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In situ characterization of enhanced thermal performance by periodic nanostructures on the surface of a microchannel

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ABSTRACT

In situ formation of nanofins (i.e. isolated and dispersed precipitation of nanoparticles) on flow conduits induces changes to the surface morphology for heat exchange, increasing effective surface area and enhancing their cooling performance. Furthermore, the existence of these nanostructures has a significant influence on heat transfer, as opposed to the thermo-physical properties (e.g. thermal conductivity) of cooling fluids enhanced by addition of nanoparticles. This study conducted an *in situ* analysis exploring the effects of nanoscale surface modifications on forced convective heat transfer and thermal performance. A numerical heat transfer analysis, based on the conductive/convective heat transfer between a fin base and deionized water (DIW), was performed with the assumption that periodic nanostructures existed on the heated microchannel surface. Predictions of enhanced thermal performance (between 37% and 143%) from the resultant increase in the effective heat transfer area were validated for flows over these nanostructures (artificial nanofins) fabricated by SFIL (step and flash imprint lithography) inside a microchannel. Polydimethylsiloxane microchannels are bonded to a silicon wafer containing thin-film thermocouple arrays deposited onto artificial nanofins, which had been fabricated *a priori* for the *in situ* characterization of thermal performance. It was found that the convective heat transfer Nusselt number (Nu) increased from 61% to 110%. Additionally, a theoretical analysis of the pressure drop was also successfully achieved for a comprehensive understanding of the heat transfer characteristics at the fluid-wall interface of a microchannel.

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1. Introduction

Cooling by heat exchange is essential for the performance of electronic parts and various mechanical systems. As electronics and equipment become smaller, overheating can lead to severe malfunctions and damage. Microchannels for high-performance heat sinking were first proposed by Tuckerman and Pease [1], and has emerged as one of the more promising techniques. They demonstrated that a silicon microchannel 50 μm wide and 300 μm deep can remove 790 W/cm^2 of heat with a temperature rise of 71 °C between the substrate and the coolant [1]. Since this first report, similar findings have been reported with the many of the subsequent studies focused on the comparison between theoretical models and experimental results. The resulting friction factors and Nusselt numbers (*Nu*) within these studies were inconsistent with predictions, and controversy arose because several research groups reported higher *Nu* numbers versus predictions [2–4] whereas others reported lower values [5,6]. Furthermore, many researchers have reported that *Nu* in fully developed laminar flow has a positive relationship with Reynolds number (*Re*) in microchannels with hydraulic diameters ranging from 50 to 600 μm. These results disagree with conventional heat transfer predictions that *Nu* is constant regardless of *Re*. Kandlikar et al. attribute this discrepancy to entrance region effects, uncertainties in experimental measurements, and ambiguity in the determination of the thermal boundary conditions [7].

Previous studies have argued that anomalous behaviours in microchannel heat transfer experiments can be explained by surface roughness effect. For instance, Kandlikar et al. found that local *Nu* is dependent on the surface roughness of very small diameter tubes (smoother surfaces yield lower heat transfer coefficient values), whereas surface roughness is insignificant in heat transfer in

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Nomenclature

Symbols

c_p	specific heat capacity (J/kg K)
f	friction factor
$f{\cdot}Re$	friction constant
k	thermal conductivity of fin material (W/m K)
h	convective heat transfer coefficient (W/m^2 K)
h_m	height of microchannel (m)
p	pressure (Pa)
$\ddot{q}$	heat flux (W/m^2)
u	flow velocity (m/s)
$\vec{v}$	3D velocity field (m/s)
D	fin diameter (m)
D_h	hydraulic diameter (m)
L	fin length (m)
Nu	Nusselt number
Pe	Pêclet number
Re	Reynolds number
T_f	fluid temperature (K)
T_w	wall temperature (K)
T_∞	temperature of free stream (K)
Q_{tot}	total heat flux (W/m^2)
α	thermal diffusivity (m^2/s)
μ	dynamic viscosity (Pa s)
ν	kinematic viscosity (m/s^2)
ρ	fluid density (kg/m^3)
ω	measurement uncertainty
∇	spatial gradient (vector) operator
$\bullet$	vector dot product

Abbreviations

DAQ	data acquisition
DIW	de-ionized water
DRIE	deep reactive ion etching
EBL	electron beam lithography
GCI	grid convergence index
PDMS	polydimethylsiloxane
SFIL	step and flash imprint lithography
TFT	thin-film thermocouple
UV	ultra violet

larger diameter tubes [8]. Qu et al. reported that *Nu* determined by experiments were much lower than those from numerical predictions [5]. They demonstrated that surface roughness on the walls of microchannels might have contributed to these results by decreasing the velocity and temperature gradient near the wall. A model that illustrates roughness-viscosity effects was proposed by the authors. Furthermore, Liu et al. numerically evaluated various types of grooved microchannels to investigate the role of surface microstructures on forced convective heat transfer in microchannels [9]. They showed that heat transfer was enhanced by the vortices that appeared in the grooves.

Meanwhile, there have been numerous efforts to improve heat transfer by adopting nanofluids. It has been agreed that convective heat transfer coefficient is enhanced by using nanofluids by various research groups including Pak and Cho [10], Xuan and Li [11], Wen and Ding [12], Lee and Mudawar [13], Jung et al. [14], Yu et al. [15], Anoop et al. [16], and Yu et al. [17]. However, developing theoretical models for the enhancement of heat transfer of nanofluids is incomplete as well as controversial. To illustrate, Buongiorno et al. and Heyhat and Kowsary suggested that Brownian diffusion and thermophoresis are the only dominant slip mechanism in nanofluids [18,19] whereas Nelson et al. and Yu et al. put emphasis on surface modification by precipitation of nanoparticles on convective heat transfer [15–17,20]. In particular, it was reported that surface roughness had a significant effect on velocity and temperature profiles near the wall, as well as the resultant *Nu* number [5]. As a result, morphological surface modification techniques were tried, resulting in significant enhancements to heat transfer within a microchannel [21–24]. Recent advancements in nanotechnology have enabled the application of nanostructures (such as nanopores and nanofins) to the surface of microchannels [9,25]. Other experimental studies have shown that grooved microstructures provide a more uniform heat transfer when compared to those with a plane surface [26], and isolated precipitation of nanoparticles onto the heated wall enhanced thermal performance due to an increase of the effective heat transfer area [15–17]. It can be concluded from these results that surface geometry plays the dominant role in the enhancement of convective heat transfer for microchannels.

In this study, enhanced thermal performance by periodic nanostructures fabricated over heat exchange surfaces was numerically and experimentally investigated. In particular, the purpose of this study is to confirm heat transfer enhancement by surface modification at the nanoscale and to check the applicability of nanofins for enhanced cooling performance. The numerical analysis for heat conduction/convection between fin base and fluid was performed using deionized water (DIW) with the assumption of nanostructures existing on the heated microchannel surface. To experimentally verify the numerical results, surface nanostructures were fabricated on the heat exchange surface using step and flash imprint lithography (SFIL). Temperature measurements were performed using the miniaturized metrology technique (temperature nanosensors with thin-film thermocouple, also known as TFT) in order to prevent disturbance to the flow. Poly dimethyl siloxane (PDMS) microchannels were fabricated by the soft lithography technique and integrated with a TFT array deposited on a pin-fin array (which was fabricated in advance) for measuring the wall temperature gradient in the flow direction and to obtain the wall heat flux during the flow of coolants.

2. Numerical modeling

2.1. Heat transfer through nanofins

A numerical analysis was performed to predict the enhancement level of heat transfer by the artificially fabricated nanostructures that were integrated into the wall of the microchannels (as shown in Fig. 1). 3D analysis, including all of the pin geometry, is desirable for an accurate assessment – but was impossible due to the high number of nanofins on the wall (approximately 23 million) versus computational capability, as well as the nanofin's extremely small size when compared to that of the microchannel (0.14 µm diameter, 0.35 µm height as an average). Thus, the numerical simulation was carried out based on the conductive/convective heat transfer analysis for a pin-fin array.

The alignment of the fin arrays (N rows × V columns) are shown in Fig. 1, and the pin-fin arrays' in-line alignment are under a cross-flow condition. The total heat flux (Q_{tot}) was determined by solving the conductive and convective heat transfer equation around the fin arrays in line, as follows: [27]

$$Q_{tot} = \left[\frac{\pi}{4}NVkD^2 m \tanh(mL) + h_w\left(A - \frac{\pi}{4}NVD^2\right)\right] \cdot (T_w - T_\infty) \quad (1)$$

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