
H	heat sink total height, m
h	convection coefficient W/(m^2 °C)
$\bar{h}_{ipfm,odd}$	mean convection coefficient of odd number of fins, W/(m^2 °C)
$\bar{h}_{ipfm,even}$	mean convection coefficient of even number of fins, W/(m^2 °C)
$\bar{h}_{ipfm}$	mean convection coefficient of IPFM heat sink, W/(m^2 °C)
K_i	inlet friction factor, dimensionless
$K_{i,1}$	inlet friction factor in IPFM design of region 1, dimensionless
K_o	outlet friction factor, dimensionless
$K_{o,3}$	outlet friction factor in IPFM design of region 3, dimensionless
k	conductivity, W/(m °C)
k_f	air conductivity, W/(m °C)
L	heat sink length, m
L'	equivalent length of L_1 region, m
$L_1/L_2/L_3$	3 section length of IPFM design, m
L_{fd}	fully developed length, m
$L_{fd,2/3}$	fully developed length in region 2/3 of IPFM design, m
L_{hy}	hydraulic developing length, m
$L_{hy1,2}$	hydraulic developing length in region 1/2 of IPFM design, m
Nu	Nusselt number
$\overline{Nu}$	averaged Nusselt number
$\overline{Nu}_{1/2/3}$	mean Nu number of IPFM design in region 1/2/3
P_h	peripheral length, m
Pr	Prandtl number, dimensionless
Q	power input, W
Re	Reynolds number, dimensionless
Re_b	$\frac{\rho v s}{\mu}\frac{s}{L}$ modified Reynolds number, dimensionless
R_{th}	thermal resistance, °C/W
s	fin spacing, m
s_1	fin spacing (2 s) of region 1 in IPFM, m
s_2	fin spacing (1 s) of region 2 in IPFM, m
T	temperature, °C
T_∞	ambient temperature, °C
t	fin thickness, m
V	airflow volume, CFM
v_f	frontal velocity, m/s
v	channel velocity, m/s
W	heat sink width, m
x^+	$\frac{x}{Re\, D_h}$, dimensionless flow length, dimensionless
${x_1}^+$	$x^+ \frac{b-s}{b+s}$, modified dimensionless flow length in region 1 of IPFM design, dimensionless
x^*	$\frac{x}{Re\, Pr\, D_h}$, dimensionless flow length, dimensionless

Greek letter

α^*	channel aspect ratio, dimensionless
ΔP	pressure drop, Pa
ΔP_a	acceleration pressure drop, Pa
ΔP_i	inlet pressure drop, Pa
$\Delta P_{i,1}$	inlet pressure dropin region 1 of IPFM design, Pa
ΔP_o	outlet pressure drop, Pa
$\Delta P_{o,3}$	outlet pressure drop in region 3 of IPFM design, Pa
ΔP_{dev}	developing region pressure drop, Pa
$\Delta P_{dev,1/2}$	developing pressure drop in region 1/2/3 of IPMF design, Pa
ΔP_{fd}	fully developed region pressure drop, Pa
$\Delta P_{fd,2/3}$	fully developed pressure dropin region 2/3 of IPFM design Pa
ΔP_{tot}	total pressure drop, Pa
ΔP_{ipfm}	total pressure drop of IPFM design, Pa
η	fin efficiency, dimensionless
η_o	overall efficiency, dimensionless
μ	dynamic viscosity, kg/(m s)
ρ_i	heat sink inlet fluid density, kg/m^3
ρ_o	heat sink outlet fluid density, kg/m^3
σ	free area ratio of channel, dimensionless
$\sigma_1/\sigma_2/\sigma_3$	free area ratio of region 1/2/3 in IPFM design, dimensionless

pipe flow. For rectangular and flat channels, by introducing the hydraulic diameter D_h ($4A_c/P_h$), the friction factor f subjected to fully developed flow can be evaluated. This estimation is accurate for large scale cooling devices since the majority portion of flow path is fully developed. However, in PC or server applications, the small fin spacing will cause significant pressure loss in heat sink inlet and outlet for contraction and expansion. In coping with the entrance and exit losses for developing flow in parallel plate channels, Kays and London [7] and Webb [8] had proposed correlations for inlet K_i and outlet K_o loss coefficients for parallel plate channels. The presented information is based on the assumption of fully developed flow; however, the developing flow region occupies a significant length and must be included in design in electronic cooling heat sink applications. Thus, Guyer [9] proposed a concept of apparent frictional factor f_{app} to evaluate the friction coefficient. By defining a dimensionless length parameter x^+, it is

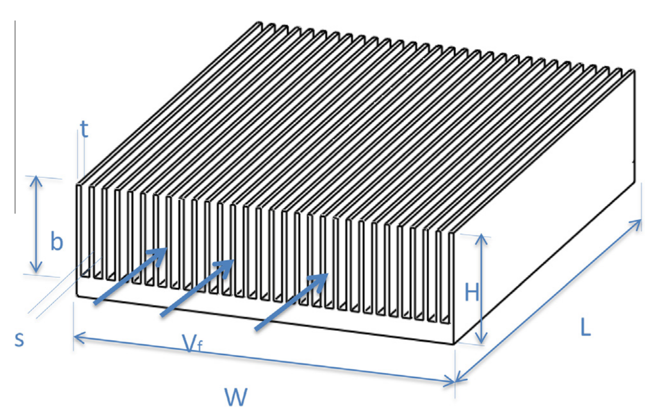

Fig. 1. Schematic of the configuration of typical air-cooled plate heat sink used in electronic cooling.