Influence of sinusoidal flow on the thermal and hydraulic performance of microchannel heat sink

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Abstract In this paper, the effect of sinusoidal flow on the thermal and hydraulic performance of microchannel heat sink (MCHS) is numerically investigated. This investigation covers Reynolds number in the range of $100 \leq Re \leq 1000$ and pure water is used as a working fluid. The three-dimensional steady, laminar flow and heat transfer governing equations are solved using finite volume method (FVM). The water flow field and heat transfer performance inside the sinusoidal microchannels is simulated and the results are compared with the straight microchannels. The effect of using sinusoidal microchannels on temperature distribution, Nusselt number, friction factor and thermal resistance is presented in this paper. It is found that with same rectangular cross-section, sinusoidal microchannels have a better heat transfer performance compared to the straight microchannels.

1. Introduction

Lately number of research based on microchannel have grown broad and boosted the technology to grow rapidly. The introduction of MCHS has even increased with attention to study this newfound discovery more in detail and broad in order to increase the efficiency of it. At the very beginning the MCHS cooling concept was proposed by Tuckerman and Pease [1] in 1981, who pointed out that decrease in liquid cooling channel dimensions to the micron scale, will lead to increase in heat transfer rates.

Al-Bakhit and Fakheri [2] noted that in microchannel heat exchangers, short lengths and comparatively thick walls through which heat is conducted avoid the existence of fully thermal developed flow over a large portion of the heat exchanger. They concluded that there is a significant change in the overall heat transfer coefficient in the developing region. Gamrat et al. [3] performed three-dimensional numerical analysis of convective heat transfer in microchannels. By considering the thermal entrance effect, the numerical analysis did not find any significant effect on the heat transfer in microchannels with the hydraulic diameter down to 100 $\mu$m. Peng and Peterson [4] studied experimentally the pressure drop and convective heat transfer for water flow in rectangular straight microchannels. They found that the cross-sectional aspect ratio has a great influence on the flow friction and convective heat transfer in both laminar and turbulent flows. T.L Ngo [5] performed numerically and experimentally investigation on the curvy and zigzag microchannel heat exchanger (MCHE). The results showed that the Nusselt number of zigzag microchannels is 24–34% higher than that of the curvy microchannels, but the friction factor is four to five times larger depending on the Reynolds number. Numerical analysis on the fluid flow and heat transfer characteristics in rectangular
cross-section MCHS with distinct inlet and outlet arrangements was carried out by Chein and Chen [6]. They mentioned that better performance in velocity and temperature can be found in the heat sinks having coolant supply and collection vertically via inlet/outlet ports opened on the heat sink cover plate. Numerical analysis on the laminar forced convection and thermal radiation in a participating medium inside a helical pipe was carried out by Zheng [7]. The effects of thermal radiation on the convective heat transfer were investigated. He found and deduced that the thermal radiation could enhance the total heat transfer rate. By using finite element method Zhang [8] executed numerical analysis on the hydrodynamic and thermal characteristics of microchannel networks. A comparison of the cooling performance among step, parallel, and spiral microchannels was also conducted based on the same heat flux and flow rate. They found that step microchannel alleviate the lowest fluid pressure drop and the most uniform temperature distribution over the substrate.

It should be noted that all the previous studied of MCHS are reported in the open literature used conventional microchannel which generally engaged with straight microchannel. Furthermore, to the best knowledge of the authors, the case of sinusoidal MCHS has not been given great attention by the researchers in the past and this has motivated the present study. The present study numerically investigates the three-dimension heat transfer and fluid flow characteristics in sinusoidal MCHS with rectangular cross-section using water as a working fluid. The Reynolds number is ranged from 100 to 1000 and heat flux applied at the bottom plate of MCHS is 1000000 W/m².

1. MCHS configuration and mathematical formulation

In this paper, sinusoidal MCHS with a constant rectangular cross-section is considered. The physical configuration of the sinusoidal MCHS is schematically shown in Figure 1 and schematic of the each unit of microchannel i.e. a) conventional channels; b) sinusoidal channels are shown in Figure 2. Heat supplied to the aluminum MCHS substrate through a bottom plate, was removed by flowing water through a number of 24 micro channels. In this study, the effect of sinusoidal flow of the MCHS on the heat transfer and fluid flow characteristics is examined.

In order to solve the Navier–Stokes and energy equations to investigate the effect of sinusoidal flow on the MCHS performance, the following assumptions were made: (i) both fluid flow and heat transfers are in steady state; (ii) the flow is isochoric which means the volume of the water remains constant throughout the process; (iii) the water flow is laminar; (iv) the water that flows in and out of the MCHS does not change phase during the process; (v) properties of both fluid and heat sink material are temperature-independent; (vi) fluid flow and heat transfer are three-dimensional; (vii) all the surfaces of heat sink exposed to the surroundings are assumed to be insulated except the bottom plate of heat sink where constant heat flux boundary condition simulating the heat generation from external sources is specified.

The governing equations and its boundary conditions in Cartesian coordinates for three-dimensional laminar incompressible flow for the current problem are:

Continuity Equation: \[ \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z} = 0 \]  

Eq. 1
X- Momentum equation: \( U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} + W \frac{\partial U}{\partial z} = -\frac{\partial p}{\partial x} + \frac{1}{\text{Re}} \left( \frac{\partial^2 U}{\partial x^2} + \frac{\partial^2 U}{\partial y^2} + \frac{\partial^2 U}{\partial z^2} \right) \) Eq. 2

Y- Momentum equation: \( U \frac{\partial V}{\partial x} + V \frac{\partial V}{\partial y} + W \frac{\partial V}{\partial z} = -\frac{\partial p}{\partial y} + \frac{1}{\text{Re}} \left( \frac{\partial^2 V}{\partial x^2} + \frac{\partial^2 V}{\partial y^2} + \frac{\partial^2 V}{\partial z^2} \right) \) Eq. 3

Z- Momentum equation: \( U \frac{\partial W}{\partial x} + V \frac{\partial W}{\partial y} + W \frac{\partial W}{\partial z} = -\frac{\partial p}{\partial z} + \frac{1}{\text{ReD}} \left( \frac{\partial^2 W}{\partial x^2} + \frac{\partial^2 W}{\partial y^2} + \frac{\partial^2 W}{\partial z^2} \right) \) Eq. 4

Energy Equation: \( U \frac{\partial T}{\partial x} + V \frac{\partial T}{\partial y} + W \frac{\partial T}{\partial z} = -\frac{1}{\text{PrD}} \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \) Eq. 5

At the microchannels sections, the inlet water temperature is taken as 293K and the inlet water velocity is calculated based on the required Reynolds number. The Reynolds number considered is ranged from 100 to 1000. The transverse velocities at the inlet are assumed to be zero. On the aluminum substrate, the velocities are zero, and it is assumed to be an adiabatic surface. The heat flux that applied at the bottom plate is 1000000 W/m². The properties of water and solid used in the computation are \( \rho = 998.2 \text{ kg/m}^3 \), \( c_p = 4182 \text{ J/kg.K} \), \( \mu = 0.001003 \text{ kg/m.s} \), \( \kappa = 0.6 \text{W/m.K} \), and \( \kappa_s = 202.4 \text{ W/m.K} \). The numerical computation is carried out by solving the governing conservation equations (Eqs. 1-5) along with the boundary conditions. From numerical computation, the temperature distribution, Nusselt number, friction factor and thermal resistance were determined. This leads to identify the thermal and hydraulic performance of the sinusoidal flow in MCHS.

2. Results and Discussion

The variation of average wall temperature for sinusoidal and straight MCHSs is presented in Figure 3. For both of MCHSs, it is found that the high-temperature region occurs at the edge of the heat sinks since there is no heat dissipation by fluid convection. The low temperature region occurs in the region where microchannels are placed, especially near the center of the heat sink due to the high heat transfer coefficient. It can clearly be seen from Figure 3 that the temperature of straight conventional MCHS is higher compared to the sinusoidal MCHS due to the poor fluid mixing. From the results obtained in Figure 3, lower thermal resistance is expected in sinusoidal MCHS compared to straight MCHS.

The Figure 4 presented the Nusselt numbers with the number of channels for straight and sinusoidal MCHSs at the Reynolds number of 1000. It can be seen from Figure 4 that the middle channel (channel 12) has the highest Nusselt number value. The Nusselt number for other channels is seen to decrease depending on their distances from the wall. The Nusselt number for both types of MCHSs is almost symmetrical with respect to the centerline of the heat sink. In sinusoidal MCHS, it is found that the Dean vortices can quickly develop along the flow direction and disturb the boundary layer. Thus, better heat transfer performance is found in sinusoidal MCHS compared to straight MCHS as shown in Figure 4.

![Figure 3: Temperature distribution](image1)

![Figure 4: Nusselt number](image2)

The variation of the friction factor with Reynolds number for both sinusoidal and straight MCHSs is shown in Figure 5. The results revealed that the friction factor decreases with the increase of Reynolds number for both types of MCHSs. It can be inferred that the changing of Dean vortices
patterns along the flow direction through the sinusoidal MCHS appears to give an increase in the friction factor compared to straight MCHS for all values of Reynolds number. The thermal resistance for sinusoidal and straight MCHSs as function of pumping power is shown in Figure 6. It is observed that sinusoidal MCHS reduces the thermal resistance due to its lower wall temperature distribution compared with that of the straight MCHS. Therefore, the sinusoidal MCHS could greatly enhance the cooling performance of MCHS compared with the traditional straight MCHS.

Figure 5: Friction factor
Figure 6: Thermal resistance

3. Conclusion
Numerical simulations on the heat transfer and fluid flow characteristics in sinusoidal MCHS are presented in this paper. The effect of sinusoidal flow in MCHS on the thermal and flow fields was intensively investigated and compared with the straight MCHS. It was found that the heat transfer performance in sinusoidal MCHS is much better compared to straight MCHS and can be proposed as next invention cooling equipment for removing high heat flux instead of using conventional straight microchannel.

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