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Investigation of nanofluid mixed convection in a shallow cavity using a two-phase mixture model

M. Goodarzi [a], M.R. Safaei [b], K. Vafai [c, *], G. Ahmadi [d], M. Dahari [b], S.N. Kazi [b], N. Jomhari [a]

[a] Department of Software Engineering, Faculty of Computer Science & Information Technology, University of Malaya, 50603 Kuala Lumpur, Malaysia
[b] Department of Mechanical Engineering, Faculty of Engineering, University of Malaya, 50603 Kuala Lumpur, Malaysia
[c] Department of Mechanical Engineering, University of California, Riverside, CA 92521, USA
[d] Department of Mechanical and Aeronautical Engineering, Clarkson University, Potsdam, NY 13699-5700, USA

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ABSTRACT

Laminar and turbulent mixed convection heat transfer of water/Cu nanofluids in a rectangular shallow cavity was studied utilizing a two-phase mixture model. The upper movable lid of the cavity was at a lower temperature compared to the bottom wall. Simulations were performed for Grashof numbers of 10^5 (laminar flow) and 10^{10} (turbulent flow) for Richardson numbers from 0.03 to 30, and nanoparticle volume fractions of 0.00–0.04. The two-dimensional governing equations were discretized using a finite volume method. The effects of nanoparticle concentration, shear and buoyancy forces, and turbulence on flow and thermal behavior of nanofluid flow were studied. The model predictions for very low solid volume fraction ($\varphi \approx 0$) were found to be in good agreement with earlier numerical studies for a base fluid. It is shown that for specific Grashof (*Gr*) and Richardson (*Ri*) numbers, increasing the volume fraction of nanoparticles enhances the convective heat transfer coefficient and consequently the Nusselt number (*Nu*) while having a negligible effect on the wall shear stress and the corresponding skin friction factor.

1. Introduction

Advances in nanofluids acting as a new heat transfer medium have introduced new and exciting potentials. The common working fluids used in industries such as water, ethylene glycol and oil typically have lower thermal conductivity compared to metals and metal oxides. By adding high-conductivity solid materials to base fluids it is possible to enhance the mixture's heat transfer performance. The notion of adding micro-sized solid materials to base fluids was proposed decades ago. However, because micro-particles have the tendency to settle in the suspension, it can result in potential adverse effect. Additional problems could be that micro-sized abrasive solid materials erode and corrode pipes and damage pumps or other devices. Nanofluids comprised of nano-sized particles suspended in base fluids could mitigate the issues of erosion, corrosion, fouling and blocking. An increase in thermal conductivity without causing a major pressure drop is a principal advantage of nanofluids. As a result, the performance of numerous heat transfer devices can be augmented, directly leading to the higher capacity of operating units. Nanofluids are also utilized in electronic cooling applications [1].

The practical application of mixed convection heat transfer in various areas such as solar collectors, double-layer glass, building insulation, electronic cooling, food drying, and sterilization among others, has been reported in literature. Mixed convection heat transfer occurs in several ways. One way is to move the walls within a cavity in the presence of hot or cold fluid. Shear stresses are thus produced, forming hydrodynamic and thermal boundary layers in the enclosed fluid, eventually leading to a forced convection condition. Numerous studies have been conducted in this area. Among the notable works are those by Khanafer and Vafai [2], Chung and Vafai [3] and Sharif [4]. Another technique is to introduce hot or cold fluid from one side through the isothermal walls, and have the fluid exit from the other side. A number of researchers have imposed a constant heat flux on the wall as the fluid passes through the channel, and subsequently analyzed the heat transfer effect [5–9].

In recent years studies on nanofluid flow and heat transfer in cavities and enclosures have attracted considerable attention. The majority of studies focus on the laminar flow regime. Muthtamilselvan et al. [10] employed a finite volume method to examine the mixed convection heat transfer of Cu/water nanofluid in a lid-driven rectangular cavity. Two of the cavity's vertical walls were

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* Corresponding author. Tel.: +1 951 827 3125; fax: +1 951 827 2899.
E-mail address: vafai@engr.ucr.edu (K. Vafai).

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Nomenclature

AR	aspect ratio
K_b	Boltzmann constant (1.3807×10^{-23} J K^{-1})
x,y	Cartesian coordinates (m)
H	cavityheight (m)
Cu	copper
d_f	diameter of the base fluid molecule (m)
d_p	diameter of nanoparticle molecule (m)
Y^+	dimensionless distance from the wall
U^+	dimensionless velocity
Y_p	distance from the wall-adjacent cell to the wall (m)
f_{drag}	drag function
Gr	Grashof Number ($g\beta_m \Delta T W^3 \ \upsilon_m^{-2}$)
$\vec{g}$	gravitational acceleration (m s^{-2})
h	heat transfer coefficient (W m^{-2} K^{-1})
$\bar{u}$	mean velocity (m s^{-1})
n	number of phases
Nu	Nusselt Number ($h_m W k_m^{-1}$)
Pr	Prandtl Number ($\upsilon_m \alpha_m^{-1}$)
P	pressure (N m^{-2})
Ra	Rayleigh Number ($Gr\ Pr$)
V_{pf}	relative velocity (slip velocity) (m s^{-1})
Re	Reynolds Number ($V_m W \upsilon_m^{-1}$)
Ri	Richardson Number ($Gr\ Re^{-2}$)
$\vec{a}$	Secondary-phase (Particle) acceleration (m s^{-2})
h_k	sensible enthalpy for phase k (J kg^{-1})
C_p	specific heat capacity (J kg^{-1} K^{-1})
T	Temperature (K)
t	time (s)
Y	the local coordinate normal to the wall
k	thermal conductivity (W m^{-1} K^{-1})
K	turbulent kinetic energy (m^2 s^{-2})
K_p	turbulence kinetic energy at the wall-adjacent cell (m^2 s^{-2})
K_t	turbulent 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})
W	Width of the cavity (m)

Greek symbols

ρ	density (kg m^{-3})
ε	dissipation rate of turbulent kinetic energy (m^2 s^{-3})
μ	dynamic viscosity (Pa S)
υ	kinematics viscosity (m^2 s^{-1})
σ_D	Prandtl dispersion coefficient
α_m	thermal diffusivity ($\mu_m \rho_m^{-1}$)
β	thermal expansion coefficient (K^{-1})
$\upsilon_{t,m}$	turbulent Eddy viscosity (m^2 s^{-1})
σ_T	turbulent thermal diffusivity (m^2 s^{-1})
ϕ	volume fraction of nanoparticles
τ_w	wall shear stress (Pa)

Subscripts

f	base fluid
c	cold wall
Dr	drift
eff	effective
h	wall
Z	indices
0	inlet conditions
lid	lid
M	mean
m	mixture
np	nanoparticles
P	point P
W	point W
F	primary phase
rms	root mean square
p	secondary phase
T	thermal
t	turbulent
W	wall

insulated; the bottom horizontal wall's temperature was maintained at T_c while the temperature of the top moving wall was T_h. Their results show that solid volume fraction and aspect ratio affect heat transfer and fluid flow within the cavity. Also they found that the average Nusselt number varies linearly with respect to solid volume fraction.

Abu-Nada and Chamkha [11] investigated the steady natural convection of CuO–EG–water nanofluid inside a rectangular enclosure using a finite volume method. In their study, the Rayleigh number varied from 10^3 to 10^5, the nanoparticle volume fraction varied from 0% to 6%, and the aspect ratio varied from 0.5 to 2. Flow streamlines and temperature contours were evaluated along with the average and local Nusselt numbers. They found that at low aspect ratios (AR), the average Nusselt number improved with an increase in nanoparticle volume fraction.

Karimipour et al. [12] recently studied the periodic mixed convection of copper/water nanofluid in a rectangular cavity with AR = 3. The examined cavity had two vertical adiabatic walls. The temperature of the upper wall that oscillated at a speed of $U = U_0 \times \sin(\omega t)$ was less than the lower wall's temperature. They demonstrated that due to the oscillating wall, heat transfer improved in the cavity. Khanafer et al. [13] investigated the unsteady mixed convection of air in a sinusoidal lid-sliding cavity utilizing finite element method. Their study indicated that the Grashof and Reynolds numbers had a significant impact on the nature and structure of flow in the cavity.

Oztop and Abu-Nada [14] analyzed the natural convection for different nanofluids in a partially heated square enclosure. They studied a wide range of Rayleigh numbers ($10^3 \le Ra \le 5 \times 10^5$), heater heights, heater locations, aspect ratios and solid volume fractions. As expected they found that an increase in heater size and Rayleigh number led to better heat transfer and fluid flow throughout the cavity. In addition, they found that the nanofluid is a key factor in heat transfer performance. They reported that the copper/water nanofluid had the highest heat transfer rate among the investigated cases.

Ghasemi and Aminossadati [15] used a finite volume method to assess the free convection in an inclined square enclosure with two insulated vertical walls and two horizontal walls at different temperatures. Pure water and CuO–water with $0.01 \le \Phi \le 0.04$ were used in their study. The Rayleigh number varied between 10^3 and 10^7 and the inclination angle ranged between 0 and 90° to examine the impact of these factors on heat transfer and fluid flow in the enclosure. They found that at low Rayleigh numbers where heat transfer occurs mainly by conduction, the flow patterns and temperature contours are similar at 30–90-degree inclination angles. However, for Rayleigh numbers above 10^5, the temperature and flow patterns at a 0-degree inclination angle are different from the