Numerical Simulation of Heat Transfer in Gradually Varying Microchannel Heat Sink

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Abstract. Microchannel heat sink has broad relevance in microelectronics and optical devices. Study in this paper focuses on enhancing the cooling characteristics of microchannel heat sink used in such devices. Instead of conventional, rectangular or circular cross-section channel designs, this study uses microchannels with gradually varying cross-section along the length. Three channels geometries employing entirely diverging or converging designs as well as combined diverging-diverging design are studied. The cooling performance is ascertained by using nanofluid and DI water as coolants flowing at different Reynolds numbers by analysing thermal characteristics such as temperature variation and Nusselt number using commercial software ANSYS® Fluent. Study demonstrates that the nanofluid is more efficient for the heat sink application irrespective of the channel geometry.

1. Introduction

Microchannels are commonly used in the fields like biomedical, microfluidic, MEMS to either act as a conduit to carry fluids between reaction chambers or to provide a continuous flow over a confined space for circulation. Microchannel heat sink is devised to transport away heat generated on the hot surface of micro-scale appliances by dissipating it to the fluid medium. It is based on the principle of heat exchanger. However, owing to its small size and high surface area to volume ratio, it shows much higher heat transfer augmentation due to scaling laws. Tuckerman and Pease [1] in 1981 firstly executed the idea of microchannel heat sink where water was used to cool silicon chips. Yong and Teo [2] found that heat sink performance improves up to 60% when rectangular microchannel was changed periodically in a converging-diverging manner along the length in comparison to straight channel.

Mohammed et al. [3] numerically analyzed the thermal and hydraulic characteristics in rectangular wavy microchannel heat sink and compared with the straight microchannels. It was discovered that wavy channel advantage in the performance due to enhanced heat transfer is much higher than the loss caused by raised pressure drop. Also, high amplitude of wavy channel amplifies the friction factor and wall shear stress. Dede and Liu [4] designed microchannel heat sink with multi-pass branch for the high heat flux applications and compared experimental, numerical and analytical results. It was shown that for high power devices, multi-pass microchannel produces low thermal resistance and pressure drop. Al-Neama et al. [5] investigated the effect of changing the number of path channels on hydraulic and thermal characteristics of rectangular serpentine microchannel heat sink both experimentally and numerically, and showed that single path multi-serpentine microchannel performs better.
Chai et al. [6] showed that the performance of the microchannel heat sink, attached with offset ribs on sidewalls, enhances significantly. Comparing rectangular, backward triangular, isosceles triangular, forward triangular and semicircular shape offset ribs revealed that for $Re < 350$, performance of the forward triangular rib is best and rectangular rib is worst while for $Re > 400$, whereas the semicircular rib gives the best performance and backward triangular rib performs worst. Ma et al. [7] induced porous wall concept in a pin finned microchannel heat sink to explore the flow boiling characteristics. It was observed experimentally that heat transfer amplifies while pressure drop and instability in two-phase flow reduces. Hussain et al. [8] proposed hybrid micro-channels heat sink designs with pillars and jet impingement mechanisms and showed its much higher cooling efficiency.

Wu and Cheng [9] determined through experimental results that geometric parameter, surface hydrophilic properties and surface roughness plays significant role in convective heat transfer in microchannel heat sink. Mohd-Ghazali et al. [10] used liquid ammonia to compare the cooling efficiency of circular and square microchannel. It was determined that thermal resistance of the square channel is higher than circular channel for equal pumping power and hydraulic diameter. Numburu et al. [11] predicted that the Nusselt number increases with the use of nanofluid as coolant while comparing turbulent flow behaviour and heat transfer characteristic of heat sink for three different nanofluids ($CuO$, $Al_2O_3$, $SiO_2$) through numerical analysis. Zargartalebi and Azaiez [12] observed the improvement in cooling efficiency of pin finned heat sink due to the addition of nanoparticles in coolant. It was demonstrated that nanofluid viscosity, particle size and surface energy play important role in improving the heat sink performance significantly due to their agglomeration and deposition.

In the present analysis, comparative study for thermal analysis is performed numerically for deionized (DI) water and nanofluid as coolant in a converging, diverging and diverging-converging microchannel heat sink. Heat transfer characteristics are computed for different Reynolds number using computational fluid dynamics based simulation software ANSYS® Fluent.

2. Geometric designs and numerical modelling

In this study, three different rectangular microchannel geometries are chosen with constant height of 0.125 mm, whereas width of the channel is varied gradually along the length. The largest width is kept 2.30 mm and the smallest is 1.04 mm in a total 20 mm long microchannel. Continuously increasing width generates purely diverging channel while continuously decreasing width provides purely converging channel, while diverging-converging channel is equally distributed along the length. Figure 1 illustrates the schematic geometrical design of the microchannels. Hydraulic diameter, $D_h$, is the characteristic dimension of the microchannel, usually defined for non-uniform section channels, to evaluate some characteristics of the system.

**Figure 1:** Schematic designs of converging/diverging (left) diverging-converging (right) microchannels.

For gradually varying rectangular cross-section channel along the length, it is calculated as:

$$D_h = \frac{4 \times W_{avg} \times H}{2 \times (W_{avg} + H)}$$

(1)

$$W_{avg} = \left[ \frac{W + W_e}{2} \times \alpha \right] + \left[ \frac{W + W_e}{2} \times (1 - \alpha) \right]$$

(2)

$$\alpha = \frac{L_{avg}}{L}$$

$$L_{avg} = \frac{L}{2}$$

where $L$ is total length of channel.
where, $W_{avg}$, $W_i$, $W_e$ and $H$ are the average width, inlet width, outlet width and height of the channel, respectively. And, $\alpha$ is the length varying factor lies between 0 and 1. For purely converging flow channel, $\alpha = 0$; for purely diverging flow channel, $\alpha = 1$ and for equally divided diverging-converging channel, $\alpha = 0.5$.

Three-dimensional, steady state, laminar flow is assumed to simulate the fluid flow in microchannel. ANSYS® Fluent software, which is based on the finite volume method, is used to solve the requisite governing equations for the hydraulic and thermal analysis of the coolant in microchannel heat sink. Boundary conditions and settings to simulate the fluid flow were validated and flow characteristics were computed in the similar channel geometry for nanofluid and water [13, 14]. For investigating cooling performance of the selected heat sink, 300 kW/m² heat flux is applied at the bottom wall surface while keeping adiabatic condition on the other wall surfaces. In addition, nano-particle agglomeration and deposition are neglected as well as single-phase flow conditions are assumed for nanofluid flow in the channel. Al$_2$O$_3$ nanoparticles of 20 nm diameter are dispersed in the mixture of ethylene glycol and water (60:40 ratio) in 6% by volume to prepare nanofluid [11]. Table 1 comprises the thermo-physical properties of the water, ethylene glycol-water mixture (EGW) and nanofluid.

Governing equations of the laminar incompressible fluid flow and convective heat transfer phenomenon in the microchannel includes conservation of mass, conservation of momentum, and conservation of energy equations; which are given below:

\begin{align}
\nabla \cdot (\rho \vec{V}) &= 0 \\
\n\nabla \cdot (\rho \vec{V}) &= -\nabla p + \nabla \cdot (\mu \nabla \vec{V}) \\
\n\nabla \cdot (c' T) &= \nabla \cdot (k \nabla T)
\end{align}

SIMPLE (Semi Implicit Method for Pressure Linked Equation) scheme is used to couple pressure-velocity in the pressure-based solver. For spatial discretization second order upwind scheme is selected. To satisfy convergence, residuals are kept under $1 \times 10^{-5}$ for momentum and continuity equation while it is chosen $1 \times 10^{-6}$ for energy equation. Reynolds number ($Re$) indicates the ratio of inertia force and viscous force. It compares the quantitative role of the corresponding forces acting on the fluid flowing and thus indicates whether the flow is laminar or turbulent. It is defined as:

\begin{equation}
Re = \frac{\rho \vec{V} D_h}{\mu}
\end{equation}

where, $\rho$, $\mu$, $\vec{V}$ represent the density, viscosity and mean velocity of the fluid, respectively. Moreover, average Nusselt number ($Nu$) represents the average quantitative ratio of convection heat transfer and conduction heat transfer carried out by the fluid in the entire fluid domain. It is defined as:

\begin{equation}
Nu = \frac{h_{avg} D_h}{k_f}
\end{equation}

where, $h_{avg}$, is the average convection heat transfer coefficient.
Table 1. Thermo-physical properties of various fluids.

| Type of fluid | Density (kg/m$^3$) | Viscosity (mPa-s) | Specific heat (J/kg-K) | Thermal Conductivity (W/m-K) |
|---------------|--------------------|------------------|------------------------|-----------------------------|
| Nanofluid [11]| 1259.3             | 9.670            | 2645.35                | 0.425                       |
| EGW [11]      | 1086.3             | 5.380            | 3084.00                | 0.349                       |
| DI Water      | 998.2              | 1.003            | 2415.00                | 0.600                       |

3. Results and discussion

To analyze the cooling capacity of nanofluid and DI water in three geometrically different microchannels, various characteristic parameters are computed numerically. Figure 2 illustrates the average Nusselt number and average wall temperature obtained for converging channel heat sink. The enhanced cooling effect showed by the nanofluid is clearly evident in the lower wall temperatures observed than in case of water. As Reynolds number increased the average wall temperature decreased by about 22 K in case of water as coolants while this reduction is only 11 K for nanofluid. This is because of the fact that high Reynolds number also indicates high mass flow rate. The higher the coolant mass flow rate the higher the cooling will be. In contrast, as Reynolds number increased Nusselt number also increased for both nanofluid and water. It can be attributed to the fact that the higher the Nusselt number the higher the convective heat transfer and cooling will be in the heat sink. In addition, the increase in Reynolds number increases the Nusselt number by about 36% for water and about 65% for nanofluid. Thus, use of nanofluid is clearly helpful in enhancing heat transfer characteristics of the microchannels. This observation is also true for other two microchannel designs.

Figure 2: Average Nusselt number (left) and average wall temperature (right) for converging microchannel.

Figure 3: Average Nusselt number (left) and average wall temperature (right) for diverging microchannel.
Figure 3 shows the variation of average Nusselt number and average wall temperature for diverging channel when Reynolds number is varied. Average wall temperature value decreased by about 33 K for water and about 13 K for nanofluid. Nusselt number shows continuous rise with Reynolds number and is enhanced by about 51% when nanofluid is employed. It can be seen in figures 2 and 3 that for water the diverging channel has a higher Nusselt number than converging, but the rate of increase in the two are different. Converging channel has higher rate of change in Nusselt number than diverging channel. In addition, Nusselt numbers are also low than the converging channel.

Figure 4 shows the variation of average Nusselt number and average wall temperature for diverging-converging channel when Reynolds number is varied. And, it is observed that when the Reynolds number increased, average wall temperature decreased by about 37 K for water and about 15 K for nanofluid. Moreover, Nusselt number is increased by about 55% and about 13% for nanofluid and water respectively.

![Figure 4](image_url)

**Figure 4**: Average Nusselt number (left) and average wall temperature (right) for diverging-converging microchannel.

Based on the results presented above, it can be observed that for the same Reynolds number the wall temperature reduces much more when nanofluid is used in comparison to water. For example, the wall temperatures in case of nanofluid under the given heat flux input condition and at $Re = 50$ are maintained at about 18 K in converging channel, about 19 K in diverging channel, and about 36 K in diverging-converging channel lower than DI water. This reduction of wall temperature is attributed to the higher specific heat capacity of the nanofluid in comparison to the DI water. Although the reduction in wall temperature obtained for the nanofluid is more in comparison to water but the difference produced decreases as the Reynolds number increases. That is, efficiency in the wall temperature of nanofluid and water is high at low Reynolds number and low at high Reynolds number. Additionally, the reduction in the wall temperature is more for the diverging-converging channel followed by the diverging and converging channel.

Furthermore, increase in Nusselt number with the Reynolds number is because convection heat transfer amplifies with the mass flow rate and so with the Reynolds number that consecutively enhance Nusselt number. For the same Reynolds number in the range of 50-250, increase in the Nusselt number for nanofluid in comparison to water is 50-90%, and 25-70% in converging and diverging-converging channels, respectively. Thus, high heat capacity of the coolant augments the heat transfer coefficient of the fluid and therefore the convection heat transfer. Furthermore, thermal conductivity of the nanofluid is small as compared to water which indirectly assists to enhance the Nusselt number. Combined effect of the high heat capacity and low thermal conductivity of the fluid attributes to high Nusselt number and more convection heat transfer for nanofluid than water as coolant. Additionally, average Nusselt number for converging channel is more than the diverging and diverging-converging channels with some abnormalities in the results of diverging channel for water as coolant. This is contrary to wall temperature analysis, where lowest wall temperature obtained for
diverging-converging channel. This happens due to the changing cross-section of the channel which further influences the mean velocity of the fluid, which is flow accelerates in the converging channel while it decelerates in the diverging channel.

4. Conclusion
Comparative numerical analysis for effect of nanofluid and DI water as coolant in microchannel sink sinks employing diverging, converging and diverging-converging cross-sections was studied numerically. Average Nusselt number and average wall temperature at the bottom hot surface were computed for all the three channels at different Reynolds numbers. Convergence test was conducted to ascertain veracity of the numerical approach adopted. Results showed that at \( Re = 50 \), the nanofluid helps in maintaining the wall temperature 18 K, 19 K and 36 K lower than DI water in converging, diverging and diverging-converging microchannels respectively. It was also observed that for both kinds of coolants, the increase in Reynolds numbers increased the Nusselt number leading to a decrease in wall temperature in all the three channel types. By substituting the water with nanofluid, average wall temperature reduced by between 11 K and 37 K for Reynolds number range studied here. Similarly, average Nusselt number increased up to 70-90% in case of nanofluid. Future works will focus on experimental validations of the results presented here.

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