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An experimental and numerical investigation of chevron fin structures in serpentine minichannel heat sinks

Ahmed F. Al-Neama [a,b,*], Zinedine Khatir [a], Nikil Kapur [a], Jonathan Summers [a], Harvey M. Thompson [a]

[a] Institute of Thermofluids, School of Mechanical Engineering, University of Leeds, LS2 9JT, United Kingdom
[b] Department of Mechanical Engineering, Faculty of Engineering, University of Mosul, Iraq

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ABSTRACT

Water-cooled micro/minichannel heat sinks are an important component in managing the temperature of electronic components, particularly where high density of heat rejection is required. This study examines the potential to decrease the thermal resistance and enhance convective heat transfer of a serpentine heat exchanger by introducing chevron fins which create secondary flow paths. This novel design is found to significantly reduce both the pressure drop across the heat exchanger and the total thermal resistance by up to 60% and 10%, respectively, and enhance the average Nusselt number by 15%. A three-dimensional conjugate heat transfer model is developed and validated against experimental measurements, before being used to carry out a parametric study involving the chevron oblique angle, secondary channel width and heat flux. The design of the serpentine minichannel with chevron fins is then optimised in terms of the minichannel width, minichannel number and chevron oblique angle. A 50 point Optimal Latin Hypercubes Design of Experiment is constructed within the design variable space, using a permutation genetic algorithm, and accurate metamodels built using Radial Basis Functions. A Pareto front is constructed which enables designers to explore appropriate compromises between designs with low pressure drop and those with low thermal resistance.

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1. Introduction

The inexorable miniaturisation of electronic components and increase in electronic packaging density is driving the development of efficient cooling methods to preserve component lifespan and reliability. Single-phase micro/minichannel heat sinks are one promising option with the ability to dissipate high heat fluxes over small areas [1]. The use of water-cooled microchannel heat sinks with straight rectangular microchannels was first introduced by Tuckerman and Pease [2] in 1981. In their experiments, they demonstrated that heat fluxes of up to 790 W/cm^2 could be dissipated with a large pressure drop of 214 kPa across the heat sink for a substrate temperature rise of 71 °C. They found that the heat transfer coefficient can be increased by decreasing the hydraulic diameter of the channels at the expense of increased pressure drop. After this pioneering work, several studies have investigated the fluid flow and heat transfer characteristics of microchannel heat sinks, see e.g. the recent review of Ghani et al. [3]. Flow boiling (two-phase flow) microchannel heat sinks, on the other hand, have also been widely studied by researchers due to their ability to dissipate high heat fluxes with lower pumping power compared with single-phase liquid microchannel heat sinks due to utilization of the latent heat of vaporization of the liquid [4]. However, at higher heat fluxes, microchannel flow boiling suffers from pressure fluctuation and flow instabilities, which reduces and degrades the heat transfer characteristics in microchannel heat sinks [5].

Fedorov and Viskanta [6], Qu and Mudawar [7] and Li et al. [8] carried out numerical studies of the fluid flow and heat transfer properties of a 1 cm^2 silicon wafer microchannel heat sink with straight rectangular microchannels that had previously been studied experimentally by Kawano et al. [9]. Li et al. [8] highlighted the importance of including the dependence of the thermophysical properties of the fluid (i.e. density and viscosity) on temperature to accurately capture the linear increase in the channel wall temperature, as included in the study of Fedorov and Viskanta [6].

This rise in surface temperature limits the efficiency of the conventional straight microchannel heat sinks; to enhance the convective heat transfer and achieve a more homogeneous temperature distribution, secondary flows can be produced by adding smaller channels between the main flow channels. Steinke and Kandlikar

* Corresponding author at: Institute of Thermofluids, School of Mechanical Engineering, University of Leeds, LS2 9JT, United Kingdom.
E-mail address: ahmedfalneama@gmail.com (A.F. Al-Neama).

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Nomenclature

A_{base}	base area of minichannel [m^2]
A_{ch}	cross-sectional area of minichannel [m^2]
A_{eff}	effective heat transfer area per minichannel [m^2]
A_{fin}	surface area of fin [m^2]
A_h	bottom heated area of the MCHS [m^2]
Cp_f	specific heat of fluid [J/kg K]
D_h	hydraulic diameter [m]
h	convective heat transfer coefficient [W/m^2 K]
H_b	substrate thickness [m]
H_{ch}	minichannel height [m]
k	turbulent kinetic energy [m^2/s^2]
k_f	thermal conductivity of fluid [W/m K]
k_s	thermal conductivity of copper block [W/m K]
k_T	turbulent thermal conductivity [W/m K]
L	heat sink Length [m]
L_{ch}	minichannel length [m]
l_f	chevron fin length [m]
l_{sc}	secondary microchannel length [m]
n	number of minichannel
N_{cf}	number of chevron fin
Nu	Nusselt number
ΔP	total pressure drop [Pa]
p_f	fin pitch ($= l_f + l_{sc}$) [m]
P_{cf}	perimeter of chevron fin [m]
P_w	the wetted perimeter [m]
Q_{in}	volumetric flow rate [m^3/sec]
q	heat transfer rate [W]
Re	Reynolds number
R_{th}	total thermal resistance [K/W]
$T_{f,avg}$	fluid bulk temperature [°C]
$T_{f,in}$	inlet fluid temperature [°C]
$T_{f,out}$	outlet fluid temperature [°C]
$T_{w,avg}$	average minichannel base temperature [°C]
$T_{w,tci}$	minichannel base temperature at thermocouple location (i = 1–4), [°C]
V_{ch}	velocity in minichannel [m/s]
W	heat sink width [m]
W_{ch}	minichannel width [m]
W_{sc}	secondary microchannel width [m]
W_w	fin width [m]

Greek symbols

η_f	fin efficiency
ρ_f	fluid density [kg/m^3]
μ_f	dynamic viscosity of fluid [kg/m s]
μ_T	turbulent viscosity [kg/m s]
ε	channel surface roughness [μm]
θ	oblique angle [degree]
ω	specific dissipation rate [1/sec]

Subscripts

avg	average
f	fluid (Water)
in	inlet
max	maximum
out	outlet
s	solid
tci	location of the thermocouple along the flow channel
Γ	interface between the fluid and solid

[10] and Kandlikar and Grande [11] suggested several techniques to promote heat transfer in microchannel heat sinks including increasing the surface area of heat transfer and improving the mixing flow using interrupted and staggered strip-fin designs. Among these, two techniques have been proposed by Steinke and Kandlikar [10] to generate secondary flow in a microchannel heat sink. The first was to add smaller secondary channels which induce flow between the main channels, while the second makes use of the Venturi effect. Both methods enhance the convective heat transfer by increasing fluid mixing without inducing significantly larger pressure losses. Advanced microchannel structures, such as stacked microchannels [12], double-layered microchannels [13,14], tree-shaped microchannel networks [15,16], strip-fin microchannels [17,18] and micro-pin fins [19–22] have also been proposed to enhance temperature uniformity and reduce the pressure drop.

Xu et al. [23,24] carried out an experimental and then a numerical study on silicon microchannel heat sinks, comprising ten parallel triangular microchannels along the flow direction with five transverse trapezoid microchambers, separating the whole straight microchannels into six independent zones. The transverse microchannel redevelops the thermal boundary layer at the onset of each zone, which significantly improved the heat transfer coefficient. In addition, they observed that the pressure drop (ΔP) decreased by 27% for the interrupted microchannel design compared to the conventional microchannel heat sink. Tsuzuki et al. [25] used Computational Fluid Dynamics (CFD) to optimize the geometry of wavy, zigzag and S-shaped fins. In their study, the parametric dependence of fin angle, guiding wing, thickness, length, and roundness were studied. They evaluated the thermal and hydraulic performance by calculating the heat transfer and pressure drop, and showed that the fin angle was the most influential parameter on the performance of the microchannel heat exchanger. Kuppusamy et al. [26] studied microchannel heat sinks with alternating slanted passages which was proposed by Steinke and Kandlikar [10], and showed that the latter led to reductions in the average thermal boundary layer thickness, enhancing the thermal performance with a slight reduction in ΔP. Comparing their results with a conventional microchannel heat sink, they showed that the slanted passages enabled thermal resistance and pressure drop to be reduced simultaneously by 76% and 6% respectively.

Recently, Ghani et al. [27] studied numerically the fluid flow and heat transfer characteristics of microchannel heat sinks with rectangular ribs and secondary oblique channels in alternating directions at different Reynolds number (Re) ranging from 100 to 500. This type of heat sink has been compared with microchannels with secondary oblique channels, microchannels with rectangular ribs and straight rectangular microchannels. The proposed design provides a larger heat transfer area in comparison with the other three microchannel geometries while also reducing the pressure drop caused by ribs by around 50%. Three parameters were selected to explore the effects of geometrical parameters on the hydrothermal performance of the heat sink: (i) the relative secondary channel width, (ii) the relative rib width, (iii) the angle of the secondary channel. The results revealed that the average Nusselt number and friction factor increase as the angle of the secondary channel decreases and decrease as the relative secondary channel width increases, while the friction factor increases as the relative rib width increases.

Lee et al. [28,29] modified conventional straight fin microchannels by breaking the continuous fins into oblique sections. Their

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