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Pitch and aspect ratio effects on single-phase heat transfer through microscale pin fin heat sinks

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ABSTRACT

Heat transfer and pressure drop of single-phase liquid flow is characterized in eight micro pin fin heat sinks with varied pitch and aspect ratios. The pins are diamond shaped with respect to the flow and have transverse pitch-to-diameter (S_T/D_h) and aspect (H_{pin}/D_h) ratio variations in the range of 1.7–3.0 and 0.7–3.2, respectively. The fluid used is PF-5060 over a Reynolds numbers (based on pin fin hydraulic diameter) range of 8–1189. Flow visualization is performed on all the heat sinks and flow transition into unsteady vortex shedding is observed only in those with specific pitch and aspect ratios. Flow visualization reveals upstream propagation of the onset of vortex shedding along the length of heat sink with an increase in Reynolds number. The existence of vortex shedding in micro pin fin heat sinks affects the prediction error of heat transfer correlations in literature. To address this gap, together with data from a prior study using liquid nitrogen [1], separate correlations are developed to predict Nu in the steady and unsteady regimes. The resulting correlation for the unsteady regime shows significantly decreased dependency of Nusselt on the Prandtl number compared to the non-vortex-shedding condition.

1. Introduction

Single-phase and phase-change heat transfer within microchannel heat sinks has emerged as an important thermal management solution for applications such as computer chip cooling and high power electronics and avionics [2–5]. Microscale pin fin heat sinks (μPFHS) geometry were first introduced by Tuckerman in 1984 [6]. Two decades later, Peles et al. in 2005 [7] documented lower thermal resistance using pin fin microchannel heat sinks. They attributed the lower resistance to increased flow mixing and reduced flow mal-distribution. Several experimental studies by the same research group [8–10] on μPFHSs with different pin fin shapes and pitch and aspect ratios showed that heat transfer increased proportionally with Reynolds number and increased flow confinement (e.g. by decreasing pitch ratio) with trade off in higher pressure drop.

When compared with flow over a bank of tubes, in the case of pin fin heat sinks, the boundary layers on either end of the pin fins within the heat sinks would affect the heat transfer rate. Koşar and Peles [9] found that top and bottom wall effects (end walls) on heat transfer diminished for $Re_{Dh} > 100$ and existing correlations developed for flow across bank of tubes like those presented by Zukauskas [11] would predict the results with good agreement. Zukauskas [11] presented correlations based on three flow regimes- a predominantly laminar flow regime for $R_{Dh} < 10^3$, a mixed regime between $5 \times 10^2 < Re_{Dh} < 2 \times 10^5$, and a predominantly turbulent regime for $Re_{Dh} > 2 \times 10^5$. The author defined mixed flow as a pattern of flow in which a laminar boundary layer developed on the tube under the influence of a turbulent flow and with intense vortical flow in the rear. Zukauskas also noted that the critical Reynolds at which flow transitioned from predominantly laminar to a mixed flow regime could vary depending on tube longitudinal and transverse pitches.

Short et al. [12] characterized flow and heat transfer in millimeter-sized pin fin heat sinks with air flow. They observed a change in slope of friction factor, f, with Reynolds number at $Re_{Dh} = 1000$. For $Re_{Dh} < 1000$, f was strongly dependent on Reynolds number and pin longitudinal spacing (S_L). For $R_{Dh} > 1000$, f was weakly dependent on pin fin height and Reynolds number. The authors attributed this behavior to transition from laminar-like to fully turbulent flow. In contrast with Short et al. [12], Prasher et al. [2] observed a change in the trends of f with Reynolds number for water flow in micro pin heat sinks at $Re_{Dh} \cong 100$. The experimental f data in their study were a strong function of Reynolds number for $Re_{Dh} < 100$ and for higher Re_{Dh}, f was not very sensitive to Reynolds number and could be correlated with $Re_{Dh}^{-0.1}$. The same trends as in Prasher et al. [2] study were reported

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Nomenclature

A	heat sink bottom heated area, $= W \times L$ (m^2)
$A_{\mu PFHS}$	heat sink heated fluid surface area (m^2)
A_{pin}	wetted surface area of a pin fin, $= 4W_{pin} \times H_{pin}$ (m^2)
$A_{pin,cross}$	pin fin cross section area (m^2)
A_{min}	minimum flow area within the pin fin array (m^2)
c_p	specific heat capacity ($J/kg \cdot K$)
D_{ch}	channel hydraulic diameter calculated at the inlet of μPFHS based on W and H_{pin} (m)
D_h	hydraulic diameter based on pin fin size (or tube size in bank of tubes), $= \sqrt{2}W_{pin}$ (m)
D_{Lc_Amin}	heat sink hydraulic diameter calculated based on A_{min} and P_{min} (m)
f	Darcy friction factor
f_G	geometry independent friction factor
f	frequency
h	average heat transfer coefficient ($W/m^2 \cdot K$)
H	height (m)
k	thermal conductivity ($W/m \cdot K$)
L	heat sink heated length (m)
m	fin parameter
$\dot{m}$	mass flow rate (kg/s)
MAE	mean absolute error
n	power index of Pr in heat transfer correlations
N_{pin}	number of pin fins in the heat sink
N_{row}	number of pin fin rows in flow direction
Nu_{Dh}	average Nusselt number based on D_h
Nu_{Lc_Amin}	average Nusselt number based on D_{Lc_Amin}
Nu_{ch}	average Nusselt number based on D_{ch}
P_{min}	wetted perimeter associated to A_{min} (m)
Pr	Prandtle number
$p - value$	a measure to determine statistical significance of an independent variable in the regression model
ΔP	pressure drop (Pa)
ΔP_{exp}	measured pressure drop (Pa)
$\mu PFHS$	micro pin fin heat sink
q''	heat flux (W/m^2)
Re_{Dh}	Reynolds number based on D_h
Re_{Lc_Amin}	Reynolds number based on D_{Lc_Amin}
Re_{Dh_CHE}	Reynolds number based on a channel hydraulic diameter which was obtained through compact heat exchanger approach, details can be found in [8]
$Re_{\max\_ch}$	Reynolds number calculated based on D_{ch} and U_{max}
R^2	a statistic term, coefficient of determination
S_D	diagonal pitch (m)
S_L	longitudinal pitch (m)
S_T	Transverse pitch (m)
St	Strouhal number, $= \frac{fD_h}{U_{\max}}$
T_{bulk}	average bulk fluid temperature between the inlet and exit (°C)
T_{wall}	heat sink wall temperature (°C)
U_{max}	maximum velocity across pin fin array (m/s)
W	heat sink heated width (m)
W_{pin}	pin fin side width (m)

Greek symbols

α	Aspect ratio, $= H_{pin}/D_h$
β	Pitch ratio, $= S_{L\ or\ T}/D_h$
η	fin efficiency
δ_h	hydrodynamic boundary layer thickness (m)
δ_t	thermal boundary layer thickness (m)
μ	viscosity ($Pa \cdot s$)
ρ	density (kg/m^3)

Subscripts

cr	critical, associated to the onset of unsteady vortex shedding
exp	experimental value
L	longitudinal
$noVS$	without transition to unsteady vortex shedding
pin	attributed to the pin fin array
$pred$	predicted value
T	transverse
$wall$	estimated at the wall
VS	with transition to unsteady vortex shedding

by Koşar et al. [8] where the slope change was seen at $Re_{Dh} \cong 60$. Hence the transition to mixed flow introduced by Zukauskas [11] appears to occur at much lower Reynolds numbers in μPFHSs, irrespective of the pin geometry or arrangement.

Brunschwiler et al. [3] reported a hydrodynamic flow regime transition within in-line μPFHSs by observing higher rate of increase for measured pressure drop across the heat sinks beyond a certain flow rate. Infrared images from the surface of one of the tested in-line μPFHS after flow transition showed lower temperature non-uniformity. The authors suggested that this better temperature uniformity was likely due to transition to vortex shedding. In a later study by the same research group, Renfer et al. [13] performed quantitative visualization using micro-particle image velocimetry on in-line circular micro pin fin heat sinks with pin fin diameter of 100 μm, $S_{L\ or\ T}/D_h = 2$, and with variation in aspect ratio $H_{pin}/D_h = 1\ \&\ 2$. For the heat sink with smaller aspect ratio no vortex shedding was observed for Re_{Dh} up to 330. The heat sink with $H/D_h = 2$ exhibited unsteady vortex shedding, at and above $Re_{Dh} = 160$. Laser-induced fluorescence and IR thermography were performed in a follow-on study to characterize the effect of vortex shedding on heat transfer [14]. The results showed that local Nu number increased by up to 230% in the presence of vortex shedding.

Single-phase cryogenic (LN_2) heat transfer using four diamond-shaped μPFHSs was characterized by Rasouli and Narayanan [1] in Re_{Dh} ranging from 108 to 507. The global heat transfer trends showed that, at a given Reynolds number (Re), heat sinks of same pin fin size and aspect ratio but larger pitch ratio resulted in higher Nusselt number (Nu). Through qualitative infrared visualization of the surface temperature using a surrogate fluid (PF-5060), they observed periodic variations in surface temperature in the heat sinks with coarser pin fin arrangement, suggesting the existence of vortex shedding.

While prior experimental studies have documented the effect of vortex shedding on heat transfer for limited pin fin geometries, the bounds of the geometrical parameters such as aspect ratio and pitch ratio that result in vortex shedding are as yet unclear. Moreover, a generalized correlation that includes the effect of vortex shedding in μPFHS does not exist. This gap in open literature forms the motivation behind the present study. Single-phase heat transfer and pressure drop of Performance Fluid (PF-5060) flow in eight heat sinks is investigated in Re_{Dh} from 8 to 1189. Flow transition into unsteady vortex shedding is gleaned using high-speed flow visualization. A map of geometrical parameters that demarcates vortex-shedding and non-vortex-shedding regimes is proposed. It is shown that the existence of vortex shedding in micro pin fin heat