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Experimental study on liquid flow and heat transfer in micro square pin fin heat sink

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ABSTRACT

The heat sinks, with the total heat transfer area of $20 \times 20\ mm^2$ and an array of 625 staggered micro square pin fins of $559 \times 559\ \mu m^2$ or $445 \times 445\ \mu m^2$ cross section by 3 mm height were fabricated on a copper test section. Using deionized water as coolant liquid, the flow and heat transfer performance of the high pin fins were studied with the Reynolds number ranging from 60 to 800. For the $445 \times 445\ \mu m^2$ cross section pin fin heat sink, the heat dissipation could reach $2.83 \times 10^6\ W/m^2$ at the flow rate of 57.225 L/h and the surface temperature of 73.4 °C. The experimental data also showed that the pressure drop and the average Nusselt number increased with the fin Reynolds number. The heat resistance decreased with, with its decreasing rate inversely proportional to, the pressure drop. Since four chosen previous correlations overestimate the flow and heat transfer performance of the present sinks, we also proposed two new correlations for the average Nusselt number and pressure drop prediction.

1. Introduction

With the rapid development of semiconductor industry, the scale of heat dissipation rate from electronic devices increases drastically. Nowadays, the average power dissipation is about $100\ W/cm^2$, and the maximum local power dissipation could reach $500\ W/cm^2$ [1], which is unreachable by traditional direct air cooling methods due to the maximum heat dissipation capability limitation. Since Tuckerman and Pease [2] proposed the first micro channel heat sinks and exhibited the large heat dissipation ability of micro channels, numerous investigations have been conducted on the convection heat transfer in micro channels [3–7]. Due to the small width and height, most flow patterns in micro channels were laminar. The mass transfer between the layers could be ignored so the heat transfer mode in the fluid interior is considered as heat conduction only. When the channel length is short or the flow velocity is high, there would not be enough time for interior fluid heat transfer. Some investigations have employed different methods to increase the interior layers mixing and heat transfer [6,8].

Micro scale pin fins (staggered or in-line) are the common geometry used to increase the surface area and the passage flow turbulence. A number of researchers have studied the heat transfer as well as the flow friction in these areas and presented a lot of experimental data and relational formulas [9–13]. According to the results of these formulations, the average heat transfer coefficient in short fins is larger than in long tubes. Using LIGA (Lithography Electroforming Micro Molding) micromachining process, Marques and Kelly [14] fabricated a micro pin fin heat exchanger and tested its performance experimentally. The results showed that the micro fin heat exchangers offered the potential to control the surface temperature in high heat flux applications more effectively. Kosar et al. [15] experimentally investigated the pressure drop and friction factors in different aspect ratio micro pin fins and proposed a modified friction factor correlation. Peles et al. [16] theoretically and experimentally studied the heat transfer and pressure drop phenomena over a bank of micro pin fins with a focus on the effect of geometrical and thermo-hydraulic parameters on the total thermal resistance. They found that the micro pin fin heat radiator could take away $790\ W/cm^2$ when the pressure drop was two atmospheric and the surface wall temperature increment was 30.7 °C. Compared with the micro channels, the heat transfer performance of micro fins was better while the pressure drop was much bigger. Siu-ho et al. [17] experimentally investigated the pressure drop and heat transfer characteristics of a single-phase copper micro pin fin heat sink. The results indicated that the micro pin fin heat sink could be very effective at meeting the needs of high-heat-flux electronic cooling applications. The previous friction factor and heat transfer correlations might not be usable for situations beyond their original ranges of validity. Therefore, new predictive tools specifically tailored for the single-phase flow and heat transfer in the micro pin fin heat sink might be required. Qu et al. [18–20] compared thermal-hydraulic performance of a single-phase micro pin fin heat sink against a micro channel heat sink. He found that the micro-channel heat sink had a higher convection thermal resistance with a lower pressure drop at high cooling flow rates. In their study, the experiments were conducted at 30 °C and 60 °C coolant inlet temperature and

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Nomenclature

A	area of heat surface (m^2)
A_1	bottom areas of heat sink (m^2)
c_p	specific heat (J/kg K)
D_c	equivalent diameter of a fin pitch (m)
D_e	equivalent diameter of a fin size (m)
f	friction factor
H	height of pin fin (m)
h	heat transfer coefficient (W m^{-2} K^{-1})
k	thermal conductivity (W m^{-1} K^{-1})
L	length of pin fin region (m)
l	distance between the thermocouple and down surface of heat sink (m)
m	mass flux (kg s^{-1})
N	row number of pin fins
N_t	the number of pin fins
Nu	Nusselt number
p	pressure drop (Pa)
p_d	inlet/outlet pressure drop (Pa)
P_{fin}	Perimeter of micro pin fin cross-section (m)
Pr	Prandtl number
Q	total heat dissipation (W)
q	heat flux (W m^{-2})
Q_f	volume flow rate (L/h)
Re	Reynolds number based on the fin size
Re_c	Reynolds number based on the channel size
R_{th}	heat resistance (W °C^{-1})
S_L	streamwise distance between pin fins (m)
S_T	transverse distance of pin fins (m)
T	temperature (°C)
u	flow velocity (m s^{-1})
W	width (m)
W_c	channel width (m)
W_{fin}	fin width (m)
Δp	pressure drop (Pa)

Greek symbols

m_{fin}	fin efficiency coefficient
η_{fin}	fin efficiency
μ	viscosity (Pa s)
ξ	fin/channel ratio
π	3.1415
ρ	density (kg m^{-3})

subscripts

w	surface
cu	copper
f	fluid
fin	square fin
i	thermocouple (i = 1–4)
in	inlet
m	average
max	maximum
min	minimum
out	outlet

with six mass velocities for each inlet temperature to test the characters of micro pin fin. Their data showed that the pressure drop was over predicted by previous correlations, and the average Nusselt number increased with the average Reynolds number. They developed new correlations to predict the pressure drop and the heat transfer in the micro pin fin heat sink, based on the thermal performance, hydraulic performance, and the cost of manufacturing.

Jasperson et al. [21] also compared the micro channel and micro pin fin heat sinks. It is found that the average convection thermal resistance decreased with the flow rate for the micro pin fin heat sink, while it was almost independent on the flow rate in the micro channel heat sink. The pressure drop in the micro pin fin heat sink was larger than that in the micro channel heat sink when the flow rate was fixed.

An effective design and a performance assessment of a micro pin fin heat sink require a fundamental understanding of the fluid flow and heat transfer aspects of micro pin fins arrays. Most existing studies were empirical due to the complex nature of the fluid flow and heat transfer in micro pin fins. Therefore, many corrections were proposed to provide valuable insight into their flow and heat transfer aspects of micro pin fins. However, no one correction is conclusive.

The heat transfer and pressure drop performance of pin fins could be affected by many factors. However, pin fin height-to-diameter ratio, operating condition might be the most important ones. It is clear that, for micro pin fins, the larger the flow rate, the lower convection thermal resistance. In engineering applications, in sake of higher heat dissipation, a larger flow rate and higher power pump are needed. As we known, the pressure drop would be smaller if we get higher micro pin fins. This paper expands the micro pin fin studies by investigating the pressure drop and heat transfer in staggered micro square high pin fins. The objectives are: 1, to provide new heat transfer and pressure drop data for liquid singe-phase flow in intermediate pin-fins (pin-fin height-to-diameter ratio of 6) with $559 \times 559\ \mu m^2$ or $445 \times 445\ \mu m^2$ cross section by 3 mm height; 2, to develop new heat transfer and pressure drop correlations for the micro pin fins and to compare with the previous.

2. Description of the experimental setup and test section

2.1. Experimental setup

The schematic diagram of experimental setup, which consists of a cooling system, a heating and control unit, a date collecting & processing unit, is shown in Fig. 1. In the cooling system, a low temperature sink was used to keep the inlet temperature of the experimental section at a fixed value. Deionized water, which was driven from the low temperature sink, flowed through a ripple damper and a 50 μm filter, and then was shunted into two parts by three way valves. The one part passed though the flowmeter, and later flew into the micro pin fin test section. After taking away the heat dissipation, it mixed with the other part and returned to the low temperature sink.

The heating and control unit was designed to heat the test section. The key part of this unit was the heat source simulator. The heat source simulator was made of a pure copper block and heated by eight 300W power cartridge heaters which were inserted from the button of the copper block (Fig. 2). Those cartridge heaters were controlled by an adjustable power meter. The top of copper block was trapped to 20 × 20 × 20 mm cube, while the lower part of copper is 80 mm in diameter and 110 mm in height. To minimize heat loss and to keep the 1-D heat transfer characteristic in the direction perpendicular to the top surface, all the surfaces other than the top one were insulated by Aspen Aerogels insulation (thermal conductivity, k = 0.012 W/m K). This heating and control