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Unsteady Natural Convection in A Water Filled Cavity with Hot Partitioned Wall

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Abstract

The main objective of this work is to study numerically the unsteady natural convection phenomena in water filled rectangular enclosure with hot partitioned wall. The fluid flow and the heat transfer described in terms of continuity, linear momentum and energy equations were predicted by using the finite volume method. Streamlines, isotherms and local Nusselt number time evolution are presented for all investigated values. The aspect ratio of the geometry, Prandtl number are fixed at 0.24, 6.64, respectively, for different partitions lengths; however the Rayleigh number values were ranging from 106 to 3,77x109 in order to observe the transition regime. Representative results illustrating the effects of the partition length for the heat transfer and the thermal boundary layer are also reported and discussed. The obtained results show that the presence of the partition on the hot wall affects both heat transfer and fluid flow. It was found that the average Nusselt number increases with increase in the Rayleigh number. Also, as the dimensionless partition length increases, the flow speed within the partitioned enclosure decreases. Moreover, the features of the unsteady flow induced by the presence of partitions are characterized and the mechanisms responsible for the unsteadiness are discussed.

Keywords

Natural Convection, unsteady regime, partial partitions, Rayleigh Number, Nusselt Number, Partitions Length

1. Introduction

Natural convection flows in a differentially heated cavity are usually encountered in various industrial applications such as heat exchangers, nuclear reactors, cooling of electronic equipment this is due to the importance of this phenomenon in natural and technological processes. The problem admits a variety of solutions, both steady-state and unsteady. This is due to the nonlinearity of the Navier-Stokes equations and the convective heat transfer and thus the studies of natural convection flows in the cavity have been extensively reported in the literature over the past decades.

Batchelor [1] has shown that the heat transfer through the cavity is dominated by conduction for sufficiently small Rayleigh numbers. However De Vahl Davis [2] et al has focused on the steady natural convection flow in the cavity. On the other hand, Patterson et al [3] studied unsteady natural convection in a rectangular cavity with instantaneous cooling and heating of two opposite vertical sidewalls. They concluded that the flow had a strong dependence on the Prandtl number and cavity aspect ratio.

Yucel et al [4] indicated in their numerical analysis of laminar natural convection in filled air enclosures with fins attached to an active wall that with increasing number of fins the heat transfer first reaches a maximum and then approaches a constant, which is not affected by the number of fins. At low Rayleigh numbers, the heat transfer rate increases with the increasing number of fins and the fin length. But, at higher Rayleigh numbers, the heat transfer rate can be decreased or increased by properly choosing the number of fins and the fin lengths.

The transition of the thermal boundary layer from start-up to a quasi-steady state in a side-heated cavity is observed using a shadowgraph technique is also investigated by Xu et al [5]. A significant feature of the transition revealed from the present flow visualization is the formation of a double-layer structure along the sidewall at the entrainment development stage. It was believed that the reverse flow in the double-layer structure is the likely cause responsible for the unstable travelling waves at the quasi-steady state. They performed a direct numerical simulation of unsteady natural convection in a differentially heated cavity with a thin fin of different lengths on a sidewall at the Raleigh number of 3.8 ×109. They found that the fin length significantly impacts on the transient thermal flow around the fin and heat transfer through the finned sidewall in the early stage of the transient flow development.

Numerical investigations of transient natural convection flow through a fluid-saturated porous medium in a rectangular cavity with a convection surface condition were conducted by Pakdee et al [6]. The exposed surface allows convective transport through the porous medium, generating a thermal stratification and flow circulations. They found that the heat transfer coefficient, Rayleigh number and

Darcy number affect considerably the characteristics of flow and heat transfer mechanisms and the flow pattern is found to have a local effect on the heat convection rate.

The two-dimensional laminar natural convective transient flow characteristics in a differentially heated air-filled tall cavity with gradual heating are investigated both experimentally and numerically for various parameters such as Rayleigh number and temperature difference by Kolsi et al [7]. The results revealed that as the Rayleigh number increases the flow becomes unstable. Also, the flow characteristics are observed to be multi-cellular and time variant especially at high Rayleigh numbers.

Rahman et al [8] investigated the unsteady natural convection heat transfer in an isosceles triangular enclosure filled with Al2O3 nanoparticles. Their study was performed using the finite element method and the most important of their finding was the increasing of the heat transfer with addition of nanoparticle and increasing of the Rayleigh number.

Transient laminar natural convection of air in a tall cavity was studied numerically by Zhu et al [9]. They found that the overall Nusselt numbers for the Rayleigh numbers covering the range from 103 to 105 reveal a good agreement with measured data. When Ra takes the value in the range from 105 to 6*105, the overall Nusselt numbers have a relative deviation from their experimental data.

Wright et al [10] studied a flow visualization of natural convection in a tall air filled vertical cavity using a smoke patterns and interferometers. Experiments results showed that as Ra exceeded 104 the flow became irregular and the core flow became increasingly unsteady.

In addition to the above-mentioned earlier studies, comprehensive investigations of the natural convection flows in the cavity with a partition on the sidewall have been reported in the recent literature. The effects of the size, material and position of the partitions on the natural convection flows in the cavity have been paid much attention. In most of these studies, the thickness of the partition is considered to be sufficiently small in comparison with the fin partition and thus the fin length is the major geometric parameter for controlling the natural convection flows in the cavity.

Frederick et al [11] reported in their study that the heat transfer through the finned sidewall is reduced as the fin length increases due to the depression of the natural convection flows adjacent to the finned sidewall.

The transition from a steady to an unsteady flow induced by an adiabatic fin on the side walls was also carried out by Xu et al [12] they demonstrated that the fin may induce the transition to an unsteady flow and the critical Rayleigh number for the occurrence of the transition is between 3.72 x106 and 3.73x 106. Thus Paul et al [13] treated the Effect of an adiabatic fin on natural convection heat transfer in an air filled triangular enclosure by numerical simulations.

Sojoudi et al [14] concluded that less time is needed for higher Ra to reach the steady stage and value of dimensionless temperature above bottom heat source is lower for higher Ra in their study of unsteady air flow and heat transfer in a partitioned triangular cavity of differentially heated from inclined walls and heat source placed at the bottom wall

Transient flows in a differently heated cavity with a fin at different positions on the sidewall were studied by Ma et al [15] for Ra=108. After that Ma et al did the same investigation for a wide range of high Rayleigh numbers from 108 to 1011 and a Prandtl number of Pr = 6.63. Their results show that the development of natural convection from the startup is dependent on the Rayleigh number and the fin position. They applied a simple scaling analysis for the transient flow around the fin and they quantified the dependence of the unsteady flow on the Rayleigh number and the fin position [16].

A numerical study of steady laminar conjugate natural convection in a square enclosure with an inclined thin fin of arbitrary length was studied by Ben-Nakhi et al [17] for Pr=0.707. It was found that the thin fin inclination angle and length, and solid to fluid thermal conductivity ratio have significant effects on the local and average Nusselt number at the heated surfaces. The presence of an inclined thin fin reduces the average Nusselt number in an unordered way. They observed that the addition of a fin to an enclosure has two counter acting effects: restraining natural convection and increasing heating surface.

The effect on heat transfer of partial horizontal partitions placed on the hot wall of a differentially heated square cavity filled with air for varied width and for varied position along the height axis of the hot wall is studied numerically using the finite element method by Nag et al[18] for Rayleigh numbers ranging from l03 to 106. They suggested a correlation for the average Nusselt number, the Rayleigh number and the width of the horizontal partition for a particular position of the partition along the height. They found that the effect of a horizontal partition is significant.

Zahmatkesh [19] undertook a numerical study to clarify how presence of a thin fin may affect natural convection heat transfer in a thermally stratified porous layer. It was found that the natural convection is intensified with decreasing the fin length, moving the fin towards the right wall, decreasing the aspect ratio of the porous layer, and diminishing the stratification parameter concluding that the fin can be used as a control element for natural convection heat transfer in thermally stratified porous layers.

Liu et al [20] investigated experimentally natural convection in a differentially heated water filled cavity with two horizontal adiabatic fins on the sidewalls using the shadowgraph technique. Conducting a corresponding numerical simulation was in a good agreement between the simulation and the experiment.

Heat transfer in a differentially heated square cavity with cold partition at the bottom wall containing heat generating fluid at Pr=0.71 for Rayleigh numbers characterizing internal and external heating from 103 to 106 was studied numerically by Oztop et al[21]. They concluded that the heat transfer has been an increasing function of the Rayleigh number. Generally in the presence of a cold partition the heat transfer is reduced and the heat reduction is gradually increased with increasing partition height and thickness. Also the heat transfer is reduced more effectively when the partition is closer to the hot or cold wall.

Williamson et al [22] present a numerical study of transition to oscillatory flow in a two dimensional rectangular cavity differentially heated with a conducting partition in the centre, for Rayleigh number ranging from 0.6 to 1.6×1010 at Prandtl number Pr=7.5. They found that the thermal coupling of the boundary layers on either side of the conducting partition causes flow to become absolutely unsteady for a Rayleigh number at which otherwise similar non-partitioned cavity flow is steady.

Ambarita et al[23] studied numerically a laminar natural convection heat transfer in an air filled square cavity with two insulated baffles attached to its horizontal walls, they observed that the two baffles trap some fluid in the cavity and affect the flow fields. In addition it was found that the Nusselt number is an increasing function of Ra, a decreasing one of baffle length, and strongly depends on the baffle position.

Varol et al [24] did an experimental and numerical study on laminar natural convection in an air filled cavity heated from bottom due to an inclined fin, they observed that the heat transfer can be controlled by attaching an inclined fin onto wall and the presence of inclined fin affects the heat transfer and fluid flow.

Latest, The transition from steady to unsteady coupled thermal boundary layers induced by a fin on the partition of a differentially heated cavity filled with water was numerically investigated by Xu [25] For Rayleigh numbers from Ra=107 to 2 × 1010. It has been demonstrated that the fin may induce a transition to unsteady coupled thermal boundary layers and the critical Rayleigh number for the occurrence of the transition is between 3.5 and 3.6 × 108. It has been found that the flow rate through the cavity with a fin is larger than that without a fin under unsteady flow, indicating that the fin may improve unsteady flow in the partition cavity. Further, The physical mechanism of flow instability and heat transfer of natural convection in a differentially heated water filled cavity with thin fins on the hot wall using the energy gradient theory and the effects of the fin length, the fin position, the fin number, and Ra on heat transfer are investigated numerically by Dou et al[26]. They found that the effect of the fin length on heat transfer is negligible when Ra is relatively high. When there is only one fin, the most efficient heat transfer rate is achieved as the fin is fixed at the middle height of the cavity. The fins enhance heat transfer gradually with the increase of Rayleigh number under the influence of the thin fins.

The aim of this investigation is to simulate the unsteady natural convection in a differentially heated cavity with a different length of partitions on the hot wall. The present work is an extension of the work already established by Xu et al [12] which they treated similar cavity. The side walls are assumed to be differentially heated and the flat top and bottom walls are considered as adiabatic. The thermal and flow behavior and heat transfer characteristics have been studied for various Rayleigh number and partition lengths. The working fluid media is water with Prandtl number of 6.64 and Rayleigh number ranging from 106 to 3.77 x 109 $10^6 \leq Ra \leq 3.77 \times 10^9$. The wave features of the thermal flows around the partition are characterized and discussed in this paper.

2. Analysis And Modelling

The two-dimensional rectangular computational domain and boundary conditions are shown in Fig. 1. The domain which is H=0.24 m high by L=1m long is considered. One partition is placed in the mid height of the hot wall, the partitions length has been changed (l=L/16, L/8, L/4, L/2); however, the partition thickness was fixed at w=H/12, $T_h = 303.55K, T_c = 287.55K, T_0 = 295.5K$ and $\Delta T = 16K$

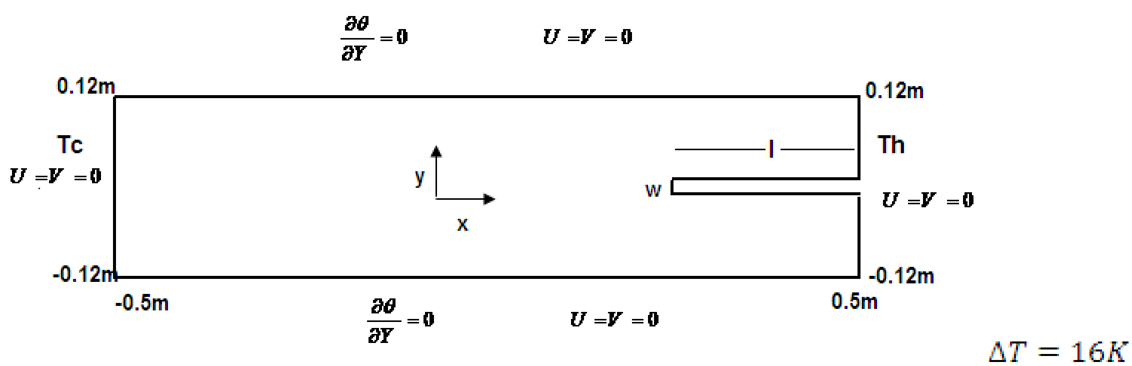


Fig.1. Computational domain and boundary conditions

The natural convection in the cavity as depicted in Figure 1 is described by the differential equations expressing conservation of mass, momentum and energy. The Boussinesq approximation is invoked for the fluid properties to relate density changes to temperature change and to couple in this way the temperature field to the flow field. The governing equations for transient natural convection can be expressed as follows:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (1)$$

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \qquad (2)$$

$$\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+g\beta(T-T_0) \qquad (3)$$

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{k}{\rho C_p}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \qquad (4)$$

To normalize the equations (1)-(4) in to non-dimensional form, the following parameters are defined:

$$Ra=\frac{g\beta L^3\Delta T}{\eta\alpha} \quad \text{and} \quad \Pr=\frac{\eta}{\rho} \qquad (5)$$

$$X=\frac{x}{L}\ ,\quad Y=\frac{y}{L}\ ,\quad U=\frac{uL}{\alpha}\ ,\quad V=\frac{vL}{\alpha}$$

$$P=\frac{pL^2}{\rho\alpha^2}\ ,\quad \theta=\frac{T-T_0}{\Delta T}\ ,\quad \Delta T=T_h-T_0\ ,\quad \tau=\frac{\alpha t}{L^2}$$

The non-dimensional numbers defined in the above Ra and Pr are the Rayleigh number and Prandtl number, respectively. Normalizing the dimensional equations, the following non-dimensional governing equations can be found as:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \qquad (6)$$

$$\frac{\partial U}{\partial \tau}+U\frac{\partial U}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial X}+\Pr\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) \qquad (7)$$

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ra\Pr\theta \qquad (8)$$

$$\frac{\partial \theta}{\partial \tau}+U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \qquad (9)$$

The average Nusselt number is defined as follows:

$$Nu = -\int_0^L \frac{h.y}{k} dy \tag{10}$$

2.1 Boundary Conditions

For the set of above equations, the boundary conditions are no slip for all walls and for energy equation; the side walls have been maintained at differentially heated condition while the other walls are considered as adiabatic. The boundary conditions and the flow domain are depicted in figure 1; they can be written mathematically as non dimensional form:

Bottom surface	U=V=0, $\frac{\partial \theta}{\partial Y} = 0$	(11a)
Top surface	U=V=0, $\frac{\partial \theta}{\partial Y} = 0$	(11b)
Left surface	U=V=0, $\theta = -0.5$	(11c)
Right surface	U=V=0, $\theta = 0.5$	(11d)

2.2 Numerical Procedure

The spatial derivatives in the Navier-Stokes equations are discretized with the finite volume method on a staggered grid. The solution domain is covered with finite volumes, having a grid point for a pressure and the temperature in the centre of each volume, grid points for the u-velocity in the middle of the west and east side, grid points for the v-velocity in the middle of the north and south side of each volume. The equations are integrated over each volume, after which mass, momentum and heat fluxes are discretized with finite differences. The PISO finite volume scheme was used with a second order centre difference scheme approximating the diffusion term, a QUICK scheme approximating the advection terms and a second order implicit time marching scheme for the unsteady term. A standard under relaxation technique is enforced in all steps of the computational procedure to ensure adequate convergence.

2.3 Grid

A fine non uniform grid was constructed in the proximity of the partition. Several grids were created for the geometry with partition on the wall in order to test the grid independence. The non uniform grid system with an expansion factor from the wall surface to the interior is adopted in the boundary layer zones. The rest of the flow domain (the central region of the cavity) is uniformly meshed. The results of the grid dependence test for the case with a partition are illustrated in Figure 2;

it is clearly seen the calculated time series of the temperature at point (0.498m,0.09m) in the early stage . In spite of strong perturbations due to the presence of the partition on the hot wall, the results calculated using the two grids are basically identical. This implies that either of the two finer grids may be used to resolve the quasi-steady state flow in the differentially heated cavity at a high Rayleigh number of 3.77 X10^9.

The figure 3 shows the grid distribution adjacent to a partition. It was clearly observed that a finer grid has been constructed around the partition. Good agreement between the current simulation and the literature reported on [12] are obtained, therefore the present numerical procedure may be applied to explore the transient natural convection flows in a differentially heated cavity over a range governing parameters (e.g different Rayleigh numbers and partition length)

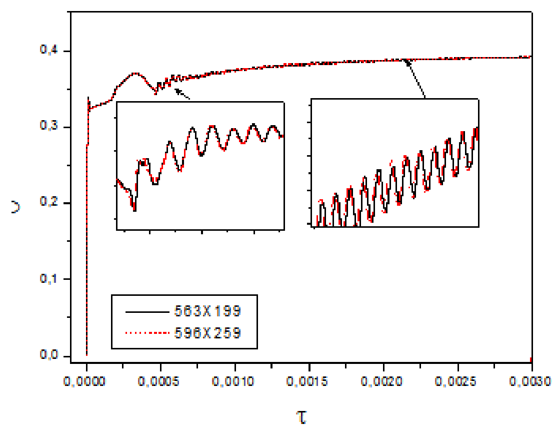


Fig.2. Temperature time series calculated by using different grids for the case of Ra= 3.77*10^9, and partition length of L/4. at the point (0.498,0.09) plotted.

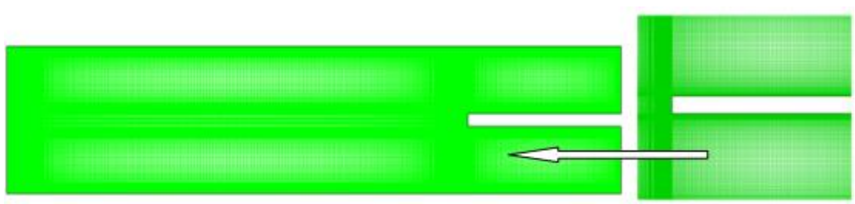


Fig.3. Grid distribution around a partition on the hot wall

3. Results and discussion

In this section, a numerical simulation of the unsteady natural convection flow in the partitioned enclosure is taken account. In order to obtain insights into the steady and unsteady flow adjacent to a partitioned wall, numerical simulations for different Rayleigh number values and partitions length are carried out. The partition is an important factor to determine the transition of natural convection flows in the cavity to a steady or unsteady regime. Figure 4 illustrates time series of the calculated temperatures at the point (x= 0.498, y= 0.09) identified in the vertical boundary layer for different partitions length for high Rayleigh number ($Ra=3.77*10^9$). It was clearly seen that the peaks are generated called leading edge effect (LEE). The temperature is oscillatory in unsteady state.

It means that the transition from a steady to unsteady flows occurs as the Rayleigh number increases. It was also noticed that the partition (L/16) induces stronger perturbations than the other partitions.

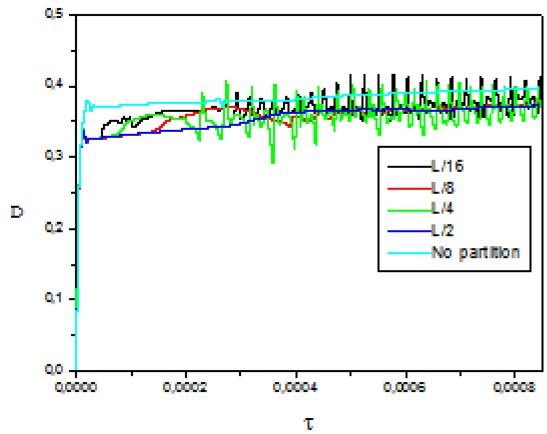


Fig.4. Temperature versus time, plotted at point (0.498, 0.09) for Ra $=3.77*10^9$ at different partition lengths

In order to examine the transient natural convection flows; figure 5 presents evolution of the temperature for different time for the partitioned enclosure with L/4 partition and fixed Rayleigh number. It is clearly seen that the perturbations are developed around the partitions. It was also noticed that when the time increases, the base flow down-stream of the partition loses its stability by forming small vortices and by moving the fluid in a wave form. The thermal flow around the partition triggers instability of the downstream thermal boundary layer. It can be concluded that as time increases, the fluid in the cavity becomes stratified and the flow adjacent to the partitioned wall becomes oscillatory.

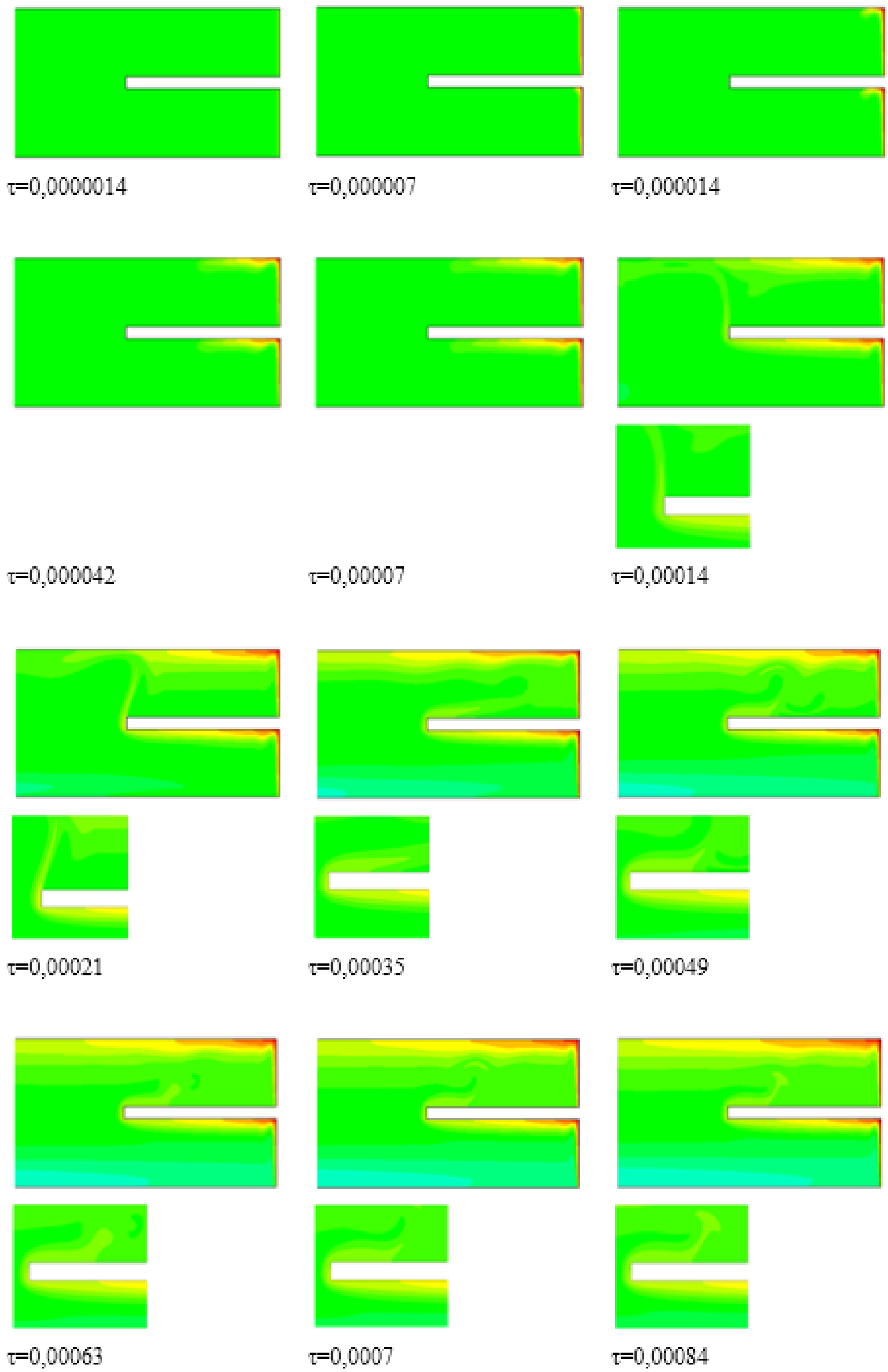
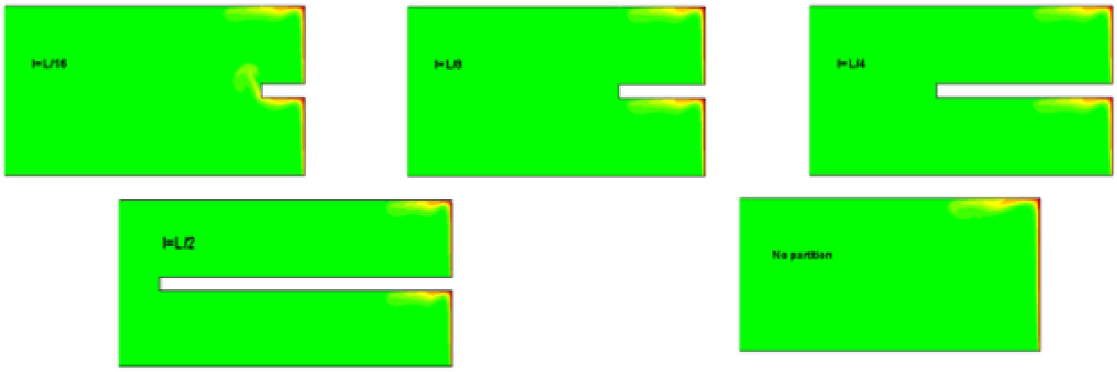
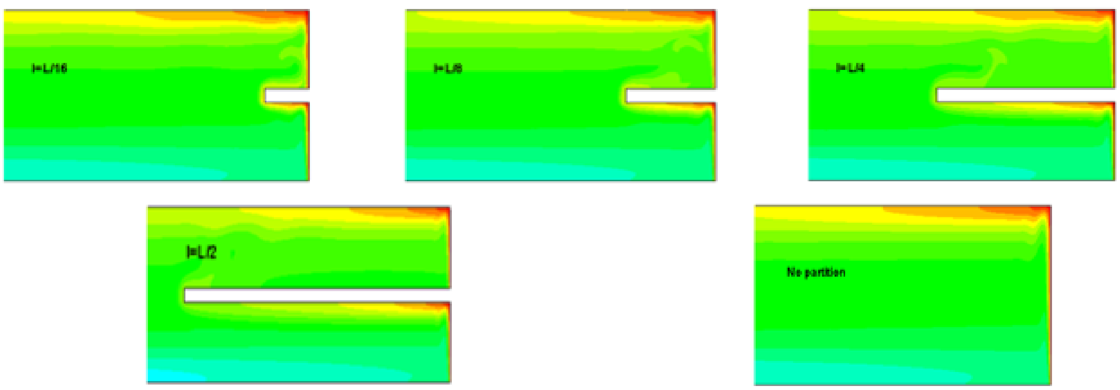


Fig.5. Temperature distribution at different time for l=L/4 and Rayleigh number equal to $3.77*10^9$

Figure 6 shows the thermal flow contours in both early and quasi steady stage for all partitions length tested. It was clearly seen that the partition length has an important effect thus the flow depends systematically on the partition length. The flow may very smooth near the partition and the oscillation of the shedding flow disturbs the thermal boundary layer which eventually approaches a periodic flow. This phenomenon was observed in the cavity with the L/2 partition length. As time increases, the position of separation moves further away from the hot wall.



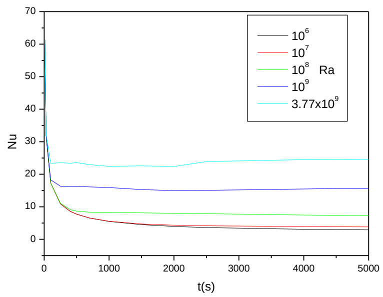
(a)τ=0,000035



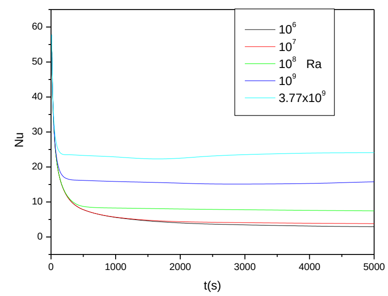
(b) τ=0,0007

Fig.6. Thermal flows, (a) in the early stage, (b) in the quasi-steady stage, for different partitions length.

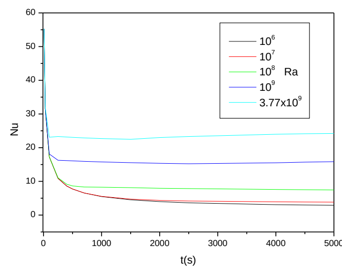
For the purpose of observing quantitatively natural convection flows and to describe heat transfer through the cavity, the mean Nusselt numbers along the hot partitioned wall versus time was plotted for different Rayleigh numbers value and different partitions length. As shown in figure 7, the perturbations induced by the oscillation of the thermal flow around the partition have more room to grow. As a consequence, the heat transfer through the partitioned hot wall is notably enhanced so the presence of partition on the hot wall shows a promising result for the enhancement of the heat transfer through a differentially heated cavity. Thus the local Nusselt number is much smaller near the partition, although oscillatory downstream of the partition. The Nusselt numbers significantly reduce with time in the early stage and then approach an oscillatory state in the cases with and without partition.



(a)



(b)



(c)

Fig.7. Mean Nusselt number, Nu along the hot wall versus time for different Rayleigh number and different partition length (a) l=L/8, (b) l=L/4 and (c) l=L/2

Conclusion

In order to control the heat transfer through the sidewall, the problem of transient natural convection in a differentially heated cavity with a partition on the heated sidewall has been investigated in this paper. It was important to note that the partition may significantly change the transient natural convection flow. The transition of the natural convection may be classified into three stages: an initial stage, a transitional stage and a quasi steady stage. The numerical results have demonstrated also that the partitions on the hot wall may play an important role in controlling the transient natural convection and heat transfer through a differentially heated cavity in the present Rayleigh number ranging from 10^6 to 10^9. It was also noticed that the perturbation induced by the partition enforce the convection and enhance the heat transfer through the hot wall. However the dependence on the partition length and heat transfer through the partitioned cavity is obtained.

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Nomenclature

g	gravitational acceleration (m/s^2)
h	convective heat transfer coefficient (W/m^2 K)
H	height of the enclosure (m)
L	width of the enclosure (m)
Nu	Nusselt number
p	pressure (N/m2)
Pr	Prandtl number
Ra	Rayleigh number
T	temperature (K)
T_h	temperature of the hot surface (K)
T_c	temperature of the cold surface (K)
T_0	initial temperature (K)
T_m	average temperature (k)
ΔT	Temperature variation, $T_h - T_c$ (k)
U	velocity component in x direction (m/s)
V	velocity component in y-direction (m/s)
x, y	Cartesian coordinates (m)
W	partition thickness
L	partition length
Greek symbols	

k	thermal conductivity of fluid (W/m k)
α	thermal diffusivity (m2/s)
β	coefficient of volumetric expansion (1/K)
θ	dimensionless temperature, (T -To)
μ	dynamic viscosity (N s/m2)
ρ	fluid density (kg/m3)

Subscripts

C	cold
H	hot
0	initial