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Numerical study of mixed convection in lid-driven cavity with traingular alveolus

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***Abstract*—In this work, the main objective is to study numerically laminar mixed convection in a lid-driven rectangular air filled cavity, provided with a triangular alveolus on the bottom wall. The finite volume method is used to analyze the influence of the number of the alveolus with different Grashof and Reynolds numbers on the flow field and heat transfer characteristics. Numerical simulations are discussed in terms of the streamlines, the temperature distributions and the average Nusselt number along the bottom wall. The results show a close relationship between the flow régime through the Grashof and Reynolds numbers. Furthermore, the number of alveolus (N) have a significant effect on the thermal and flow fields inside the enclosure.**

Keywords—mixed convection; driven cavity; alveolus; finite volume method

I. INTRODUCTION

Convection heat transfer in enclosures received considerable attention during the last three decades owing to its importance in many practical applications particularly in solar collectors, lubrication systems, electronic equipment cooling, solar energy systems, and crystal growth.

In the literature, there is a significant number of investigations on the convection in closed cavities. Among these studies include one conducted by De Vahl Davis [1] for a numerical study on natural convection in a square cavity, it provided a set of reference solutions stable for Rayleigh number $\leq 10^6$. Sarris et *al*. [2]**,** studied the influence of Rayleigh number and aspect ratio in a rectangular cavity with a sinusoidal temperature on the top wall. Cheng [3] has provided a study of the problem of lid-driven square cavity for several Richardson and Prandtl numbers. Basak et *al*. [4] have shown the existence of a transition between the conduction and convection with increased Gr number and natural to forced convection with increased Re number. Öğüt [5] showed that the angle of inclination of the cavity affects the heat transfer and flow for large numbers of Ri. Torrance et *al*. [6] have studied the fluid flow in a rectangular driven cavity, their results indicate that natural convection predominates for higher Grashof numbers. In summary, it is namely revealed that the core flow is highly influenced by the mechanical motion of the walls which in turn affect significantly the heat transfer.

Other studies have included an alveolus or partitions on one of the wall of the enclosure. As Yang et *al*. [7] have studied experimentally cooling of a dimpled surface, the results show that the heat transfer improves with the increase in the curvature of the wall. Aminossadati et *al*. [8] studied the mixed convection in a horizontal channel provided with alveolus and heat source, the results show that the heat transfer rate increases when the heat source is located on the right side wall. Also, Allegrini et *al*. [9] studied the mixed convection in a channel with a small alveolus on the bottom wall. Other have studied the mixed convection in cavities mounted with blocks, such as Najam et *al*. [10] and Amahmid et *al*. [11]. Dogan et *al*. [12] conducted a study of the combination of natural convection and radiation for several different forms of fins mounted on a plate. Chen and Cheng [13] investigate the steady laminar buoyancy-driven and convection heat transfer characteristics within three different across-shape concave enclosures. Also, Xia et *al*. [14] studied numerical simulation of heat transfer in three-dimensional micro channel contains circular alveolus, their results led to the selection of two ranges optimal configurations. Al-Salem et *al*. [15] showed that the heat transfer decreases with increasing of magnetic field.

It's worthy of note that there are other interesting works on this subject with various other applications. An overview of this topic can be found in the publications of Moraga [16], Prasad [17], Al-Salem [18], Cheng [19], Muthtamilselvan [20] and Kefayati [21].

Based on the literature survey conducted by the authors and up to their knowledge, there has not been any attention given to investigate the momentum and energy transport processes in a shaped enclosure with a moving lid subjected to a sinusoidal temperature on the heat transfer characteristics for various pertinent parameters such as Reynolds and Grashof numbers.

II. DESCRIPTION AND FORMULATION OF THE PROBLEM

The physical problem considered in this study is the 2D flow heat fields in a concave shaped enclosure filled

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with air (Pr=0.71) (Fig. 1). It consists in a lid driven cavity with two vertical and thermally insulated sides and two horizontal active surfaces. The upper wall subjected to a sinusoidal temperature moves from left to right with a constant velocity u_o while the shaped bottom wall is assumed to be isothermal at a warm temperature T_h. In these figures, H_1, L_1 and H_2, L_2 shows the height and the length of the enclosure and the micro-cavity (alveolus), respectively. For commodity, these parameters are defined in terms of geometric ratios such as: $a = H_1/L_2=0.25$; $b = L_1/L_2=1.5$, and $C= L_2/H_2$. Knowing that each value of C (4, 2, 1 and 3/4), correspond to a configuration with a specific number of alveolus (N=1, 2, 3 and 4). The sinusoidal temperature is expressed as follow: $T = T_c+(T_h-T_c)\sin(\pi x/L_o)= T_c+(T_h-T_c)\sin(\pi X)$.

The numerical model for heat transfer and fluid flow in the partitioned rectangular enclosure was developed under some assumptions as steady state, laminar and incompressible Newtonian fluid. Viscous dissipation and compressibility effects are neglected. Also, the fluid properties are assumed constant except the density in the buoyancy term of the momentum equations, which can be approximated by the standard Boussinesq model. The mathematical formulation governing the two dimensional fluid flow and heat transfer can be written on dimensionless form in Cartesian coordinates (X, Y) as follow:

Continuity equation:

$$\left(\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}\right)=0 \quad (1)$$

Momentum equations:

$$\left(U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}\right)=-\frac{\partial P}{\partial X}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) \quad (2)$$

$$\left(U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}\right)=-\frac{\partial P}{\partial Y}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+\frac{Gr}{\mathrm{Re}^2}\theta \quad (3)$$

Energy equation:

$$\left(U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}\right)=\frac{1}{\mathrm{Pr\,Re}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \quad (4)$$

Assuming the non-slip flow, the relevant dimensionless boundary conditions can be written as follows:

$$\begin{array}{lll} \theta=\sin(\pi X) & U=1,\ V=0 & \text{(moving lid : top wall)} \\ \partial\theta/\partial X=0 & U=V=0 & \text{(upper side walls)} \\ \theta=1 & U=V=0 & \text{(bottom wall)} \end{array} \quad (5)$$

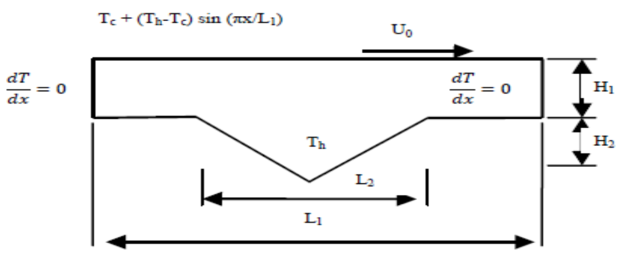

Fig. 1. Schematic diagram of physical domain.

Table 1. Grid indenpendency.

Grid	Average Nusselt
60x60	2.350
80x80	2.363
100x100	2.363
120x120	2.368
140x140	2.369
160x160	2.311

Table 2. Comparison of average Nu with [5].

Average Nu	Ögüt [5]	Present work
Ri=0.1	3,850	3,890
Ri=1	2,602	2,631
Ri=10	2,139	2,163
Ri=100	1,884	1,906

The dimensionless quantities appearing in Eqs. (1)-(4) are the Grashof, Reynolds and Prandtl numbers respectively, defined as : $\mathrm{Gr}=g\beta(T_h-T_c)L_o^3/(\nu^2)$, $\mathrm{Re}=u_oL_o/\nu$ and $\mathrm{Pr}=\nu/\alpha$. Further, one can also define a dimensionless quantity so called Richardson number as follows $Ri=Gr/Re^2$. In the above equations, P, θ are the dimensionless pressure and temperature, while (X, Y) and (U, V) are the dimensionless Cartesians coordinates and corresponding velocity components, respectively.

III. Numerical approach

The mass, momentum and energy balance equations (1)-(4) subjected to the specified boundary conditions Eqs. (5) are solved numerically using a developed solver based on a control-volume method under non-uniform grid system in x and y directions. The described solver uses a pressure correction based on iterative SIMPLER algorithm (for more details, see [22]). To check the convergence of the sequential iterative solution, the normalized residual is calculated for the mass, momentum and energy equations. The convergence is obtained when the residual becomes smaller than 10^{-7}.

A. Grid dependency

A curvilinear grid was generated to solve the problem treated. Reliable results have been obtained with various grid combinations (60×60 to 160×160). For each grid size, average Nusselt number is calculated and summarized in Table 1 for $Gr=10^4$ and Re=100. Throughout this investigation, average Nusselt remains almost the same for grids finer than 100x100 witch satisfies the grid dependency. Hence, considering both the accuracy and the computational costs, most computations reported in the current work were performed with a multiple grid system of 120x120.

B. Code validation

In this step, we validated the results obtained in the case of driven square cavity for Öğüt [5] with different Richardson numbers Ri = 0.1, 1, 10 and 100, the comparison is made with

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the average Nusselt number. According to the Table 2, our results show excellent agreement. This allows validating our numerical simulation procedure.

IV. RESULTS AND DISCUSSION

In this section, simulations have been carried out for different number of alveolus N (1, 2, 3, 4). Further, the Grashof number varies in the range, i.e., 10^3 - 10^7 and Reynolds number varies from 10 to 1000 will be investigated.

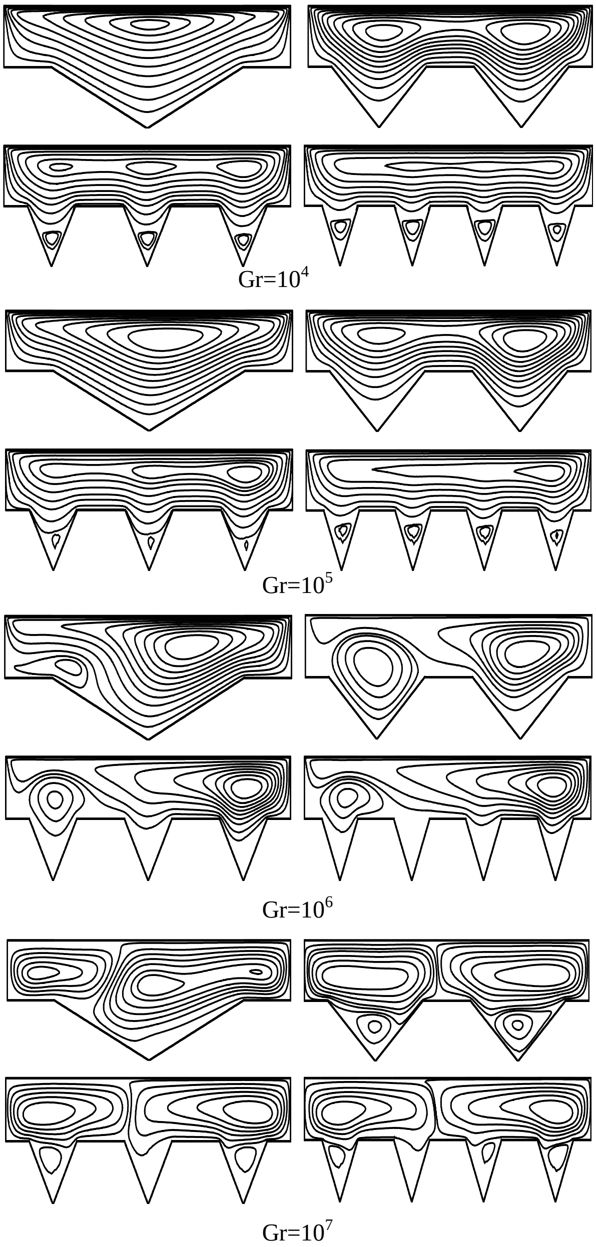

Fig. 2. Streamlines contours for differents N=1, 2, 3, 4 and Gr numbers.

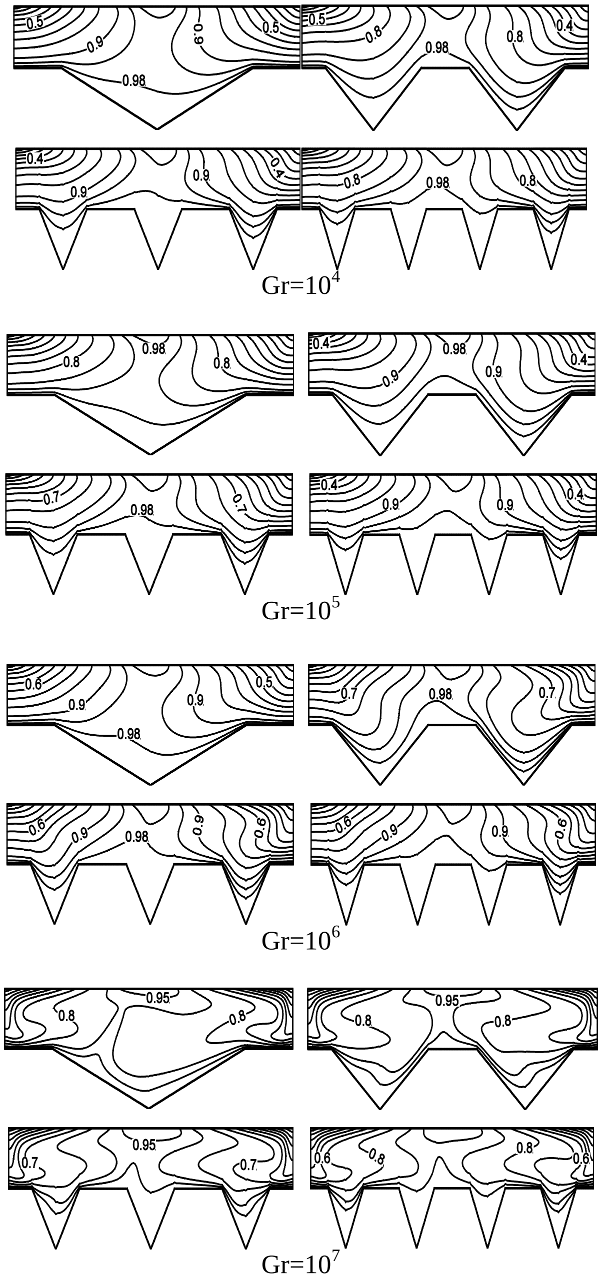

Fig. 3. Isotherms contours for differents N=1, 2, 3, 4 and Gr numbers.

A. *Flow and thermal fields*

Representatives isotherms and streamlines plots are shown in Figs. 2-3, for different number of alveolus and Gr number with Re=100. At low Grashof number, $Gr=10^4$, The fluid flow is characterized by a single clockwise cell extending over the whole of the cavity with a secondary anti-clockwise circulation vortices for each micro-cavity respectively. As Gr increases, the secondary recirculation formed in the bottom right corner of the micro-cavity (induced by the buoyancy force) begins to grow. When the $Gr=10^7$, the flow fields becomes two pairs of counter-rotating circulation eddies, signifying that natural convection transfer is the dominant mode transfer.

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Isothermal lines extending between the cold surface and the adiabatic surfaces are normal to the insulated walls in accordance with the literature without surface radiation. For relatively low Grashof numbers, the isotherms plots are smooth curves indicating that the conduction is the dominant heat transfer mechanism. The distributions of isotherms are almost invariant. However, the increase in the Grashof number ($\geq 10^6$) caused by the increased buoyancy forces alters the flow pattern so the isotherms are distorted indicating that, the convective heat transfer is the dominant mode. Consequently, the heat convection is weakened significantly when Gr is highest.

As the number of alveolus (N) increases, the minor cells with lower intensity are formed in each micro-cavity. Moreover, with the increase of the number of alveolus (N), the core of the main circulation stretched and flattened, and the isotherms are exclusively at the upper part of the cavity, subsequent increase in N (number of alveolus) reduces the convection currents in the alveolus.

B. Heat transfer

The analysis of the intensity of heat exchange within the cavity is made through a dimensionless average (Nu) Nusselt number through the hot bottom wall, deduced from the integration of the local quantity (Nu_x) :

$$\overline{Nu} = \int_0^L Nu_X \, dX \qquad Nu_X = -\frac{\partial \theta}{\partial Y}$$

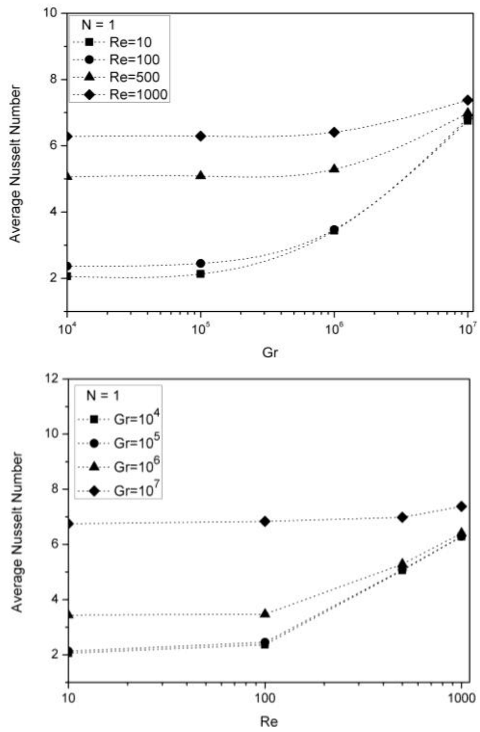

Fig. 4. Average Nusselt numbers versus Gr and Re numbers for N=1 (single alveolus).

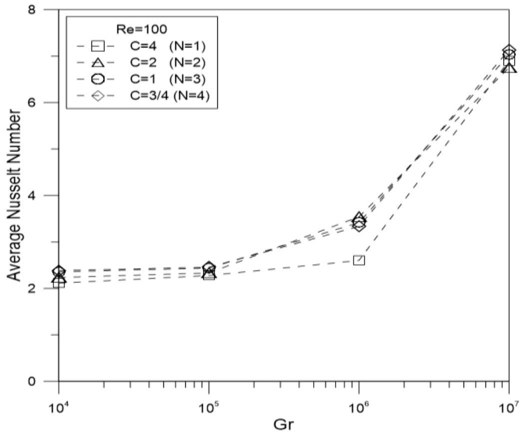

Fig. 5. Average Nusselt numbers versus Gr number for different number of alveolus (N) at Re=100.

Figure 4 illustrates the variation of the average Nusselt number on the hot bottom wall, with various Grashof and Reynolds numbers fo number of alveolus N = 1. When the heat transfer is only due to conduction (Gr < 10^5 and Re < 100), the Grashof (Reynolds) number does not entail any significant change and the mean Nusselt number remains unchanged. Also, when the heat transfer is partly or mainly due to convection, the mean Nusselt is found to be affected significantly by the Grashof Number. Indeed, increasing Gr produces the higher buoyancy-induced flow within the enclosure.The Increasing Re due to high lid speed, the convective transport grows, which lead to better mixing of the fluid, and consequently the higher heat transfer rate. The mixed convection plays the dominant role over the free convection.

Furthermore, Fig. 5 depicts the effect of number of alveolus on the average Nusselt number, As the number N increases the average Nu increases slowly, indicating that the heat transfer is not susceptible to changes of parameter C, i.e. the number of alveolus (N).

V. CONCLUSION

Steady mixed convection flow and heat transfer in a lid-driven rectangular cavity with the presence of triangular alveolus (micro-cavity) under laminar regime have numerically investigated. A detailed analysis for the distribution of streamlines, isotherms, average Nusselt number along the bottom hot wall were carried out to investigate the effect of the dimensionless parameters and aspect ratio such as ; Reynolds number, Re, Grashof number, Gr and number of alveolus, C (N). From this numerical study, the following major conclusions have been drawn:

- For low Grashof number, the flow behavior is mono-cellular although a smalls vortices appear in the micro-cavity, as the Gr increases gradually, the flow becomes bi-cellular and increases in the distortion of isotherms lines, Due to buoyancy force dominance.
- The increase of the number of alveolus (N) at the heated bottom wall causes small vortices appear at alveolus in addition to the large vortices occupying the

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whole cavity and the isotherms back slightly at the lower part of the cavity.

- Beyond two alveolus (N = 2), the average Nusselt number do not change substantially with number of micro-cavities (N).
- Moreover, the mixed convection is very sensitive to the variation of the flow regime through the Re and Gr numbers, the increase of the latter have a direct impact on increasing of the thermal exchanges in the enclosure.

In summary, concave enclosure may play an important role in the flow and heat transfer characteristics. Further, this study suggest several guidelines for thermal design in engineering process and emphasizes the interest of the alveolus shape for a better thermal efficiency.

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Nomenclature

a, b	aspect ratio $a = H_1/L_2$, $b = L_1/L_2$
C	Aspect ratio, $C = L_2/H_2$
C_P	specific heat , J. kg^{-1}. K^{-1}
H_1, H_2	height of the cavity and the alveolus, m
L_1, L_2	width of the cavity and the alveolus, m
g	gravitational acceleration, m.s^{-2}
Gr	Grashof number
N	Number of alveolus
Nu	Nusselt number
k	thermal conductivity, W.m^{-1}. K^{-1}
p	Pressure, Pa
P	dimensionless pressure
Pr	Prandtl number
Re	Reynolds number
Ri	Richardson number
T	Temperature , K
u, v	velocity components in x, y directions, m.s^{-1}
U,V	dimensionless velocity components
U_0	velocity of the top wall, m.s^{-1}
x, y	Cartesians coordinates, m
X, Y	dimensionless coordinates

Greek symbols

α	thermal diffusivity, m^2. s^{-1}
β	thermal expansion coefficient, K^{-1}
θ	dimensionless temperature
ν	kinematic viscosity, m^2. s^{-1}

Subscripts

h	hot temperature
c	cold temperature

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