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Effects of pin fin shape and configuration on the single-phase heat transfer characteristics of jet impingement on micro pin fins

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ABSTRACT

Experiments to investigate the effects of cross flow area and pin fin shapes on the single-phase impingement point heat transfer coefficients of jet impingement on micro pin fins have been performed. The micro pin fins were circular, hydrofoil, square, and elliptical pin fins with characteristic lengths ranging from 50 μm to 125 μm. Jet parameters were fixed with an orifice diameter of 2.0 mm and a stand-off ratio of 0.865. Using R134a as the working fluid, Reynolds numbers based on jet diameter were varied from 8000 to 80,000. Cross flow effects were found to have little to no significance on the heat transfer coefficients while area enhancement played a major role on the overall heat transfer enhancement. When comparing the various micro pin fins, the circular pin fins along with the square pin fins displayed the highest heat transfer coefficients for a given jet velocity.

1. Introduction

In the past few decades, the microelectronics industry has been growing very rapidly. This growth is mainly driven by advances in microelectronics fabrication technologies and the ever increasing desire to improve performance of microelectronics through device scaling. Since the early 1960s, the number of transistors on a chip has been increasing steadily and now exceeds one billion. This trend follows Moore's law, which predicts that component density and performance of integrated circuits would double every 18 months. However, the miniaturization of the electronics semiconductors along with their increasing operating frequencies has led to an exponentially spiraling increase of the devices' heat flux. Unless cooled properly below their typical maximum operating temperature (e.g., 85 °C), resulting high surface temperatures will lead to the degradation in performance and failure of the electronic chips. Besides ensuring their reliability, there are many advantages of operating electronics at low temperature including faster switching times, reduction in leakage current, and increased circuit speed.

One of the most promising electronics cooling technologies is the use of single-phase liquid jet impingement. Jet impingement, as an effective cooling technique, has matured through many years of research and industrial applications. Some of the literature on jet impingement can be found in [1–4], and most recently [5]. One of the disadvantages of conventional jet impingement, however, is its low heat transfer area [6]. A few researchers have looked into methods to improve the performance of jet impingement by introducing highly engineered enhanced structures on the impingement surface. Several enhancement surfaces have been proposed in the literature from roughened surfaces [7,8] to extended surfaces (i.e., fins) [9–16]. All these enhanced surfaces have one goal in common which is to increase the product hA by the augmentation of the heat transfer area and the enhancement of the heat transfer coefficients through flow mixing and interruption of the boundary layer growth. In most investigations, significant heat transfer enhancements have been recorded; enhancement factors as high as 9.7 have been demonstrated [12].

Even though a few researchers have looked into the enhancement heat transfer of jet impingement on individual pin fin shapes and configurations, the literature to compare the effects of different pin fin geometries and configurations on the heat transfer coefficients is limited. Using infrared thermography, Li et al. [17] studied the thermal performance of heat sinks with confined impinging jets. Nine square pin fin heat sink models and five plate fin heat sink models were tested. The Reynolds number, based on jet diameter, varied from 5000 to 25,000. The convective thermal resistance, based on the total heat transfer area, was used as a performance criterion. The results showed that increasing Reynolds number, fin width, and fin height caused a decrease in the thermal resistance. In general, the thermal resistance of pin fin

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Nomenclature

A	area, m^2
d	jet diameter, m
D	pin fin diameter, m
h	average heat transfer coefficient, W/m^2 K
I	current, A
Nu	Nusselt number based on diameter d
Q	heat input, W
q''	heat flux, W/cm^2
Re_d	Reynolds number based on diameter d
t	thickness, m
T	temperature, K
U	velocity, m/s
V	voltage, V

Greek symbols

ϵ_A	porosity
μ	viscosity , kg/m s
ρ	density, kg/m^3

Subscripts

conv	convective
f	fluid
h	heater
i	inlet
j	jet
l	liquid
s	solid
w	wall

heat sinks were found to be lower than those of parallel plate heat sinks with 0.3–0.5 °C/W less thermal resistance when the height-to-heat sink length ratio increased to 0.5625.

In a recent paper, Kim et al. [18] compared the thermal performance of optimized plate fin and pin fin heat sinks for various heat sink sizes. All the pin fins, however, had the same cross sectional shape (square) but with different pitches. Their optimization study was based on a volume averaging method for predicting the pressure drop and the thermal resistance and was validated against experimental data from two types of fin designs: plate-fin and in-line square pin–fin heat sinks. In general, the optimized plate fin heat sinks displayed lower thermal resistance than the optimized pin fin heat sinks for a given pumping power.

In this study, we investigate the effects of cross flow area (area available to flow moving across the heat sink perpendicular to the impinging jet) and pin fin shapes on the heat transfer coefficients of single-phase jet impingement on micro pin fins. The literature on the effects of different pin fin cross sectional shapes on the parallel flow heat transfer coefficients is abundant [19–23]; however, only few have looked into the case of jet impingement. The micro pin fins considered in the current study consist of circular, hydrofoil, square, and elliptical pin fins. Two circular pin fins of different diameters, 125 μm and 75 μm, were considered to investigate the effects of cross flow (pitch was kept constant) on the heat transfer coefficients. In order to have a representative comparison of cross sectional shapes, the cross sectional areas of the various pin fins were kept approximately constant.

2. Micro pin fin devices

The micro devices used in the current study are the result of several design iterations with the aid of finite element analysis tools. The micro devices were fabricated using MEMS microfabrication techniques, consisting of steps of thin film deposition, photolithography, etching, CMP, bonding, and dicing. Fig. 1 shows a CAD drawing of the devices. Each micro device consists of an array of micro pin fins etched on a silicon substrate. Four fluid exit holes and a pressure transducer port are incorporated into the device. Fig. 2 (not to scale) shows a microfabrication process flow cross sectional view of the micro devices. The silicon base from where the micro pin fins protude has a thickness of 20 μm. Underneath the silicon substrate, a 880 nm thick silicon dioxide layer was deposited for electrical insulation. A 2×2 mm^2 titanium heater with a thickness of 100 nm was formed underneath the silicon dioxide layer. Power to the heater is achieved via the two 1-μm thick aluminum pads shown on the figure. For structural consideration, a 1-mm predrilled Pyrex wafer was bonded to bottom of the micro devices. Detailed description of the microfabrication process flow can be found in [24].

The micro pin fins investigated in the current study consist of circular pin fins (Fig. 3(A)), square pin fins (Fig. 3(B)), hydrofoil pin fins (Fig. 3(C)), and elliptical pin fins (Fig. 3(D)). The height of the pin fins and pin fin array pitch were kept fixed at 230 μm and 250 μm, respectively, and there were a total of 64 pin fins on each micro device. As shown on Fig. 4, the micro pin fin arrays were designed in such a configuration (streamlined, radially) to reduce pressure drop. 3D SEM images of the micro devices are shown on Fig. 5.

3. Experimental apparatus, procedure, and data reduction

Fig. 6 shows a CAD assembly of the test section setup. The test section assemby consists of the micro device sandwiched between two blocks. Through the top block (Lexan), a 2.0-mm through-hole was drilled to form the jet orifice. On the face of the block adjacent to the micro device, a 1.5-mm deep recess was cut into the block, making the jet stand-off ratio (distance from jet exit to base of pin fin array over jet diameter) equal to approximately 0.865. The transparent nature of Lexan block made it easier for device alignement and allowed for some degree of flow visualization. The bottom block, made out of Delrin, had a pocket machined for the micro device and also had four fluid exits to allow the impinging fluid to exit the test section fixture. A pressure tap was also drilled into the bottom block. To ensure a complete leak-free system, several o-rings were installed on both the top and the bottom blocks of the test section fixture. Electrical contact to the micro device heater was achieved through two small holes drilled through the bottom block of the fixture.

Fig. 7 shows a schematic of the experimental setup. With the micro device fixture integrated into the experimental loop, single-phase heat transfer experiments at different Reynolds numbers were conducted using R134a. For a given Reynolds number, power (product of measured voltage and current) applied to the heater

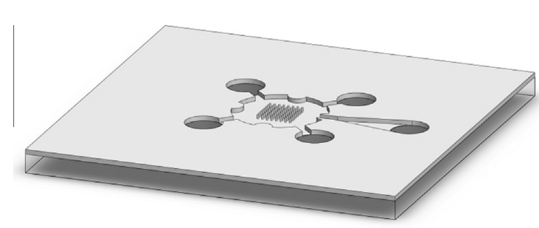

Fig. 1. CAD drawing of a micro device.