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Heat Transfer Performance of Homogenous Nanofluids Under Mixed Convection in a Vented Cavity with Linearly Varying Wall Temperature

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Abstract: The present work consists to investigate numerically the mixed convection heat transfer in a multiple vented enclosure, confining alumina-water nanofluid, with inlet and outlet ports due to the suction of leaving flow. The bottom wall is subjected to a linearly varying decreasing heat, whereas the other boundaries are assumed to be thermally insulated. The consequence of varying the Reynolds number, $200 \leq Re \leq 5000$, and nanoparticles concentration, $0 \leq \phi \leq 0.07$, on the dynamical and thermal characteristics of the flow and heat transfer performances are investigated and discussed. The obtained results show that the presence of nanoparticles enhances the heat transfer across the cavity and promotes the reheating of this latter. Morever, a comparative study shows that the decreasing temperature profile has the same thermal performance as that of a uniform temperature profile.

Key words: Mixed convection, nanofluid, decreasing heating, multiple vented cavity, suction.

1. Introduction

Enhancing mixed convection effects in an enclosure is an important issue for engineering applications, such as electronic cooling devices, furnaces, lubrication technologies and drying technologies. Fluids such as water, mineral oils and ethylene glycol are often adopted for heat transfer mechanism. But these fluids have a rather low thermal conductivity and may not give an efficient heat transfer for the growing demand of technological progress. One way to avoid this problem is to add solid nanoparticles, having a high thermal conductivity, into the base fluid. The resulting mixture is called nanofluid and has favorable characteristics of heat transfer.

In some practical applications, since natural or forced convection is not able to provide cooling effectiveness in many thermal processes, the mixed convection cooling mechanism has been recommended in such systems. Therefore, the mixed convection heat transfer in multiple ventilated cavity using nanofluid is a fertile area of research. In recent years, many studies are devoted to the study of nanofluid mixed convection in rectangular geometries. Among these studies we quote those related to the case of cavities with one or two lid- driven walls. In this context, Muthtamilselvan et al. [1] treated mixed convection heat transfer numerically in an enclosure filled with Cu-water nanofluid. The bottom wall is heated uniformly and not uniformly while the top wall is movable at constant velocity. The presented results show that the solid volume fraction plays a significant role on the flow ant thermal fields and the Nusselt number distributions for different flow configurations. Also, it is found that the Richardson number strongly

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affects the fluid flow and heat transfer in the cavity. Another work was performed by Hassan et al. [2]. The authors studied numerically mixed convection in a square cavity filled with a nanofluid and whose the bottom horizontal wall is maintained at a constant temperature while the upper wall is isolated and mobile. The results showed that the thermal and dynamic flow fields are very sensitive to the Richardson number, Ri, and that the presence of nanoparticles in the basic fluid improves the heat transfer. A numerical study of mixed convection in two-sided facing lid-driven cavity partially heated from below was carried out by Moumni et al. [3]. Their results showed that the heat transfer rate increases by increasing Richardson and Reynolds numbers and the addition of nanoparticles to the base fluid. Dahani et al. [4] conducted a numerical investigation of mixed convection flow in a two sided lid driven square cavity filled with Al_2O_3-water nanofluid and heated from below and cooled from above. The obtained results reveal that the nanoparticles addition may also reduce the range of inclination generating an unsteady flow, and change the threshold corresponding to the transition from the monocellular to the multicellular flow structure.

Mixed convection of nanofluids in ventilated cavities was also the object of interest during the last years. Such problems are encountered in many industrial applications such as heat exchangers, lubrication technology, chemical processing equipment, solar energy collectors and cooling of electronic equipment. Several researchers have studied the mixed convection of nanofluids within ventilated cavities subjected to uniform heating. In this perspective, Sourtiji et al. [5] studied numerically mixed convection flows through an Alumina-water nanofluid in a square cavity with imposed temperature. The inlet port is located on the upper part of the left wall, whereas the location of the outlet port was varied along the four walls. It is observed that the average Nusselt number is an increasing function of Reynolds number, Richardson number and the nanoparticle volume fraction. Recently, numerical investigations are carried out on mixed convection in a vented rectangular enclosure filled with Al_2O_3-water nanofluid and cooled by an injected or sucked imposed flow were presented by Bahlaoui et al. [6,7]. It was found that the heat transfer rate increases by increasing the volume fraction of nanoparticles and the suction mode favors the heat transfer in comparison with the injection one. Shirvan et al. [8] investigated numerically the mixed convection in ventilated cavity filled with Cu-water nanofluid. It was illustrated that the inlet port placed in the lower part of the left wall and the outlet port placed in the top part of the right wall is the optimal design for heat transfer. In the same topic, mixed convection of nanofluid in a ventilated square cavity discretely heated from below was solved numerically by Moumni et al. [9]. These authors demonstrated that regardless of the Richardson and Reynolds numbers and solid volume fraction, the highest heat transfer enhancement occurs in the left heat source. Simulation of nanofluid in a ventilated cavity with circular cooling obstacle was investigated by Boulahia et al. [10]. They concluded that by increasing the solid volume fraction, the heat transfer rate is enhanced at different Richardson numbers and outlet port positions. Very recently, Arroub et al. [11,12] analyzed the ventilation mode effect (suction or injection) on the thermal performances inside a rectangular cavity heated from below by different heating modes. They found that the increase of the nanoparticles volume fraction contributes to an enhancement of the heat transfer and the mean temperature within the cavity. Also, it was revealed that the fluid suction mode yields the best heat transfer performance and the improvement in heat transfer is more pronounced for cavities with low aspect ratio. In addition, a better heat transfer through the cavity is obtained by applying a decreasing heating mode.

It is clear that so far, the problem of nanofluid mixed convection within a multiple vented cavity submitted to non-uniform heating conditions have not been subject of any study yet. Therefore, the aim of the current study is to simulate numerically mixed convection heat transfer characteristics of water-based Al_2O_3 nanofluid flowing through a multiple ventilated rectangular cavity heated by a linearly decreasing temperature. The consequence of varying the Reynolds number, $200 \leq Re \leq 5000$, the nanoparticle concentration, $0 \leq \phi \leq 0.07$, on the dynamical and thermal characteristics of the flow and heat transfer performance have been investigated and discussed.

2. Problem Definition and Mathematical Modeling

The geometry considered is depicted in Fig. 1. It consists of a multiple ventilated (three ports) rectangular cavity of height, H′, and length, L′, with an aspect ratio A = 2, and non-uniformly (with a linearly decreasing temperature profile) heated from the bottom wall. The other boundaries are assumed to be perfectly insulated. The cavity is filled with a nanofluid (Al_2O_3-water). The nanofluid enters into the cavity by suction from the two openings located on the lower part of the vertical walls and a third opening located in the middle of the upper horizontal wall provides ventilation. These ports have a constant relative height, B = 1/4. Thus, it is assumed that both the base flow and nanoparticles are in thermal equilibrium and no slip occurs between them. The dispersed particles have a spherical shape and their size is assumed to be uniform. The properties of the nanofluid are assumed to be constant except for the density which is estimated by the Boussinesq approximation. The nanofluid is assumed Newtonian and incompressible. The flow is laminar and two-dimensional. Therefore, using following dimensionless variables:

$$A = L'/H' \;,\; B = h'/H' \;,\; x = x'/H' \;,\; y = y'/H' \;,$$

$$u = u'/u'_o \;,\; v = v'/u'_o \;,\; t = t'u'_o/H' \;,$$

$$T = (T' - T'_C)/(T'_H - T'_C),\; \Psi = \Psi'/u'_o H',\; \Omega = \Omega' H'/u'_o$$

$$Pr = \nu_f/\alpha_f,\; Ra = g\,\beta_f (T'_H - T'_C) H'^3/\alpha_f \nu_f,\; Re = u'_o H'/\nu_f$$

The non-dimensional governing equations describing the system behavior can be written, in terms of Ψ-Ω formulation, as follows:

$$\frac{\partial\Omega}{\partial t} + u\frac{\partial\Omega}{\partial x} + v\frac{\partial\Omega}{\partial y} = \frac{Ra}{Re^2 Pr}\left[\left(\frac{\phi}{\left((1-\phi)\frac{\rho_f}{\rho_{np}} + \phi\right)}\right)\frac{\beta_{np}}{\beta_f} + \frac{1}{\left(\frac{\phi}{(1-\phi)}\frac{\rho_{np}}{\rho_f} + 1\right)}\right]\frac{\partial T}{\partial x} + \frac{1}{Re}\left[\frac{\frac{\mu_{nf}}{\mu_f}}{\left(\phi\frac{\rho_{np}}{\rho_f} + (1-\phi)\right)}\right]\left(\frac{\partial^2\Omega}{\partial x^2} + \frac{\partial^2\Omega}{\partial y^2}\right) \quad (1)$$

$$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \frac{1}{Re\,Pr}\left(\frac{\frac{\lambda_{nf}}{\lambda_f}}{(1-\phi) + \phi\frac{(\rho c_p)_{np}}{(\rho c_p)_f}}\right)\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \quad (2)$$

$$\frac{\partial^2\Psi}{\partial x^2} + \frac{\partial^2\Psi}{\partial y^2} = -\Omega \quad (3)$$

The dimensionless horizontal and vertical velocities are converted to:

$$u = \frac{\partial\Psi}{\partial y} \;;\; v = -\frac{\partial\Psi}{\partial x} \quad \text{and} \quad \Omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \quad (4)$$

The nanofluid effective density, effective thermal conductivity, thermal diffusivity, heat capacitance, thermal expansion coefficient and the effective dynamic viscosity of the nanofluid are, respectively, given by:

$$\rho_{nf} = \phi\rho_s + (1-\phi)\rho_f \quad (5)$$

$$\lambda_{nf} = \lambda_f\left[1 + 4.4\,\mathrm{Re}_{np}^{0.4}\,\mathrm{Pr}^{0.66}\left(\frac{T}{T_{fr}}\right)^{10}\left(\frac{\lambda_{np}}{\lambda_f}\right)^{0.03}\phi^{0.66}\right] \quad (6)$$

$$\alpha_{nf} = \frac{\lambda_{nf}}{(\rho c_p)_{nf}} \quad (7)$$

$$(\rho c_p)_{nf} = \phi(\rho c_p)_s + (1-\phi)(\rho c_p)_f \tag{8}$$

$$(\rho\beta)_{nf} = \phi\rho_s\beta_s + (1-\phi)\rho_f\beta_f \tag{9}$$

$$\mu_{nf} = \frac{\mu_f}{1 - 34.87\left(\frac{d_{np}}{d_f}\right)^{-0.3}\phi^{-1.03}} \tag{10}$$

Where Re_{np} is the nanoparticle Reynolds number, T_{fr} is the freezing point of the base liquid, d_{np} is the diameter of the suspended nanoparticles and d_f is the equivalent diameter of a base liquid molecule.

″f″, ″np″ and ″nf″ indicate fluid, nanoparticles and nanofluid respectively.

2.1 Boundary Conditions

The hydrodynamic and thermal boundary conditions associated to our problem is given explicitly as follows:

$u = v = 0$ on the rigid walls

$\frac{\partial T}{\partial n} = 0$ on the adiabatic walls

$T = (-\frac{2a}{A})x + (1+a)$ on the lower horizontal heated wall

$T = 0$ at the upper inlet port

$u = -1$, $v = 0$, $\Psi = -y$ and $\Omega = 0$ at the left outlet port

$u = 1$, $v = 0$, $\Psi = y$ and $\Omega = 0$ at the right outlet port

$\Psi = -B$ between the left outlet port and the upper inlet port

$\Psi = B$ between the left outlet port and the upper inlet port

Where ″*n*″ indicates the normal direction to the considered adiabatic wall.

- The boundary conditions concerning u, v, Ψ and Ω are unknown at the inlet opening while the temperature T is unknown at the two outlet openings, left and right. Values of these variables are extrapolated at each time step by considering zero second derivatives at these openings.
- $a = 0.5$ for decreasing heating temperature profile.

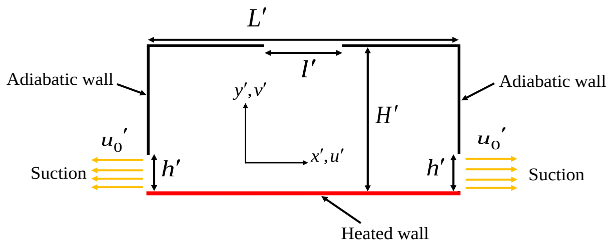


Fig. 1: Geometry and coordinates system.

2.2 Heat Transfer

The estimation of the heat transfer enhancement and efficient cooling is done based on the average Nusselt number, Nu, calculated on the heated bottom wall of the cavity such as:

$$Nu = -\frac{1}{A}\left(\frac{\lambda_{nf}}{\lambda_f}\right)\int_0^A \left.\frac{\partial T}{\partial y}\right|_{y=0} dx \tag{11}$$

3. Numerical Approach

The numerical algorithm used to solve the dimensionless governing equations (1)-(3), is based on the finite difference technique. The second-order upwind approach is used to discretize the advection terms to avoid possible instabilities frequently encountered in mixed convection problem. Also, the second order central difference scheme is adopted to approximate the first and second derivatives of the diffusive terms. Thereafter, the solution of the vorticity and energy equations, (1) and (2), is performed iteratively with the Alternating Directions Implicit (ADI) method. The stream function is related to the vorticity by the Poisson equation. This equation is solved by a Point Successive Over-Relaxation method (PSOR) with an optimum over-relaxation coefficient equal to 1.88 for the grid (201×101) retained in this work.

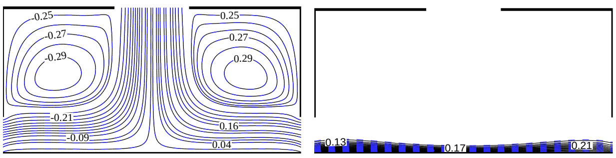
4. Results and Discussion

In this section, the Rayleigh number was kept at a constant value Ra = 10^6 and the Reynolds number Re ranging between 200 and 5000. The values of these parameters involve values of the Richardson number, $\mathrm{Ri} = \mathrm{Ra}/\mathrm{Re}^2\,\mathrm{Pr}$, varying in the range [$6.45\times10^{-3}$, 4.03] which simulates natural convection, mixed convection and forced convection dominating regimes. In the following, effects of nanofluids in the case of the decreasing heating mode on the flow and the thermal structures and on effective cooling are investigated. As illustrated in Fig. 2a, obtained for a low value of Re (Re = 200), the streamlines (on the left) show a flow structure symmetrical to the cavity median. This structure consists of a forced flow descends vertically from the upper opening and leaves the cavity through the lower openings located on the vertical walls. Also, we note the existence of two closed cells of the same size and opposite directions overcoming the open lines of forced convection. These cells are separated by vertical streamlines and their formation is mainly due to the shear and buoyancy force effects. The heat transfer between these cells and the hot wall takes place via horizontal open lines. The effect of the addition of nanoparticles on the dynamic structure is not yet visible since the streamlines with or without the presence of nanoparticles is confused. Thus, is should be noted that the symmetry observed in this dynamic structure comes from the thermal and dynamic boundary conditions symmetrically imposed. The corresponding isotherms (in the right) are condensed in the vicinity of the hot wall in the form of a beam called a thermal boundary layer. The narrow thickness of this layer indicates a good heat exchange between the active wall and the flowing fluid. This is justified by the fact that the descending fluid comes into direct contact with the hot wall before exiting. Consecutively, a great part of the space offered in the cavity is at a uniform cold temperature. A progressive increase of the Reynolds number up to 5000 (Figs. 2b-d), leads to the intensification of forced flow while the dynamic structure remains qualitatively unchanged. The isotherms show that the thickness of the thermal boundary layer decreases with Re because of the increasing effect of the flow rate which accentuates forced convection and disadvantages the rise of the fluid under the natural convection effect. Consequently, forced convection becoming more and more predominant. Finally, it is noted that the presence of nanoparticles (ϕ = 0.07) does not alter the dynamic and thermal fields regardless of the intensity of the forced flow.

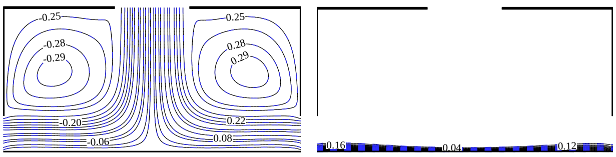
In order to illustrate the thermal performances when the suction mode is applied, variations versus the Reynolds number, Re, of the mean Nusselt number, Nu, evaluated on the hot wall are presented in Fig. 3 for different values of ϕ. This figure shows, in general, a monotous increase of Nu with Re for different values of ϕ. It should be noted that this increase becomes more important from Re = 500. This trend is justified by the intensification of flow thanks to the inertial force. This leads, therefore, to intense heat exchange, especially by dominant forced convection. In addition, Nu is observed to increase by increasing the volume fraction of nanoparticles. This behavior is due to the increased heat diffusion resulting from the increased thermal conductivity of the fluid. It is noted that this increase becomes more noticeable for larger values of Re (Re $\geq$ 1000). More precisely, for the favorable case obtained for high values of Re (Re = 5000), Nu, increases from 55.49 (pure fluid) to 59.27 (nanofluide ϕ = 0.07) with a rate of 6.81% thanks to the nanoparticles addition effect.

The effect of the addition of nanoparticles on heat transfer improvement is shown in Fig. 4 in terms of variations, with Re, of the parameter E_{nf}, for ϕ = 0.04 and ϕ = 0.07. We noted that E_{nf} increases clearly with Re for values of this one greater than 1000. However, for Re < 1000, E_{nf} remains insensitive to any variation of Re. Therefore, it can be said that the improvement of thermal transfer by the addition of nanoparticles is more favoured by Re but only for a dominant forced convection regime. Thus, for a fixed value of Re, the increase in ϕ obviously contributes to the growth of E_{nf}. As an indication, for Re = 200, increasing ϕ from 0.04 to 0.07, the thermal improvement increases from 5% to 7%.

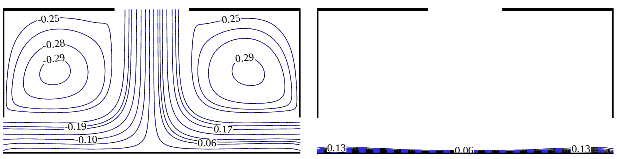
With: $$E_{nf} = \left[\frac{Nu_{nf} - Nu_f}{Nu_f} \right] \times 100 \qquad (12)$$



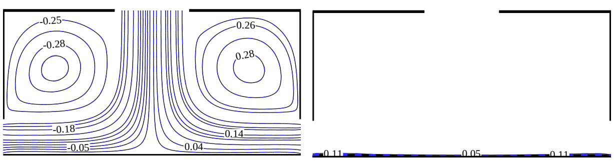
a) *Re = 200 (Ψ_{min} = -0.296 ; Ψ_{max} = 0.296)*



b) *Re = 600 (Ψ_{min} = -0,293 ; Ψ_{max} = 0,293)*



c) *Re = 1000 (Ψ_{min} = -0,290 ; Ψ_{max} = 0,290)*



d) *Re = 5000 (Ψ_{min} = -0,285 ; Ψ_{max} = 0,285)*

Fig. 2 Streamlines and isotherms, for ϕ = 0 (—) and ϕ = 0.07 (- - -) for different values of Re: a) Re = 200, b) Re = 600, c) Re = 1000 and d) Re = 5000.

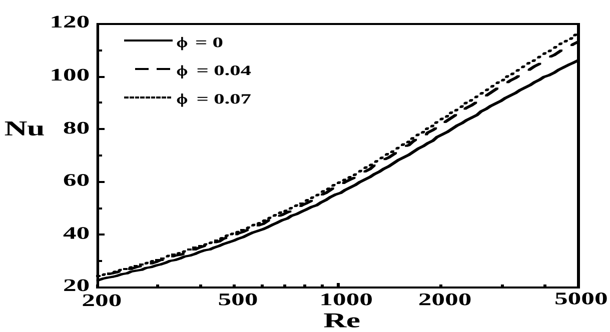


Fig. 3: Variations, with Re, of the average Nusselt number, Nu, for different values of ϕ.

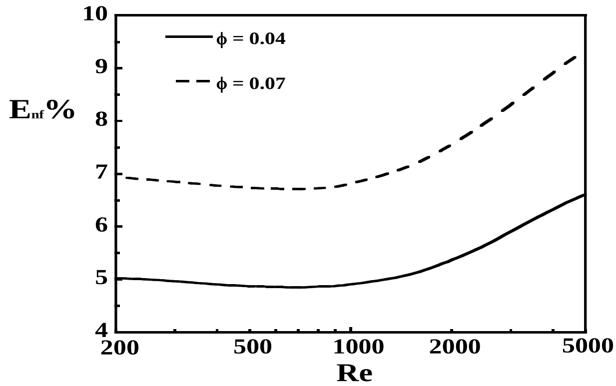


Fig. 4: Variations, with Re, of the parameter, E_{nf} (heat transfer induced by the addition of nanoparticles) for different values of ϕ.

In many practical applications (domestic or industrial), it is of great importance to know the impact of the governing parameters on the average temperature, $\overline{T}$, of the fluid within the cavity. Thus, as is clearly shown on Fig. 5, the evolution of $\overline{T}$ is characterized by a continuous decrease by increasing Re for different values of ϕ. This decrease results from the flow structure which presents an open lines of forced convection in total and direct contact with the active wall. This aspect favours the removal of heat, carried by these lines towards the outside of the cavity. In addition, this energy released and accentuated by the increase of Re contributes to the cooling of the cavity and subsequently to the fall of its average temperature. Also, the addition of nanoparticles to the base fluid involves a slight increase in average temperature. This behaviour results from an improved heat exchange within the cavity due to increased thermal conductivity of nanofluid.

To more understand the heat exchange distribution between the hot wall and the following fluid, we present in Fig. 6a and Fig. 6b the profiles of the local Nusselt number, Nu_{loc}, along the hot wall, for Re = 200 and Re = 5000 and different values of ϕ. Fig. 6a obtained for Re = 200 (dominant natural convection) shows that for all the values of ϕ, the local heat transfer reaches its maximum just near the left outlet opening and its minimum near the right outlet opening. This phenomenon is justified by the decreasing profile of the heating temperature applied to the bottom wall. Similarly, there is a bump on the Nu_{loc} curve at X = 0.9. This remark can be justified by the fact that forced flow comes vertically from the inlet opening towards the hot wall, which further favors the exchange in this position. On the other hand, the effect of ϕ on the improvement of Nu_{loc} is almost absent since we are in a predominantly natural convection regime which is insensitive to the aforementioned effect. In the case of high Re (Re = 5000), the Nu_{loc} profile is illustrated in Fig. 6b for different ϕ. We observe that the Nu_{loc} profile keeps almost the same trend as that displayed in the case of Re = 200 with a very significant quantitative improvement due to the intensification of forced flow. The effect of adding nanoparticles to the pure fluid on local heat transfer has been very visible since the increase of ϕ favors Nu_{loc}. This positive impact is very noticeable when moving away from the edges of the hot wall. Generally, it can be concluded that Nu_{loc} depends on the nature of the heating profile, the structure of the flow and the volume concentration of the nanoparticles.

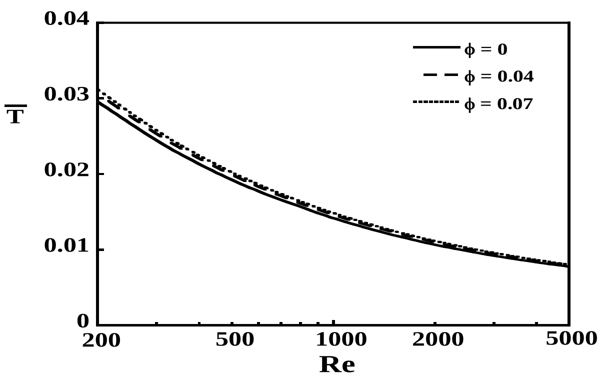


Fig. 5: Variations, with Re, of the mean temperature, $\overline{T}$, for different values of ϕ.

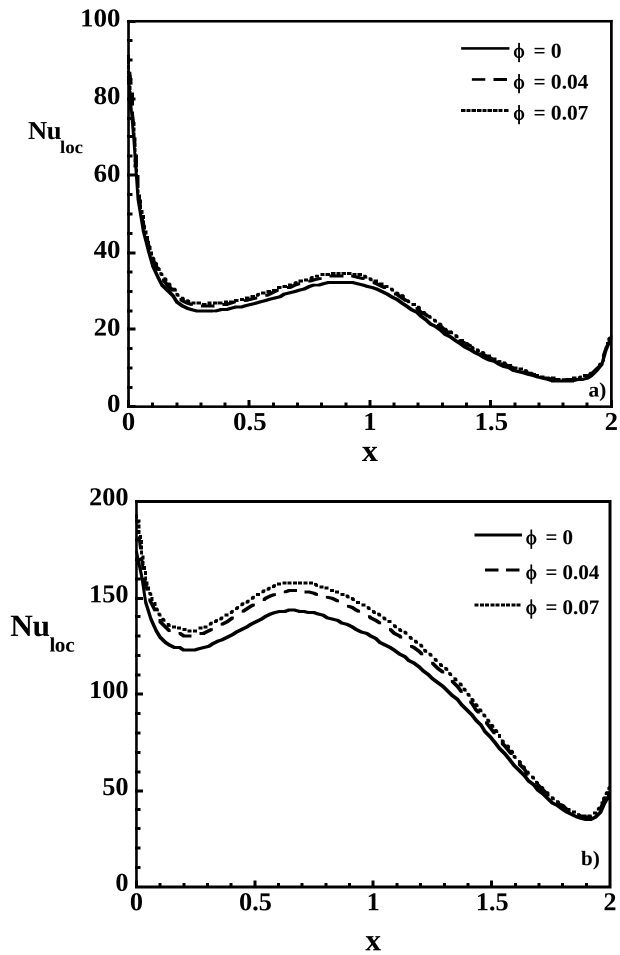


Fig. 6: Variations of the local Nusselt number, Nu_{loc}, along the hot wall for different values of ϕ : a) Re = 200 and b) Re = 5000.

Our goal in the next sections is to to illustrate the heating temperature profil effect on the heat perferformance inside the cavity. Thus, variations with Re of the average temperature $\overline{T}$ for pure fluid (ϕ = 0) and nanofluid (ϕ = 0.07) are illustrated in Fig. 7 for both decreasing and uniform heating temperature profiles. It is clearly seen that the heating mode does not affcet qualitatively the average temperature whatever the nanofluid concenctraion and the externel flow velocity.

Fig. 8 depicts the variations of the average Nusselt number Nu, versus the Reynolds number Re, over the heated bottom wall for both decreasing and uniform heating temperature distributions in the absence (ϕ = 0) and the presence (ϕ = 0.07) of nanoparticles. This figure shows a perfect concordance of the results relating to both heating temperature profile whatever the values of Re and ϕ. This notable behaviour marked on the Nu and $\overline{T}$ variations is mainly due to the symmetry of the thermodynamic boundaries conditions.

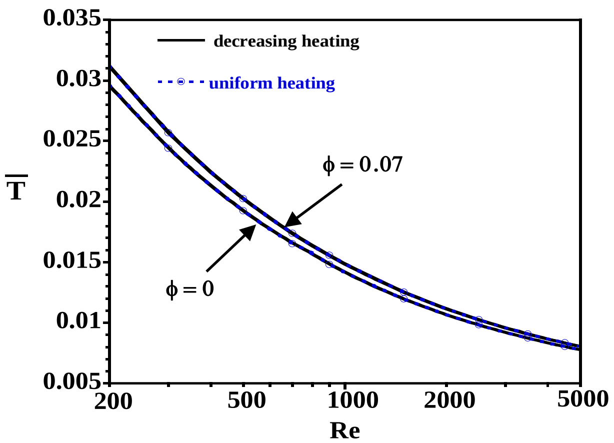


Fig. 7: Variations, with Re, of the mean temperature, $\overline{T}$, for both decreasing and uniform heating profile and two values of ϕ.

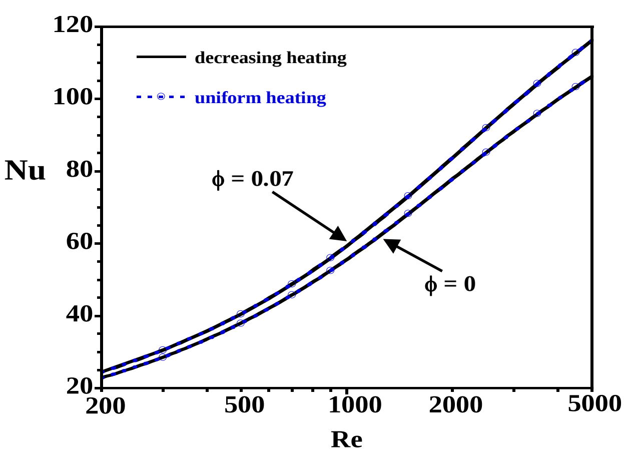


Fig. 8: Variations, with Re, of the average Nusselt number, Nu, for both decreasing and uniform heating profile and two values of ϕ.

5. Conclusions

A numerical analysis has been performed to study the laminar two dimensional mixed convection flow of Al_2O_3-water nanofluid in a horizontal multiple ventilated enclosure heated from below and cooled by suction of an external flow. The study is carried out by considering linearly decreasing heating. The results of the study show that the combined effect of Re and ϕ affects the dynamic and the thermal structure of the flow. Also, it is found that the same parameters contribute to the increase in the heat transfer rate and the mean temperature within the enclosure. Similarly, it is obtained that Re promotes the cooling of the cavity while ϕ promotes its reheating. Also, it is important to mention that a comparison was made between the thermal performances of decreasing and uniform heating type. Results demonstrate that the two heating modes have a same thermal performance. For this reason, we have limited ourselves to decreasing heating.

References

[1] M. Muthtamilselvan, D. H. Doh, Mixed convection of heat generating nanofluid in a lid-driven cavity with uniform and non-uniform heating of bottom wall, Applied Mathematical Modelling, 38 (13), (2014) 3164-3174.

[2] M. N. Hasan, K. Samiuzzaman, S. H. Haque, S. Saha, M. Q. Islam, Mixed convection heat transfer inside a square cavity filled with Cu-water nanofluid, Procedia Engineering, 105 (2015) 438-445.

[3] H. Moumni, H. Welhezi, R. Djebali, E. Sediki, Accurate finite volume investigation of nanofluid mixed convection in two-sided lid driven cavity including discrete heat sources, Applied Mathematical Modelling, 39 (14), (2015) 4164-4179.

[4] Y. Dahani, A. Amahmid, M. Hasnaoui, Effect of nanoparticles on the hysteresis loop in mixed convection within a two-sided lid-driven inclined cavity filled with a nanofluid, Heat Transfer Engineering, 40 (1-2), (2018) 128-146.

[5] E. Sourtiji, M. Gorji-Bandpy, D. D. Ganji, S. F. Hosseinizadeh, Numerical analysis of mixed convection heat transfer of Al_2O_3-water nanofluid in a ventilated

cavity considering different positions of the outlet port, Powder Technology, 262 (2014) 71-81.

[6] A. Bahlaoui, A. Raji, M. Hasnaoui, M. Naïmi, Mixed convection in a nanofluid filled vented rectangular cavity: suction and injection heat performance, Computational Thermal Sciences, 6 (2), (2014) 129-142.

[7] A. Bahlaoui, A. Raji, M. Hasnaoui, M. Naïmi, Mixed convection heat transfer enhancement in a vented cavity filled with a nanofluid, Journal of Applied Fluid Mechanics, 9 (2), (2016) 593-604.

[8] K. M. Shirvan, M. Mamourian, R. Ellahi, Numerical investigation and optimization of mixed convection in ventilated square cavity filled with nanofluid of different inlet and outlet port, International Journal of Numerical Methods for Heat & Fluid Flow, 27 (9), (2017) 2053-2069.

[9] H. Moumni, H. Welhezi, E. Sediki, Numerical investigation of heat transfer enhancement in a square ventilated cavity with discrete heat sources using nanofluid, High Temperature, 55 (3), (2017) 426-433.

[10] Z. Boulahia, A. Wakif, R. Sehaqui, Heat transfer and Cu-water nanofluid flow in a ventilated cavity having central cooling cylinder and heated from the below considering three different outlet port locations, Frontiers in Heat and Mass Transfer, 11 (2018) 1-10.

[11] I. Arroub, A. Bahlaoui, A. Raji, M. Hasnaoui, M. Naïmi, Cooling enhancement by nanofluid mixed convection inside a horizontal vented cavity submitted to sinusoidal heating, Engineering Computations, 35 (4), (2018) 1747-1773.

[12] I. Arroub, A. Bahlaoui, A. Raji, M. Hasnaoui, M. Naïmi, Varying heating effect on mixed convection of nanofluids in a vented horizontal cavity with injection or suction, Heat Transfer Engineering, 40 (11), (2019) 941-958.