Numerical investigation of Two-phase Turbulent forced convection heat transfer and flow of nanofluids in a non-parallel wall minichannel heat sink

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Abstract. A Miniaturisation is an advance micro-technology of size reduction of handheld electronic devices which significantly enhance their performance though with a substantial increase in a thermal generation. Quick and efficient dissipation of the heat flux ensures reliability and prevent premature failure. This work employs simultaneous passive techniques of using nanofluid and corrugated minichannel configuration on forced convection heat to improve hydro-thermal performance and examine the effect of nanofluids on heat transfer and pressure drop. The CFD analysis is conducted by FLUENT software using two-phase mixture model under a uniform heat flux for two different water-based nanofluids with volume fractions of 0.005–0.03 and the range of Reynolds numbers 5000–10000. Validation of the numerical results with existing data in the literature displays a good agreement with a fair deviation. The results indicated that the heat transfer rates and wall shear stress increase with the increase of the nanofluid volume concentration. Besides, Heat transfer coefficient appreciates by about 39% at 3vol% with an increase of Re from 5000 to 10000 for both nanofluids. In contrast, the friction factor increases and causes a fair pressure drop. Hence passive enhancement techniques used indicated high suitability for hydro-thermal performance in compact electronic devices.

1. Introduction
Thermal management of heatsinks and heat exchangers have become a dynamic process due to ever-changing needs and constraints in the devices. Electronic devices and heat exchangers are becoming smaller in size. At the same time, they are expected to give better efficiency and use less energy. These requirements make their thermal performance a difficult task. Despite the challenge posed by thermal management in electronics and process industries, some researchers conducted significant investigation either experimentally or numerically using either a single-phase or multiphase [1-6].

Nanofluid has shown the remarkable result as thermal fluid because of improved thermophysical properties relative to the base fluid [7-11]. Passive techniques to improve heat transfer, such as combining geometry variation with highly conductive nanofluids has been regarded as reliable options [12]. Dominic et al. [13] investigated heat transfer and pressure loss via experimental and numerical methods considering wavy divergent and wavy uniform cross-sections using Alumina-Water for the range of Re 700-3300. They reported higher heat transfer performance about 9 % in the divergent wavy minichannels and 30–38 % lower pressure drops than the wavy uniform minichannels. Anticipating
variation of the result between the single-phase and multiphase, researchers [14, 15] examined the effect of these distinct methods in the nanofluids hydrothermal analysis. Also, micro and minichannels are preferred geometries for such investigations [16-18] Ghasemi et al. [19] conducted an extensive review on numerical methods employed in the hydraulic and thermal analysis of fluids. The better prediction observed in the two-phase model compared to the single-phase since nanofluid is a heterogeneous mixture. Though Sidik et al. [20] reviewed the significance of turbulent flow and concluded that turbulent flow been widely applied in the industry could enhance heat transfer using nanofluids as thermal fluids. Ahmed et al. [21] examined heat transfer of water-based Al2O3 and CuO nanofluids by a two-phase numerical method in the triangular duct having vortex generators. They found significant thermal enhancement at 3 vol.% of Al2O3-water nanofluid and Re 16000 with the highest overall heat transfer performance of 45.7%. However, Albojamal and Vafai [22] reported reasonable agreement of the newly proposed single-phase method with the experimental result, while the two-phase methods either over-estimate or gave unrealistic values.

Despite the reported successes and shortcomings in the past literature, numerical studies on two-phase using nanofluid in diverging-converging minichannel were grossly limited. Thus, there is a need for more understanding about its potential in hydro-thermal performance in cooling electronic devices. The hydrothermal performance examination of aqueous Al2O3 and CuO nanofluids through a diverging-converging minichannel heat sink (DCMCHS) is the objective of the paper. It involves studying the influence of the nanofluids on heat transfer coefficient, wall temperature, friction resistance and overall pressure drop using the Eulerian-Eulerian two-phase mixture model.

2. Mathematical formulation

2.1. Thermophysical Properties of Fluids

In this work, α-Alumina oxide and Copper Oxide nanoparticles and pure water as base fluid formed the nanofluids; with a volume fraction of 0.005 to 0.03. Table 1 expressed the thermophysical properties of the nanoparticles and water.

| Materials          | Unit       | Water (H2O) | Alumina (Al2O3) | Copper oxide (CuO) |
|--------------------|------------|-------------|-----------------|--------------------|
| Density            | (kg/m³)    | 996         | 3970            | 2220               |
| Specific heat      | (J/kg K)   | 4178        | 765             | 540                |
| Thermal conductivity | (W/mK)   | 0.615       | 36              | 32                 |
| Dynamic Viscosity  | *10^-6 (kg/ms) | 798     | -               | -                  |

The density and specific heat capacity are computed by Equations 1, and 2, respectively.

\[ \rho = (1 - \phi) \rho_{bf} + \phi \rho_p \]  
\[ C_{Pnf} = \frac{\phi (\rho_p C_p) + (1-\phi) (\rho_{bf} C_{bf})}{\rho_{nf}} \]

Hamilton and Crosser [23] incorporate the shape factor of the nanoparticle (n=Ψ) in estimating the thermal conductivity of the nanofluid (Knf) using the following model:

\[ \frac{k_{nf}}{k_{bf}} = \frac{k_p + (n-1) k_{bf} - \phi(n-1)(k_{bf} - k_p)}{k_p + (n-1) k_{bf} + \phi(k_{bf} - k_p)} \]

Batchelor [24] proposed an equation to calculate the nanofluids viscosity with constant properties based on experimental data and depends only on the volume fraction:

\[ \frac{\mu_{nf}}{\mu_{bf}} = \left ( 1 + \frac{5}{2} \phi + 6.2 \phi^2 \right ) \]

Where, \( \rho, C_p, k, \) and \( \phi \) denote the density, specific heat capacity, thermal conductivity, viscosity, and concentration of the nanoparticle, respectively. Besides, subscripts nf, p, and bf denote, the nanofluid, nanoparticle and the base fluid, respectively. For the spherical nanoparticle, n = 3.
2.2. Model geometry and boundary conditions
The study examined the steady forced convection heat transfer and flow in a Divergent-Convergent minichannel heat sink (DCMCHS) as depicted in Figure 1. Muhammad et al. [25] provide the details of the geometry. The base experienced a constant heat flux of 850 kW/m², while the top surface and sidewalls were well insulated, and they experienced the no-slip condition. The nanofluid assumed to be Newtonian fluid and flows through the channels of an aluminium heat sink attached to the chip bottom. At the inlet of the DCMCHS, “velocity-inlet” assigned and the working fluid enters at a temperature of 303K in the turbulent continuum, while “Pressure outlet” imposed at the outlet with p= 0 Pa (gauge pressure). Based on the conditions mentioned above, we expressed the Navier-Stokes and energy equations as follows:

Continuity equation:
$$\nabla \cdot \left( \rho_m \cdot \vec{v}_m \right) = 0$$  \hspace{1cm} (5)

Momentum equation:
$$\nabla \cdot \left( \rho_m \cdot \vec{v}_m \cdot \vec{v}_m \right) = -\nabla P_m + \nabla \cdot \left( \rho_m \cdot \nabla \vec{v}_m \right) + \nabla \cdot \left( \sum_{k=1}^{n} \phi_k \rho_k \vec{v}_{dr,k} \cdot \vec{v}_{dr,k} \right)$$  \hspace{1cm} (6)

Energy equation:
$$\nabla \cdot \left( \sum_{k=1}^{n} \phi_k \vec{v}_m (\rho_k E_k + P_m) \right) = \nabla \cdot k_m \nabla T$$  \hspace{1cm} (7)

Volume fraction equation for the secondary phase (p):
$$\nabla \cdot \left( \phi_p \cdot \rho_p \cdot \vec{v}_m \right) = -\nabla \cdot \left( \phi_p \cdot \rho_p \cdot \vec{v}_{dr,p} \right)$$  \hspace{1cm} (8)

Where n is the number of phases, and $\rho_m$ and $\mu_m$ signify the mixture’s density and viscosity, respectively, while $\vec{v}_m$ is a mass-averaged velocity.

2.3. Turbulent flow model
This analysis employed the realisable k-ε turbulence model proposed by Shih et al. [26]; it is a modification of initial work by Lauder and Spalding [27]. Its characterised by the kinetic energy k and $\varepsilon$ for dissipation rate of this turbulent energy, and respectively expressed as follows:

$$\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho k v_j) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t \sigma_k}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_m + S_k$$  \hspace{1cm} (9)

$$\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_j} (\rho \varepsilon v_j) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t \sigma_\varepsilon}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_1 S \varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\varepsilon \varepsilon}} + C_1 \frac{\varepsilon}{k} C_3 \varepsilon G_b + S_\varepsilon$$  \hspace{1cm} (10)

Where $G_k$ and $G_b$ represent turbulence kinetic energy generation due to average velocity gradient and buoyancy, correspondingly. $Y_m$ is the incompressible turbulence involvement of the unstable dilatation to the total dissipation rate. From equation 10, the values of the model constants are: $C_{1\varepsilon}$, $C_2$, $\sigma_k$ and $\sigma_\varepsilon$ are 1.9,1.0 & 1.2, respectively.

Figure 1 The computational domain. (a)The geometry, (b) Structured mesh of geometry
3. Numerical approach

3.1. Solution scheme
A commercial CFD solver ANSYS FLUENT v17 modelled the three-dimensional forced convection heat transfer and nanofluid flow. The fluid (water) and the solid (nanoparticle) phases coupled using dual-way coupling through the numerical work. Table 2 summarised the methods and schemes used in this work. The convergence of all the variables set when the normalised residual readings are lesser than $10^{-6}$.

| Discretisation subject | Solution method/Scheme |
|------------------------|------------------------|
| Pressure               | PRESTO                 |
| Pressure-Velocity      | COUPLED                |
| Momentum               | Third-order MUSCL      |
| Energy                 | Third-order MUSCL      |
| Volume fraction        | QUICK                  |

3.2. 3-D Grid sensitivity test
The simulation work used five different hexahedral mesh generated by edge-sizing along with the cartesian coordinates. Different grids dimensions from 600000 (50*40*300) to 1228800 (64*64*300) engaged to ensure non-reliance of the simulation results on the dimension and the magnitude of the cells. The Nusselt numbers (Nu) error on the mesh with 900000 and 1228800 elements were below 0.5%. The friction factor shows a similar trend between these grids; hence, all the simulations in this study used the mesh with 900000 elements for the economy of computing time and memory.

3.3. Validation of numerical results
The numerical data validation achieved by comparison with well-known correlations to confirm the ability of the solver to accurately predicts the outcomes. Using Phillips et al. [28] and Blasius relation [1] for the fully developed turbulent region used for local Nusselt number and average frictional resistance; respectively, and using pure water conditions at Reynolds number 20000. The deviation of local Nusselt number with the correlation is about 3%, as shown in Figure 2 (a). In comparison, in Fig. 2 (b) the numerical values for the friction factor deviate from the theoretical results by about 20% and 4% lesser at 5000 and 20000 Reynolds numbers, correspondingly; because of the assumptions in the mathematical formulation of the simulation.

![Figure 2](image-url)

**Figure 2.** Validation of (a) Local Nusselt number (Nu) and (b) Friction coefficient (f)
4. Results

This section presents the results obtained using a two-phase mixture model for volume concentrations ($\phi$) 0.5 to 3 volume %. Reynolds numbers ranged 5000 – 10000 and uniform heat flux of 850 kW/m$^2$.

4.1. Heat transfer analysis

The nanofluids volume fraction and Reynold number significantly influence the Heat transfer coefficient enhancement, as demonstrated in Figure 3. An increase in volume concentration and Reynolds number enhances the coefficient of heat transfer (HTC) due to the rise in effective thermal conductivity. Al$_2$O$_3$-H$_2$O having the highest effective thermal conductivity value than CuO-H$_2$O shows a better enhancement, though, its effective density and increased frictional resistance affect its HTC enhancement.

![Figure 3](image)

**Figure 3.** Variation of HTC with concentrations at different Re (a) Al$_2$O$_3$-H$_2$O and (b) SiO$_2$-H$_2$O

4.2. Flow analysis

Figure 4 demonstrate the pressure drops change as a function of volume concentration. The advancement of nanofluid thermophysical properties mainly viscosity, compared to water, causes an upsurge in the pressure drop from about 7% to near 13 %, correspondingly, for the concentrations of 0.5 % and 3 %. The pressure drops escalations perhaps because the velocity decreases when the concentration and viscosity increase as well as enlargement and shrinkage of the flow route, which interrupt the hydrodynamic boundary layer and augments the heat transfer. The pressure drop is nearly the same for both the nanofluids since the friction factor increment is not significant to cause significant variation.

Wall shear stress surges with increment in the fluid velocity. It assists in boundary layer disturbance and improves the heat transfer mechanism, as shown in Figure 5; though, wall shear stress decreases with the rise in volume fraction due to increment of nanofluids viscosity.

4.3. Effect of Temperature on the minichannel walls

Figure 6 illustrates the effect of nanofluids on the wall temperature. The increase in volume fraction and Re reduces the wall temperature. Though there is a difference in effective thermal conductivity of the nanofluids, water as the primary phase dominates the flow, so not much difference noticed between the Al$_2$O$_3$/Water and CuO/Water. Hence at a concentration of 3% and Re 10000, the wall temperature raises by 8%, while the outlet temperature increases by merely 1%, compared with 12% observed for 0.5% concentration at Re 5000, thus, indicates the convective heat transfer efficiency of the DCMCHS at higher concentration and fluid velocity.

Figure 7 and Figure 8 highlighted the Temperature contours within the channel and at the channel outlet, respectively. The variation of temperature is visible from the walls towards the channel core. The thermal boundary layer has been disturbed which causes the temperature on the wall to declines.
indicated that the nanofluid flow affects the heat transfer and the enhancement is significant.

Figure 4 Pressure drop variation with concentration for (a) Al₂O₃-H₂O and (b) SiO₂-H₂O

Figure 5 Wall shear stress variation with concentration for Al₂O₃-H₂O (a) 3vol% and (b) 0.5vol%

Figure 6. Effect of wall temperature on concentration for (a) Al₂O₃-H₂O and (b) SiO₂-H₂O
Figure 7. Temperature contours at the channel for \(\text{Al}_2\text{O}_3\)-H\(_2\)O at Re 10,000 and concentrations of (a) 3% and (b) 0.5%

Figure 8. Temperature contours at the outlet for \(\text{Al}_2\text{O}_3\)-H\(_2\)O at Re 10,000 and concentrations of (a) 3% and (b) 0.5%

5. Conclusions
Two-phase analysis of the nanofluids thermal and hydraulic performance via a non-parallel wall minichannel was conducted. Reynolds number significantly influenced the heat transfer advancement for both the base fluid and the nanofluids. The fluid flow slowing and speeding up due to the channel corrugation cause high turbulence at the channel centre thereby enhances flow mixing hence removes the heat from the walls due to convection and transport them through the channel core to the outlet chamber at a reduced temperature. The two-phase model was able to predict the mixture effect between the oxide’s nanoparticles and the base fluid, though the base fluid influence is much significant, hence the variation of heat transfer and flow between the nanofluids may not be much substantial.

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