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Original Research Paper

A modified two-phase mixture model of nanofluid flow and heat transfer in a 3-D curved microtube

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ABSTRACT

This study numerically investigated the laminar mixed-convection heat transfer of different water-copper nanofluids inside a microtube with curvature angle of 90°, using a finite volume method. The Reynolds number of modeling was 10, nanoparticles volume fractions were chosen from 0.0% to 6.0% and Richardson numbers varied from 0.1 to 10. The findings were depicted for dimensionless axial velocity, coefficient of friction and Nusselt number profiles as well as dimensionless temperature contours. The validity of model was excellent compared to former numerical and experimental studies. The results showed that the heat transfer and hydraulics behavior of nanofluids in curved geometries is to some extent different with other geometries and flat surfaces due to presence of buoyancy and centrifugal forces at the same time. Especially, in the regions near and after 45° curvature angle, the behavior of heat transfer and nanofluid flow is unpredictable. In this region, increasing the nanoparticles volume concentration or transition from forced convection regime to free convection state, cause a decrease in Nusselt number and friction factor. That's while for the entrance region of microtube, the results are completely opposite; increasing the Richardson number or nanoparticle concentration enhances the heat transfer as well as friction factor. Also, the velocity profile variations in the vertical and horizontal diameter of microtube is significant in areas of 60° ($\pi/3$) and the heterogeneity of this profile increases by rising Rayleigh number and volume fraction of solid particles.

1. Introduction

Novel technologies have made electronic devices very highly efficient and more compact, but this necessitates high heat flux removal. Two highly effective ways of removing excess heat are to increase the surface area per unit flow volume and to reduce the hydraulic diameter. Increased surface area will improve coefficient of heat transfer in a microstructure. Therefore, smaller channels, heat sinks and tubes are more eligible for heat removal [1,2].

On the other hand, increasing coolant thermal conductivity can enhance thermal performance of micro heat sinks. Lately, more attention was paid to new types of coolants containing a mixture of a nanoparticle, such as metal oxide or metal and a conventional fluid such as pure water since this combination was shown to considerably improve heat transfer. Recent numerous studies have proved that these coolants, namely nanofluids, have greater thermal conductivity than pure fluids and thus can enhance heat transfer [3–5]. Consequently, nanofluids are good options for high performance heat sinks in electronic devices [6,7].

Researchers are interested in nanofluid flow and heat transfer in microtubes and microchannels, but at the same time they are concerned about the forced convection heat transfer in straight, smooth ones [8–10]. Karimzadehkhouei et al. [11] experimentally studied the laminar forced convection heat transfer of Al_2O_3 and TiO_2/water nanofluid by a horizontal smooth microtube with outer and inner diameters of 717 μm and 502 μm, respectively. They found that heat transfer enhances as mass fraction and *Re* of nanofluid increase in the turbulent and transition regimes, no

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Nomenclature

K_b	Boltzmann constant (1.3807×10^{-23} J K^{-1})
x, y	Cartesian coordinates (m)
Cu	copper
d_f	diameter of the base fluid molecule (m)
d_p	diameter of nanoparticle molecule (m)
Gr	Grashof number $\left(Gr_d = \frac{g\beta_{eff}d^3}{\nu_{eff}^2}(T_H - T_C)\right)$
$\vec{g}$	gravitational acceleration (m s^{-2})
H	heat transfer coefficient (W m^{-2} K^{-1})
N	number of phases
Nu	Nusselt number
Pr	Prandtl number
P	pressure (N m^{-2})
Re	Reynolds number $\left(Re = \frac{\rho_{eff} \times d \times V_m}{\mu_{eff}}\right)$
Ri	Richardson number $\left(Ri = \frac{Gr}{Re^2}\right)$
H	sensible enthalpy (J kg^{-1})
C_p	specific heat capacity (J kg^{-1} K^{-1})
T	temperature (K)
t	time (s)
k	thermal conductivity (W m^{-1} K^{-1})
u, v	velocities components in X and Y directions (m s^{-1})
$\dot{q}$	wall heat flux (W m^{-2})

Greek symbols

ρ	density (kg m^{-3})
μ	dynamic viscosity (Pa s)
υ	kinematics viscosity (m^2 s^{-1})
α	thermal diffusivity
β	thermal expansion coefficient (K^{-1})
ϕ	volume fraction of nanoparticles

Subscripts

f	base fluid
c	cold
eff	effective
h	hot
Z	indices
0	inlet conditions
m	mixture
np	nanoparticles

significant heat transfer augmentation was seen in the laminar regime, particularly for $Re < 1000$.

Malvandi and Ganji [12] did a theoretical study on laminar forced convection heat transfer of alumina-water nanofluid within a vertical microtube along with a uniform magnetic field. Supposing a fully developed flow, the governing equations are reduced to 2-point boundary value equations with end point singularities and solved numerically. The study demonstrated that magnetic field enhances rate of heat transfer particularly for the smaller nanoparticles.

Salman et al. [13] studied the laminar forced convection of water-based nanofluids comprising of Al_2O_3 and SiO_2 in a microtube both experimentally and numerically. The SiO_2 showed the highest friction factor and highest Nusselt number, followed by Al_2O_3 and water. Likewise, it was reported that experimental and numerical findings for nanofluids match well with the previous theories. Study of fluid flow and heat transfer in Newtonian and non-Newtonian fluids are one of the main researchers' interests [14,15]. Rahimi-Gorji et al. [16] studied nanofluids comprising of water and ethylene glycol as base fluid and Cu, Al_2O_3, Ag, TiO_2 nanoparticles. They found out that increasing the concentration of solid nanoparticles causes reduction of temperature differences of cooling fluid and microchannel wall.

The literature review shows that combination of microtubes and nanofluids results in higher heat transfer performance than the traditional fluids used in conventional systems [17–19]. Nevertheless, additional advancement is needed to satisfy the requirements for other categories of the microtubes. A specific case yet to be comprehended is the heat transfer and nanofluid flow within curved microtubes. That's while, flow in curved tubes have many industrial applications in different types of heat exchangers and thermal devices [20,21].

On the other hand, study of behavior and structure of multiphase flows helps to solve lots of problems related to areas of industry, education and biomedical in more accurate ways [22,23]. Therefore, in this study, dilute mixture of water and copper nanoparticles were examined in this work within a microtube with curvature angle of 90°. FLUENT software was used to study laminar mixed convection heat transfer. Nanofluid thermophysical properties were taken from the obtainable formulations in literature. Validation of model was assessed by comparing the existing literature and the simulation results. The focus of this study was on the heat transfer of water-based nanofluids with various volume fractions of solid nanoparticles in microtube, in different Richardson numbers. The findings can be applied to the use of coolants in electronic devices, e.g. Micro-Electro Mechanical System (MEMS), Very-Large-Scale Integrated (VLSI) circuits and high power Light Emitting Diodes (LEDs) [24].

Table 1
Thermophysical properties of the base fluid and Cu nanoparticles [28].

	Copper (Cu)	Water
ρ (kg m^{-3})	8933	997.1
k (W m^{-1} K^{-1})	400	0.613
C_p (J kg^{-1} K^{-1})	385	4179
β (K^{-1})	0.0000167	0.00021
μ (Pa s)	–	0.000891

2. Governing equations for laminar nanofluids

The governing equations for the mixture's continuity, momentum, energy and turbulence are used for flow analysis. Nanoparticles are presumed to be spherical having a diameter of 10 nm that move at a mean velocity similar to the base fluid, whereas the other properties are constant. The followings are the governing equations [25–27]:

Continuity equation:

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

where

$$\vec{V}_m = \frac{\sum_{Z=1}^{n} \varphi_Z \rho_Z \vec{V}_Z}{\rho_m} = V_Z \tag{2}$$

and

$$\rho_m = \sum_{Z=1}^{n} \varphi_Z \rho_Z \tag{3}$$