Study of Heat Transfer by Laminar Natural Convection of a Nanofluid in a Solar Water-Heater Enclosure

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Abstract :

We numerically studied the natural convection of a water-copper nanofluid in a solar water heater enclosure in which heating occurs through a solar collector wall at hot constant temperature T_c. This solar collector is connected to a solar thermal storage tank of rectangular form, with the left vertical wall being maintained at constant cold temperature T_f. The unheated parts of the enclosure were considered adiabatic. For the purpose of analyzing the effect of the use of a nanofluid on heat transfer by natural convection, the volume fraction of the particles is varied in the range of 0 to 0.25. The permanent forms of the Navier–Stokes equations in two dimensions and the conservation equations of mass and energy were solved by the finite volume method. The SIMPLE algorithm is used for pressure-velocity coupling. The Rayleigh number Ra was varied in the range 10^3-10^6. The streamlines and the isotherms and the variation of the average Nusselt number at the heated wall are shown for various combinations of the Rayleigh number Ra and different values of the volume fraction of the nanoparticles.

Keywords: Laminar Flow, Solar Water Heater, Nanofluids

1 Introduction

Today, a multitude of industrial applications are based on heat transfer by the natural convection mode such as boilers, heat exchangers, cooling systems for electronic components and solar thermal water-heaters. This is why so many numerical and experimental research works where undergone all over the world which focused on improving heat transfer by natural convection in these energy systems. Recently, the ideas for heat transfer improvement especially touched the physicochemical nature of the convective fluid, because the thermal conductivity of the fluid is relatively low compared to that of the solid. This idea is then to insert within the fluid nanometer sized solid particles to increase the thermal conductivity of the resulting mixture, which is called a nanofluid.

Several recent studies have addressed this issue. Wang et al. [2] measured the thermal conductivity of nanofluids containing nanoparticles Al_2O_3 and CuO scattered in various base fluids. The study showed that the thermal conductivity of these nanofluids increases with the increase of the volume fraction of the nanoparticles in the base fluid, and for a given volume fraction, increasing the thermal conductivity is different depending on the various base fluids. Das et al. [3] studied the influence of temperature on the thermal conductivity for the nanofluids (Al_2O_3+water) and (CuO + water). The results show that the thermal conductivity of nanofluids increases linearly with increasing temperature. Khanafer et al. [4] have numerically studied natural convection inside a rectangular enclosure filled with nanofluids. One of the vertical walls of the enclosure is kept at a low temperature, and the other wall is kept at a high temperature, while the horizontal walls are insulated. The results show that for the range of the Grashof number $10^3 \leq Gr \leq 10^6$, the average Nusselt number increases with the increase of the volume fraction of the nanoparticles. Hakan et al. [5] studied the effect of using different nanofluids on the temperature field in a partially heated cavity. The results show that heat transfer increases with the increase of the value of the Rayleigh number. Mansour et al. [6] have studied numerically heat transfer by natural convection in a T-shaped cavity filled with a nanofluid (Cu + water). The results show that the average Nusselt number increases with the increase of the Rayleigh number and the volume fraction of copper nanoparticles.

Despite the large number of research works which studied the heat transfer by natural convection, there remains ambiguity about heat transfer by laminar natural convection with nanofluids in the field of solar energy, this is why the present study was undertaken. The study is numerical and concerns the improvement of heat transfer by laminar natural convection in a solar thermal water-heater enclosure with the use of water-copper nanofluids. The heating process occurs through the solar collector wall, assumed at a hot constant temperature T_c. The unheated parts of the enclosure were considered adiabatic. The left vertical wall of the enclosure is maintained at a constant cold temperature T_f. The volume fraction of copper nanoparticles is varied from 0 (corresponding to pure fluid) to 0.25 and different Rayleigh numbers ($10^3 \leq Ra \leq 10^6$) were considered. The results are presented in the form of isothermal lines, streamlines, and curves which show the variation of the average Nusselt number for different values of Rayleigh number and volume fractions of nanoparticles.

2 Mathematical Formulation

The objective of this part is to translate the physical model into the form of mathematical equations which will enable us to calculate all the thermal and dynamic characteristics of flow resulting from the heat transfer by laminar natural convection inside the solar water heater enclosure.

2.1 Physical Model and Governing Equations

This study focuses on the laminar natural convection of the nanofluid water-copper in a solar thermal water heater enclosure, which is shown in Figure 1.

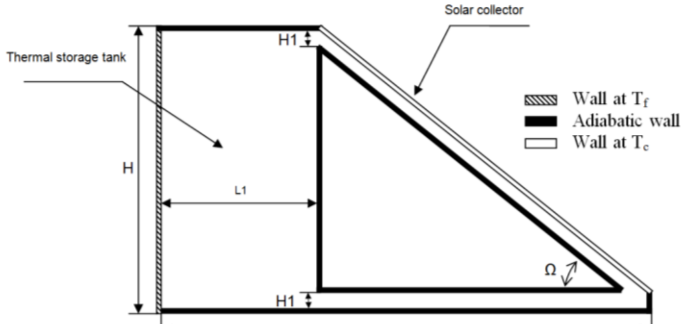

Figure 1. Sketch of the enclosure

We consider that the enclosure is infinitely long in the z direction. The solar collector wall is maintained at a hot constant temperature T_c. The vertical wall at the left of the solar thermal storage tank was considered as cold wall at a constant temperature T_f. The unheated parts of the enclosure were assumed adiabatic.

For the analysis reported in this study, the volume fraction is varied in the interval [0, 0.25]. The study is performed in the range of the Rayleigh number of 10^3 to 10^6. For a simple mathematical formulation of the problem, we made some assumptions: that the fluid is Newtonian and incompressible, the flow is stationary and two-dimensional, and the density in the buoyancy term obeys the Boussinesq approximation while the physical properties of the fluid are constant.

The fluid density varies linearly with temperature and is given by the following formula:

$$\rho = \rho_0[1 - \beta(T - T_0)] \quad (1)$$

We can now write the modeling equations in dimensional form, which govern heat transfer by natural convection in Cartesian coordinates as follows:

Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad (2)$$

Momentum equations

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho_{nf}}\frac{\partial p}{\partial x} + \nu_{nf}\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \qquad (3)$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{\rho_{nf}}\frac{\partial p}{\partial y} + \nu_{nf}\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) - \mathrm{g}\,\beta_{nf}\left(T - T_f\right) \qquad (4)$$

Energy equation

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha_{nf}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \qquad (5)$$

In order to make the above equations dimensionless the following non-dimensional parameters have been used:

$$X = \frac{x}{H}\,, \quad Y = \frac{y}{H}\,, \quad U = \frac{u}{\left(\frac{\alpha_{nf}}{H}\right)}\,, \quad V = \frac{v}{\left(\frac{\alpha_{nf}}{H}\right)}\,, \quad \theta = \frac{T - T_f}{T_c - T_f}\,, \quad P = \frac{p}{\rho_{nf}\left(\frac{\alpha_{nf}}{H}\right)^2} \qquad (6)$$

After making use of these non-dimensional values in the above conservation equations (2)-(5), we get the following:

Continuity equation in non-dimensional form

$$\frac{\partial \mathrm{U}}{\partial X} + \frac{\partial \mathrm{V}}{\partial Y} = 0 \qquad (7)$$

Momentum equations in non-dimensional form

$$\mathrm{U}\frac{\partial \mathrm{U}}{\partial X} + \mathrm{V}\frac{\partial \mathrm{U}}{\partial Y} = -\frac{\partial \mathrm{P}}{\partial X} + \mathrm{Pr}_{nf}\left(\frac{\partial^2 \mathrm{U}}{\partial X^2} + \frac{\partial^2 \mathrm{U}}{\partial Y^2}\right) \quad (8)$$

$$\mathrm{U}\frac{\partial \mathrm{V}}{\partial X} + \mathrm{V}\frac{\partial \mathrm{V}}{\partial Y} = -\frac{\partial \mathrm{P}}{\partial Y} + \mathrm{Pr}_{nf}\left(\frac{\partial^2 \mathrm{V}}{\partial X^2} + \frac{\partial^2 \mathrm{V}}{\partial Y^2}\right) + \mathrm{Ra}_{nf}\,\mathrm{Pr}_{nf}\,\theta \qquad (9)$$

Energy equation in non-dimensional form

$$\mathrm{U}\frac{\partial \theta}{\partial X} + \mathrm{V}\frac{\partial \theta}{\partial Y} = \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) \quad (10)$$

The density and specific heat of the nanofluid are calculated according to [7] using the formula:

$$\rho_{nf} = (1 - \varphi_v)\rho_f + \varphi_v\rho_s \qquad (11)$$

$$\left(\rho C_p\right)_{nf} = (1 - \varphi_v)\left(\rho C_p\right)_f + \varphi_v\left(\rho C_p\right)_s \qquad (12)$$

The thermal expansion coefficient of the nanofluid is obtained from the formula [7]:

$$(\rho\beta)_{nf} = (1-\varphi_v)(\rho\beta)_f + \varphi_v(\rho\beta)_s \quad (13)$$

In this study, the Brinkman model is used for the viscosity of the nanofluid, it is obtained from the following equation [7]:

$$\mu_{nf} = \frac{\mu_f}{(1-\varphi_v)^{2.5}} \quad (14)$$

The effective thermal conductivity of the nanofluid is determined using the Maxwell model [7]. For a suspension of nanoparticles of spherical shapes in a base fluid, the expression is:

$$\frac{k_{nf}}{k_f} = \frac{k_s + 2k_f + 2\varphi_v(k_s - k_f)}{k_s + 2k_f - \varphi_v(k_s - k_f)} \quad (15)$$

The thermal diffusivity of a nanofluid is given by the formula:

$$\alpha_{nf} = \frac{k_{nf}}{(\rho C_p)_{nf}} \quad (16)$$

2.2 Heat Transfer

The heat transfer is characterized by the average Nusselt number which is calculated for the heated wall using the formula:

$$\overline{Nu} = \frac{\bar{h}\sqrt{(H-H_1)^2 + (L-L_1)^2}}{k_f} \quad (17)$$

2.3 Dimensionless Boundary Conditions

The different boundary conditions in dimensionless form are shown in Figure 2.

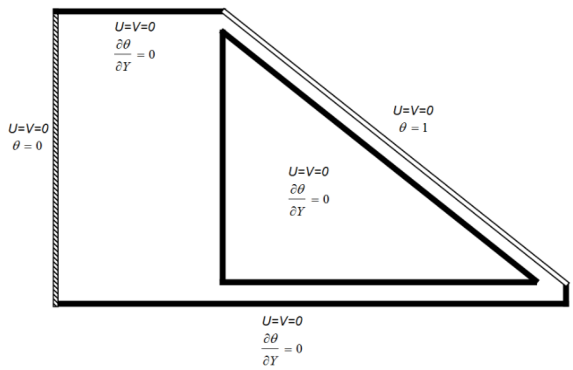

Figure 2. Boundary conditions in dimensionless form

3 Numerical Procedure

3.1 Validation of the Simulation Model

In order to validatethe simulation model developed in this work, we compared the numerical results of Basak and Chamkha [9] with those obtained by the present numerical simulation. The comparison curves are shown in Figure 3.

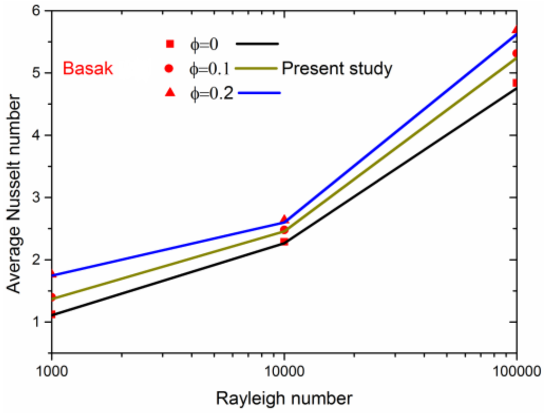

Figure 3. Comparison of the average NUSSELT number with numerical study of Basak [9]

One can notice that the computed average Nusselt number as a function of Rayleigh number is in excellent agreement with that obtained by Basak and Chamkha [9]. This validates the simulation method used in the present computations.

3.2 Grid Independence Study

The equations of the mathematical model are solved numerically using the finite volume method [1]. All numerical simulations of this study are performed using the commercial software Fluent. The influence of the number of nodes on the accuracy of the results in the case of (Ra = 10^3 and $\varphi_v = 0$) is illustrated in Figure 4 which shows the heat transfer through the active wall, i.e. the heated wall, of the enclosure.
From the 22096-nodes grid onwards the average Nusselt number becomes constant.Therefore, all the results of this study were obtained using an unstructured mesh of 22096 nodes with a triangular cell type, as shown in Figure 5.

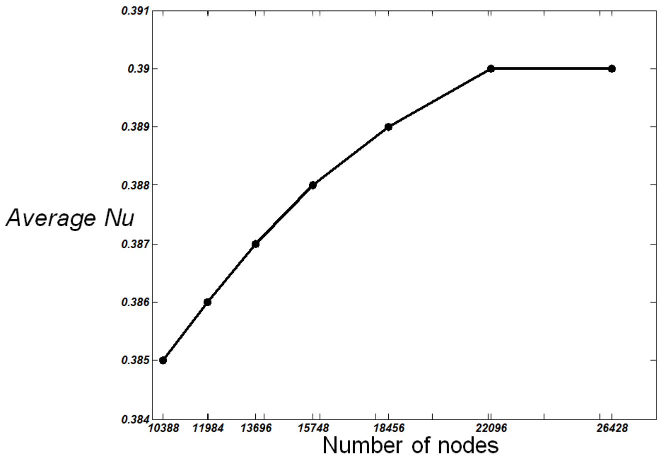

Figure 4. Average NUSSELT number along the heated wall for $\varphi_v = 0$ and Ra = 10^3

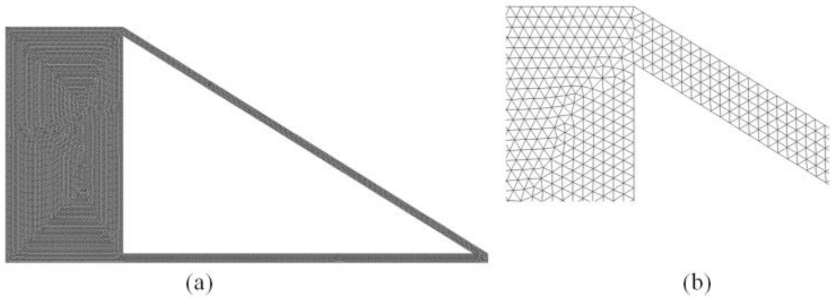

Figure 5. The mesh used in the computations (a) and detail (b)

The calculation algorithm used in Fluent is the one based on pressure. It solves the equations of the mathematical model sequentially. The SIMPLE algorithm is used for pressure-velocity coupling. The discretization of the convective terms in the conservation equations is made with the "QUICK" scheme, while the centered scheme is used to discretize the diffusive terms. The interpolation of the pressure is linear and uses the distances between nodes and those between nodes and the faces of the control volume.
The convergence of the set of algebraic equations obtained after discretization is reached when the sum of normalized residuals at each node of the computational domain becomes less than 10^{-3}.

4 Results and Discussion

In this study, we investigated the effects of using the copper-water nanofluid with volume fraction varied from 0 to 0.25 on fluid flow and heat transfer by laminar natural convection inside the solar water-heater enclosure in the range of the Rayleigh number from 10^3 to 10^6.

4.1 Thermal Fields

This field is shown by the temperature contours in Figure 6 for a Rayleigh number which varies in the range of 10^3 to 10^6, and volume fraction φ_v varied from 0 to 0.25.
Due to molecules of fluid flowing by natural convection in the solar collector, the heat is recovered through the hot wallandthen transported to the upper portion of the solar thermal storage tank.The heat is then discharged through the cold wall of the tank and the cold fluid is transported back to the bottom of the solar collectorand the heating cycle is repeated again and again.
By analyzing the results showing the effect of varying both the volume fraction and the Rayleigh number on the thermal field of nanofluid flow we can draw the following conclusions:
- For Ra fixed and φ_v varied from 0 to 0.25:
It is noted that for each value ofRa, the isotherms are almost identical.
- For φ_v fixed and Ra varied from 10^3 to 10^6:
Figure 6 shows the isotherms obtained for different values of Ra. By comparing them,it may be noted that when theRa increases,the horizontal temperature gradient becomes higher. The isotherms approach each otherin the area near the exit of the solar collector, that is to say that the temperaturegradients become higher nearthe hot fluid inlet in the solar thermal storage tank. At the solar collector inputwe note that the thermal boundary layers become thinnerand the isotherms become stratified for Ra = 10^6.

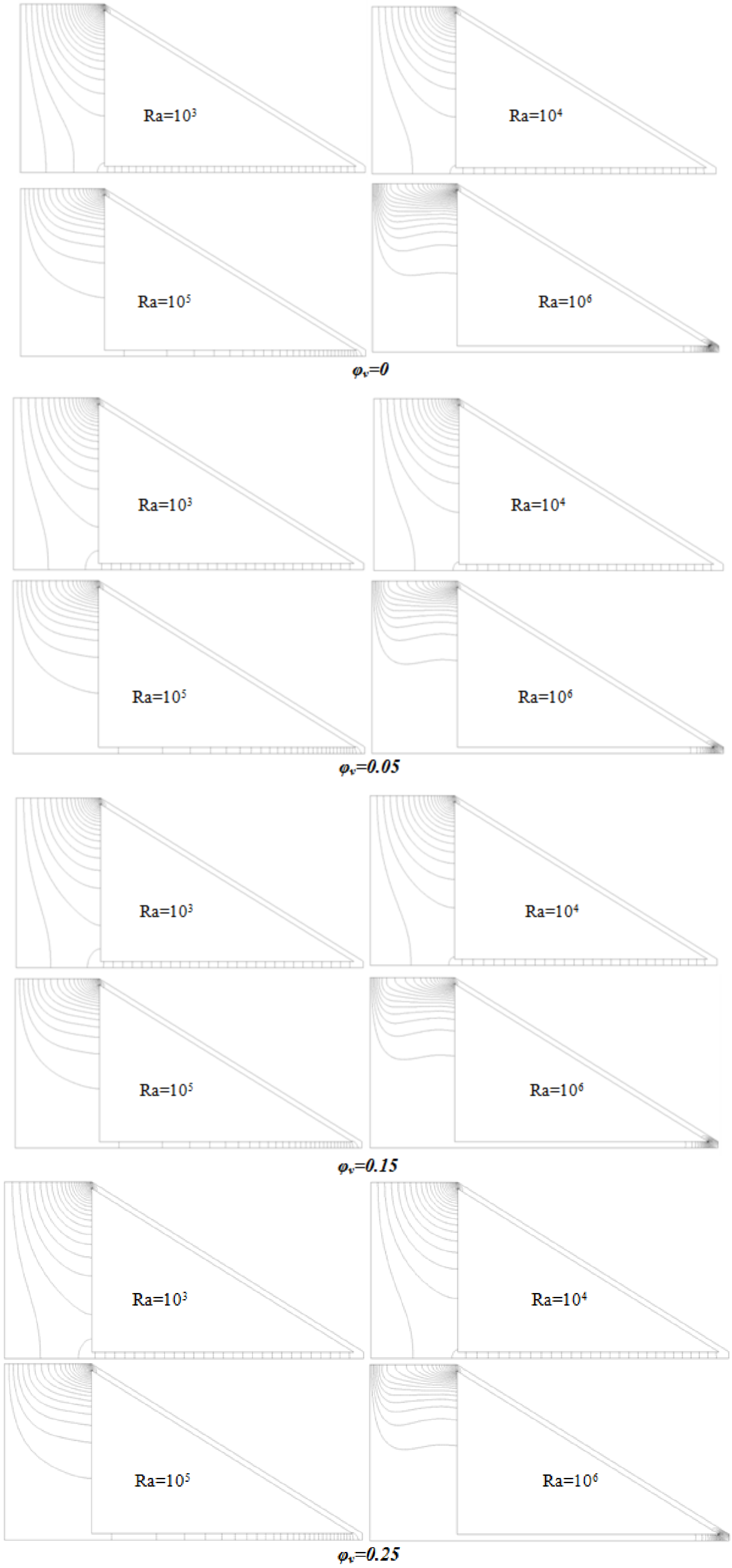

Figure 6. Temperature contours

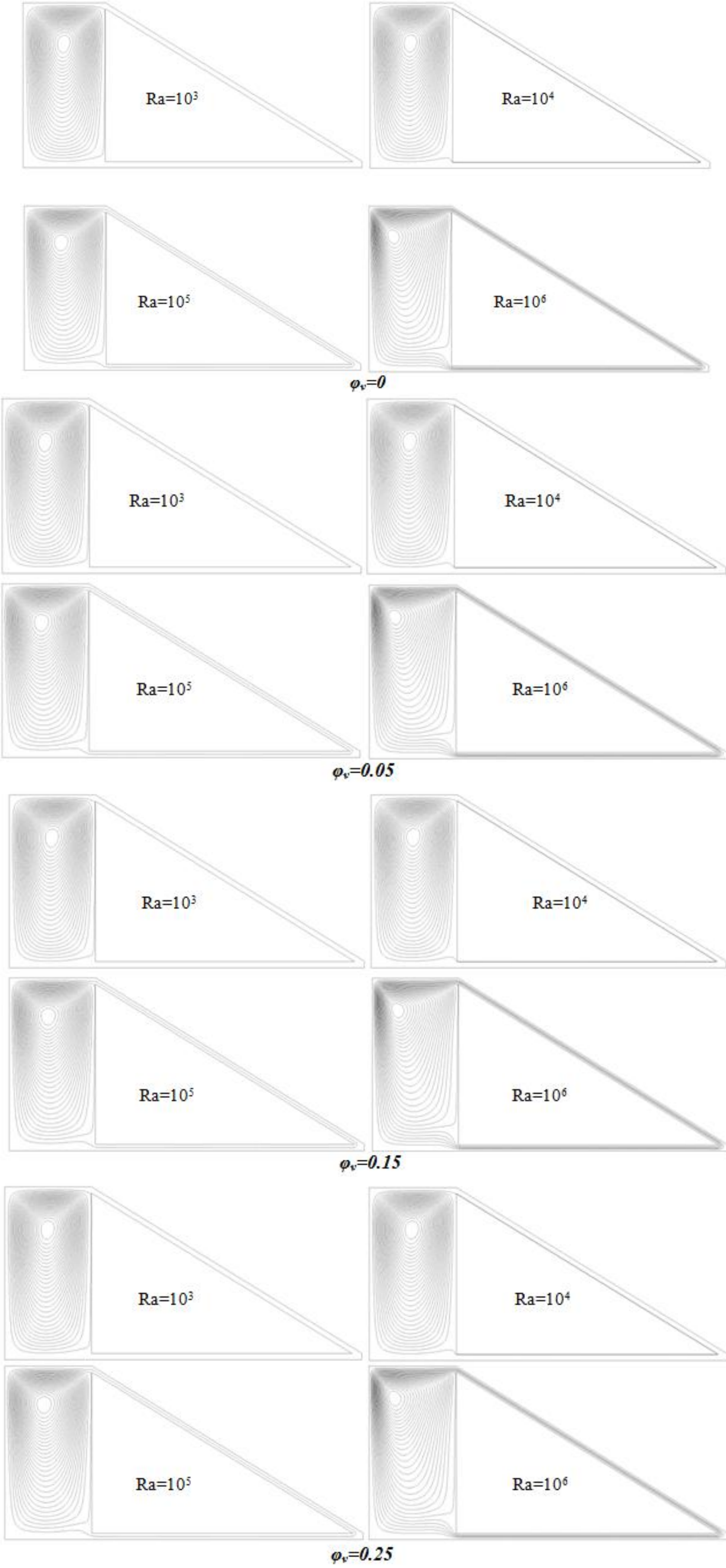

Figure 7. Streamlines

4.2 Dynamic Fields

This field is represented by the contours of the streamlines in Figure 7,for a Rayleigh number in the range between 10^3 and 10^6,and volume fraction φ_v varied from 0 to 0.25.For all Rayleigh numbers and all volume fractions,we note the formation of a fluid recirculation cell. The fluid rotates in the thermal storage tank in the anti- clockwisedirection.
By analyzing the results showing the effect of varying each of volume fractions and Rayleigh number on the dynamic field of nanofluid flow we can draw the following conclusions:
- For Ra fixed and φ_v varied from 0 to 0.25:
It is noted that for each values of the Ra, the streamlines are almost identical.
- For φ_v fixed and Ra varied from 10^3 to 10^6:
For a given value of $\varphi_v$it is noted that with the increaseof Rayleigh number,the intensity of recirculation inside the enclosure increasesand the center of the fluid recirculation cell moves to the upper portionof the cold wall. For Ra =10^6 it is noted that the diameter of the rotating cell decreases.

4.3 Nusselt Number

The evolution of the average Nusselt numberas a function of the Rayleigh numberfor different values of φ_v is shown in Figure 8.For Ra= [10^3 to 10^4]we noted that the average Nusselt number remains constantfor all values of φ_v.Whereas for Ra=[10^4 to 10^6] the average Nusselt number increases with increasing$\varphi_v$and for a given value of φ_v,the average Nusselt number increases when Ra increases.

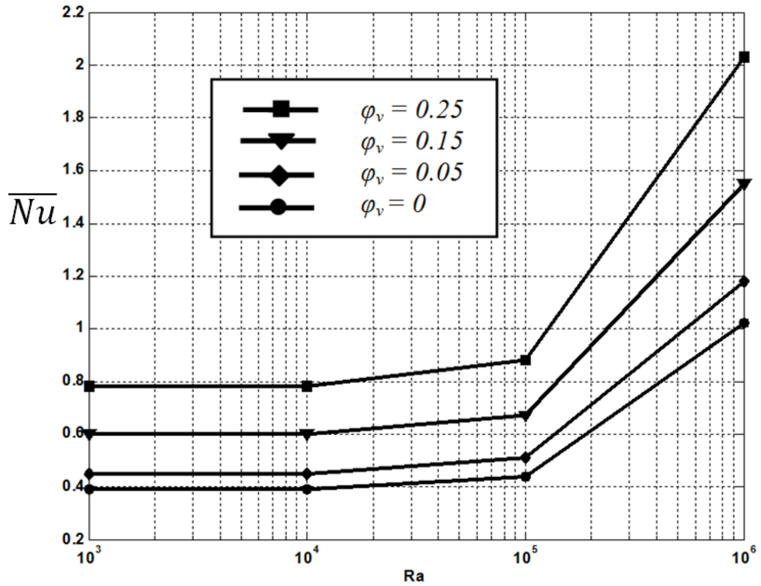

Figure 8. Average NUSSELT number at the active wall

5 Conclusions

In this numerical study we modeled the heat transfer by laminar natural convection in a solar thermal water- heater enclosure in order to study the improvement of heat transfer due to the use of a water-copper nanofluid instead of the pure fluid (water). The results obtained clearly show that the use of nanofluids can influence greatly on the heat transfer enhancement in this geometry. The use of a water-copper nanofluid as coolant with volume fraction equal 0.25 increases the heat transfer by 100%

compared to the use of pure water. Also for all the values of Ra and φ_v we noted the formation of a rotary cell in the thermal storage tank. The intensity of recirculation of this cell becomes greater when increasing the value of Ra. Increasing the velocity of recirculation also implies improved heat transfer. The results show that the volume fraction $\varphi_v = 0.25$ allows the water-copper nanofluid to receive the maximum heat through the hot wall in this solar thermal water heater enclosure.

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Nomenclature

Symbol

g acceleration of gravity, m s^{-2}
L1 width of thermal storage tank, m
H the enclosure height, m
H1 solar collector output section height, m
L enclosure width, m
k thermal conductivity of the fluid, $Wm^{-1}K^{-1}$
p pressure, Pa
P dimensionless pressure
u horizontal velocity, m s^{-1}
v vertical velocity, m s^{-1}
U dimensionless horizontal velocity
V dimensionless vertical velocity
T_f temperature of cold surface, *K*
T_c temperature of hot surface ,*K*
x, y coordinates, m
$\bar{h}$average heat transfer coefficient, $Wm^{-2}K^{-1}$
$\overline{Nu}$ averageNusseltnumber,
Ra Rayleigh number, $g \beta_f H^3 (Tc-Tf)/ \nu_f \alpha_f$
Pr Prandtl number, $Pr= \nu_f/ \alpha_f$

Greek symbols

α thermal diffusivity, $m^2 s^{-1}$
ν kinematic viscosity, $m^2 s^{-1}$
ρ density, kg m^{-3}
φ_v volume fraction of the nanoparticle
θ temperature dimensionless
Ω inclination angle of the solar collector
β thermal expansion coefficient at constant pressure, K^{-1}

Indices / Exhibitors

*f*base fluid
*nf*nanofluid
*s*solid