Entropy generation study of TiO$_2$ nanofluid in microchannel heat sink for Electronic cooling application

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Abstract. Development of Micro-electro-mechanical systems (MEMS) in the recent years has motivated and necessitated the study of flows in micro-scale geometries such as microchannel. Thermal management in ultra-densely packed electronic devices is highly essential to increase the reliability of the component without compromising packaging. The present study provides an experimental and numerical investigation on laminar forced convection in parallel microchannel heat sink accompanied with integrated Aluminium bulk heat spreader and ultrafine TiO$_2$ nanoparticle based nanofluid for different wt. % ranging from 0.1-0.35 under different power ratings. Numerical study is performed to understand the flow hydrodynamics in microchannel to investigate the temperature distribution in bulk heat spreader with increased flow rates by implementing the thermo-physical properties. Furthermore, a study on Exergy and entropy generation for different fluids is also discussed. The experimental studies reveal that parallel microchannel increases the effectiveness of integrated cooling with a marginal temperature deviation between the heat sink and Aluminium bulk for a distance of 1.5 mm. Implementation of TiO$_2$ nanofluid registered as a better working fluid than the pure fluid for all the experimental settings.

1. Introduction

The rapid advances in miniaturized electronic components developed high performance devices with enhanced efficiency. But the complexity of their function has tremendously increased leading it to dissipate more heat which leads to reduction of reliability of the system. Thus efficient cooling is needed to operate the device in a thermally stable condition by implementing a robust heat transfer system. In recent years many efficient and compact heat transfer methods were proposed, among them most effective is by using microchannel heat sink which was initiated by Tuckerman and Pease [1].

In 1995, Choi [2] first formulate the term nanofluid to refer the fluids with ultratine particles in common liquids called nanoparticle. They estimated the potential benefits of the fluids including drastic reduction in pumping power of heat exchanger and increased thermal conductivity. Nanofluid [3] which exhibits higher thermal conductivities resulting in enhanced heat transfer rates and increased entropy generation due to increased volume fractions of nanoparticle. Wu et al. [4] numerically observed effectiveness of nanofluid Al$_2$O$_3$/H$_2$O on performance characteristics of microchannel heat sinks. Increasing entropy generation or reducing thermal efficiency always affects thermal devices due to increase in irreversible losses. Thus, in the design of microchannel heat sinks, it is essential to find the entropy generation or Exergy. Entropy generation analysis in a uniformly heated rectangular microchannel was investigated.
by Abbassi [5] considering the fully developed laminar flow. He concluded that best thermal performance was achieved when the Nusselt Number is high and entropy generation is low. Sohel et al. [6] analytically showed that in a turbulent flow condition Al₂O₃-water reduce entropy generation rate and the smaller channel diameter shows less entropy generation. Entropy generation rate for different shaped microchannel was numerically investigated by Alfaryjat et al. [7] and concluded that, as the Reynolds Number increases entropy generation rate escalates. Results also showed that square microchannel heat sink has lowest entropy generation rate. An analytical analysis was performed by Singh et al. [8] on entropy generation for three different tube sizes due to Al₂O₃-water nanofluid. They concluded that for both turbulent and laminar flow regimes an increment in the entropy generation ratio was found for Al₂O₃-water nanofluid in both the microchannel and conventional channels.

A numerical investigation on entropy generation of a rectangular conventional channel as well as heat transfer performance was carried out by Xie et al. [9] due to the flow of Al₂O₃-water nanofluid. They concluded that the average heat transfer entropy generation rate was found to decrease with increase in Reynolds number and as the nanofluid volume fraction is increased a rise in rate of average friction entropy generation is found. Due to laminar flow the entropy generation was explored by Mohammadian et al. [10] using Al₂O₃-water nanofluid and results showed that a decrement is found in the fluid frictional entropy generation as the particle size increases although an increment in the heat transfer contribution is found. They also concluded that high volume fraction nanofluid is needed to reduce the entropy generation. Finally, the overall performance of microchannel heat sinks is optimized by minimizing entropy generation procedure technique was given by Khan et al. [11].

Based on above literature review it is clear that entropy generation in microchannel heat sinks has very limited numerical and experimental analysis. Also, from the literature review it is contemplated that nanofluid effectively elevates the heat transfer in microchannel with substantial pressure drop. In many cases increase in nanofluid concentration augments the efficiency of the microchannel heat sink with reduced entropy generation. The point of this study is to examine experimentally and numerically entropy analysis on laminar forced convection in parallel microchannel heat sink accompanied with integrated Aluminium bulk heat spreader and ultrafine TiO₂ nanoparticle for different flow rates.

2. Methodology

2.1. Geometry description

The A laminar forced convection in rectangular microchannel is considered for the present study. The simplistic diagram of the channel is shown in Fig.1. High thermal conductivity material is utilized for the channel and the flow inside the channel is assumed to be three-dimensional flow. The microchannel having bottom area of 10 mm² and height of 6.5 mm is used as the heat sink. An acrylic sheet is kept on the top surface of the channel for increased resistance. The microchannel is constructed with finite thickness on its bottom to adapt heat source or electronic component. Copper material is used for the solid region of the microchannel. The fluids used for cooling the microchannels are water and TiO₂ nanofluid. A heat spreader of 1 mm thickness is positioned 0.5 mm above the source and 2 mm near the domain of the fluid flow with an area of 30 mm² and with a hydraulic diameter of the micro channel as 375µm. The effect of heat transfer and fluid flow due to the involvement of the surface roughness at the inner walls of the rectangular channel is also considered. It should be noted that the surface roughness is introduced to reproduce the outcome of the microchannel fabrication.

2.2. Experimental Set up

The experimental set up comprises of Copper rectangular microchannel encompasses with Aluminium block as shown in Fig.2. Peristaltic pump is used to provide pulsating flow to the microchannels at the channel inlet and to cool down the fluid to the room temperature the outlet is connected to secondary heat exchanger before circulation. Experiment is performed taking a test block with rectangular copper Channel surrounded by Aluminium block which in turn connected to thermocouples and flow circuit is
maintained. A DC source is connected to cartridge heater which is fitted with the channel block where the temperature of heater is gradually increased by varying the voltage supply. Under this condition power input is fixed and copper channel is set to reach a maximum temperature of 60°C which is undesirable for densely packed electronic components. The data from thermocouples are achieved with higher flow rates. The fluid is allowed to flow through the channel after filling the inlet plenum and the corresponding change in the temperature along length of the channel and surrounding temperature drop in Aluminium are recorded. After 120 sec of run time, the maximum energy is extracted from the source and difference in temperature of the fluid was found to be 4°C between outlet and inlet.

2.3. Governing Equations and Boundary Conditions
A single phase model has been adopted for the numerical studies of laminar forced convection heat transfer of the rectangular microchannel. This model includes conduction and convection heat transfer simultaneously. Thus steady state mass, momentum equation and energy equations are solved for TiO$_2$ nanofluid. Steady state heat conduction equation is solved for the solid copper region. The boundary conditions are shown in Fig 4. The fluid is considered as single phase, incompressible laminar without viscous dissipation of energy. Laminar fully developed flow and uniform zero pressure is assumed at the outlet.
2.4. Thermo physical properties of nanofluid are:
The numerical study is investigated with the nanofluid consisting nanoparticle of TiO$_2$ and water used as the base fluid. The nanoparticle are contemplated to be in thermal equilibrium with the base fluid. Also, it is assumed that the mixture obtained is homogenous. The different physical properties of used nanofluid have been calculated using the following equations. The density and specific heat of the nanofluid (Eqn 1 and Eqn 2) are determined using references [12] and [13] respectively.

\[
\rho_{nf} = \rho_{p} (1 - \phi) + \rho_{p,\phi}
\]

\[
C_{p, nf} = \left(\frac{\rho_{p, C_{p, p}} (1 - \phi)}{\rho_{p}} + \rho_{p,\phi} C_{p, p}\right) \frac{1}{\rho_{p}(1 - \phi) + \rho_{p,\phi}}
\]

Eqn (3) is used to calculate the effective viscosity of the Nanofluid using Brickman [14] equation.

\[
\mu_{nf} = \frac{\mu_{p}}{(1 - \phi)^{2.5}}
\]

2.5. Data Reduction
In order to understand heat transfer characteristics of the current model the non-dimensional form of heat transfer enhancement Nusselt number is expressed in Eqn 4.

\[
Nu = \frac{hD_{h}}{k_{nf}}
\]

Where $D_{h}$ is the hydraulic diameter of the channel, $h$ is heat transfer coefficient and $k_{nf}$ is the thermal conductivity of the nanofluid.

\[
D_{h} = \frac{4W_{ch}H_{ch}}{2(W_{ch} + H_{ch})}
\]

Here $W_{ch}$ and $H_{ch}$ are the width and height of the channel. The local heat transfer coefficient is expressed by Eqn (6).

\[
h_{l} = -\frac{Q_{h}}{A_{c}(T_{w} - T_{f})}
\]

Where $A_{c}$ is the area of the channel and $Q_{h}$ is the amount of heat given, $T_{w}$ is corresponding wall temperature and $T_{f}$ is fluid inlet temperature. Darcy-Weisbach equation is used to determine the friction factor can be expressed by Eqn (7).

\[
f = \frac{2D_{h} \Delta P}{\nu_{m}^{2}L\rho}
\]

\[
P_{p} = \frac{m}{\rho} \Delta P
\]

Here $\Delta P$ are the experimentally measured pressure difference values at inlet and outlet plenum.

\[
\nu_{m} = \frac{m}{\rho_{nf}(nA_{ch})}
\]

Where $n$ is number of channels, $m$ is the flow rate in the microchannel.

2.6. Exergy Analysis
The outlet Exergy of the microchannel heat sink is calculated by using Eqn (10).
\[ Ex_{\text{out}} = C_{nf} \left[ (T_{nf, \text{out}} - T_e) - T_e \ln \left( \frac{T_{nf, \text{out}}}{T_e} \right) \right] \]

Where \( C_{nf} \) is heat capacity of the nanofluid. Exergy gain acquired by the cooling fluid is calculated through Eqn (11), where \( P_p \) represents the pumping power used in the system given by Eqn (8).

\[ Ex_{\text{gain}} = C_{nf} \left[ (T_{nf, \text{out}} - T_{nf, \text{in}}) - T_e \ln \left( \frac{T_{nf, \text{out}}}{T_{nf, \text{in}}} \right) \right] - P_p \]

2.7. Entropy Analysis
To determine the entropy generated by heat transfer and the fluid friction can be expressed as follows:

\[ S_{g,HT} = \frac{q^2 \pi D_f^2 L}{kT_{\text{in}} T_{\text{out}} N_{\text{it}}} \]

\[ S_{g,FF} = \frac{8 m^3 L}{\pi^2 \rho^2 \left( \frac{T_{\text{in}} + T_{\text{out}}}{2} \right)} D_f^5 \]

2.8. Numerical Method
The solid and fluid regions are discretized based on finite volume method. The entire microchannel is considered for the computational purpose. SIMPLE (Semi-Implicit Method for Pressure Linked Equations) method was selected for the coupling of pressure and velocities. The resulting algebraic system of equations are solved by using Gauss Seidal iterative method with successive over relaxation (SOR) to enhance the convergence time. The numerical model is verified with grid independence study and validated with standard experimental result to ensure the fidelity of the mathematical model and the approach of the methodology. The convergence criterion is set to \( 10^{-6} \).

2.9. Grid Independence Study
The grid system employed in the numerical analysis has 568190 elements in the x, y and z directions respectively. The sensitivity of the numerical results is checked with four different grids 299904, 568190, 856111 and 980209 elements respectively. The results from the last three grids are very close to each other as shown in Fig. 3. Taking into account less computational time and resources needed for performing the simulation, the second grid system is used in the current work.
3. Results and Discussion

3.1. Validation Study
The developed numerical model is validated with conducting laminar forced convection experiment for the flow rate of 210 ml/min. using DI-water with a constant heat flux condition. The obtained results provide considerable agreement with the experimental result which is presented in Fig 6. It shows that the temperature along the channel length of the heat sink increases for the simulated condition.

3.2. Exergy Study
The outlet Exergy for different volume fractions of TiO$_2$ for increase in flow rate is calculated by Eqn 10 and presented in Fig 7. It is perceived that the maximum outlet Exergy was developed by using TiO$_2$ nanofluid with 0.25% volume fraction generating outlet Exergy of 8W which is 60% more than that of pure fluid for the same flow rate of 210 ml/min. This increase in outlet Exergy is highly contributed to increased thermal conductivity of the working fluid at higher TiO$_2$ volume fractions. On other hand increase in flow decrements the Exergy gain which is presented in Fig 8 and a maximum gain of 8 W is developed for 0.25% of TiO$_2$ at a lower flow rate of 210 ml/min with the increase in flow rate the Exergy gain slightly started to reduce. For the same flow rate condition the pure fluid developed a gain of 4 W. This trend is contributed increase in sink temperature for increased flow rate.

![Figure 7. Outlet Exergy for increase in flow rate with different nanofluids.](image1)

![Figure 8. Exergy gain for increase in flow rate with different nanofluids.](image2)

3.3. Entropy Study
The entropy generated in the system is discussed in the Fig 9 and 10. Fig 9 depicts the thermal entropy generation and Fig 10 represents the entropy generated from fluid friction which is calculated from the Eqn. 12 and Eqn. 13. It is perceived that increasing thermal conductivity of the fluid combined with increased flow tend to reduce the thermal entropy generation. The thermal entropy is generated by 0.25% TiO$_2$ is about 0.033 W/K for a flow rate of 135 ml/min and at 270ml/min it reduced till 0.023W/K. In contract frictional entropy increases with the increased flow rate and the contribution of the fluid entropy is lower when compared with the thermal entropy generation. It majorly takes into account of increase in friction along the flow length. The maximum frictional entropy is developed for highest mass flow rate when compared the increment of 0.25% which is 6% more than that of 0.1 and 9% more than pure fluids. Frictional entropy generation shows relatively lower than the thermal entropy which is primarily due to the inclusion of lower volume fraction associated with a less flow rate. Furthermore the used heat sink has low axial length along the direction of flow. Moreover use of the TiO$_2$ nanofluids develops increase in thermal conductivity which shows the predominance in the thermal entropy generation.
It is seen that the decrease in channel decreases the time period shown in the Fig 11. Due to this, the direct cooling of copper channel influences the Aluminium block temperature. But the temperature drop in the Aluminium block is very gradual when compared to the copper channel.

Since the materials under investigations possess high thermal conductivity, the temperature difference measured between Aluminium and copper channel core is more or less similar when the flow was initiated. Due to this, the direct cooling of copper channel influences the Aluminium block temperature. Fig. 12 depicts the decrease in temperature for different powers ranging from 14W to 20W. For a constant mass flow rate, 20W power consumes more fluid to attain the temperature near to room temperature and correspondingly it drops for higher flow rate and the maximum cooling is observed for 14W. Also, with the increase in volume fraction a maximum drop in channel temperature for a flow period of 2 min using pure fluid at a flow rate of 210 ml/min with pulsating flow. As soon as the flow is activated, a steep drop in temperature of about 12°C was observed within 8sec in copper channel and the drop reduces as the flow progresses the temperature of the channel decreases nearly 7°C more than the inlet fluid. But the temperature drop in the Aluminium block is very gradual when compared to the copper channel.

3.5. Fluid Hydrodynamics Study
The flow hydrodynamics of the performed domain is presented in the Fig 13 (a) which shows the dedicated planes to capture the fluid streamlines throughout the fluid region. Fig 13 (b) to (d) depict the streamline plot along the channel for the different flow rate ranging from 135 ml/min to 189 ml/min.
Maximum fluid velocity is seen in centre of the fluid region throughout the length of the flow which is shown at Fig 13 (b).

| Flow Rate (ml/min) | Velocity Streamline | Pressure Contours |
|--------------------|---------------------|-------------------|
| 135                | ![Velocity Streamline](image1.png) | ![Pressure Contours](image2.png) |
| 135                | ![Velocity Streamline](image3.png) | ![Pressure Contours](image4.png) |
| 162                | ![Velocity Streamline](image5.png) | ![Pressure Contours](image6.png) |
| 189                | ![Velocity Streamline](image7.png) | ![Pressure Contours](image8.png) |

**Figure 11:** Velocity streamline, Pressure contours for pure fluid DI – water for different flow rate.

The maximum velocity of 1.205 m/s was observed at central microchannel throughout the normal of flow direction. Further increase in flow rate develops a similar trend with increased number of channels which is shown from Fig 13 (c) to (d). Variation of centerline velocity for water with axial position follows same trends and will be maximum inside the channel for all the flow rates independent of the
channel width. The major change in fluid flow develops near the channel inlet which results chaotic flow is seen for all the flow rates causing maximum pressure drop which is shown in Fig 13 (e). This is further increased for higher flow rate shown in Fig 13 (f) to (g). Furthermore flow near channel exit distributed more or less similarly along the height of the channel but soon it recombines to develop centreline velocity throughout the fluid region.

4. Conclusion
Numerical simulations were performed on a rectangular microchannel accompanied with heat spreader for various flow rates and heat flux for the working fluid containing both water and TiO$_2$. For the same heat flux, the heat removal rate of very low concentration TiO$_2$ nanofluid was found superior to pure fluids. The simulation results are summarized below:

1. The heat removal rate of TiO$_2$ nanofluid was found superior than pure fluid and TiO$_2$ with 0.25% volume fraction developed uniform cooling throughout the heat sink.
2. Simultaneous cooling on Aluminium block was observed with a maximum temperature gradient of 10°C at lower flow rate and it decreased with the increase in flow rate and TiO$_2$ volume fraction.
3. It is observed that the maximum outlet Exergy was developed by using TiO$_2$ nanofluid with 0.25% volume fraction. It generated an outlet Exergy of 8W which is 60% more than that of pure fluid for the same flow rate of 210 ml/min.
4. The thermal entropy is generated by 0.25% TiO2 is about 0.033 W/K for a flow rate of 135 ml/min and at 270ml/min it reduced till 0.023W/K. The frictional entropy is developed is increased at higher flow rates for all the working fluids subjected.
5. The maximum frictional entropy is developed for highest mass flow rate when compared the increment of 0.25 wt. % which is 6% more than that of 0.1wt. % and 9% more than that of pure fluids.

Based on the results it shows that Aluminium surrounding with heat sink cooling involving reduced TiO$_2$ nanofluid as working fluid will be effectively suitable for handling high flux density without compromising packaging for highly compact electronic devices.

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