

Thermal Performance Analysis of Nanofluids in Microchannel Heat Sinks

Manay E., Sahin B., Yilmaz M., Gelis K.

Abstract—In the present study, the pressure drop and laminar convection heat transfer characteristics of nanofluids in microchannel heat sink with square duct are numerically investigated. The water based nanofluids created with Al_2O_3 and CuO particles in four different volume fractions of 0%, 0.5%, 1%, 1.5% and 2% are used to analyze their effects on heat transfer and the pressure drop. Under the laminar, steady-state flow conditions, the finite volume method is used to solve the governing equations of heat transfer. Mixture Model is considered to simulate the nanofluid flow. For verification of used numerical method, the results obtained from numerical calculations were compared with the results in literature for both pure water and the nanofluids in different volume fractions. The distributions of the particles in base fluid are assumed to be uniform. The results are evaluated in terms of Nusselt number, the pressure drop and heat transfer enhancement. Analysis shows that the nanofluids enhance heat transfer while the Reynolds number and the volume fractions are increasing. The best overall enhancement was obtained at $\phi=2\%$ and $\text{Re}=100$ for CuO-water nanofluid.

Keywords—Microchannel Heat Sink, Nanofluid, Heat transfer enhancement, pressure drop

I. INTRODUCTION

TRADITIONAL transfer fluids used in heat exchangers are mainly water, ethylene-glycol and oil. Traditional heat transfer fluids used in heat exchangers are mainly water, ethylene-glycol and oil. The heat transfer performance of the traditional fluids is low; this causes heat transfer enhancement efficiency to be lower, in addition prevents heat exchanger to have smaller dimensions. In order to enhance the thermal conductivity of the traditional fluid and heat transfer characteristics, the solid particles are suspended into the base fluid used in the heat exchangers. Adding particles into the base fluid enhances the thermal conductivity, because the thermal conductivity of the solid metal particles is higher than the base fluid.

The new heat transfer fluids engineered by dispersing nanoparticles having smaller diameter than 100 nm into the base fluid are called as nanofluid. Using nanofluids in heat exchange systems enhances thermal conductivity. Heat transfer coefficient increases by the increase of the thermal conductivity [1, 2]. Heat transfer coefficient increases not only by the increase of the thermal conductivity but also by the increase of Reynolds number and particle volume fraction [3]. Adding nanoparticles into the base fluid enhances heat transfer for the cases which the particle volume concentration is lower than 2 vol. %. Pressure drop increases remarkably by the increase of the Reynolds number; the particle volume concentration increases the pressure drop slightly [4].

In heat transfer processes, the use of the microchannels having higher heat transfer capacity are inevitable due to the decrease of the hydraulic diameter of the channels, increase of the heat transfer surface area and less fluid amount requirement with respect to the macro channels or heat pipe systems. The cooling systems that use conventional enhanced surfaces have reached to the limits in new generation electronic devices. The cooling and heat exchange systems in which nanofluids are used will lead to the increase in the performance of the electronic devices and their available life limitations. The reason of preferring microchannels is due to the high heat flux, low working fluid amount and small dimensions. In the industrial applications, water or other traditional heat transfer fluids are used in microchannels. Instead of traditional fluids, using nanofluids in microchannels will be an effective system heat exchange system for thermal applications.

Different types of nanofluids have been used in microchannels. In the literature, the well-known nanofluids are the ones which are created by Al_2O_3 , CuO, CNT and diamond nanoparticles. Akbarinia et al. [5] investigated the effect of the use of the Al_2O_3 -water nanofluids in square cross sectioned microchannels on heat transfer enhancement. It was stated that the nanoparticle concentration had no significant effect on Peclet number and Poiseuille number at the constant inlet velocities. A numerical study on the thermal behavior and the thermal performance of the Al_2O_3 -water nanofluid in microchannel was conducted by Chen and Ding [6]. In parallel microchannel heat sinks, Al_2O_3 -water nanofluids increased heat transfer along the channel, and the heat transfer enhancement increased with increasing particle volume concentration and decreased with increasing particle diameter [7]. The surface friction coefficient increased with increasing particle volume concentration [8]. Lee and Mudawar [9] investigated the single phase and two phase heat transfer characteristics of Al_2O_3 and HFE7100 based nanofluids in microchannels. It was observed that the nanofluids increased heat transfer remarkably, and the increase of the nanoparticle concentration increased the pressure drop in laminar and fully developed flow.

Manay, E. Author is with the Bayburt University, Mechanical Engineering Department, 69000, Bayburt/Turkey (phone: +90-458-211-11-53; fax: +90-458-211-11-72; e-mail: emanay@bayburt.edu.tr).

Sahin, B., Author is with the Erzurum Technical University, Mechanical Engineering Department, 25240, Erzurum/Turkey (phone: +90-442-666-25-27; fax: +90-442-666-25-37; e-mail: bayram.sahin@erzurum.edu.tr).

Yilmaz, M., Author is with the İnönü University, Mechanical Engineering Department, 442810, Malatya/Turkey (e-mail: m.yilmaz@inonu.edu.tr).

Gelis, K., Author is with the Atatürk University, Natural Applied Sciences, 25240, Erzurum/Turkey (e-mail: kadirgelis@hotmail.com).

CuO-water nanofluids provide higher heat transfer than the pure water in microchannels, and the use of CuO-water nanofluid enhances the thermal performance of the microchannel.

This is because of the fact that the thermal conductivity increases, and the thermal diffusion of the nanoparticles is affected. By the use of the CuO particles in the base fluid, lower microchannel wall temperatures are achieved [10, 11]. Kalteh et al. [12] studied two phase heat transfer of CuO-water nanofluid in microchannel between isothermally heated two parallel plates. It was specified that the relative velocities and the temperatures between single phase and two phase could be neglected. It was also observed that Nusselt number increased with increasing Reynolds number and particle volume concentration. The effects of Brownian and Reynolds number, particle volume concentration, channel aspect ratio and porosity on heat transfer and temperature distribution were investigated by Ghazvini and Shokouhmand [13] using fin and porous media approaches.

The use of nanofluids in microchannels enhances heat transfer than the pure water used in the same microchannel. Another advantage of the nanofluids is the fact that they reduce the thermal resistance and the temperature difference between the nanofluid and the channel wall. This decrease in both the thermal resistance and the temperature difference is directly proportional with volumetric fraction of the nanofluid [14, 15]. Two different numerical methods were used to predict heat transfer characteristics of the nanofluids in microchannels having equilateral trapezoid cross section by Kleinstreuer and Li [16]. It was reported that a slight increase in the pumping power enhanced the thermal performance of the microchannel remarkably. The comparison among water based Al_2O_3 , SiO_2 , Ag and TiO_2 nanofluids with respect to heat transfer performance in microchannel was made by Mohammed et al. [17]. It was observed that the highest heat transfer coefficient was achieved by SiO_2 -water nanofluid. It was also seen that the pumping power increased and the effectiveness decreased by the increasing Reynolds number due to the high shear stress between microchannel walls and the nanofluid.

The objective of this study is to numerically investigate the thermal performances of water based Al_2O_3 and CuO nanofluids in microchannel heat sink and to compare the results of both nanofluids in terms of heat transfer performance under laminar and steady state flow conditions.

II. NUMERICAL SOLUTION

A. Governing Equations

In this study, the microchannel heat sink having square ducts is presented in Fig. 1. As seen from the Fig. 1, the channel height is $100\ \mu\text{m}$, the width is $100\ \mu\text{m}$ and the length is $20 \times 10^3\ \mu\text{m}$. The numerical computations are performed for steady state conditions. The Reynolds number range used in this numerical study is 100-1,000, which was based on the hydraulic diameter of the channel over the test section (D_h), and the particle volume fractions are 0, 0.5, 1, 1.5 and 2%. The radiative heat transfer effects are neglected. Also, the upper side of the channel is assumed to be adiabatic, and the

constant heat flux is applied to the bottom plate of the microchannel heat sink. In the solutions conducted by the two phase flow approach under the conditions of laminar flow and steady state, Mixture Model Theory is used.

The governing equations (continuity, momentum and energy) are presented below [18].

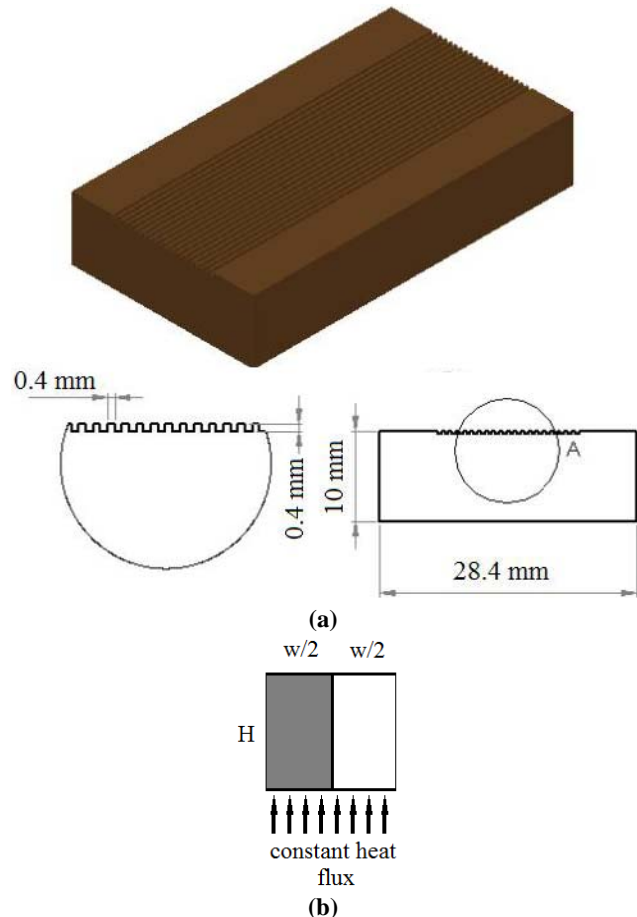


Fig. 1 Computational domain of microchannel heat sink (a) physical domain, (b) numerical domain

Continuity Equation:

$$\nabla \cdot (\rho_m \vec{V}_m) = 0 \quad (1)$$

In Eq. 1, \vec{V}_m represents the mass-averaged velocity and is calculated by Eq. 2,

$$\vec{V}_m = \frac{\sum_{k=1}^n \phi_k \rho_k \vec{V}_k}{\rho_m} \quad (2)$$

where ρ_m is the density of the mixture and is obtained by Eq. 3.

$$\rho_m = \sum_{k=1}^n \phi_k \rho_k \quad (3)$$

Momentum Equation:

$$\begin{aligned} \nabla \cdot (\rho_m \vec{V}_m \vec{V}_m) &= \nabla P + \nabla \cdot [\mu_m (\nabla \vec{V}_m - \nabla \vec{V}_m^T)] \\ &+ \nabla \cdot \left(\sum_{k=1}^n \phi_k \rho_k \vec{V}_{dr,k} \vec{V}_{dr,k} \right) \end{aligned} \quad (4)$$

In Eq. 5, n and μ_m represent the number of the phases ($n=2$ for this study) and the viscosity of the mixture, respectively.

$$\mu_m = \sum_{k=1}^n \phi_k \mu_k \quad (5)$$

In Eq. 6, $\vec{V}_{dr,k}$ is the drift velocity of the secondary phase and is calculated as below.

$$\vec{V}_{dr,k} = \vec{V}_k - \vec{V}_m \quad (6)$$

Energy Equation:

$$\nabla \cdot \left(\sum_{k=1}^n \phi_k \vec{V}_k (\rho_k E_k + P) \right) = \nabla \cdot (k_{eff} \nabla T) \quad (7)$$

The relation between the relative and the drift velocity is given by Eq. 8.

$$\vec{V}_{dr,p} = \vec{V}_{pq} - \vec{V}_m - \sum_{k=1}^n c_k \vec{V}_{qk} \quad (8)$$

The drag coefficient (f_{drag}) is obtained with the below correlation proposed by Schiller and Naumann;

$$f_{drag} = \left\{ 1 + 0.15 \text{Re}^{0.687}, \text{Re} \leq 1000 \right. \quad (9)$$

$$\left. f_{drag} = \left\{ 0.0813 \text{Re}, \text{Re} > 1000 \right. \right.$$

The acceleration of the mixture is calculated by Eq.10.

$$\vec{a} = \vec{g} - (\vec{V}_m \cdot \nabla) \vec{V}_m \quad (10)$$

The volume fraction equation for the secondary phase is given as below;

$$\nabla \cdot (\phi_p \rho_p \vec{V}_m) = -\nabla \cdot (\phi_p \rho_p \vec{V}_{dr,p}) \quad (11)$$

Local convection heat transfer coefficient is obtained by Eq. 12 where T_y and $T_{n,f}$ are surface and nanofluid temperature, respectively.

$$h(x) = q'' / (T_y(x) - T_{n,f}(x)) \quad (12)$$

By integrating Eq. 12 for defined surface area, average convection heat transfer coefficient is obtained.

$$\bar{h} = \frac{1}{A} \int h(x) dA \quad (13)$$

Nusselt number is calculated by Eq. 14 where D_h and $k_{n,f}$ are the hydraulic diameter and the thermal conductivity of the nanofluid, respectively.

$$Nu = \frac{\bar{h} D_h}{k_{n,f}} \quad (14)$$

B. Boundary Conditions

At the channel inlet, the nanofluid enters the channel with a uniform velocity and temperature. Two phases are assumed to be interpenetrating, so it is assumed that the velocities of the base fluid and particles are equal. Velocity inlet and outflow boundary conditions are applied at the channel inlet and the channel outlet, respectively. No-slip boundary condition is applied for both phases on channel walls. The upper side of the channel is assumed to be adiabatic. The constant heat flux (5000 W/cm²) and symmetry boundary condition are applied to the bottom plate and both side of the domain, respectively.

C. Numerical Method and Grid Independence

The computational fluid dynamics (CFD) calculations are made by using commercial code of FLUENT version 6.1.22. In all the numerical calculations, segregated manner is selected as solver type due to its advantage which helps preventing from convergence problems and oscillations in pressure and velocity fields of strong coupling between the velocity and pressure. A two phase flow approach called as Mixture Model Theory is used for simulations. The second order upwind numerical scheme and SIMPLE algorithm are used to discretize the governing equations. The converging criterions are taken as 10e-7 for the energy and 10e-6 for other parameters. In momentum and continuity equations, the thermo-physical properties are assumed to be temperature independent and, the flow is three-dimensional and steady-state.

For grid independence, the Nusselt number is used as criterion. The number of volume mesh is varied from 30.000 to 960.000 for obtaining the most appropriate mesh number. When the number of the volume mesh reaches to 450.000, no remarkable change in the Nusselt number (nearly % 0.2) is observed.

III. RESULTS AND DISCUSSION

In numerical studies, the validation of the data obtained is one of the most important and required procedure. In this study, the numerical method used is validated via the comparison of the results with available literature. For validation, the study of Kalteh et al. [12] was used. Kalteh et al. [12] numerically studied the Cu-water nanofluid for heat transfer characteristics, and used Eulerian-Eulerian two phase flow approach under steady state, laminar flow and constant wall temperature conditions. As seen from Fig. 2, the results of Eulerian-Eulerian multiphase flow model and Mixture model were in good agreement. This means that the used numerical method is applicable for the present study.

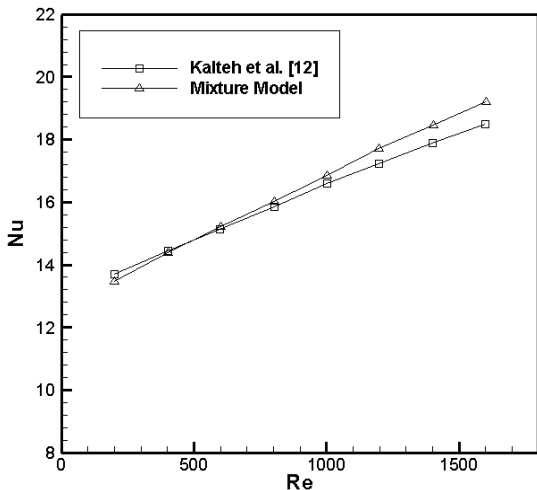
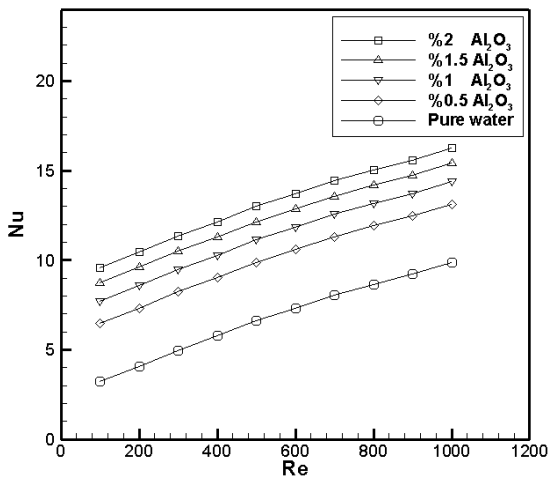
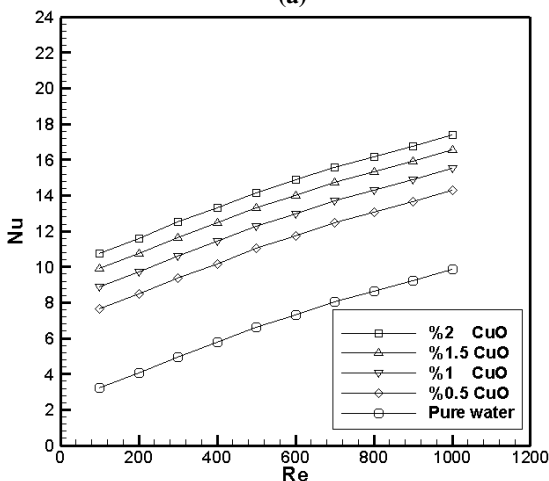


Fig. 2 Comparison of mixture model and Kalteh et al. [12]



(a)

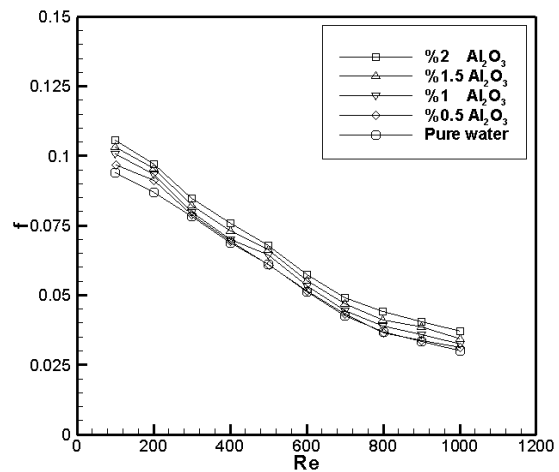


(b)

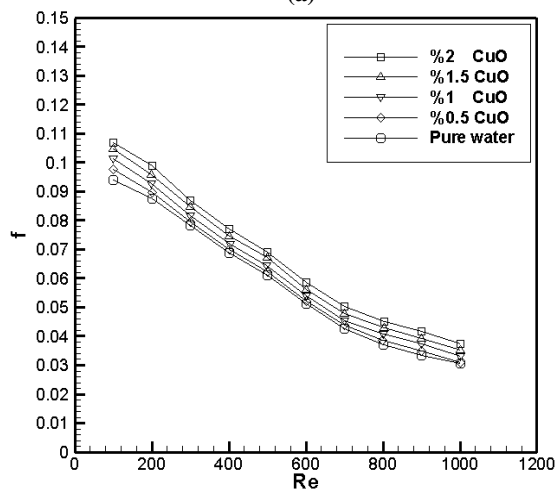
Fig. 3 Nusselt number variations versus Reynolds number; (a) Al₂O₃-water, (b) CuO-water

In Fig. 3a and b, Nusselt number variations versus Reynolds number are presented for Al₂O₃ and CuO nanofluids. The presence of the particles in the base fluid enhances heat transfer remarkably. As seen from the figures, Nusselt number increases with increasing Reynolds number. Also, Nusselt number increases with increasing particle volume concentration. CuO-water nanofluid provides higher heat transfer than Al₂O₃-water nanofluid.

In figure 4a and b, the friction factor variations versus Reynolds number are shown. The results were presented for both nanofluid and pure water. It is observed that adding nano particles into the base fluid does not increase friction factor remarkably. The friction factor decreases with increasing Reynolds number. Also, the friction factor tends to slightly increase with increasing nano particle volume concentration.



(a)



(b)

Fig. 4 Friction factor variations versus Reynolds number; (a) Al₂O₃-water, (b) CuO-water

In heat transfer studies, it is important to evaluate the effects of both the heat transfer and the pressure drop. Therefore, the overall enhancement for both Al₂O₃ and CuO nanofluids is

presented in Figure 5 versus the Reynolds number to determine the overall gain. If the overall heat transfer enhancement is bigger than one, it is said that there is a net energy gain and, this case is useful with respect to both the heat transfer and the pressure drop. As concluded from Figure 5, the heat transfer enhancement is provided for all given cases. The best overall heat transfer enhancement is found at $\phi=2\%$ and $Re=100$ for CuO-water nanofluid. The lowest overall heat transfer enhancement is found at $\phi=0.5\%$ and $Re=1.000$ for Al_2O_3 -water nanofluid.

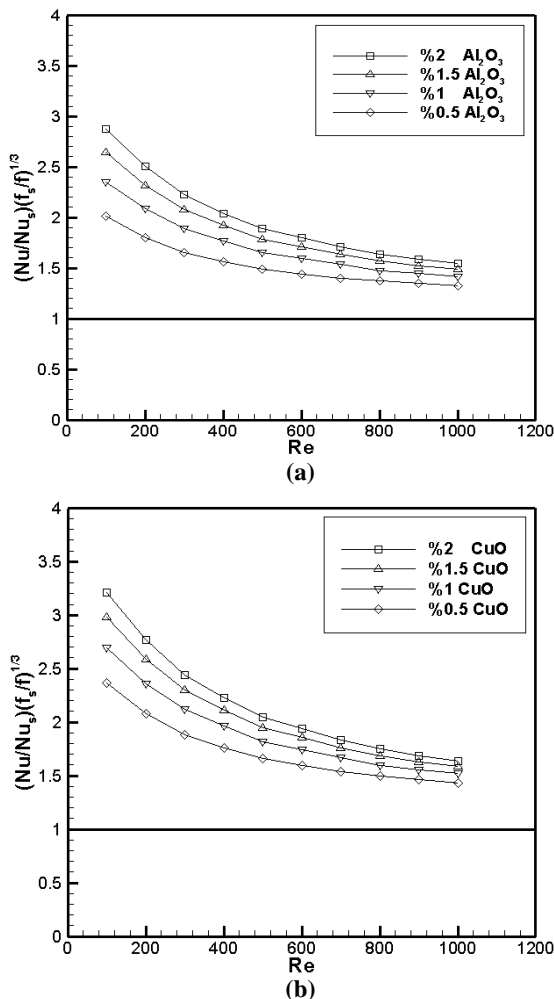


Fig. 5 The overall heat transfer enhancement versus the Reynolds number.; (a) Al_2O_3 -water, (b) CuO-water

IV. CONCLUSION

The objective of this study was to numerically investigate the thermal performances of water based Al_2O_3 and CuO nanofluids in microchannel heat sink and to compare the results of both nanofluids in terms of heat transfer performance under laminar and steady state flow conditions. At the end of the numerical analysis conducted, it was concluded that Mixture model theory could be applied to the nanofluid flows in microchannels for laminar flows. The presence of the nanofluids instead of traditional fluids

provided higher heat transfer than the pure water. The nanofluids enhanced heat transfer remarkably. Heat transfer increased with increasing Reynolds number as well as particle volume concentration. CuO-water nanofluid provided higher heat transfer than that of Al_2O_3 . No significant increase in friction factor was observed by the addition of the nano particles into the pure water. For both nanofluids used, heat transfer enhancement was obtained, and heat transfer enhancement decreased with increasing Reynolds number. The best heat transfer enhancement was achieved at $\phi=2\%$ and $Re=100$ for CuO-water nanofluid. At $Re=100$ and $\phi=2\%$, the heat transfer enhancement was 2.87 and 3.21 for Al_2O_3 and CuO nanofluids, respectively. The lowest heat transfer enhancement $\phi=0.5\%$ and $Re=1.000$ for both nanofluids.

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