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Numerical Investigation of Laminar Natural Convection in an Inclined Cavity with a Wavy Wall

Said Mekroussi, Nord-Eddine Sad Chemloul
Department of Mechanical Engineering, University Ibn Khaldoun, Tiaret 1400, Algeria

Abstract— ***Natural convection in two-dimensional enclosure with three flat and one wavy wall is numerically investigated, which are formed by horizontal adiabatic walls and vertical isothermal walls. The tests were carried out for different inclination angles, amplitudes and Rayleigh numbers, while the Prandtl number was kept constant. The results obtained show that the variation of Nusselt number depends on the mathematical expression describing the shape of the wave and the hot wall undulation and the angle of inclination affect the flow and heat transfer rate in the cavity. With increase in amplitude, the average Nusselt number on the wavy wall is appreciably high at low Rayleigh number. The trend of local Nusselt number is wavy.***

Index Terms— **natural convection, inclined cavity, square cavity.**

I. INTRODUCTION

The study of natural convection in enclosures has mainly been devoted to the classical Rayleigh-Benard problem (hot bottom wall and cold top wall) and also to the case of a square cavity with one vertical wall heated and the opposite one cold. However, in a variety of engineering applications, enclosures are inclined to the direction of gravity. Hence, buoyancy forces have both components relative to the walls of the enclosure which modify strongly the flow structure and the heat transfer therein. The effect of inclination on natural convection in an enclosure has been discussed by several investigators ([1-2] among others). The first studies were devoted to the stability problem of such a flow [3]. Experimental research was undertaken for the estimation of Nusselt numbers, flow structures and critical Rayleigh (Ra) number. [4] revealed experimentally first a minimum and then a maximum heat transfer rate with increasing angle of inclination, but their work was limited to Rayleigh numbers of up to 10^4. Adjlout et al. [5] reported a numerical study of the effect of a hot wavy wall in an inclined differentially heated square cavity. Tests were performed for different inclination angles, amplitudes and Rayleigh numbers for one and three-undulations. The trend of the local heat transfer is wavy. The mean Nusselt number decreases comparing the square cavity. Hammady et al. [6] measured local and mean Nusselt numbers at various inclination angles and Rayleigh numbers from 10^4 to 10^6. They found a strong dependence of the heat flux on the inclination angle and Rayleigh number. They also presented limited numerical results for Rayleigh equal to 10^5. The use of different boundary conditions has shown that the flow and the heat transfer are seriously affected when the variable properties such as the thermal conductivity, the viscosity and the heat capacity are varied Zhong et al. [7]. Saidi et al. [8] presented numerical and experimental results of flow over and heat transfer from a sinusoidal cavity. They reported that the total heat exchange between the wavy wall of the cavity and flowing fluid was reduced by the presence of vortex. Vortex plays the role of a thermal screen, which creates a large region of uniform temperature in the bottom of the cavity.

Kumar and Shalini [9] investigated the effects of surface undulations on natural convection in porous enclosure with global cumulative heat flux boundary conditions for different undulation numbers and thermal stratification level. They indicated that the local Nusselt numbers are very sensitive to thermal stratification. Kumar [10] made a numerical work for the natural convection in a porous enclosure which their vertical wavy surfaces under constant heat flux. Das and Mahmud [11] analyzed the free convection inside both the bottom and the ceiling wavy and the isothermal enclosure. They indicated that, only at the lower Grashof number, the heat transfer rate rises when the amplitude wave length ratio changes from zero to other values. Recently, Dalal and Das [12] made a numerical solution to investigate the inclined right wall wavy enclosure with spatially variable temperature boundary conditions. The natural convection along a vertical wavy surface has been studied theoretically for the application of a cooling fin [13]. The result shows that the mean Nusselt number for a wavy surface is constantly smaller than

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that of corresponding flat plate. Indeed, in the renewable energy, some effort has been made in order to reduce the natural convection in the solar collector.

In this study, the effect of inclination is studied in detail for various Rayleigh numbers, ranging from 10^3 to 10^8, paying attention to their effect on the streamlines, isotherms, and local Nusselt number of the hot wall wavy. The dependence of the mean Nusselt number on the Rayleigh number is examined for angles ϕ equal to 0°, 30°, 60°, 90°, 120°, 150° and 180°. The effects of inclination angle on the Nusselt-Rayleigh studied for tow different geometrical (one and three undulations) which have Prandtl number equal to 0.71. The fluid used is air. The validity of this results proposed in this article is done through its application to natural convection in the square cavity. The number of papers available shows the great interest in this topic. The flow is laminar for Rayleigh numbers (Ra) less than 10^8. For the laminar flow [14-17], and for the transitional flow, [18], [19] and [16].

II. ANALYSIS

The problem treated is a two-dimensional heat transfer in an inclined square cavity. The hot wall is wavy with a constant temperature T_h, the cold wall is opposite to the latter with a constant temperature T_c while the other sides are insulated. The Rayleigh number is varied up to 10^8 while Prandtl number is fixed to be 0.71. Fig. 1 show the geometrical features of the cavity used.

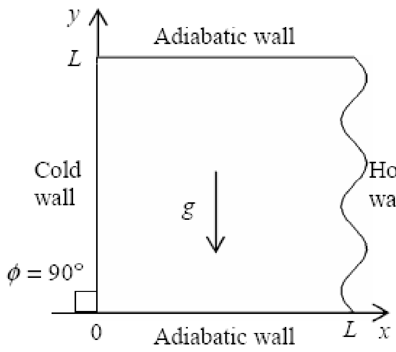

Fig 1. Geometrical features of the undulated cavity

A. *Mathematical Formulation*

The continuity, momentum and energy equation can be written for a two dimensional laminar flow of an incompressible Newtonien fluid. For these equations it is assumed that there is no viscous dissipation, the gravity acts in vertical direction, and fluid properties are constant. However, the Boussinesq approximation is accepted. Thus, governing equations are obtained as:

Continuity equation

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

Momentum Equation

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

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

Energy equation

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \quad (4)$$

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B. *Boundary Conditions*

The geometry of the laminar convection natural in a cavity is shown in Fig. 1. For this problem, the boundary conditions are:

$$\begin{aligned}
&u(x,0)=u(x,L)=u(0,y)=u(L,y)=0,\\
&v(x,0)=v(x,L)=v(0,y)=v(L,y)=0,\\
&T(0,y)=T_c,\text{on the cold wall,}\\
&T(L,y)=T_h,\text{on the hot wall,}\\
&\frac{\partial T}{\partial y}(x,L)=\frac{\partial T}{\partial y}(x,0)=0
\end{aligned} \quad (5)$$

Where x and y are the distances measured along the horizontal and vertical directions, respectively ; u and v are the velocity components in the x - and y - directions, respectively ; T denotes the temperature; v and α are kinematic viscosity and thermal diffusivity, respectively; p is the pressure and ρ is the density ; T_h and T_c are the temperatures at hot and cold walls, respectively; L is the side of the cavity. Using the following change of variables,

$$\begin{aligned}
&X=\frac{x}{L},\ Y=\frac{y}{L},\ U=\frac{uL}{\alpha},\ V=\frac{vL}{\alpha},\\
&P=\frac{pL^2}{\rho\alpha^2},\ \theta=\frac{T-T_c}{T_h-T_c},\ Pr=\frac{v}{\alpha},\\
&Ra=\frac{\beta g L^3\left(T_h-T_c\right)}{\alpha v}
\end{aligned} \quad (6)$$

The governing equations (1) - (4) reduce to non - dimensional form:

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

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

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

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

With boundary conditions

$$\begin{aligned}
&U(X,0)=U(X,1)=U(0,Y)=U(1,Y)=0,\\
&V(X,0)=V(X,1)=V(0,Y)=V(1,Y)=0,\\
&\theta(0,Y)=0,\text{on the cold wall,}\\
&\theta(1,Y)=1,\text{on the hot wall,}\\
&\frac{\partial \theta}{\partial Y}(X,1)=\frac{\partial \theta}{\partial Y}(X,0)=0
\end{aligned} \quad (11)$$

Here X and Y are dimensionless coordinates varying along horizontal and vertical directions, respectively; U and V are, dimensionless velocity components in the X- and Y directions, respectively; θ is the dimensionless temperature; P is the dimensionless pressure; Ra and Pr is Rayleigh and Prandtl numbers, respectively. The shape of the wavy vertical wall is taken as sinusoidal. The expression of the wavy wall is given by

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$$f(y) = a\,sin\left(\frac{2\pi y}{\lambda}\right) \quad (12)$$

The heat transfer coefficient in terms of the local Nusselt number is defined by:

$$Nu_L = -\frac{\partial\theta}{\partial n} \quad (13)$$

Where n-denotes the normal direction on a plane. The average Nusselt numbers at the hot wall are computed as follows

$$Nu_{av} = \int_0^1 Nu_L \, dY \quad (14)$$

Where the temperature gradient is obtained using a second order scheme.
The maximum horizontal velocity u_{max} is obtained for $y = 0.5$ at x_{max}; the maximum vertical velocity v_{max} is given at $y = 0.5$ and y_{max}.

C. *Numerical Solution Procedure*

The numerical method used to solve Eqs. (1) and (4) is the SIMPLE algorithm [20] (semi-implicit method for pressure linked equations). Five grid sizes are chosen for the present analysis, (21×21, 41×41, 61×61, 81×81 and 128×128). So a grid number of (41 × 41) is chosen for $10^3 \le Ra \le 10^6$ and (128 × 128) for $10^7 \le Ra \le 10^8$. The calculations were performed on Pentium IV, 1 Go RAM machine. The computer code based on the mathematical formulation discussed earlier and the SIMPLE method were validated for various cases published in the literature.

III. RESULTS AND DISCUSSION

A parametric study was carried out to determine the influence of the inclination angle (ϕ), the Rayleigh number Ra, the amplitude (a) and the number of undulations on the flow field of air. Inclination angles (ϕ) range of the cavity from 0° to 180°. The Rayleigh number was varied between 10^3 and 10^8 to cover the large range. The influence of the amplitude (a) was examined for the values 0.05, 0.06, 0.075 and 0.08. All these cases were computed for one and three undulations.
The details concerning the flow fields for the square heated cavity can be found in the literature. The transition from the motionless conduction dominated regime to the convection dominated regime takes place after $Ra = 10^3$. For $10^7 \le Ra \le 10^8$, the fluid motion takes place at the boundary layer near the walls. The results analysis will be divided in two parts: the laminar flow and the transitional flow.

A. *The Laminar Flow* $10^3 \le Ra \le 10^6$

Table 1 Reports the average Nusselt number at the hot wall, the maximum vertical velocity, the maximum horizontal velocity and the positions where they occur. The numerical results obtained in this paper are very close to the values found in the literature.

Table 1. Comparison of laminar flow with previous works for the square heated with imposed temperatures

		$Ra = 10^3$	$Ra = 10^4$	$Ra = 10^5$	$Ra = 10^6$
u_{max}	De ValDavis [14]	3.649	16.178	34.730	64.630
	Hotman et al. [15]	-	16.180	34.740	64.837
	Kuznik et al. [16]	3.636	16.167	34.962	64.133
	Shu et al. [17]	3.648	16.182	34.721	64.855
	Present work (1 und)	3.504	15.891	34.460	65.096
	Present work (3und)	3.469	16.030	35.522	68.874
y_{max}	De ValDavis [14]	0.813	0.823	0.855	0.850
	Hotman et al. [15]	-	0.825	0.837	0.850
	Kuznik et al. [16]	0.809	0.821	0.854	0.860

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	Shu et al. [17]	0.810	0.820	0.850	0.850
	Present work (1 und)	0.805	0.830	0.854	0.853
	Present work (3und)	0.806	0.829	0.852	0.851
v_{max}	De ValDavis [14]	3.697	19.617	68.590	219.360
	Hotman et al. [15]	-	19.629	68.639	220.461
	Kuznik et al. [16]	3.686	19.597	68.578	220.537
	Shu et al. [17]	3.696	19.628	68.462	220.072
	Present work (1 und)	3.570	19.464	63.488	196.724
	Present work (3und)	3.537	18.920	60.534	177.795
x_{max}	De ValDavis [14]	0.178	0.119	0.066	0.038
	Hotman et al. [15]	-	0.120	0.883	0.039
	Kuznik et al. [16]	0.174	0.120	0.067	0.038
	Shu et al. [17]	0.180	0.120	0.07	0.04
	Present work (1 und)	0.195	0.195	0.073	0.049
	Present work (3und)	0.195	0.146	0.097	0.065
Nu_a	De ValDavis [14]	1.117	2.238	4.509	8.817
	Hotman et al. [15]	-	2.244	4.521	8.825
	Kuznik et al. [16]	1.17	2.246	4.518	8.792
	Shu et al. [17]	1.18	2.244	4.519	8.814
	Present work (1 und)	1.119	2.254	4.622	9.122
	Present work (3und)	1.127	2.252	4.577	8.945

B. *Transitional flow* $10^7 \leq Ra \leq 10^8$

Table 2 summarizes the numerical values of Nu_a, u_{max}, v_{max} and y_{max} compared with literature results. From this table, the numerical values are in good agreement with those from literature.

C. *Local Heat Transfer*

Figure 2 shows the distribution of local Nusselt number on the wavy wall for different amplitudes for the two configurations (one and three undulations) where we note that the shape of the profiles is the same as that of the undulation. For the first configuration Fig. 2 (a) reveals that the profiles have the same allure. For $0 \leq y \leq 0.15$ and $0.4 \leq y \leq 0.9$ the Nusselt number increases slightly when the amplitude decreases, for $0.15 < y < 0.40$ the Nusselt number increases slightly when the amplitude increases. From $y = 0.9$ the Nusselt number is almost the same value.

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Table 2. Comparison of Transitional flow with previous Works for the square heated with imposed temperatures.

		$Ra = 10^7$	$Ra = 10^8$
u_{max}	Markatos et al. [18]	-	514.3
	Le Qyuéré [19]	148.850	321.876
	Kuznik et al. [16]	148.768	321.457
	Present work (1 und)	157.186	355.359
	Present work (3und)	169.155	359.394
y_{max}	Markatos et al. [18]	-	0.941
	Le Qyuéré [19]	0.879	0.928
	Kuznik et al. [16]	0.881	0.940
	Present work (1 und)	0.872	0.926
	Present work (3und)	0.862	0.913

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v_{max}	Markatos et al. [18]	-	1812
	Le Qyuéré [19]	699.236	2222.39
	Kuznik et al. [16]	702.029	2243.36
	Present work (1 und)	633.728	1884.01
	Present work (3und)	503.469	1375.03
x_{max}	Markatos et al. [18]	-	0.0135
	Le Qyuéré [19]	0.021	0.120
	Kuznik et al. [16]	0.020	0.121
	Present work (1 und)	0.012	0.0123
	Present work (3und)	0.037	0.0123
Nu_a	Markatos et al. [18]	-	32.045
	Le Qyuéré [19]	16.523	30.225
	Kuznik et al. [16]	16.408	29.819
	Present work (1 und)	17.491	34.801
	Present work (3und)	17424	35.041

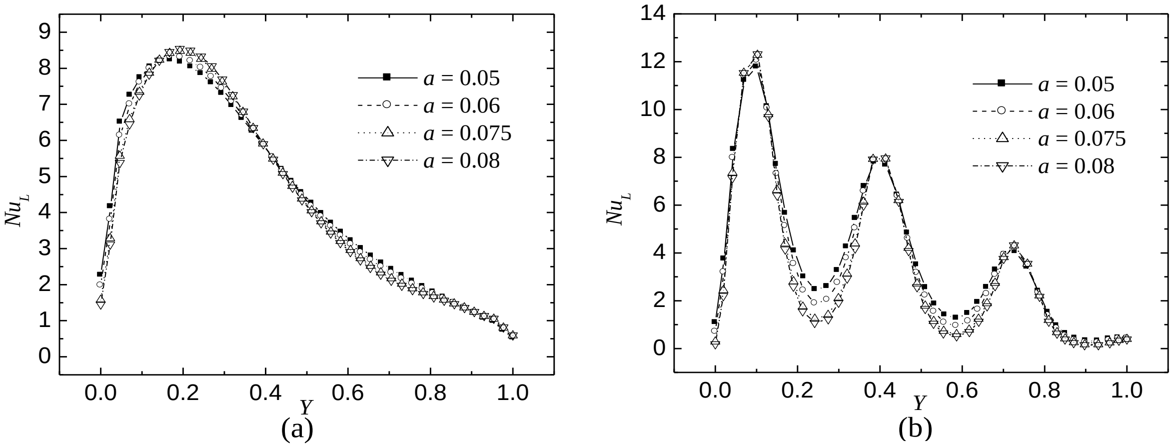

Fig 2. Variation of local Nusselt number for $\phi = 90°$ and $Ra = 10^5$: (a) one undulation ; (b) three undulations.

Figure 3 shows the influence of inclination on the variation of local Nusselt number along the wave for the two configurations tested. In the case of a single undulation (Fig. 3a) reveals that for a tilt angle ϕ varying from 30° to 150 ° Nusselt curves have the same shape and exhibit a maximum in the part of undulation outward. The greatest value of this maximum corresponds to $\phi = 90°$, this result was also found by [5]. For $\phi = 180°$ the maximum Nusselt number is located in the middle of the undulation, while for $\phi = 0°$ the shape of the Nusselt number variation is that of the undulation. For the second configuration (three undulations, Fig. 3 (b)) the remarks for a single ripple remain valid. The number of the peaks in the local Nusselt number curves corresponds to the undulation number of the hot wall.

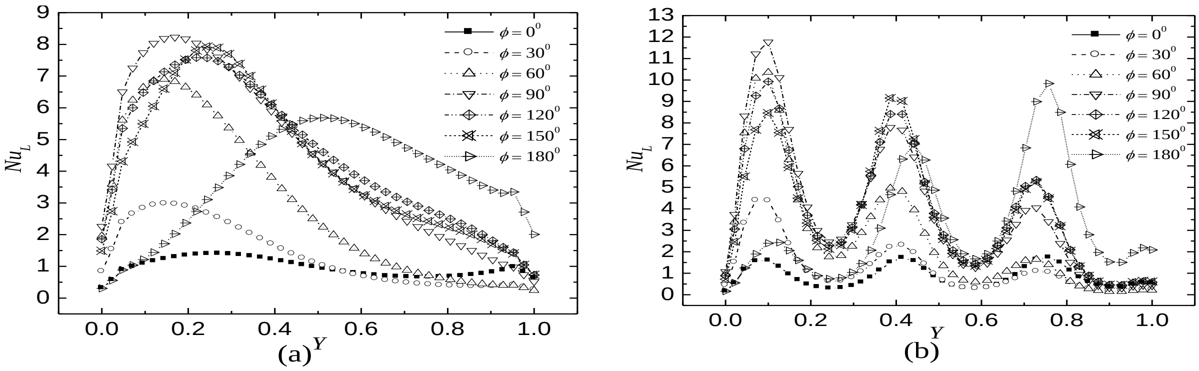

Fig 3. Effect of angle of inclination ϕ on the local Nusselt number for $Ra = 10^5$ and $a = 0.05$: (a) one undulation; (b) three undulations.

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D. *Local Number Nusselt Comparison*

Figure 4 shows a comparison between our results and those of Adjlout et al. [5] for the variation of local Nusselt number based on the height of the cavity. We note that the shape of the variation of Nusselt obtained is different from [5] because of the mathematical expression describing the wavy wall. The maximum Nusselt obtained is higher than that of [5] with a difference of 22.64%.

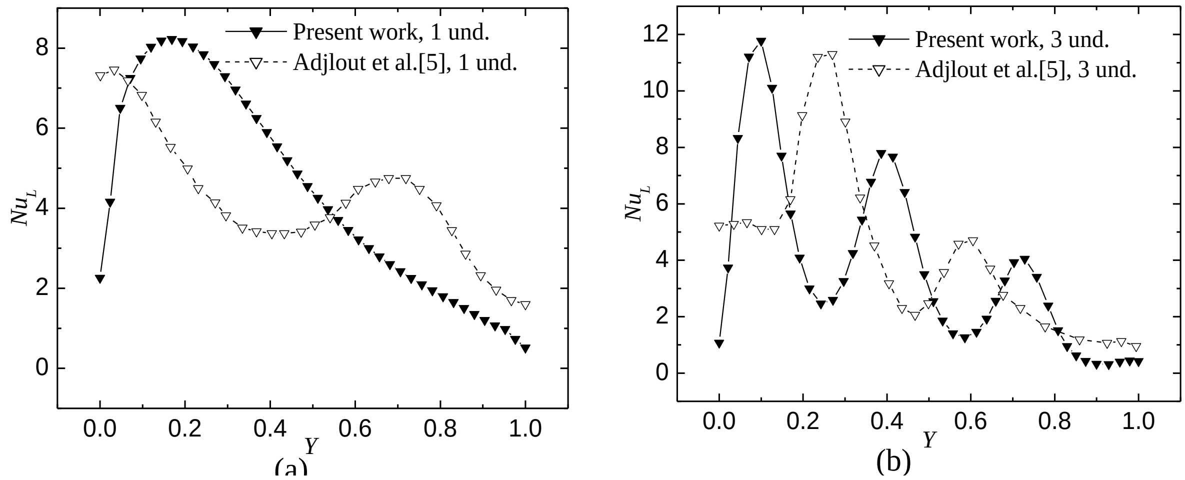

Fig 4. Comparison of local Nusselt number between present work and [5] for $\phi = 90°$, a = 0.05 and Ra=10^5 : (a) for one undulation ; (b) three undulations.

E. *Effect of Amplitude*

Figure 5 shows the influence of Rayleigh number and amplitude on the average Nusselt number. We found considerable influence of both parameters on the latter, the average Nusselt number increases as the Rayleigh number increases regardless of the location and type of orientation of the undulation. There is an increase in the average Nusselt number with increasing of amplitude; we can explain the dominance of convective effects undulation outward-oriented over a wave directed inwards. Fig. 5a obtained for a single undulation, shows a superposition of all the curves up to $Ra = 10^6$, beyond was a slight increase in the Nusselt number for a = 0.05. For $Ra = 10^8$ the Nusselt number takes the same value regardless of amplitude. Fig. 5b obtained for three undulations shows that the Nusselt number increases with the number of Rayleigh. We note that the amplitude has no effect on the Nusselt number.

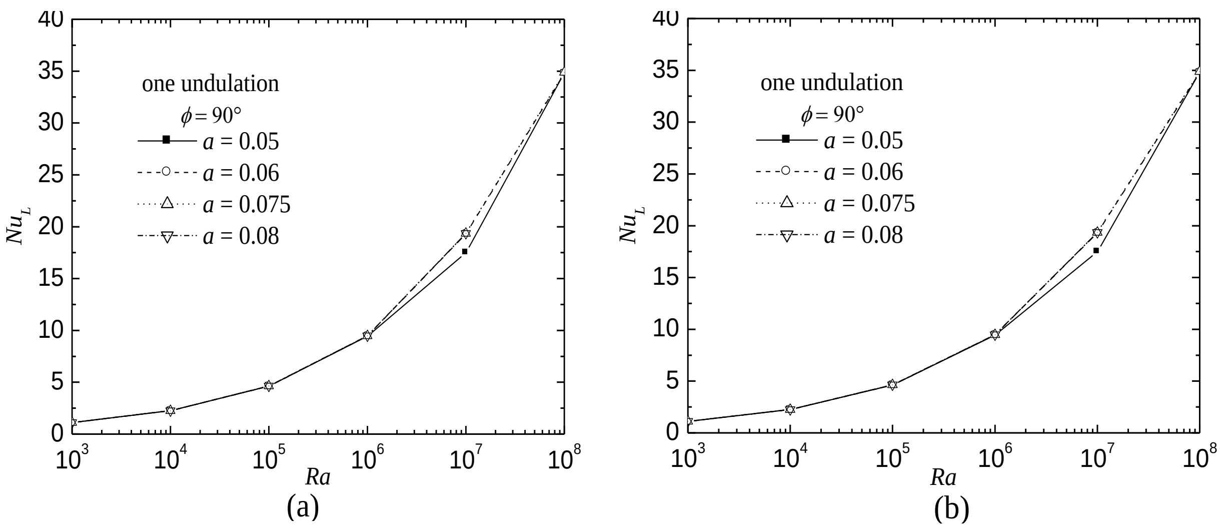

Fig 5. Variation of local Nusselt number with Rayleigh number for different amplitude and for $\phi = 90°$.

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F. *Average Nusselt Number*

Figure 6 compares the results obtained in this work and those of Adjlout et al. [5] and Zhong et al. [7] on the variation of Nusselt number as a function of the inclination for $Ra = 10^5$. We note that the shape of the variation of Nusselt obtained is the same as that of [5] and [7] and is very close to that of [7]. Between $0° \le \phi \le 90°$ and $140° \le \phi \le 160°$ the curve of variation of Nusselt number obtained is located above those of [5] and [7], for $90° \le \phi \le 140°$ and $\phi \ge 160°$ the shape obtained is below that of [5] and [7].

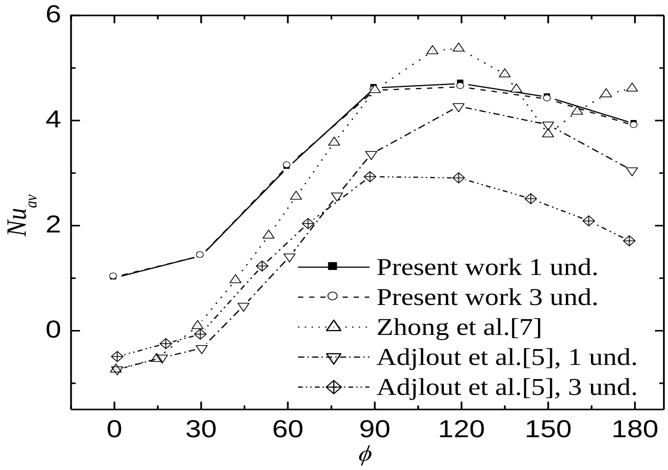

Fig 6. Comparison of the average Nusselt number at $Ra = 10^5$**.**

Figure 7 display a comparison of the average Nusselt number at the number of Rayleigh between the configuration of our study area (one and three undulations) and that of Adjlout et al. [5] and Zhong et al. [7] for an angle of inclination of 90°. This comparison is for a Rayleigh number up to 10^5. The influence of the wall undulations is clearly seen in the latter figure by a clear decrease in Nusselt number comparing with the square cavities. The difference between the averaged Nusselt number of [7] and the cavity with three undulations increases with an increase in the Rayleigh number. On the other hand, the configuration with one undulation has a mean Nusselt number higher than the configuration with three undulations.

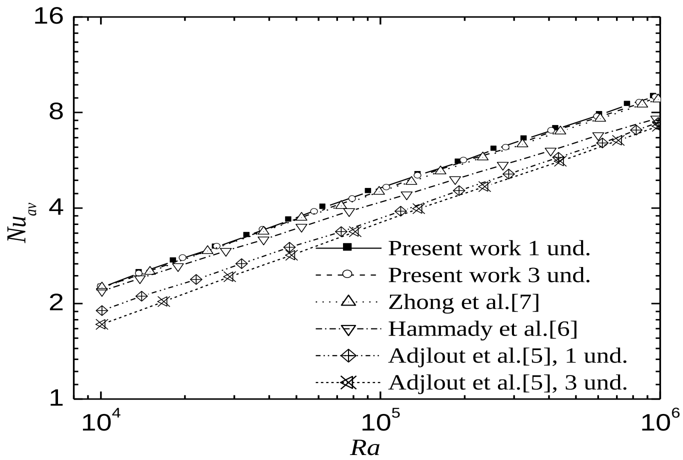

Fig 7. Comparison of the average Nusselt number results for $\phi = 90°$**.**

G. *Influence* of the amplitude of the undulation

Figure 8 shows the influence of amplitude on the variation of Nusselt and the length of undulations for both configurations. It is found that the average Nusselt number increases slightly with increasing amplitude in the case of a single undulation Fig. 8a, and decreases in the case of three undulations. Fig. 8b shows that the length of the undulation increases with the amplitude for the two configurations. This leads to an increase in surface heat

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exchange and therefore a decrease in the number of average Nusselt number.

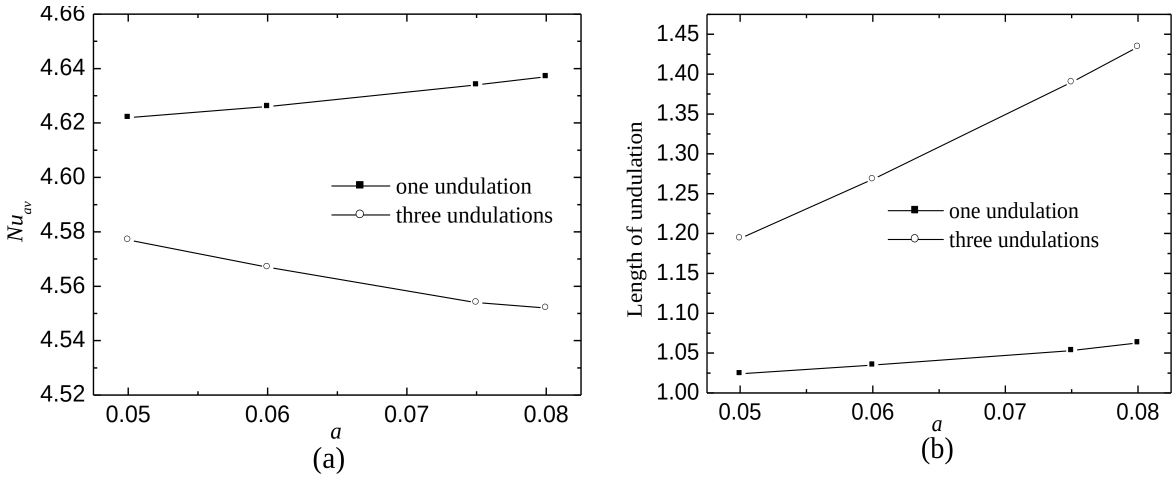

Fig 8. Variation of: (a) average Nusselt number and (b) length of undulation with the amplitude for $Ra = 10^5$ and $\phi = 90°$.

IV. CONCLUSION

In this work we have studied numerically the two-dimensional natural convection in laminar flow in a cavity in one and three undulations. The effect of the amplitude of the undulation and the angle of inclination of the cavity was analyzed. This simulation was done with the CFD code FLUENT). Dynamic validation of this simulation showed good agreement between our results and those of [14-19]. The results show that:

- The only wall undulation has an advantage over the three undulations. Indeed, for the same amplitude and same angle of inclination of the cavity, increasing the Nusselt number with Rayleigh number in the case of a single undulation is more than three significant undulations.
- The angle of inclination has a significant influence on the Nusselt number for both configurations tested in contrast to the amplitude.
- A comparison of results with those of Adjlout et al. [5], shows that the variation of Nusselt number depends on the mathematical expression describing the shape of the wave.

NOMENCLATURE

a : amplitude height of the wavy wall, [m]
g : magnitude of gravitational, [ms^{-2}]
Gr: Grashof number ($g\beta L(T-T)/\nu$), [–]
L : cavity height/width, [m]
Nu_L : Nusselt local number, [–]
Nu_{av} : average Nusselt number, [–]
P : dimensionless pressure, [–]
p: pressure, [Pa]
Pr: Prandtl number, [–]
Re : Reynolds number, [–]
Ri : Richardson number, [–]
T : temperature, [K]
u, v: velocity components along x and y respectively [ms^{-1}]
U, V: dimensionless velocity components along X and Y, respectively, [–]
U_{Lid} : sliding left wall velocity, [ms^{-1}]
x, y: co-ordinates, [m]
X,Y: dimensionless co-ordinate, [–]

Greek symbols

α : thermal diffusivity, [m^2s^{-1}]
β : thermal dilatation coefficient, [K^{-1}]
ν : kinematic viscosity, [m^2s^{-1}]
ρ : fluid density, [Kg m^{-3}]
ϕ: inclination angle, [°]
θ : dimensionless temperature, [–]
λ : wave length of the wavy wall, [m]

Subscripts

c : cold wall
h : hoot wall

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