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Investigation of microchannel heat sink with new micro pin-fin design for heat transfer enhancement#

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ABSTRACT

A novel design of different arrangements of micro pin-fins is proposed and comparative analysis has been done to study the heat transfer enhancement evaluated by pressure drop, dimensionless temperature hoist at the exit and Nusselt number. Furthermore, the variables that material of heat sink, five different shapes and three different arrangement of micro pin fins are investigated to determine the MCHS with optimum heat transfer capacity. The predicted results demonstrate that the amount of Nu for the novel deign 59.9% larger than unfinned structure at Re=1000. The addition of micro pin-fins increases heat transfer surface area as well as interrupts and makes thermal boundary layer redevelop, resulting effective heat transfer. Different arrangements improve the effect and show better thermal performance. The results also illustrate that the pin-fin with triangular shaped cross-section yield better performance.

Keywords: microchannel heat sink, numerical analysis, heat transfer

NONMENCLATURE

Symbols	
D	Hydraulic diameter
h	heat transfer coefficient
Nu	Nusselt number
P	Pressure
T	Temperature

1. INTRODUCTION

With the increasing density of electronic components in chips, they would produce much more heat than before, thereby thermal management has become a serious issue. Therefore, it has become a hot topic in recent years to remove the large amount of heat generated by chips effectively [1, 2].

Microfluidic-based cooling technique was proposed by Tuckerman et al. [3] in 1981. With the rapid development of fabrication of miniature devices and increasing amount of heat produced by chip, microfluidic cooling method regain public attention. Microchannel heat sink typically has several channels whose hydraulic diameter is in micron scale. These microchannels make the device have high specific surface area of coolant and low thermal resistance, which can considerably increase heat transfer performance and efficiency. Therefore, microchannel heat sink has become hot topic of cooling technique recently [4-6]. Xu et al. [4] investigated hydraulic and thermal characteristics of rectangular cross section microchannels with dimples of heat sink. They independently studied several parameters of dimpled MCHS, such as aspect ratio of rectangular cross section, spacing and depth of dimples. Compared with plain channels, dimpled channels reduced 3.2 K of temperature and 2% of pressure drop, also increase 15% value of Nusselt number. However, there was few literatures to investigate integrated microchannel heat sink with micro pin-fins. Therefore, a novel design of different arrangements of micro pin-fins is proposed and the effect of material of heat sink, shape of cross section and spacing of micro pin-fins on thermal performance have been investigated in this paper.

A novel design of MCHS is proposed to investigate the effect of overall thermal performance. The aim of this project is to examine the maximum heat transfer dissipation considering materials of heat sink, arrangements, various geometries and spacing of micro pin-fins by using simulation method. First, the effect of four common materials (silicon, aluminum, copper, and silver) for microchannel heat sink fabrication on heat transfer capacity was investigate. Then, the influence of variable factors of fins was studied, such as three different arrangements and five geometrical shapes, including circle, ellipse, square, triangle and hexagon.

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2. NUMERICAL MODELLING

2.1 Channel design and heat transfer data reduction

Three arrangements of MCHS with micro pin-fins are modelled and make a comparison. The isometric views are shown in Figure 1. Direction of fluid flow has been clearly marked and three arrangements can be named Structure A, B and C. Structure A is that one in-line micro pins are designed in the bottom surface of channels while Structure B has two rows' fins in the upper and bottom surfaces. Structure C is a novel idea that micro pin-fins are designed on whole surfaces of channels. The first fin is on the bottom surface, and each subsequent fin is on the counterclockwise side of the previous fin. Dimensions are marked in Figures 2-3. The length, width and height of the MCHS are 12 mm, 10.4 mm and 2.2 mm respectively. The height and width of each channel are set as 0.8mm and 0.5mm, the length of channels is equal to the length. The height of micro pin-fins is 0.2mm. Spacing of adjacent micro pin-fins varies from 0.3 mm to 0.9 mm with an increment of 0.3mm.

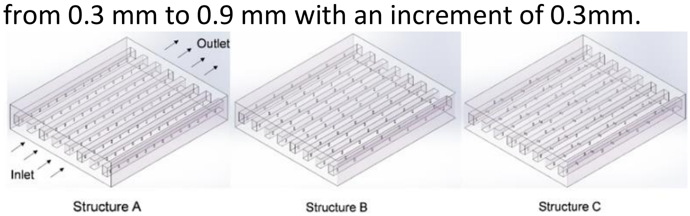

Fig. 1 Isometric view of three arrangements of MCHS with micro pin-fins

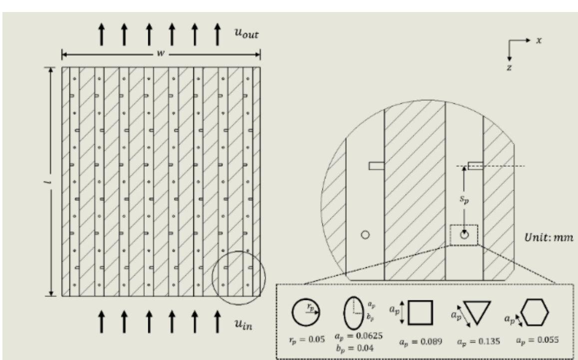

Fig.2 Cross section view of Structure C at y=1.5 mm

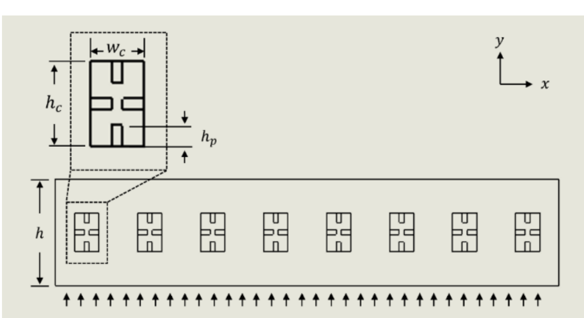

Fig.3 Front view of Structure C

A constant heat flux is applied to the bottom of the MCHS. The hydraulic diameter of microchannel is very important to calculate Reynolds number and average Nusselt number. It can be expressed as:

$$D_h = \frac{2h_c w_c}{h_c + w_c}$$

where h_c and w_c are height and width of microchannel. Hydraulic diameter of each channel can be figured out; in the current work, h_c and w_c of all models are constant, thus D_h value is also constant as 0.6154mm. Reynolds number is defined by :

$$Re = \frac{\rho_f V D_h}{\mu}$$

where V is mean velocity. In this work, uniform inlet velocity of working fluid can be calculated at a given Reynolds number:

$$u_{in} = \frac{Re\mu}{\rho_f D_h}$$

Pressure drop can be calculated by following expression:

$$\Delta p = p_{in} - p_{out}$$

Based on the boundary conditions, p_{out} is atmospheric pressure. Thus, the expression can be simplified as

$$\Delta p = p_{in}$$

Inlet temperature of fluid was set to 20 ℃ . Dimensionless temperature at the outlet of channels can be defined:

$$T' = \frac{T_{out} - T_{in}}{T_{w,avg} - T_{in}}$$

The larger outlet temperature makes larger T', resulting from the more heat dissipated by fluid, thus, it is a direct way to measure the value of T' to evaluate the heat transfer capacity.

The average heat transfer coefficient h_{avg} can be calculated by:

$$h_{avg} = \frac{qA_f}{A_c(T_{w,avg} - T_{f,avg})}$$

where $q, A_f, A_c, T_{w,avg}$ and $T_{f,avg}$ denote the heat flux per unit area, the heated area, the convection heat transfer area, the average wall temperature of solid liquid contact, and the average fluid temperature, respectively. Heat flux q that applied at the bottom plate was constant as 100kW/m^2.

Nusselt number can then be calculated by:

$$Nu = \frac{h_{avg} D_h}{k_f}$$

p_{in} is calculated by ANSYS Fluent as area weighted average pressure of inlet of all the channels. $T_{w,avg}$ can be made as area weighted average wall temperature, $T_{f,avg}$ is obtained as volume average temperature for working fluid. Therefore, average heat transfer coefficient can be obtained when $T_{w,avg}$ and $T_{f,avg}$ are calculated by software since q, A_c and A_f are known.

2.2 Governing equations

The governing equations of fluid flow in one microchannel are provided below.

In terms of incompressible flow, continuity equation can be figured out:

$$\rho\left(\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}+\frac{\partial W}{\partial Z}\right)=0$$

The thermal properties are assumed to be constant, so the momentum equation can be expressed as:

X-momentum:

$$\rho\left(U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}+W\frac{\partial U}{\partial Z}\right)=\frac{\partial}{\partial X}\left(\mu\frac{\partial U}{\partial X}\right)+\frac{\partial}{\partial Y}\left(\mu\frac{\partial U}{\partial Y}\right)+\frac{\partial}{\partial Z}\left(\mu\frac{\partial U}{\partial Z}\right)-\frac{\partial P}{\partial X}$$

Y-momentum:

$$\rho\left(U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}+W\frac{\partial V}{\partial Z}\right)=\frac{\partial}{\partial X}\left(\mu\frac{\partial V}{\partial X}\right)+\frac{\partial}{\partial Y}\left(\mu\frac{\partial V}{\partial Y}\right)+\frac{\partial}{\partial Z}\left(\mu\frac{\partial V}{\partial Z}\right)-\frac{\partial P}{\partial Y}$$

Z-momentum:

$$\rho\left(U\frac{\partial W}{\partial X}+V\frac{\partial W}{\partial Y}+W\frac{\partial W}{\partial Z}\right)=\frac{\partial}{\partial X}\left(\mu\frac{\partial W}{\partial X}\right)+\frac{\partial}{\partial Y}\left(\mu\frac{\partial W}{\partial Y}\right)+\frac{\partial}{\partial Z}\left(\mu\frac{\partial W}{\partial Z}\right)-\frac{\partial P}{\partial Z}$$

Besides, this simulation is assumed to ignore radiation and viscous energy dissipation, so energy equation is:

$$\rho c_p\left(U\frac{\partial T}{\partial X}+V\frac{\partial T}{\partial Y}+W\frac{\partial T}{\partial Z}\right)=\frac{\partial}{\partial X}\left(k_f\frac{\partial T}{\partial X}\right)+\frac{\partial}{\partial Y}\left(k_f\frac{\partial T}{\partial Y}\right)+\frac{\partial}{\partial Z}\left(k_f\frac{\partial T}{\partial Z}\right)$$

where k_f is thermal conductivity of coolant.

2.3 Simulation setups

In this simulation study, ANSYS 2022 R2 was used to simulate the thermal behaviour of MCHS with micro pin-fins. 3D models made by Solidworks are imported, labelled, and meshed in ANSYS Workbench. Second order upwind scheme and the SIMPLE algorithm are chosen to discretize the momentum and energy equations and the pressure velocity coupling. For residuals, apart from the criterion of energy was set to 10^{-6}, the criteria for continuity and velocity in x, y and z directions are set to 10^{-3}.

The assumptions are made in our numerical study: (1) Working fluid is subject to steady and incompressible flow; (2) Working fluid is laminar flow at Reynolds number ranging from 100 to 1000. since the cross-section of microchannels is very small. (3) Gravitational effect can be neglected.

The water has been applied as the coolant. For the respective of materials of heat sink, The thermal conductivity of possible candidate materials for MCHS is shown in Figure 4.

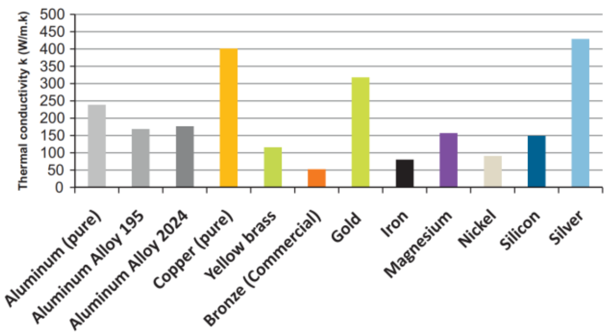

Fig.4 Possible candidate's materials for MCHS fabrication [7]

Among these materials, aluminum, copper, silicon, and silver were chosen as the simulated heat sink materials because they are the most common materials in other related papers. The related properties (ρ_s, c_{p_s}, k_s) are shown in Table 2. Pressure at the outlet of channels has been set to be same as the atmospheric pressure.

Table 2 Properties of heat sink materials in simulation

Material	Properties		
	Density ρ_s (kg/m^3)	specific heat capacity c_{p_s} (J/kgK)	thermal conductivity k_s (W/mK)
silicon	2329	710	149
aluminum	2719	871	202.4
copper	8978	381	387.6
silver	10490	235	429

2.4 Model validation

Before proceeding to the numerical model of MCHS with micro pin-fins, it is necessary to validate the model with published results. The present data was reproduced and compared to the simulation conducted by Gunnasegaran et al. [7]. The comparison was shown in Figure 5 that the relationship between temperature rise of working fluid and increasing Reynolds number for rectangular shaped microchannel. Present numerical results show great agreement with the previous numerical result, implying that the model of MCHS with micro pin-fins can carry out precise simulation.

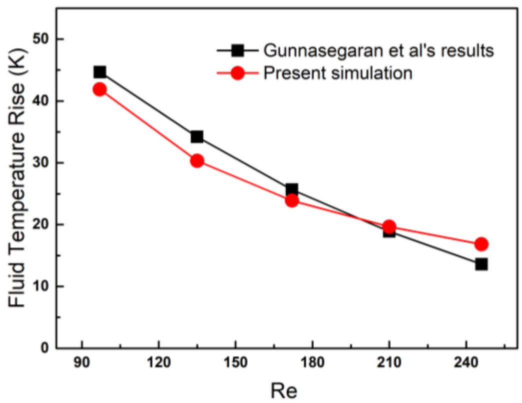

Fig. 5 Comparison of temperature rise for rectangular shaped microchannel given by Gunnasegaran et al [7].

The models are meshed based on finite volume method (FVM). It is important to carry out grid independence study to ensure not only the accuracy and reliability of results but also find the minimum number of grids to shorten calculation time. This study was carried out the model of Structure A with 0.9 mm-spacing micro pin-fins at Re=100. The grid distribution of MCHS with grid size of 1.0mm is shown in Figure 6. Structural grids of hexahedral volume elements are used to discretize the entire computational domain. Six different element number in Table 3 are evaluated. The outlet temperature of MCHS is plotted and the result of grid size validation is shown in Figure 7. It was clearly noticed that the value obtained when grid number is between 3124528 and 3751895 is so similar that the error can be ignored. Thus, in order to increase computational speed, Grid 5 (grid number: 3124528) are selected for all the numerical simulations.

Table 3 The mesh size and number of grids in the present study

	Grid 1	Grid 2	Grid 3	Grid 4	Grid 5	Grid 6
Grid size	7 mm	4 mm	3 mm	2 mm	1.8 mm	1 mm
Number of grids	534994	1118684	1602276	2729716	3124528	3751895

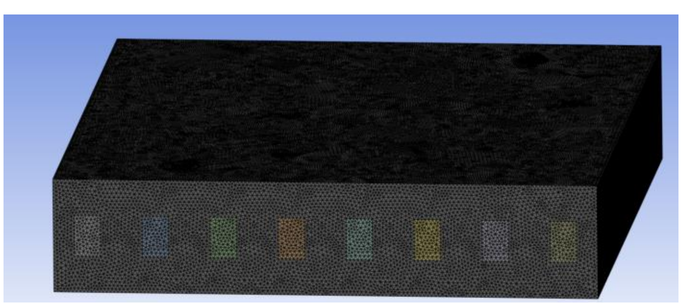

Fig. 6 Meshing of the heat sink with grid size of 1.0 mm.

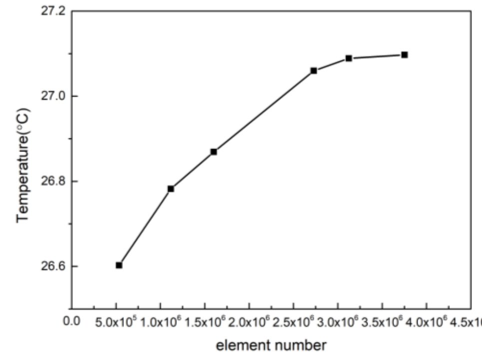

Fig. 7 The temperature with various grid sizes

3. RESULTS AND DISCUSSION

3.1 Material selection of heat sink

The findings show that different materials of heat sink have little effect on the Nu value. Silicon is the best material used for heat sink resulting from the best performance that the number of Nu is 12.17 among four materials, as shown in Figure 8.

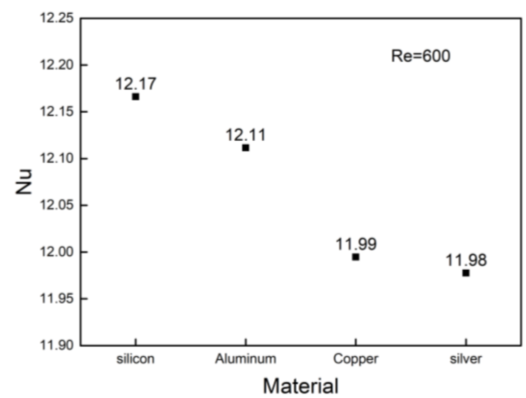

Fig. 8 comparison of Nusselt number when using silicon, aluminum, copper, or silver as heat sink at Re=600

3.2 The effect of different arrangements of micro pin-fins

The results are shown in Figure 9. The adding of pin-fins reduces the total volume of working fluid, which encounters the disruption, resulting increased velocity and larger pressure drop. Besides, local vortices would be generated behind pin-fins thereby rises inlet pressure. The assumption can be made that Structure A causes only a partial increase in the pressure of the fluid, while structures B and C affect a larger portion of the fluid due to the difference in arrangement. The mechanism can be described as an increase in heat transfer due to an increase in the heat transfer surface area (Ac). The addition of pin fin structure enables thermal boundary separation and redevelopment.

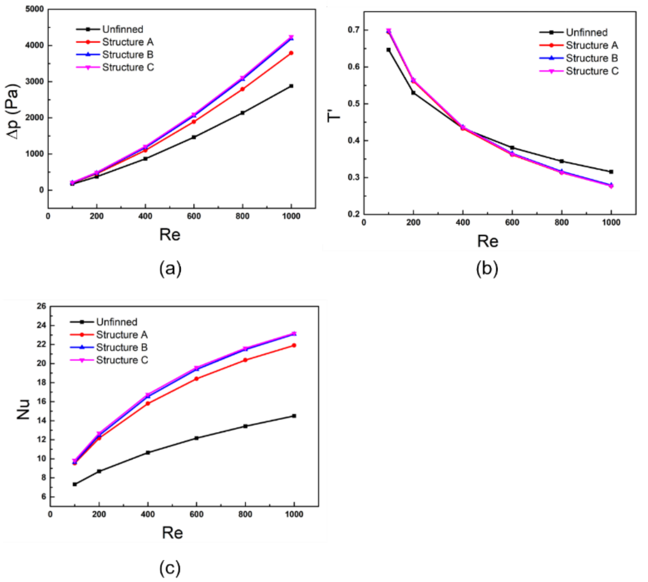

Fig.9 Effect of Re on (a) pressure drop, (b) dimensionless temperature at the exit and (c) Nu for different arrangements of MCHS.

3.3 The effect of geometry of micro pin-fins

The results are shown in Figure 10. The reason for this can be explained by the fact that triangular pin-fin produces a larger adverse pressure gradient than other shapes, it can be deduced that triangle shaped cross section exhibits the best heat transfer dissipation, although it sacrifices the higher pressure drop.

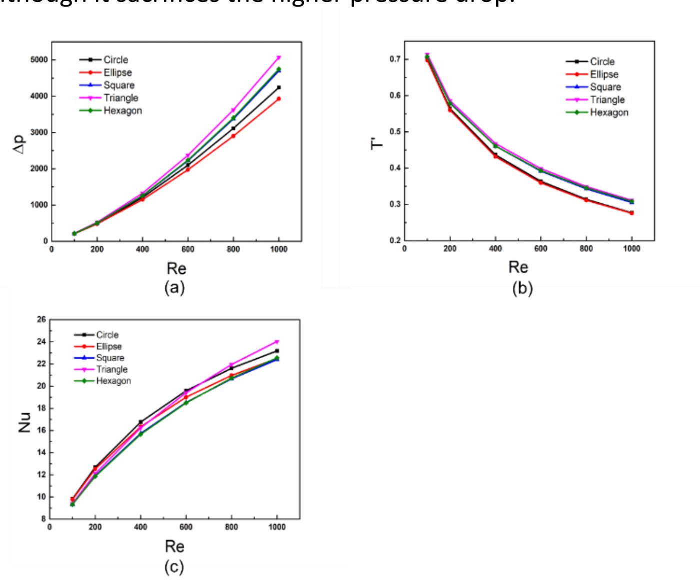

Figure 10 Effect of Re on (a) pressure drop, (b) dimensionless temperature hoist at the exit and (c) Nu for different geometry of MCHS.

It is evident from Figure 10 that the rise in Re results in an increase in pressure drop, but the rate of increase is dependent on the geometry of the micro pin-fins. Figure 10 also depicts that all geometries exhibited a negative relationship between T' and Re. The T' of micro pin fin with circular and elliptical sections displayed similar values and gradients, while square, triangular, and hexagonal shaped cross-sections also exhibited nearly identical values and gradients. It's also observed in Figure 10 that the Nu increases with increasing Re for the micro shape considered. The MCHS with circular shaped micro pin-fin has the highest Nu when Re was between 100 and 600, while Nu with triangular shaped pin-fin surpasses that of circular shaped fin when Re>600. It can be attributed to the ability of triangular and circular pin fins to generate wake at the trailing edge, promoting fluid mixing and consequently increasing the heat transfer performance.

4. CONCLUSIONS

A novel microchannel heat sink design with pin-fin structure was proposed with varying fin geometries and fin spacing. Silicon was considered as material of heat sink and the cooling medium was water. Various configurations were numerically analyzed to determine the optimal fin geometry and spacing. Numerical simulations were conducted for the range of Re=100-1000. A comparative analysis of the predicted results yielded the following conclusions: Materials of heat sink have little effect on the heat transfer enhancement. The amount of Nu for the novel deign (Structure C) is 59.9% larger than unfinned structure at Re=1000. Structure C also exhibits the best thermal performance on heat transfer among three arrangements of micro pin-fins. Besides, the triangular cross-section of micro pin-fins shows the highest overall thermal performance although it sacrifices higher pressure drop since this configuration has the largest A_c, resulting more effective heat transfer.

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DECLARATION OF INTEREST STATEMENT

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper. All authors read and approved the final manuscript.

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