A NEW APPROACH FOR THE ANALYSIS OF THE NANOPARTICLES EFFECTS ON Cu-WATER NANOFLUID MIXED CONVECTION HEAT TRANSFER AND REQUIRED POWER IN A LID-DRIVEN CAVITY

by

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In this paper, a new approach is used for numerical analysis of the sole effects of nanoparticles volume fraction of Cu-water nanofluid on laminar mixed and natural convection heat transfer in a 2-D cavity. Horizontal walls are insulated and fixed, and vertical walls are maintained at constant temperature. Vertical walls are considered for both fixed and moving conditions. Some researchers have studied flow and heat transfer of nanofluid in a lid-driven cavity, keeping fixed both Richardson and Grashof numbers. They found that by the increase of nanoparticles volume fraction, Nusselt number increases, then from this result they concluded the total heat transfer increases from the walls. It is shown that total heat transfer obtained from the Nusselt number by the mentioned approach results from not only the nanoparticles volume fraction increase but also temperature difference and walls velocity increases. Thus, this approach is not appropriate to study the sole effects of nanoparticles volume fractions on the mixed convection heat transfer. Using the new approach, it is shown that in order to have specific heat transfer rate from the walls, base fluid (water) needs less power for moving the wall than Cu-nanofluid. Therefore, the usage of Cu-water nanofluid is not recommended to increase mixed convection heat transfer in a lid-driven cavity. Moreover, using this new approach, it is shown that the increase of nanoparticles volume fraction reduces natural convection heat transfer, which is contradictory to the previous studies. Thus, its usage is not recommended for this case as well. Key words: new approach, nanofluid, convection, power, lid-driven cavity

Introduction

Analysis of flow and heat transfer in a cavity is one of the problems which have been widely studied. In recent years, heat transfer in cavities with moving walls has found great importance in the applications such as electronic equipment's cooling, solar collectors, metal casting, food processing and furnaces. Also, most of the recent studies show significant increase in the heat transfer of a nanofluid flow compared to the base fluid flow. In previous studies, many researchers have concluded that the usage of a nanofluid in a cavity increases heat transfer. Some researchers have studied nanofluid flow and natural convection heat transfer in a cavity with fixed walls [1-10]. Among them, Khanafer *et al.* [1] used different models for thermophysical properties of Cu-water nanofluid and showed that Nusselt number is increased with the increase of nanoparticles volume fraction. Then, from the increase of Nusselt

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number, they concluded that natural convection heat transfer also increases. Oztop and Abu-Nada [2] studied natural convection heat transfer in a cavity using Cu, Al_2O_3 , and TiO₂ nanoparticles. They showed that the increase of nanoparticles volume fraction increases Nusselt number and from that they also concluded the increase of natural convection heat transfer.

Some researchers have also studied nanofluid mixed convection heat transfer in a cavity with moving walls [11-19]. Among them, Tiwari and Kumar Das [15] studied the effects of nanoparticles volume fraction and Richardson number for three different moving conditions of the walls and showed that with the increase of nanoparticles volume fraction average Nusselt number of moving walls increases, and from this result they concluded that heat transfer from the walls increases. Talebi *et al.* [16], studied nanofluid mixed convection heat transfer by using of Cu-nanoparticles in a cavity with moving wall, and found that in constant Reynolds and Rayleigh numbers Nusselt number increases with nanoparticles volume fraction and then they concluded the increase of heat transfer. Muthtamilselvan *et al.* [17] studied Cuwater nanofluid mixed convection heat transfer in a cavity with moving wall. They also found that average Nusselt number increases with the increase of nanoparticles volume fraction, and from this result they concluded that heat transfer increases.

According to the review, researchers found that the increase of nanoparticles volume fraction increases Nusselt number, and from Nusselt number increase they directly concluded the increase of mixed convection heat transfer. In this study, first it is shown that this approach is not appropriate for this conclusion. Then, using the proposed new approach based on dimensional case studies, the sole effects of nanoparticles volume fractions on Cu-water natural and mixed convection heat transfer of Cu-water nanofluid in a square cavity is investigated numerically.

Problem statement

Geometry and boundary conditions of the cavity is chosen as the one studied in [15]. In this cavity, horizontal walls are adiabatic, and vertical walls are moving in opposite directions with velocity, V_p . The cold wall temperature is T_c , hot wall temperature is T_h , aspect ratio *is* AR = 1, and gravitational acceleration g is downward. Boussinesq approximation is used to calculate buoyancy forces. Cavity is filled with Cu-water nanofluid, assumed as Newtonian and incompressible.

Nanofluid physical properties

The following equations are used to calculate nanofluid physical properties. Nanofluid density is calculated from the equation [20]:

$$\rho_{\rm nf} = (1 - \phi)\rho_{\rm f} + \phi\rho_{\rm s} \tag{1}$$

and nanofluid specific heat capacity, c_p , and expansion coefficient, β , are calculated from the relations [20]:

$$(\rho c_p)_{\rm nf} = (1 - \phi)(\rho c_p)_{\rm f} + \phi(\rho c_p)_{\rm s}$$
⁽²⁾

$$(\rho\beta)_{\rm nf} = (1-\phi)(\rho\beta)_{\rm f} + \phi(\rho\beta)_{\rm s} \tag{3}$$

In previous equations, ϕ is the nanoparticle volume fraction, and subscripts *s*, f, and nf denote nanoparticles, base fluid, and nanofluid, respectively. Brinkman model [21] is used to calculate dynamic viscosity of nanofluids and stated:

$$\mu_{\rm nf} = \frac{\mu_{\rm f}}{\left(1 - \phi\right)^{2.5}} \tag{4}$$

where μ is the dynamic viscosity. Also Maxwell model [22] is used to calculate the conductivity of nanofluids and stated:

$$\frac{\kappa_{\rm nf}}{\kappa_{\rm f}} = \frac{\kappa_{\rm s} + 2\kappa_{\rm f} - 2\phi(\kappa_{\rm f} - \kappa_{\rm s})}{\kappa_{\rm s} + 2\kappa_{\rm f} + \phi(\kappa_{\rm f} - \kappa_{\rm s})}$$
(5)

where κ denotes thermal conductivity.

Governing equations

To non-dimensionalize the governing equations and boundary conditions, dimensionless variables are defined:

$$u^* = \frac{u}{V_p}, \ v^* = \frac{v}{V_p}, \ y^* = \frac{y}{L}, \ x^* = \frac{x}{L}, \ T^* = \frac{T - T_h}{T_h - T_c}, \ p^* = \frac{p}{\rho V_p^2}, \ t^* = \frac{tV_p}{L}$$
(6)

where L is the characteristic length, V_p – the characteristic velocity, t_0 – the characteristic time, and $\Delta T = T_h - T_c$ – the characteristic temperature. Superscript (*) denotes dimensionless variables. After substituting the dimensionless variables into the governing equations and boundary conditions, these equations in dimensionless forms are obtained:

continuity equation

$$\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} = 0 \tag{7}$$

momentum equation along x-direction

$$\frac{\partial u^*}{\partial t^*} + \frac{\partial (u^{*2})}{\partial x^*} + \frac{\partial (u^* v^*)}{\partial y^*} = -\frac{\partial p^*}{\partial x^*} + \frac{1}{\operatorname{Re}} \left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}} \right)$$
(8)

- momentum equation along y-direction

$$\frac{\partial v^*}{\partial t^*} + \frac{\partial (u^* v^*)}{\partial x^*} + \frac{\partial (v^{*2})}{\partial y^*} = -\frac{\partial p^*}{\partial y^*} + \frac{1}{\operatorname{Re}} \left(\frac{\partial^2 v^*}{\partial x^{*2}} + \frac{\partial^2 v^*}{\partial y^{*2}} \right) + \frac{\operatorname{Gr}}{\operatorname{Re}^2} T^*$$
(9)

energy equation

$$\frac{\partial T^*}{\partial t^*} + \frac{\partial (u^*T^*)}{\partial x^*} + \frac{\partial (v^*T^*)}{\partial y^*} = \frac{1}{\Pr \operatorname{Re}} \left(\frac{\partial^2 T^*}{\partial x^{*2}} + \frac{\partial^2 T^*}{\partial y^{*2}} \right)$$
(10)

In the energy equation, viscous dissipation term has been neglected. In the previous equations dimensionless numbers Grashof, Reynolds, and Prandtl number are defined:

$$Gr = \frac{g\beta_{nf}L^{3}(T_{h} - T_{c})}{\nu_{nf}^{2}}, \qquad Re = \frac{V_{p}L}{\nu_{nf}}, \qquad Pr = \frac{\nu_{nf}}{\alpha_{nf}}$$
(11)

Boundary conditions

Boundary conditions on the left wall:

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 $u^* = 0$, $v^* = -1$, $T^* = 0$ at $x^* = 0$, $0 \le y^* \le 1$ (12) Boundary conditions on the right wall:

$$u^* = 0, \quad v^* = 1, \quad T^* = 1 \quad \text{at} \quad x^* = 1, \quad 0 \le y^* \le 1$$
 (13)

Boundary conditions on the top and bottom walls:

$$u^* = 0, \quad v^* = 0, \quad \frac{\partial T^*}{\partial y^*} = 0 \quad \text{at} \quad y^* = 0 \quad \text{or} \quad y^* = 1, \quad 0 \le x^* \le 1$$
 (14)

From these dimensionless equations and boundary conditions it is seen that Reynolds, Grashof, and Prandtl number are three effective non-dimensional numbers.

Numerical method

Finite volume approach is used to discretize the governing mass, momentum, and energy equations, and SIMPLER algorithm is used for velocity and pressure coupling. Second-order central difference scheme is used to discretize diffusion terms and upwind scheme is employed to discretize the convective terms.

Four types of girds 20×20 , 40×40 , 60×60 , and 80×80 has been examined. The variation of dimensionless temperature vs. x^* at $y^* = 0.5$ and the dimensionless horizontal velocity at $x^* = 0.5$ vs. y^* for the case Gr = 10^4 , Ri = 1, and $\phi = 0$ are plotted (not shown here for brevity) for these grids. It is seen that a good agreement exists between the results of grids 60×60 and 80×80 . The maximum relative error for the dimensionless temperature and for the dimensionless horizontal velocity between the two grids is 0.054% and 0.049%, respectively. Thus, the grid 60×60 is appropriate for further studies.

Validation

As shown in figs. 1 and 2, the variations of Nusselt number vs. y^* and the variations of $u^* vs. y^*$ for the case $Gr = 10^4$, Ri = 0.1, L/H = 1, $\phi = 0$ (for fig. 2). The $\phi = 8\%$ (for fig. 3) obtained from the present work are compared with those of Tiwari and Kumar Das [15]. It is seen that good agreement exists between these two results, with maximum difference less than 1.5% and 1.8% for Nusselt number and u^* , respectively.

Results

In this section, the proposed new approach which is based on dimensional case studies is used to study the sole effects of the noanoparticles volume fractions and wall velocity on the mixed convection heat transfer and required power in a lid-driven cavity filled with Cu-water nanofluid. For this goal, a cavity of length L = 10 mm and height H = 10 mm is considered.

Knowing the length of the assumed cavity, L = 10 mm and properties of nanofluids, using eqs. (1)-(5), the velocity, V_p , and temperature difference, $\Delta T = T_h - T_c$ are calculated at Gr = 10^4 and Ri = 0.1, 1, and 10, and at $\phi = 0.0, 2$, and 4%, from the following relations:

$$Gr = \frac{g\beta_{\rm nf}L^3(T_{\rm h} - T_{\rm c})}{\nu_{\rm nf}^2} \Rightarrow T_{\rm h} - T_{\rm c} = \frac{Gr\nu_{\rm nf}^2}{g\beta_{\rm nf}L^3}$$
(16)

$$\operatorname{Ri} = \frac{\operatorname{Gr}}{\operatorname{Re}^2} \Longrightarrow \operatorname{Re} = \sqrt{\frac{\operatorname{Gr}}{\operatorname{Ri}}} \Longrightarrow V_p = \frac{\operatorname{Re} \upsilon_{\mathrm{nf}}}{L}$$
(17)

The results are shown in tab. 1.



Figure 1. Comparison of variations of Nusselt number *vs.* y^* of the cold wall obtained from the present work with results in [15] at Gr = 10^4 , Ri = 0.1, AR = 1, and $\phi = 0$



Figure 2. Comparison of variations of $u^* vs. y^*$ at $x^* = 0.5$ obtained from the present work with results in [15] at Gr = 10^4 , Ri = 0.1, AR = 1, and $\phi = 8\%$

Table 1. Temperature difference and wall velocity at $\phi = 0.0, 2, 4\%$, Gr = 10^4 , Ri = (0.1, 1, 10), and L = 10 mm, and H = 10 mm

φ	$Ri = 0.1, Gr = 10^4$		Ri = 1	, $Gr = 10^4$	$Ri = 10, Gr = 10^4$		
	$V_{\rm p} [{\rm ms}^{-1}]$	$\Delta T[\mathbf{K}]$	$V_{\rm p} [{\rm ms}^{-1}]$	$\Delta T [K]$	$V_{\rm p} [{\rm ms}^{-1}]$	$\Delta T [K]$	
0%	0.02832	3.893	0.00896	3.893	0.00283	3.893	
2%	0.02919	4.824	0.00923	4.824	0.00292	4.824	
4%	0.03197	6.638	0.01011	6.638	0.00319	6.638	

It is observed that at constant Grashof and Richardson numbers, with the increase of nanoparticles volume fraction, temperature difference, and wall velocity increase. Because, by using eqs. (1)-(4), the ratio of v_{nf}^2/β_{nf} increases with the increase of nanoparticles volume fraction, thus leading to the increase of temperature difference according to the eq. (16) at fixed Grashof number. Also, at fixed Grashof and Richardson numbers, Reynolds number remains fixed. As nanoparticles volume fraction increases, v_{nf} increases according to eqs. (1) and (4); thus, from eq. (17) it leads to the increase of lid velocity, v_p , at fixed Reynolds number.



Figure 3. <u>Variations of average Nusselt</u> number, Nu, of the cold wall vs. ϕ at Gr = 10⁴ and at three Ri = 0.1, 1, and 10 [15]

In fig. 3, the variations of average Nusselt number, \overline{Nu} , of the cold wall vs. ϕ at $Gr = 10^4$ and at three Richardson numbers: Ri = 0.1, 1, and 10 [15] are shown.

Because for each case study, $\Delta T = T_h - T_c$ is constant, the following relations can be obtained to calculate total wall heat transfer, Q, as a function of $\overline{\text{Nu}}$, κ_f and $\Delta T = T_h - T_c$:

$$\overline{\mathrm{Nu}} = \frac{\overline{h_{\mathrm{nf}}}H}{\kappa_{\mathrm{f}}}, \qquad \overline{h_{\mathrm{nf}}} = \frac{\overline{Q}}{T_{\mathrm{h}} - T_{\mathrm{c}}}, \qquad \overline{\mathrm{Nu}} = \frac{\overline{Q}H}{\kappa_{\mathrm{f}}(T_{\mathrm{h}} - T_{\mathrm{c}})}$$
(18)

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Thus, total heat transfer from vertical walls is calculated:

$$Q = \operatorname{Nu} \kappa_{\rm f} (T_{\rm h} - T_{\rm c}) \tag{19}$$

For each case of tab. 1, $\overline{\text{Nu}}$ from fig. 3 and $\Delta T = T_{\text{h}} - T_{\text{c}}$ from this table are substituted into eq. (19) to calculate total vertical wall heat transfer. The results are presented in tab. 2 for each case.

Table 2. Value of *Q* for $\phi = 0.0, 2$, and 4% at Gr = 10⁴, Ri = (0.1, 1, and 10), L = 10 mm, and H = 10 mm

φ	$Gr = 10^4$, $Ri = 0.1$			$Gr = 10^4$, $Ri = 1$			$Gr = 10^4$, $Ri = 10$		
	<i>Q</i> [W]	$V_{\rm p} [{\rm ms}^{-1}]$	$\Delta T [K]$	<i>Q</i> [W]	$V_{\rm p} [{\rm ms}^{-1}]$	$\Delta T [K]$	<i>Q</i> [W]	$V_{\rm p} [{\rm ms}^{-1}]$	$\Delta T [K]$
0%	75.372	0.02832	3.893	41.786	0.00896	3.893	23.186	0.00896	3.893
2%	97.283	0.02919	4.824	53.915	0.00923	4.824	29.916	0.00923	4.824
4%	140.801	0.03197	6.638	79.727	0.01011	6.638	44.238	0.01011	6.638

It is observed that at constant Richardson and Grashof numbers, with increase of nanoparticles volume fraction, total heat flux from the wall increases significantly. Maximum heat transfer increases for, Ri = 0.1, 1, and 10 are 86.7, 87.58, and 87.59%, respectively. Moreover, in these cases temperature difference, ΔT , and velocity of the walls, V_p , are increased with the increase of nanoparticles volume fraction. This means that the increase of total wall heat transfer results from not only the increase of nanoparticles volume fractions but also the increase of temperature difference and wall velocity, which in turn increase heat transfer. Therefore, keeping fixed Grashof and Richardson numbers is not an appropriate approach to study the sole effects of nanoparticles volume fractions on the total heat transfer.

Using the proposed new approach, the sole effects of nanoparticles volume fraction on total mixed convection heat transfer from the cavity are investigated. To achieve this goal, temperature difference and the wall velocity are kept fixed at three different nanoparticles volume fractions ($\phi = 0.0, 2, \text{ and } 4\%$) for the cavity dimensions previously studied, L = 10mm, H = 10 mm, then, total wall heat fluxes are calculated. The results of Q_s and the corresponding Richardson number at three different wall velocities are presented in tab. 3. Richardson numbers for the cases $V_p = 0.025, 0.0079$, and 0.0029 m/s are about 0.1, 1, and 10, respectively. Maximum total heat transfer increase for these cases is 4.93, 4.42, and 2.98% respectively, which is much different from the previous study [15]. The reason of this difference was discussed previously.

Table 3. Variations of Q and Ri vs. $\phi = 0.0, 2$, and 4% at $\Delta T = 4$ K, $V_p = 0.025, 0.0079$, and 0.0029 m/s, L = 10 mm, and H = 10 mm

φ	$\Delta T = 4 \text{ K}, V$	$v_p = 0.025 \text{ m/s}$	$\Delta T = 4 \text{ K}, V_{\text{p}}$	= 0.0079 m/s	$\Delta T = 4$ K, $V_{\rm p} = 0.0029$ m/s		
	<i>Q</i> [W]	Ri	<i>Q</i> [W]	Ri	<i>Q</i> [W]	Ri	
0%	72.699	0.132	40.283	1.32	23.844	9.798	
2%	74.569	0.113	41.216	1.13	24.252	8.398	
4%	76.475	0.1	42.148	1	24.577	7.325	

Isotherms and streamlines for those cases of tab. 3 are also drawn (not shown here for brevity), and for the case of $V_p = 0.025$ at $\phi = 4\%$ shown in fig. 4. Results show small differ-



Figure 4. Isotherms and streamlines for the case $\phi = 4\%$, $\Delta T = 4$ K, $V_p = 0.025$ m/s, L = 10 mm, and H = 10 mm

ence, while the results of the previous study [15] show large difference. The reason is that in [15], isotherms and streamlines are drawn at fixed Richardson and Grashof numbers, and as mentioned previously, temperature difference and wall velocity increase with nanoparticles volume fraction variations. So the variations in isotherms and streamlines result from not only the increase of nanoparticles volume fractions but also the increase of temperature difference and wall velocity.

As shown in this study, at constant wall temperatures and wall velocities, a slight increase in heat transfer occurs because of the increase of nanoparticles volume fraction. Now required power, *P*, for moving the wall is investigated, which is defined:

$$P = FV_p, \qquad F = \int_0^H \tau_{\rm w} dy \tag{20}$$

where τ_w is the wall shear stress and *F* is the vertical wall shear force. Keeping constant the temperature difference of the cavity, the effects of the wall velocity on the total heat flux of the cavity and required power for moving the wall are shown in figs. 5 and 6 at three different volume fractions of nanoparticles ($\phi = 0.0, 2, \text{ and } 4\%$). It is observed that total wall heat flux and required power increase with the increase of nanoparticles volume fraction.



Figure 5. Variations of total wall heat flux *vs.* wall velocity, at different volume fractions of nanoparticles for the case $\Delta T = 4$ K, L = 10 mm, and H = 10 mm, (0.1 < Ri < 10)

Figure 6. Variations of required power vs. wall velocity, at different volume fractions of nanoparticles for the case $\Delta T = 4$ K, L = 10 mm, and H = 10 mm, (0.1 < Ri < 10)

Combining the results of these two figures, the variation of total wall heat flux vs. required power are shown in fig. 7.

It is observed that for a target total wall heat flux, a nanofluid with larger volume fraction of nanoparticles needs more power, meaning that the usage of the nanofluid to increase heat transfer is not helpful compared to the base fluid. For example, to have Q = 130 W



Figure 7. Variations of total wall heat flux *vs.* required power, at different volume fractions of nanoparticles for the case $\Delta T = 4$ K, L = 10mm, and H = 10 mm

heat flux from the wall, water ($\phi = 0$) needs about 33% less required power than the nanofluid. This rate of heat transfer is obtained at $V_p = 0.08$ m/s for water and at $V_p = 0.09$ m/s for nanofluid.

To study natural convection heat transfer of Cu-water nanofluid, the cavity studied by Khanafer *et al.* [1], is considered here and critically reviewed. This cavity has length *L* and height *H*. Hot and cold vertical walls are kept at constant temperatures T_h and T_c , and top and bottom walls are insulated. Following the approach of Khanafer *et al.* [1], the effects of volume fraction of nanoparticles are critically studied at a fixed Grashof number. For this goal, at $Gr = (10^4, 2 \cdot 10^4, and 4 \cdot 10^4)$ the temperature dif-

ferences between the two vertical walls, ΔT , at $\phi = 0.0$, 2, and 4%, are calculated. After calculating $\overline{\text{Nu}}$ and having known vertical walls temperature difference, total wall heat flux is calculated using eq. (22) and the results are presented in tab. 4 for the mentioned Grashof numbers.

Table 4. Nu and Q at $\phi = 0.0, 2$, and 4% for the case: Gr = $(10^4, 2 \cdot 10^4, 4 \cdot 10^4)$, L = 10 mm, and H = 10 mm

,	$Gr = 10^4$			$Gr = 2 \cdot 10^4$			$Gr = 4 \cdot 10^4$		
φ	Nu	<i>Q</i> [W]	$\Delta T [K]$	Nu	<i>Q</i> [W]	$\Delta T [K]$	Nu	<i>Q</i> [W]	$\Delta T [K]$
0%	4.0741	9.5176	3.893	5.036	23.529	7.786	6.191	57.851	15.572
2%	4.2767	12.378	4.824	5.286	30.602	9.648	6.501	75.271	19.296
4%	4.5556	18.144	6.638	5.630	44.847	13.276	6.921	110.26	26.552

It is observed that at constant Grashof number, with the increase of nanoparticles volume fraction, total heat flux from the wall increases significantly. Maximum heat transfer increases for the cases $Gr = (10^4, 2 \cdot 10^4, and 4 \cdot 10^4)$ are 87.54, 87.53, and 87.53%, respectively. Moreover, in these cases temperature difference, ΔT , increases with the increase of nanoparticles volume fraction. This means that the increase of total wall heat transfer results from not only the increase of nanoparticles volume fractions but also the increase of temperature difference, which in turn increases heat transfer. Therefore, keeping fixed Grashof number is not an appropriate approach to study the sole effects of nanoparticles volume fractions on the total heat transfer.

Using the proposed new approach, the sole effects of nanoparticles volume fraction on the total natural convection heat transfer from the cavity are investigated. To achieve this goal, temperature difference is kept fixed at three different nanoparticles volume fractions ($\phi = 0.0, 2, \text{ and } 4\%$) for the cavity dimensions previously studied, L = 10 mm, H = 10 mm, then, total wall heat fluxes are calculated. The results of Q and the corresponding Grashof number are presented in tab. 5. It is observed that, keeping constant the temperature difference, total heat transfer rate reduces with the increase of volume fraction of nanoparticles. The maximum decrease of heat transfer rate for the cases $\Delta T = (20, 10, \text{ and } 5 \text{ K})$ is 4.49, 4.84, and 5.17%, respectively. This shows that using nanofluid to increase natural convection heat transfer in the cavity is not helpful.

φ	$\Delta T =$	20 K	$\Delta T =$	10 K	$\Delta T = 5 \text{ K}$		
	<i>Q</i> [W]	Gr	<i>Q</i> [W]	Gr	<i>Q</i> [W]	Gr	
0%	79.949	$5.2 \cdot 10^4$	32.565	$2.6 \cdot 10^4$	13.199	$1.3 \cdot 10^4$	
2%	78.845	$4.12 \cdot 10^4$	32.069	$2.06 \cdot 10^4$	12.974	$1.03 \cdot 10^{4}$	
4%	76.358	$3.2 \cdot 10^4$	30.987	$1.6 \cdot 10^4$	12.516	$0.8 \cdot 10^4$	

Table 5. Variations of Q and Gr at $\phi = 0.0, 2$, and 4% for the cases: $\Delta T = (20, 10, \text{ and } 5 \text{ K})$, $T_c = 300 \text{ K}$, $T_h = 320 \text{ K}$, L = 10 mm, and H = 10 mm

Conclusions

A new approach was used for numerical analysis of the sole effects of nanoparticles volume fraction of Cu-water nanofluid on laminar mixed and natural convection heat transfer in a 2-D cavity by employing homogenous mixture model. It was shown that the total heat transfer obtained from the Nusselt number while keeping fixed Richardson and Grashof numbers, as used in previous studies, is not appropriate approach to study the sole effects of volume fractions of nanoparticles on heat transfer rate. Because, in this approach the increase of heat transfer results from not only the increase of nanoparticles volume fraction increase but also the increase of temperature difference and walls velocity. Using the new approach, it was shown that in order to have specific heat transfer rate from the walls, base fluid (water) needs less power than Cu-nanofluid for moving the wall. Therefore, the usage of Cu-water nanofluid is not recommended to increase mixed convection heat transfer in a lid-driven cavity.

Moreover, it was shown that the total heat transfer obtained from the Nusselt number while keeping fixed Grashof numbers, as used in previous researches, is not appropriate approach to study the sole effects of volume fractions of nanoparticles on natural convection heat transfer rate in a cavity. Because, in this approach the increase of heat transfer results from not only the increase of nanoparticles volume fraction but also the increase of temperature difference. Then, using the new approach, it was shown that the increase of nanoparticles volume fraction reduces natural convection heat transfer rate, thus, its usage is not recommended for this case as well.

Nomenclature

- AR aspect ratio
- c_p heat capacity, $[Jkg^{-1}k^{-1}]$
- g acceleration of gravity, [ms⁻²]
- H height, [mm]
- h heat transfer coefficient, [Wm⁻²k⁻¹]
- L length, [mm]
- P power, [W]
- p pressure, [Nm⁻²]

Greek symbols

References

- α heat penetration, [m²s⁻¹]
- β volumetric expansion coefficient, [k⁻¹]
- κ thermal conductivity, [Wm⁻¹k⁻¹]

- μ dynamic viscosity, [Nsm⁻²]
- v kinematic viscosity, [m²s⁻¹]
- ρ density, [kgm⁻³]
- τ shear stress, [Nm⁻²]
- ϕ volume fraction of nanoparticles

Subscripts

- c cold
- f base fluid
- h hot
- nf nanofluid
- s nanoparticles

- [1] Khanafer, K., *et al.*, Buoyancy-Driven Heat Transfer Enhancement in a Two-Dimensional Enclosure Utilizing Nanofluids, *International Journal of Heat and Mass Transfer*, 46 (2003), 19, pp. 3639-3653

- [2] Oztop, H. F., Abu-Nada, E., Numerical Study of Natural Convection in Partially Heated Rectangular Enclosures Filled with Nanofluids, *International Journal of Heat and Fluid Flow*, 29 (2008), 5, pp. 1326-1336
- [3] Jou, R.-Y., Tzeng, S.-C., Numerical Research of Natural Convective Heat Transfer Enhancement Filled with Nanofluids in Rectangular Enclosures, *Int. Commun. Heat Mass*, *33* (2006), 6, pp. 727-736
- [4] Ahmed, G. R., Yovanovich, M. M., Numerical Study of Natural Convection from Discrete Heat Sources in a Vertical Square Enclosure, *Journal of Thermophysics and Heat Transfer*, *6* (1992), 1, pp. 121-127
- [5] Chen, C. J., et al., Numerical Simulation of Natural Convection of Nanofluid in a Square Enclosure: Effects Due to Uncertainties of Viscosity and Thermal Conductivity, International Journal of Heat and Mass Transfer, 51 (2008), 17-18, pp. 4506-4516
- [6] Hwang, S. K., et al., Buoyancy-Driven Heat Transfer of Water-Based Al₂O₃ Nanofluids in a Rectangular Cavity, International Journal of Heat and Mass Transfer, 50 (2007), 19-20, pp. 4003-4010
- [7] Yu, Z.-T., et al., Numerical Study of Transient Buoyancy-Driven Convective Heat Transfer of Water-Based Nanofluids in a Bottom-Heated Isosceles Triangular Enclosure, International Journal of Heat and Mass Transfer, 54 (2011), 1-3, pp. 526-532
- [8] Jou, R.-Y., Tzeng, S.-C., Numerical Research of Nature Convective Heat Transfer Enhancement Filled with Nanofluids in Rectangular Enclosures, *International Communications in Heat and Mass Transfer*, 33 (2006), 6, pp. 727-736
- [9] Cianfrini, M., et al., Natural Convection in Square Enclosures Differentially Heated at Sides using Alumina-Water Nanofluids with Temperature-Dependent Physical Properties, *Thermal Science*, 19 (2012), 2, pp. 591-608
- [10] Cho, C.-C., et al., Enhancement of Natural Convection Heat Transfer in a U-Shaped Cavity Filled with AL2O3-Water Nanofluid, Thermal Science, 16 (2012), 5, pp. 1317-1323
- [11] Akbarinia, A., Behzadmehr, A., Numerical Study of Laminar Mixed Convection of a Nanofluid in Horizontal Curved Tubes, *Appl Therm Eng*, 27 (2007), 8-9, pp. 1327-1337
- [12] Oztop, H. F., Dagtekin, I., Mixed Convection in Two-Sided Lid-Driven Differentially Heated Square Cavity, Int. J. of Heat and Mass Tran., 47 (2004), 8-9, pp. 1761-1769
- [13] Chen, C.-L., et al., Experimental and Numerical Studies on Periodic Convection Flow and Heat Transfer in 2 a Lid-Driven Arc-Shape Cavity, International Communications in Heat and Mass Transfer, 39 (2012), 10, pp. 1563-1571
- [14] Sharif, M. A. R., Laminar Mixed Convection in Shallow Inclined Driven Cavities with Hot Moving Lid on Top and Cooled from Bottom, *Appl. Therm. Eng.*, 27 (2007), 5-6, pp. 1036-1042
- [15] Tiwari, R., Kumar Das, M., Heat Transfer Augmentation in a Two-Sided Lid-Driven Differentially Heated Square Cavity Utilizing Nanofluids, *Int. J. Heat Mass Tran.*, 50 (2007), 9-10, pp. 2002-2018
- [16] Talebi, F., et al., Numerical Study of Mixed Convection Flows in a Square Lid-Driven Cavity Utilizing Nanofluid, Int. Commun. Heat Mass, 37 (2010), 1, pp. 79-90
- [17] Muthtamilselvan, M., et al., Heat Transfer Enhancement of Copper-Water Nanofluids in a Lid-Driven Enclosure, Commun. Nonlinear Sci. Numer. Simulat., 15 (2010), 6, pp. 1501-1510
- [18] Mahmoodi, M., Mixed Convection Inside Nanofluid Filled Rectangular Enclosures with Moving Bottom Wall, *Thermal Science*, 15 (2011), 3, pp. 889-903
- [19] Pourmahmoud, N., et al., Numerical Study of Mixed Convection Heat Transfer in Lid-Driven Cavity Utilizing Nanofluid: Effect of Type and Model of Nanofluid, *Thermal Science*, 19 (2013), 5, pp. 1575-1590
- [20] Xuan, Y., Li, Q., Investigation on Convective Heat Transfer and Flow Features of Nanofluids, ASME Int. J. Heat Mass Tran., 125 (2003), 1, pp. 151-155
- [21] Brinkman, H. C., The Viscosity of Concentrated Suspensions and Solutions, J. Chem. Phys., 20 (1952), 4, pp. 571-581
- [22] Maxwell, J., A Treatise on Electricity and Magnetism, 2nd edition, Oxford University Press, Cambridge, UK, 1904

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