Numerical Investigation Of Natural Convection In Air Filled Cubical Enclosure With Hot Wavy Surface And Partial Partitions

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Abstract

Natural convection in cubical enclosure with hot surface geometry and partial partitions has been analysed. The geometry is a cube with wavy hot surface (three undulations) and three partitions. The investigation has been performed for different partitions lengths and Rayleigh number while the Prandtl number kept constant. This problem is solved by using the partial differential equations which are the equation of mass, momentum, and energy. The results obtained show that the hot wall geometry with partitions affects the flow and the heat transfer rate in the cavity. It has been found also that the mean Nusselt number decreases compared with the heat transfer in the undulated cubical cavity without partitions.

Keywords: Natural convection; Cubical enclosure; Wavy hot surface; Partial Partitions

1. Introduction

Natural convection flow analysis in enclosures has many thermal engineering applications, such as cooling of electronic devices, energy storage systems and fire-safe compartment. In the numerical domain, a big progress has been made on the natural convection in complex geometries [1, 2, 3 & 4]. For many years a lot of attention has been paid to the problems of natural convection in enclosure. However, the limitations of numerical tools and
experimental techniques restrict investigators within the approximation of a two dimensional model even though fluid motion is three dimensional in nature. Bessov et al. [5] have suggested a benchmark numerical solution for the three dimensional natural convection in cubical enclosure. Three dimensional laminar flow has been also studied by Mallinson et al.[6] and Lee et al. [7] for enclosures of the length aspect ratio $A_z$ varying from 2 to the enclosure with $A_z$ =1 and 2 have been considered by Lankhorst [8] who computed steady flows for numbers of Rayleigh ranging from $10^6$ to $10^8$. Fusegi et al. [9] made a three dimensional flow analysis on natural convection in a differentially heated cubical enclosure, the detailed structures of the fields were scrutinized by using high-resolution computational results over the range of Rayleigh numbers studied, $10^3 \leq Ra \leq 10^6$, they clarified three dimensional structures of flow, vorticity and temperature in the cavity; they also compared their numerical results with the experimental measurements as studied by Bilsk [10]. The complexity of such problems increases when the hot surface of the cubical enclosure becomes wavy. Adjilout et al. [11] have also investigated the two dimensional natural convection in an inclined cavity with hot wavy wall. They simulated the fluid and heat transfer with ADI scheme; one of their interesting results was a decrease of the averaged heat transfer compared with the mean Nusselt number of the square cavity. They showed that the thermal boundary layer is considerably affected by wavy wall and they recommended investigating the hot wall geometry optimisation. Natural convection in three dimensional rectangular enclosures had been analyzed numerically by Sik Lee et al [12]; the effect of the Rayleigh number was mainly investigated. The results showed that the temperature distribution imposed on the wall reinforced the axial flow and magnified the three dimensional effect. Three dimensional structures in laminar natural convection in a cubic enclosure were investigated experimentally by Hiller et al [13]. The Rayleigh numbers ranged from $10^4$ to $2 \times 10^7$ and the Prandtl numbers from $5.8$ to $6 \times 10^3$. They showed the velocity and vorticity fields and compared the

| Nomenclature                           | Definition                                                                                                                                 |
|----------------------------------------|-------------------------------------------------------------------------------------------------------------------------------------------|
| $g$                                     | Gravitational acceleration [m/s²]                                                                                                       |
| $L$                                     | Cavity side [m]                                                                                                                           |
| $Nu$                                    | Nusselt number                                                                                                                           |
| $p$                                     | Static pressure [Pa]                                                                                                                     |
| $P$                                     | Dimensionless pressure                                                                                                                   |
| $Pr$                                    | Prandtl number, $\nu / \alpha$                                                                                                         |
| $Ra$                                    | Rayleigh number, $g \beta L^3 \Delta T / \alpha \nu$                                                                                    |
| $T$                                     | Temperature [K]                                                                                                                          |
| $T_0$                                   | Average temperature, $(T_h + T_c) / 2$ [K]                                                                                               |
| $T_h, T_c$                              | Hot and cold temperature [K]                                                                                                             |
| $\Delta T$                              | Temperature variation, $T_h - T_c$ [K]                                                                                                   |
| $u,v,w$                                 | Dimensionless fluid velocities                                                                                                           |
| $X,Y,Z$                                 | Dimensionless Cartesian coordinates, $x/L, y/L, z/L$                                                                                     |
| $x,y,z$                                 | Coordinates [m]                                                                                                                          |

| Greek symbols                           | Definition                                                                                                                                 |
|----------------------------------------|-------------------------------------------------------------------------------------------------------------------------------------------|
| $\alpha$                               | Thermal diffusivity of air [m²s⁻¹]                                                                                                       |
| $\beta$                                | Thermal expansion coefficient [K⁻¹]                                                                                                      |
| $\theta$                               | Dimensionless temperature                                                                                                                 |
| $\nu$                                  | Kinematic viscosity [m²/s]                                                                                                                 |
| $\rho$                                  | Air density [Kg/m³]                                                                                                                        |

| subscripts                             |                                                                                                                                          |
|----------------------------------------|-------------------------------------------------------------------------------------------------------------------------------------------|
| $a$                                     | Average                                                                                                                                  |
experimental observations with numerical calculations found in the literature. Gunes [14] studied numerically in his thesis three dimensional natural convection cooling in enclosures and vertical channels with spatially periodic heat sources. It was established that the convection reduces the maximum operating temperature for all configurations. Yu et al [15] carried out a numerical study of a three dimensional laminar natural convection in a vented enclosure. They presented the local and overall heat transfer from the heat source and the substrate, in terms of Nusselt numbers and the surface temperatures to illustrate the vent effects. Akrour et al [16] made numerically a three dimensional steady flow on natural convection in a differentially heated cubical enclosure. The variation of Nusselt numbers on the hot and cold walls were also presented to show the overall heat transfer characteristics inside the enclosure. They found that the three dimensional data demonstrate reasonable agreement with the experimental measurement. Natural convection heat transfer associated to fluid dynamics phenomena was studied extensively by Corzo et al [17]. The results presented revealed a good agreement not only in 2D also in 3D and for a wide range of Ra numbers. Kürekci et al [18] made an experimental and numerical study of laminar natural convection in a differentially heated cubical enclosure. They presented the local and overall heat transfer from the heat source and the substrate, in terms of Nusselt numbers and the surface temperatures to illustrate the vent effects. Sabeur et al [21] performed a numerical investigation of the influence of the hot surface geometry on a laminar natural convection in a cubical cavity filled with air differentially heated. The results obtained showed that the hot wall geometry affects the flow and the heat transfer rate in the cavity. The mean Nusselt number decreases compared with the heat transfer in the cubic cavity. Fusegi et al [22] studied numerically a transient three dimensional natural convection in a differentially heated cubical enclosure at Rayleigh number of $10^6$. They showed that the behavior of the heat transfer rate in the enclosure was considerably influenced by the presence of the internal gravity wave motion. They also [23] carried out a three dimensional numerical simulation of periodic natural convection in a differentially heated cubical enclosure; at the Rayleigh number of $8.5 \times 10^6$, they found that the period of the oscillations was consistent with the experimental measurements. In another study, the same authors [24] made numerical simulations of natural convection in a differentially heated cubical enclosure with a partition. They scrutinized the effects of the partition geometry on the three dimensional flow properties. Silva et al [25] evaluated the effect of the aspect ratio and horizontal length of a high conductivity rectangular fin attached to the hot wall of a three dimensional differentially heated cubic enclosure in laminar natural convection, a scale analysis was used to predict the domain in which the fin geometry played a significant role. Frederick [26] made a numerical study of natural convection of air in a differentially heated cubical enclosure with a thick fin placed vertically in the middle of the hot wall; he investigated the variation of overall Nusselt number with Rayleigh number and thermal conductivity ratio. He [27] studied also a numerically natural convection heat transfer in a cubical enclosure with two active sectors on one vertical wall over a wide range of Rayleigh number. He described the flow patterns and temperature distribution and proposed an expression for overall heat transfer. In another study, Frederick et al [28] investigated numerically three dimensional natural convection of air in a cubical enclosure with a fin on the hot wall. It was concluded that for $10^5 \leq Ra \leq 10^6$, a fin of partial width is more effective in promoting heat transfer than a fin of full width. Bocu et al [29] studied numerically laminar natural convection heat transfer in 3D rectangular air filled enclosures, with pins attached to the active wall. The Rayleigh numbers considered in their study ranges from $10^7$ to $10^5$; they concluded that the mean Nusselt number increases with increasing Rayleigh number for a fixed case. The present investigation is an extension of the work already established by Belkadi et al. [30] which they treated similar cavity in two dimensional study. The most part of our simulation is to show the three dimensional effects and the influence of the hot wavy wall with partial partitions on the laminar natural convection.
2. Analysis

The geometry and coordinate system are illustrated in Fig.1. The geometry is simply represented by a cubical cavity. This latter has a hot surface geometry wavy with partitions introduced at the crests with the constant temperature, \( T_h \). The opposite wall is straight and has a constant colder temperature \( T_c \). The others are thermally insulated. The study has been carried out for different partitions lengths and for two Rayleigh numbers (\( 10^5 \) and \( 10^6 \)); the Prandtl number is equal to 0.71.

2.1 Governing equations

The viscous incompressible flow inside a cubical enclosure and a temperature distribution are described by the Navier-Stokes and the energy equations. The governing equations for this case are for laminar and three dimensional flow. In the model development, the following assumptions are adopted:

- The process is in steady state.
- The Boussinesq approximation applies, which implies that except for the density in the gravitational term, all other properties in the governing equations are kept constant.
- There is no source or sink in the system.

Once above assumptions are employed into the conservation equation of mass, momentum and energy and the following dimensionless variable are introduced:

\[
X = \frac{x}{L}, \quad Y = \frac{y}{L}, \quad Z = \frac{z}{L} \tag{1}
\]

\[
U = \frac{u}{v/L}, \quad V = \frac{v}{v/L}, \quad W = \frac{w}{v/L} \tag{2}
\]

\[
P = \frac{p + \rho g y}{\rho (v/L)^2} \tag{3}
\]

\[
\theta = \frac{T_0 - T_c}{T_h - T_c}, \quad T_0 = \frac{T_h + T_c}{2} \tag{4}
\]

The dimensionless governing equations in Cartesian coordinates for the present study then take the following forms:

**Continuity**

\[
\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} + \frac{\partial W}{\partial Z} = 0 \tag{5}
\]

**X Momentum**

\[
U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} + W \frac{\partial U}{\partial Z} = - \frac{\partial P}{\partial X} + Pr \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} + \frac{\partial^2 U}{\partial Z^2} \right) \tag{6}
\]

**Y Momentum**

\[
U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} + W \frac{\partial V}{\partial Z} = - \frac{\partial P}{\partial Y} + Pr \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} + \frac{\partial^2 V}{\partial Z^2} \right) + Ra Pr \theta \tag{7}
\]

**Z Momentum**

\[
U \frac{\partial W}{\partial X} + V \frac{\partial W}{\partial Y} + W \frac{\partial W}{\partial Z} = - \frac{\partial P}{\partial Z} + Pr \left( \frac{\partial^2 W}{\partial X^2} + \frac{\partial^2 W}{\partial Y^2} + \frac{\partial^2 W}{\partial Z^2} \right) \tag{8}
\]

**Energy**

\[
U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} + W \frac{\partial \theta}{\partial Z} = \left( \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} + \frac{\partial^2 \theta}{\partial Z^2} \right) \tag{9}
\]

Knowing that \( 0 \leq Y \leq 1 \) and \( 0 \leq X \leq f(Y) \) with \( f(Y) = \left[ 1 - amp + amp (\cos 2 \pi Y) \right] \) and \( 0 \leq Z \leq 1 \) with \( n \) and
amp are respectively, the number of undulations and amplitude as proposed by Adjlout et al [11]. The geometry of the tested cavity is shown in figure 1.

Fig. 1. Physical domain

The study is completed with the definition of the following boundary conditions:

\[
\begin{align*}
\theta &= 0.5 & U &= V = W = 0 & \text{On the hot wall} \\
\theta &= -0.5 & U &= V = W = 0 & \text{On the cold wall} \\
\frac{\partial \theta}{\partial Y} &= 0 & U &= V = W = 0 & \text{On the adiabatic walls}
\end{align*}
\]

It can be seen from the above dimensionless equations that the Rayleigh number and Prandtl number are two of the important model parameters for a given flow geometry. The overall heat transfer characteristics are described by the average Nusselt number which is defined over the hot wall as follows:

\[
\overline{Nu} = \frac{1}{L} \int_{X=0}^{1} \int_{Z=0}^{1} \frac{\partial \theta}{\partial Y} dX dZ
\]  

(10)

3. Numerical method and model validation

A 3D uniform and staggered grid is used with a control volume formulation for the discretization. The central difference scheme is employed for the convective diffusive transport variables. Pressure correction and velocity correction are implemented in accordance with the SIMPLE algorithm to achieve a converged solution. The discretized algebraic equations are solved by the tri-diagonal matrix algorithm (TDMA). Relaxation factors of about 0.2-0.7 are used for the velocity components, while relaxation factors of about 0.5-0.8 are adopted for the temperature and pressure corrections. The adiabatic boundary structure condition is treated by the additional source term method. In the present study, typically 2000-8000 outer iterations are required to achieve the convergence. For the convergence criteria, the relative variations of the temperature and velocity between two successive iterations are imposed to be smaller than the specified accuracy levels of $10^{-6}$. A grid independency test was carried out, and the results are indicated in table 1. Three sets of grids $54 \times 54 \times 54$, $60 \times 60 \times 60$ and $67 \times 67 \times 67$ were employed; the case with $67 \times 67 \times 67$ grids (Figure 2) was used for taking both the accuracy and convergence rate into account.

| Grid          | $54 \times 54 \times 54$ | $60 \times 60 \times 60$ | $67 \times 67 \times 67$ | Bessov [5] |
|---------------|--------------------------|--------------------------|--------------------------|-------------|
| $Nu_1$        | 4.458                    | 4.287                    | 4.325                    | 4.339       |
4. Results and Discussion
The streamlines and isotherms in the computation domain for the case mentioned above and for $Ra = 10^5$, $10^6$ and different partitions length and in different planes $XYZ$. It was noticed in figure 3 and 4 that near the wavy geometry, the thermal boundary layer thickness increases and decreases just before the partitions or just after a trough resulting in a big decrease in the global heat transfer along the hot geometry. When Compared with the cube and the undulated cavity without partitions, the thermal boundary thickness appears to enhance with the presence of partitions. Every partition takes part to the thickening of the thermal boundary layer. That shows the induction of the partition length on isotherms and streamlines distributions. It was clearly seen that an increase in partitions length seems to increase in the thermal boundary thickness. To observe the transition between the one-roll and two roll systems, the Rayleigh number was changed gradually from $10^5$ to $10^6$. At small temperature gradients, as mentioned before, only one vortex is observed in the vertical mid-plane of the cavity (figure 3), the centre of this vortex is not in the middle of the plane but shifted toward the wavy hot wall. However, regardless of the Rayleigh number, the best performance is achieved when the partitions length reaches 0.25 which corresponds to 5amp.

Figure 6 presents the local Nusselt Number distributions at the heated wall for undulated cavity with wavy hot geometry with partitions, undulated cavity without partitions and cube ($x=1$) for $Ra=10^5$. The wavy tendency of the local Nusselt number is describing the cavities with corrugated wall. It was observed also that for the undulated cavity, the heat transfer decreases notably compared with the undulated cavity without partition and the cube as illustrated in table 2. The relative decrease of the global heat transfer for the cavity with hot surface geometry and partitions reaches 40% for $Ra=10^6$ compared with a cube. The same phenomena was also detected by Belkadi et al.[30] when they studied the identical case in 2D.
The distribution of average Nusselt number for different partition length are represented in figure 5 for all Rayleigh numbers investigated (Ra=10^5 and 10^6 respectively). It was clearly seen that there is an influence of the partition length on the mean Nusselt number so; the trend of the curve is notably decreasing with an increase in the partition length.

**Table 2** Comparison of the Mean Nusselt Number between the cubical and undulated enclosure (Ra=10^5)

| Geometry (2D) | square | Undulated cavity without partitions | Undulated cavity With partitions |
|--------------|--------|------------------------------------|---------------------------------|
| Nu_a         | 4.52   | 3.51                               | 3.36                            |

| Geometry (3D) | Cube | Undulated cavity without partitions | Undulated cavity With partitions |
|---------------|------|------------------------------------|---------------------------------|
| Nu_a          | 4.32 | 3.48                               | 3.26                            |

**Table 3** Comparison of the Mean Nusselt Number between the cubical and undulated enclosure (Ra=10^6)

| Geometry (2D) | square | Undulated cavity without partitions | Undulated cavity With partitions |
|--------------|--------|------------------------------------|---------------------------------|
| Nu_a         | 8.82   | 7.55                               | 6.97                            |

| Geometry (3D) | Cube | Undulated cavity without partitions | Undulated cavity With partitions |
|---------------|------|------------------------------------|---------------------------------|
| Nu_a          | 8.42 | 7.36                               | 6.65                            |

**Table 4** Mean Nusselt Number for tested enclosures for (Ra=10^5)

| Geometry | 1 Amp | 2 Amp | 3 Amp | 4 Amp | 5 Amp |
|----------|-------|-------|-------|-------|-------|
| Nu_a     | 3.26  | 2.70  | 2.11  | 2.04  | 1.82  |

**Table 5** Mean Nusselt Number for tested enclosure for (Ra=10^6)

| Geometry | 1 Amp | 2 Amp | 3 Amp | 4 Amp | 5 Amp |
|----------|-------|-------|-------|-------|-------|
| Nu_a     | 6.65  | 5.76  | 5.11  | 4.58  | 4.03  |

**Fig. 3.** Temperature field and Isosurfaces of the absolute values of the vorticity at Ra=10^5 for all partitions Length and at different planes: X=0.84 Y=0.2 and Y=0.8 a.1 amp; b.2 amp; c.3 amp; d.4 amp; e.5 amp
5. Conclusion
The present investigation deals with the effect of the presence of a partition on the hot wall geometry of undulated cubical cavity, air filled, and differentially heated enclosure. These partitions caused decrease of up 40% in the heat transfer with respect to the case at the same Rayleigh number. As Ra grew, boundary layer got progressively thinner, and the effects of the partitions on the temperature field became more and more localized around the partitions. It was found that the flow pattern observed in 3D enclosure is very similar in the 2D case already studied in reference [30]. The presence of partitions has a substantial effect on the flow in the 3D enclosure. This was confirmed by the results, the percentage heat transfer reduction grows with the partitions length which is a function of amplitude of the waviness hot geometry. These effects described could be useful for reducing convective losses in solar collectors if they were observed also in high aspect ratio or rectangular cavities. As the insulating effect was related to a local distortion of the temperature field, it can be concluded that multiple partitions would be required in such cavities to reach the levels of heat transfer reduction.

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