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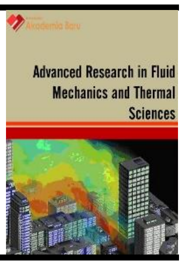

Numerical Heat Transfer Analysis in Microchannel Heat Sink with Different Aspect Ratio

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ABSTRACT

The cooling of electronic devices is essential to guarantee their functional performance and operational lifetime. Due to continued miniaturization and integration of transistors in packaged chips, the heat dissipation rate has surpassed the limits of classical air-cooled heat sinks. This has triggered a lot of research towards alternatives for high heat flux cooling. Liquid cooling with micro heat sinks is one of these candidate solutions. Cold liquid flows through microscopic channels to extract heat from the chip. In this paper, the studied is focused on the effect of aspect ratio in the Microchannel Heat Sink (MCHS) using numerical analysis, and the result obtained is discussed in this paper. The overall result of the present work shows there is a close relationship between both the numerical and analytical data.

1. Introduction

Technology involving high speeds of data processing in microprocessors of fast computers has led to a reduction in sizes of integrated chips. The need to remove a large amount of heat generated from such devices for effective functional characteristics was first investigated by Tuckerman and Pease [1]. In their work, they had studied the experimental and theoretical performance of microchannel as heat exchangers using water as the coolant and suggested the use of high aspect ratio microchannel to reduce thermal resistance.

Semiconductor device technologies development since the middle of twentieth century leads to miniaturization of many sensors and components into microelectromechanical devices. The amount of heat flux produces by these devices increase exponentially due to the downsizing and their increasing in functionality. These amounts of heat from such small volume or surface area cannot be efficiently dissipated using the conventional cooling method. Sobhan and Garimella [2] proved that

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Microchannel heat sinks (MCHS) managed to remove heat faster than conventional heat sink devices by their massive heat transfer surface-to-volume ratio. Besides that, Tullius [3] and Zhang *et al.,* [4] in their investigation also agreed MCHS can be a good candidate in the selection of advanced cooling methods which capable to dissipate heat from a tiny surface area with higher heat fluxes.

The main objective of this research is to study on the heat transfer, geometric aspect ratio MCHS by using numerical analysis. Numerical simulations are executed to determine the influence of geometric parameters on the heat transfer and flow characteristic of rectangular shaped MCHS. The validation process on the numerical analysis has been carried out which the similar physical model used by Sohel *et al.,* [5] and Steinke *et al.,* [6] was implemented. In this dimension of a physical model, the height, width, and length of the heat sink will be represented as H, W, and L respectively and Hc and Wc stand for the height and width of channels respectively. WW refers to the width of the sidewall, while Htop and Hb refer to the thickness of the top plate and heat sink substrate respectively.

In the year of 2000, an investigation regarding the convective-conductive heat transfer was done by Kuppusamy *et al.,* [7] due to a laminar boundary layer flow of the stream over dimensional reveal of rectangular chip squares which express to the limited heat assets. Li and Olsen [8] tried to approve the CFD package FLUENT with the trial data got by then some time recently. They have successfully carried out two investigations on the heated chip with a temperature of 353K and the air channel speed at 293K. Xu *et al.,* [9] explored the stream and characteristics of heat transfer of a separated crossflow square chamber set symmetrically in a planar opening for a scope of conditions.

Zeighami *et al*., [10] and Sharp and Adrian *et al*., [11] separated the effect of geometry on flow stream and heat transfer, finding that a relentlessly uniform heat flux transition achieved an extended variation from the norm of temperature course on the chip surface. Two-layered microchannel heat sink thought in electronic cooling was done by Sharp and Adrian [11]. Li *et al.,* [12] investigated the experimental and numerical Micro-Channel Heat sinks for cooling and also numerically intense on Electronic Devices. Familiar, a computational fluid dynamic (CFD) system was used to predict the temperature flow in a smaller scale channel of heat exchanger. Results shows from the numerical reproduction remained then contrasted with exploratory information.

2. Methodology

2.1 Modelling and Simulation

The methods or technique applied to analyze the flow fluid and heat transfer characteristics with passive enhancement of the MCHS. The MCHS computational domain's dimensions are based on experimental obtaining by Sohel *et al.,* were used and the data presented in Table 1.

Table 1
The dimension of the Straight microchannel heat sink

Channel Parameter	Values
Heat sink dimension, (mm)	50 x 50 x 10
Length of the channel, L (mm)	50
Width of the channel, W (mm)	0.5
Height of the channel, H (mm)	0.8
Number of channels	49

At the beginning stage, the simple MCHS and its computational domain are designed as shown in Figure 1 with its dimensions as provided in Table 1. The working fluid used for this analysis was pure water H2O and thermophysical properties of the pure water as shown in Table 2. The analysis was performed for three different Reynolds numbers which were 395 and 989. The heat flux supplied at the bottom surface of the silicon MCHS was q=1.6x105 W/m2.

Table 2
Thermophysical properties of cooling fluid in MCHS

Coolant	Pure water
Density, ρ (kgm-3)	1000
Thermal conductivity, K (Wm-1 K-1)	0.6
Specific heat, Cp (Jkg-1 K-1)	4178

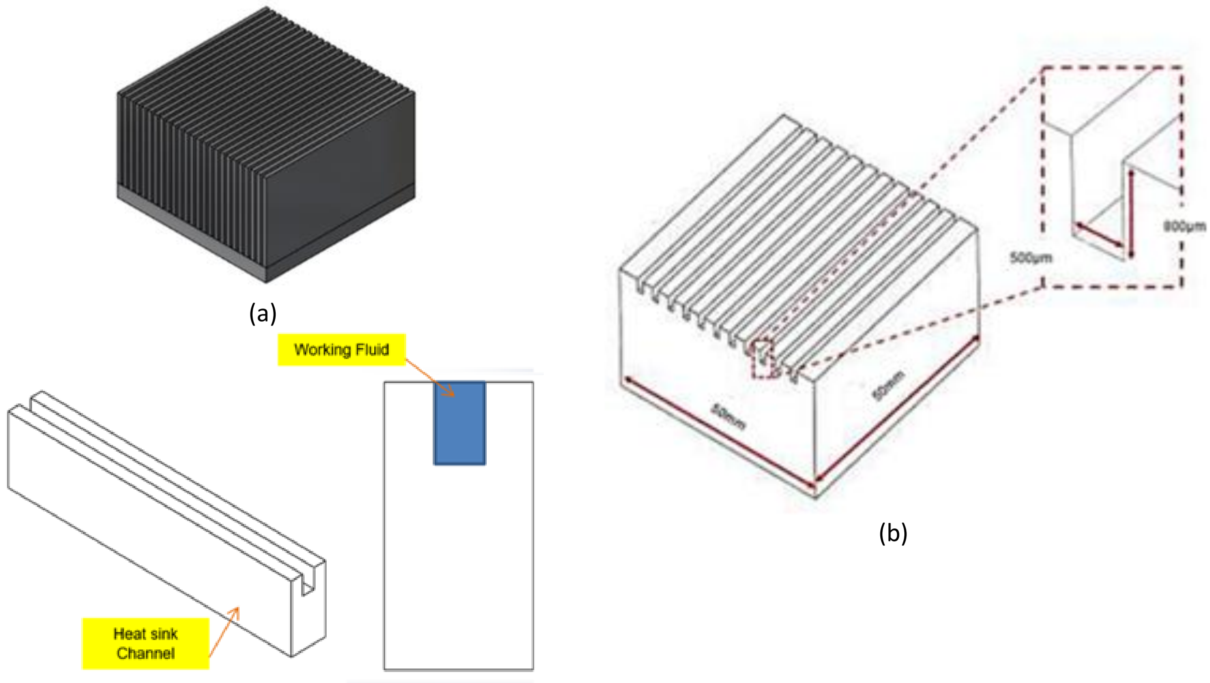


Fig. 1. (a) Picture of MCHS test piece (autodesk.com), (b) Details view of microchannel heat sink, (c) Microchannel heat sink (Schematic view)

2.2 Computation

The following stage involved is referred to computation and it is used Fluent 16.2, commercial CFD programming. Firstly, MCHS meshing will be load by the program and the data like a wall, wall coupled and mass flow inlet for heat transfer and pressure outlet will be given for all the boundary conditions. After that, the solution algorithms and numerical parameters will be determined. In this all-important initial condition for the evaluation is referred for the iteration process initiative. Then, Fluent will solve the energy equations, transport and conservation numerically. Furthermore, the initial guess values are used to solve the discretized forms of the equations. Finally, the achievement of convergence criteria will be succeeding when the residual of every cell in the domain reached zero. In general, geometry creation, structure meshing and defining up of boundary conditions are comprises in the measure which solved by numerically. Figure 2 below demonstrated the structured mesh of the straight MCHS.

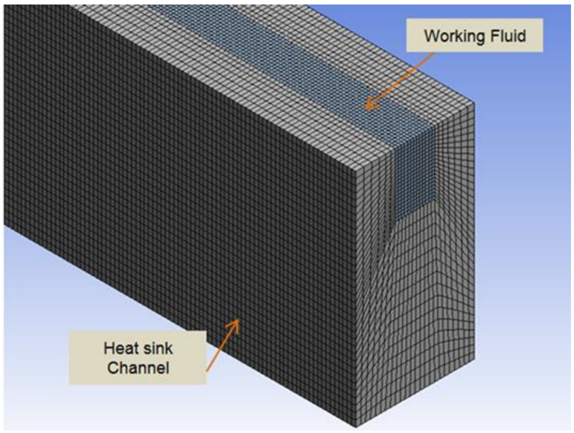


Fig. 2. Structured mesh of the straight MCHS

2.3 Governing Equation

The numerical analysis based on following assumptions.

I. The fluid flow is in steady state and incompressible condition (laminar).
II. The viscosity is depending on the fluid properties, except for temperature.
III. Gravitational force and radiation heat transfer will be neglected.
IV. The fluid without of viscous dissipation.

2.4 Boundary Conditions

The numerical calculation was performed for the entire computational domain i.e. 3D conjugate heat transfer analysis.

I. Thermal boundary condition

x-coordinate at the inlet, the fluid temperature is constant.

T_f = Tin = 293K, at x = 0 (Direction 1, D1) and x = L (Direction 2, D2), (1)

$-K_f \frac{\partial T_f}{\partial x} = 0,$ at x = L (Direction 1, D1) and x = 0 (Direction 2, D2), (2)

No axial heat transfers in solid region.

$-K_s \frac{\partial T_s}{\partial x} = 0,$ at x = 0 and x = L (3)

$\frac{\partial}{\partial y} = 0,$ at y = 0, y = W, (4)

$-K_s \frac{\partial T_s}{\partial z} = q,$ at z = 0 (5)

$-K_s \frac{\partial T_s}{\partial z} = 0,$ at z = H (6)

Fourier's Law defines the heat transfer conjugate between solid and fluid:

$$-K_f \frac{\partial T_f}{\partial n} = -K_f \frac{\partial T_s}{\partial n} \tag{7}$$

where n represents the local coordinate normal to the wall.

II. Hydrodynamic Boundary Conditions

$u = v = w = 0$ at the fluid-solid wall

$U_f = u_{in}$ at x = 0 (Direction 1, Case A) and x =L (Direction 2, Case B),

$P_f = P_{out}$ = 1 atm at x =L (Direction 1) and X=0 (Direction 2)

There is no-slip and no-penetration velocity boundary conditions were employed at the solid wall. A uniform inlet and atmospheric outlet pressure were set as fluid boundary conditions.

In order to check the validity of the constructed numerical model, verification was prepared by Sohel *et al.,* [5] solving the experimental model introduced by and the outcomes were compared. In this research, the geometry and working conditions are equal to those adopted in the experiments and the numerical analysis will use a single channel both sided walls. The configuration of these computational domains shows in Figure 3.

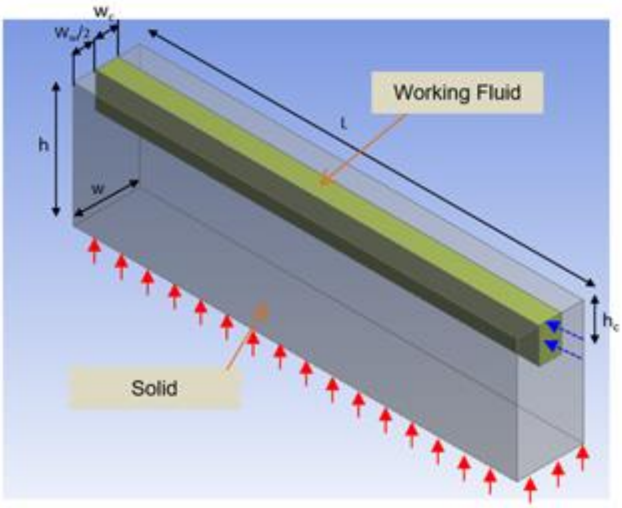


Fig. 3. Computational domain of MCHS with Single Channel

A simulation with an identical boundary condition (Uin=1.5m/s, q"= 1.6x105 W/m2) and the performance of these channels will be evaluated by using solution method. The variation results for the local Reynold number, friction and pressure drop of these computational domains are studied.

3. Results

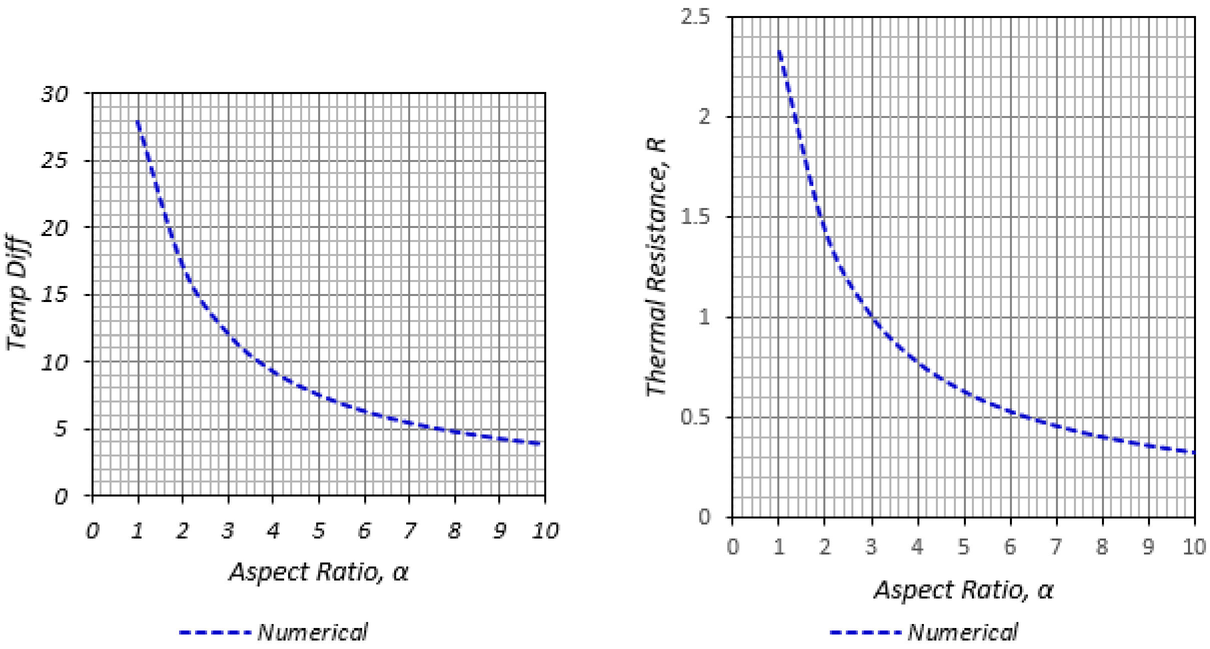
3.1 Effect of Aspect Ratio

The effect on the heat sink thermal performance at vary aspect ratios (α = H/W) of the microchannel is investigated. The footprint of the heat source is at a size of 50 mm × 1.5 mm. The height of the channel, H is varying to acquire various aspect ratios (α) while maintaining the channel width unchanged at 150 µm. When the height of the channel increases, it offers an increased surface area for enhanced heat transfer, thus the aspect ratio also increases. With an increase in channel height, the cross-section at fluid flow is increased and flow rate of fluid flow increases.

I. Variation between different aspect ratio, temperature different and thermal resistance.

The effects of aspect ratio on temperature different are shown in Figure 4(a). The temperature difference is calculated based on average wall and fluid temperature difference (T_w-T_f). It is observed that a high aspect ratio channel will enhance the contact surface area will result notable effects on reduces temperature different of the heat sink temperature.

At lowest aspect ratio, thermal resistances reach the peak and exponentially decrease when increasing aspect ratio as shown in Figure 4(b). At aspect ratios of 2, 4, and 6 thermal resistances is 1.428, 0.775, and 0.529 °C/W respectively. It showed 6% decrease of thermal resistance when the aspect ratio of channel varies from 1.0 to 10. Lower temperature different from the heat sink and significant increase in contact surface areas caused at higher aspect ratio channels the thermal resistance is lowest. However, the rate of change in thermal resistance is very negligible beyond certain aspect ratio (e.g., α = 8.0), signifying the fact that infinite increase in height of channel does not produce enhancements on comparable heat transfer.



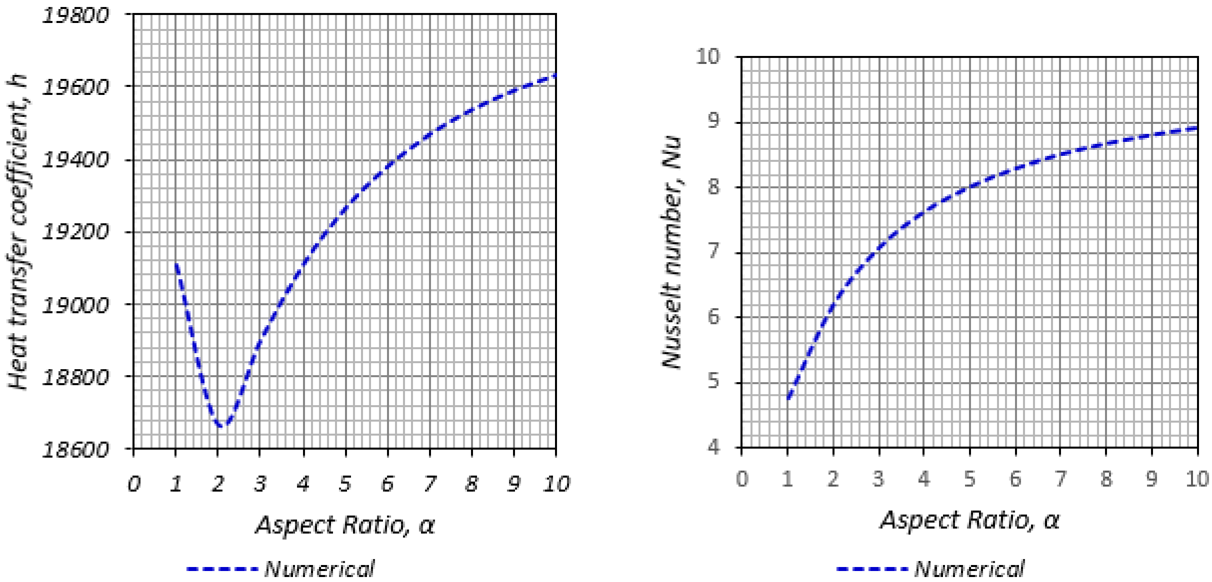
(a) (b)

Fig. 4. (a) Temperature different (b) Thermal Resistance of the microchannel for the different aspect ratio of the channel

II. Variation between different aspect ratio, heat transfer coefficient and Nusselt number

Figure 5(a) emphasizes that there is an effective enhancement of the heat transfer coefficient in the range of aspect ratio of 1 to 10. At the highest aspect ratio, heat transfer coefficient gave around 19,634 W/m2 then influence the heat transfer performance and increase the value of the Nusselt number of the heat sink. Besides that, the increases of Reynolds number also represent the improvement of the convective heat transportation capacity of the coolant and maximize the reduction of the base temperature of the heat sink.

Fully developed average Nusselt number (Nu) and different aspect ratio are demonstrated in Figure 5(b), which the Nusselt number is gradually increased with variation in aspect ratio. It is because of the larger aspect ratio of the channel, it will give the higher value of the Nusselt number and totally the contact surface will be larger and increase the heat transfer coefficient.



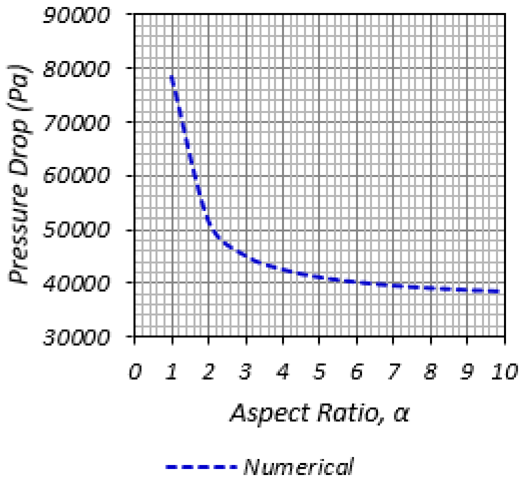
(a) (b)

Fig. 5. (a) Heat transfer coefficient of the microchannel for the different aspect ratio of the channel, (b) Fully developed average Nusselt number and different aspect ratios of the channel

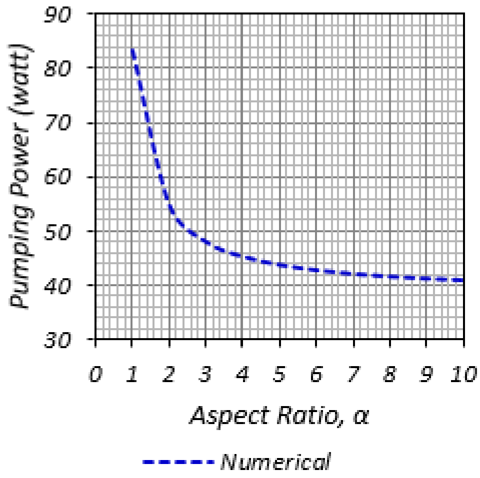
III. Variation between different Aspect ratio, Heat Pressure drop and Pumping power

Pressure drop (ΔP) is corresponding to different aspect ratios are plotted in Figure 6(a). The pressure decreases from 78.52 to 38.5 kPa, when the aspect ratio of the channel increases from 1 to 10. A high pressure drops happened due to the higher flow rate accompanied with notable pressure drops across the microchannel heat sink for the low aspect ratio channels. Low fluid flow velocity caused pressure drop decrease as the channel aspect ratio increase. On the other hand, a higher aspect ratio of microchannel offers lower thermal and resistance lower pressure drop, however, the drawback is the contacting surface area has to be increased and thus more space is required for the heat sink.

Meanwhile, the relationship between pumping power and different aspect ratio has been studied in order to identify the correlation between both. Based on Figure 6(b), the higher pumping power value will gradually reduce the value of aspect ratio of the channel. A pressure drop occurs when the coolant passes through the narrow channel of the heat sink. The system needs some extra pumping power to overcome this pressure drop. Figure 6(b) also demonstrates the decreasing in pumping power with the increasing in aspect ratio as well as cross-section area of fluid flow.



(a)



(b)

Fig. 6. (a) Relationship pressure drops of the microchannel for the different aspect ratio of the channel, (b) Relationship pumping power of the microchannel for the different aspect ratio of the channel

4. Conclusions

The influence of different aspects ratio in the channel is presented and based on results from the simulations it can be summarized as the numerical results conclude that straight MCHS can contribute better heat transfer enhancement with appropriate length and hydraulic diameter. It is also shows that an increase initial surface contact area at the channel offers increased local convective heat transfer coefficient. The ratio between heat transfer by convection (α) and heat transfer by conduction alone (λ/L) is reduced when the distance of fluid flow is increased. Besides that, with higher channel aspect ratio results in higher Nusselt number and heat transfer coefficient but lower temperature different, thermal resistance, pressure drop and pumping power. Higher channel flow rate results in a higher Nusselt number, heat transfer coefficient, pressure drop and pumping power. However, higher of channel flow rate also will result in lower temperature different and thermal resistant. Finally, the required coolant flow rate and pressure drop can be reduced by using higher thermal conductivity heat sink materials, resulting in a smaller or cheaper pump required.

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