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# Numerical investigation of two-phase laminar pulsating nanofluid flow in helical microchannel filled with a porous medium

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ABSTRACT

Numerical results of the thermal and hydraulic characteristics of pulsating laminar flow in a three-dimensional helical microchannel heat sink (HMCHS) filled with a porous medium using $Al_2O_3$–water based nanofluid are presented in this paper. The two-phase mixture model with modified effective thermal conductivity and viscosity equations were used to solve the laminar flow governing equations. The detailed results for thermal and flow fields are reported for the effects of amplitude (1–3) and frequency (5–20 rad/s). The presence of porous media led to better thermal enhancement and significant improvement in heat transfer performance was observed for sinusoidal velocity inlet conditions as compared to steady flow conditions. In addition, the proposed model showed better accuracy in predicting nanofluid flow.



## 1. Introduction

Research interest in the field of electronic thermal management has intensified in the past decade in order to address the ever increasing heat loads. Microchannel heat sink (MCHS) is a preferred choice of thermal management and also widely used since the first introduction by Tuckerman and Pease [1,2] in 1981 and followed by numerous researches [3–10] which have been carried out to further improve its performance. The secondary flow in curved tubes is of particular interest to researches as it leads to significant heat transfer enhancement. The secondary flow is induced by the centrifugal forces which act on the flowing fluid. This secondary flow is perpendicular to the main axial flow. The fluid is directed from the inner wall of the tube to the outer wall of the tube across the center and it is forced back to the inner wall again. The Dean number ($De = Re\sqrt{\frac{r}{HR}}$) which depicts the magnitude of the secondary flow for curved tubes was first introduced by Dean, a British applied mathematician and fluid dynamist [11–12].

Alam and Kim [13] numerically studied the mixing in a curved microchannel with grooves on side-walls. They observed that grooved microchannels lead to better mixing performance compared to smooth microchannels and pressure drop was lesser for grooved microchannels than smooth channels. Chu et al. [14] examined the flow characteristics in a curved rectangular microchannel with a different aspect ratio and curvature ratio. The pressure drop increased on decreasing channel width and the effect intensified at larger flow rates. Curved geometries had higher pressure drop due to the secondary flow. In another separate study, Chu et al. [15] investigated the characteristics of water through curved rectangular microchannels for various aspect ratios and curvature. The experimental results showed that classical Navier–Stokes equations were still applicable for the incompressible laminar flow for curved microchannels.

Analysis of heat and fluid flow using a porous medium has been studied extensively in the past [16–23]. Singh et al. [24] analyzed the sintered porous heat sink used for cooling high-powered compact microprocessors. They found that the system was able to dissipate 2.9 $MW/m^2$ of heat flux with a pressure drop of 34 kPa. Hooman [25] analytically studied cases of fully developed forced convection in a rectangular channel with and without a (Brinkman) porous medium. The Nusselt number increases on increasing porous medium shape parameter. Wan et al. [26] developed a novel micro heat sink thermal management system for high power LEDs. The numerical results showed that the heated surface temperature of the heat sink was low at high heat flux. The average heat transfer coefficient increased on increasing velocity. Hung et al. [27] studied the heat transfer analysis on three dimensional porous microchannel heat sinks of various configurations. They concluded that the presence of a porous medium lead to an additional pressure drop with the sandwich distribution design having better overall thermal performance with the least amount of pressure drop.

Zhao and Lu [28] carried out a two-dimensional analytical and numerical study to analyze the forced convective heat transfer characteristic in a rectangular MCHS. They observed an increase of the overall Nusselt number due to a reduction in the effective thermal conductivity ratio. Moraveji et al. [29] performed a numerical study on the cooling performance and pressure drop in a mini-channel heat sink using nanofluids. They found that the heat transfer coefficient increased on increasing nanoparticle concentration and Reynolds number. Fani et al.

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**Nomenclature**

| | |
|---|---|
| $A$ | dimensionless amplitude |
| $C_p$ | heat capacity (J/kg K) |
| D | channel depth (m) |
| $D_h$ | hydraulic diameter ($2W_{ch}D$ / $W_{ch}$ + D) (m) |
| $De$ | Dean number |
| HR | helix radius |
| $h$ | heat transfer coefficient (W/m$^2$ K) |
| $K_p$ | permeability (m$^2$) |
| λ | thermal conductivity (W/m K) |
| $L$ | length of channel (m) |
| $Nu$ | Nusselt number ($Nu = hD_h$ /k) |
| $P$ | pressure (Pa) |
| $q$ | heat transfer rate (W) |
| $r$ | inner tube radius (m) |
| $R_c$ | curvature ratio |
| $Re$ | Reynolds number ($Re = \rho VD_h / \mu$) |
| $T$ | temperature (K) |
| $t$ | time (s) |
| $V$ | velocity (m/s) |
| $W_{ch}$ | channel width |
| *Greek symbols* | |
| $\alpha$ | channel aspect ratio (D / $W_{ch}$) |
| $\kappa$ | Boltzmann constant |
| ε | porosity (%) |
| $\rho$ | density (kg/m$^3$) |
| $\mu$ | viscosity (kg m/s) |
| $Ø$ | volume fraction of nanoparticle |
| $f$ | frequency (rad/s) |
| *Subscript* | |
| ch | channel |
| f | fluid |

[30] examined the effect of nanoparticle size on thermal performance of nanofluid through a trapezoidal MCHS. The pressure drop increased on increasing concentration. Heat transfer decreased on increasing particle size. Emran and Islam [31] numerically studied the flow dynamics and heat transfer characteristics in a MCHS. They observed that the highest temperature was at the bottom of the heat sink immediately below the channel outlet and the lowest at the channel inlet.

Lotfi et al. [32] studied the forced convective heat transfer of nanofluid flow through a horizontal circular tube using the two-phase mixture model and two-phase Eulerian model. They concluded that the mixture model provided better results as compared to all the other models. Kalteh et al. [33] carried out both a numerical and an experimental study on the laminar convective heat transfer of $Al_2O_3$–$H_2O$ nanofluid flow through a wide rectangular MCHS. The two-phase Eulerian–Eulerian model was used. They observed that the two-phase numerical results were in good agreement with experimental results as compared to single-phase results. Targui and Kahalerras [34] observed heat transfer enhancement due to the addition of an oscillating component to the mean flow structure. The peak performance was obtained for the case of pulsating hot fluid. Nandi and Chattopadhyay [35] numerically studied the heat transfer characteristics of water through a two-dimensional wavy microchannel subject to pulsating inlet fluid condition. They deduced that the pulsating inlet fluid condition led to improved heat transfer performance with pressure drop within acceptable limits. The research was further continued by Akdag et al. [36] using nanofluid. They observed an increase in heat transfer performance on increasing nanoparticle volume fraction and amplitude of pulsation.

To the authors' best knowledge, no analyses were made in the past to study the combined effect of secondary flow, porous medium, nanofluid and pulsating flow on the heat transfer and fluid flow characteristics; hence, this has motivated the present study. The present study examines the three-dimensional laminar flow in HMCHS filled with a porous medium and saturated with $Al_2O_3$–water based nanofluid. The results for the effects of amplitude (1–3) and frequency (5–20 rad/s) are reported.

## 2. Mathematical model

### 2.1. Governing equations

The helical microchannel heat sink considered in this study is shown schematically in Fig. 1. Heat is transferred from the outer wall of the HMCHS to the nanofluid which flows through the channel. Several assumptions are made on the operating conditions of the HMCHS as follows: (i) the MCHS operates under steady-state conditions; (ii) the properties of the HMCHS material are temperature independent; (iii) the external heat transfer effects are ignored; (iv) the outer walls of the MCHS are adiabatic.

The two-phase mixture model governing equations for mass, momentum and energy are [37]:

Continuity equation:

$$\frac{\partial}{\partial t}(\rho_m) + \nabla \cdot \left(\rho_m \vec{v}_m\right) = 0 \tag{1}$$

where

$$\rho_m = \sum_{k=1}^{n} Ø_k \rho_k. \tag{2}$$

Momentum equation:

$$\frac{\partial}{\partial t}\left(\rho_m \vec{v}_m\right) + \nabla \cdot \left(\rho_m \vec{v}_m \vec{v}_m\right) = \nabla \rho_m + \nabla \cdot \left[\mu_m\left(\nabla \vec{v}_m + \vec{v}_m^{\,T}\right)\right] + \rho_m \vec{g} + \nabla . \left(\sum_{k=1}^{n} Ø_k \rho_k \vec{v}_{dr,k} \vec{v}_{dr,k}\right)\Big] \tag{3}$$

where $n$ is the number of phases, $Ø$ is the volume fraction and $\mu_m$ is the viscosity of the mixture:

$$\mu_m = \sum_{k=1}^{n} Ø_k \mu_k \tag{4}$$

and $\vec{v}_{dr,k}$ is the drift velocity for secondary phase $k$ defined as:

$$\vec{v}_{dr,k} = \vec{v}_k - \vec{v}_m. \tag{5}$$

Energy equation:

$$\frac{\partial}{\partial_t}\sum_{k=1}^{n}(Ø_k \rho_k h_k) + \nabla \cdot \left(\sum_{k=1}^{n} Ø_k \vec{v}_k(\rho_k h_k + p)\right) = \nabla \cdot (\lambda_{eff} \nabla T). \tag{6}$$

The representative viscosity for the nanoparticle is estimated as:

$$\mu_{np} = \frac{\dfrac{\mu_m}{Ø_{bf}\mu_{bf}}}{Ø_{np}} \tag{7}$$



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