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# Effect of inclined Wavy Surface on Heat Transfer Inside a Rectangular Cavity: Solar Applications

Younes Agrouaz\*, Tarik Bouhal, Abdelmajid Jamil,  
Abdelouahad Ait Msaad  
Université Sidi Mohamed Ibn Abdelah  
Ecole Supérieure des Technologies, EST  
Fès, Morocco  
agrouazyounes@Gmail.com

Tarik Kousksou, Mustapha Mahdaoui  
Laboratoire des Sciences de l'Ingénieur Appliquées  
à la Mécanique et au Génie Electrique (SIAME)  
Université de Pau et des Pays de l'Adour  
Pau, France

Tarik El Rhafiki  
Université Moulay Ismail  
Ecole Nationale Supérieure d'Arts et Métiers, ENSAM  
Meknès, Morocco

**Abstract**— In this present work a steady-state natural convection was numerically simulated in an inclined rectangular cavity with a sinusoidal bottom wavy wall. The vertical walls are insulated while the bottom surface maintained to higher temperature than the top surface. In this numerical simulation, Rayleigh number ( $10^3$ ,  $10^6$  and  $6 \times 10^6$ ) and inclination angle ( $30^\circ$ ,  $60^\circ$  and  $90^\circ$ ) were chosen in order to analyze the effect of these parameters on the heat transfer and the flow fields in two dimensional(2-D) enclosure filled in air ( $Pr=0.71$ ). It is found that the same Rayleigh number, the heat transfer is influenced by the inclination angle. In other words the increase of inclination angle leads to an increase in the heat transfer inside the rectangular cavity.

**Keywords**-natural convection; numerical simulation; heat transfer.

## I. INTRODUCTION

Natural convection flow and heat transfer characteristics in a rectangular cavities have received an important attention in the past decades. This importance due to its thermal performances in several applications such as cooling of electronic devices, food processing, drying technologies, solar collectors, energy storage systems, cooling technologies [1-2].

In a cavity, when the two vertical walls are differentially heated and the top and bottom walls are maintained under adiabatic conditions. The density gradient (due to temperature gradient) is horizontal and the gravity vector acts perpendicularly. These two vectors act normal to each other and the direction of the circulation depends upon their orientation. However, the situation becomes more complex when these two vectors are parallel to each other. When the bottom wall is heated and the top wall is cooled, the density increases from bottom to top.

These two vectors are parallel and opposite to each other. In this case the circulation will start after a critical Rayleigh number is reached [3]. The literature have widely studied a natural convection in an enclosures for square, rectangular and inclined walled geometries, but there is a limited number of studies with wavy inclined walled enclosures due to the

complexity of the fluid hydrodynamics inside the cavity. The complexity of this physical phenomena is a consequence of the geometry and the boundaries conditions. For example Amaresh and Manab [4] have given a parametric study of the influence of Rayleigh number, amplitude of undulation and number of undulations on the flow and the heat transfer inside the cavity. Catton [5] and Yang [6] have shown the advantage of inclined cavities on the thermal behavior of the flow regimes. On the other hand Yasin and Oztop [7] studied numerically a natural convection heat transfer in a horizontal and wavy enclosure. They showed that the heat transfer is increased with the decreasing non-dimensional wave length and Rayleigh number.

In this paper, the main objective is to extend our comprehension to the laminar natural convection in a rectangular cavity having three flat walls and the bottom wall consisting of three undulations of amplitude 0.10. The two vertical walls are maintained adiabatic and the top wall are maintained at a fixed lower temperature than the sinusoidal bottom wall.

## II. PHYSICAL MODEL AND GOVERNING EQUATIONS

### A. Description of schematic model

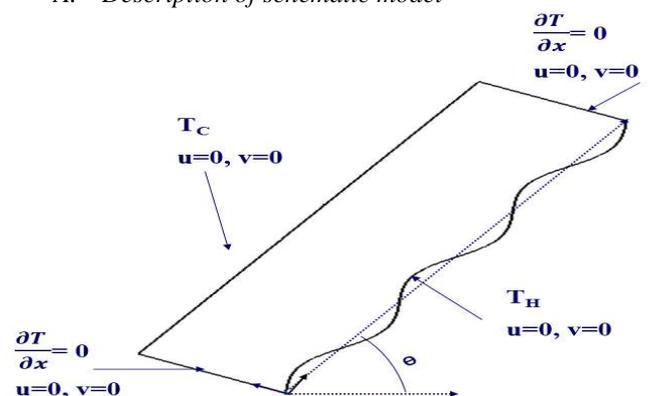


Figure.1 The description of the geometrical configuration

## NOMENCLATURE

$g$	gravity vector (m s <sup>-1</sup> )
$A_P$	linearized coefficient
$c$	specific heat capacity (J K <sup>-1</sup> Kg <sup>-1</sup> )
$Pr$	Prandtl number
$Ra$	Rayleigh number
$T$	temperature (°C)
$P$	pressure (Pa)
$u, v$	velocities (m s <sup>-1</sup> )
$x, y$	coordinates(m)
$n$	unit normal vector
$S$	surface (m <sup>2</sup> )
$V$	volume (m <sup>3</sup> )
$t$	time (s)
Greek symbols	
$\Theta$	inclination angle (°)
$\beta$	thermal expansion coefficient K <sup>-1</sup>
$\lambda$	thermal conductivity (W m <sup>-1</sup> K <sup>-1</sup> )
$\rho$	density of the fluid (Kg m <sup>-3</sup> )
$\mu$	dynamic viscosity (Kg m <sup>-1</sup> s <sup>-1</sup> )
$\phi$	transport variable
$\Gamma$	diffusion coefficient
Subscripts	
$C$	cold
$H$	hot
$i$	initial

The schematic configuration of a wavy inclined wall is given in Fig. 1. The boundaries condition are also plotted on this figure showing that inclined adiabatic wall. The cold and hot wall correspond respectively to the top and bottom of the cavity. The geometrical characteristic such as wave length, height and width are also depicted on Fig. 1.

### B. Mathematical model

The following assumptions are made in the analysis

- The Boussinesq approximation is valid, i.e., liquid density variations arise only in the buoyancy source term, but are otherwise neglected.
- The air is Newtonian.
- Viscous dissipation is neglected.
- Fluid motion in the melt is laminar and two-dimensional.

With the foregoing assumptions, the conservation equations for mass, momentum and energy may be stated as

$$\int_S \bar{u} \cdot \bar{n} dS = 0 \quad (1)$$

$$\frac{d}{dt} \int_V \rho \bar{u} dV + \int_S \rho \bar{u} \bar{u} \cdot \bar{n} dS = - \int_V \bar{\nabla} p dV + \int_S \bar{\tau} \cdot \bar{n} dS + \int_V \bar{A}_U dV \quad (2)$$

$$\frac{d}{dt} \int_V \rho c_p T dV + \int_S \rho c_p T \bar{u} \cdot \bar{n} dS = \int_S \lambda \bar{\nabla} T \cdot \bar{n} dS \quad (3)$$

Where  $\bar{u}$  is the velocity vector,  $p$  the pressure and  $T$  the temperature.  $\bar{\tau}$  is the viscous stress tensor for a Newtonian fluid:

$$\bar{\tau} = \mu \left( \bar{\nabla} u + (\bar{\nabla} u)^T \right) \quad (4)$$

The integration occurs over a control volume (CV) surrounded by a surface  $S$ , which is oriented by an outward unit normal vector  $\bar{n}$ . The source term in Eq. (2) contains two parts:

$$\bar{A}_U = \rho \beta (T - T_m) \bar{g} \quad (5)$$

where  $\beta$  is the coefficient of volumetric thermal expansion and  $\bar{g}$  the acceleration of gravity vector. The first part of the term source represents the buoyancy forces due to the thermal dilatation.  $T_m$  is the reference temperature.

## III. NUMERICAL SCHEME

The conservation Eqs.1-3 are solved by implementing them in an in house code. This code has been successfully validated in several situations involving flow and heat transfer as in [8-9]. The present code has a two dimensional unstructured finite-volume framework that is applied to hybrid meshes. The variables values are stored in cell centers in a collocated arrangement. All the conservation equations have the same general form. By taking into account the shape of control volumes, the representative conservation equation to be discretized may be written as

$$\frac{d}{dt} \int_V \rho \phi dV + \sum_i \int_{S_j} \left[ \rho u_i \phi - \Gamma \phi \frac{\partial \phi}{\partial x_i} \right] dS_i = \int_V S_\phi dV \quad (6)$$

generally, this convective phenomena, where the explicit schemes are too restrictive owing to stability limitations. Hence implicit schemes are often preferred and the simplest choice is the first order Euler scheme. The cell face values, appearing in the convective fluxes, were obtained by blending the upwind differencing scheme (UDS) and the central difference scheme (CDS) using the differenced correction approach [10-11]. The coupling of the dependent variables was obtained on the basis of the iterative SIMPLE algorithm developed by Patankar and Spalding [12-13].

Summation of the fluxes through all the faces of a given CV results in an algebraic equation which links the value of the dependent variable at the CV centre with the neighbouring values. The equation may also be written in a conventional manner as

$$A_P \phi_P = \sum_{nb} A_{nb} \phi_{nb} + b_\phi \quad (7)$$

The coefficients  $A_{nb}$  contain contributions of the neighboring (nb) CVs, arising out of convection and diffusion fluxes as defined by Eqs. (1)-(3). The central coefficient  $A_P$  on the other hand, includes the contributions from all the neighbours and the transient term. In some of the cases, where sources term linearization was applied, it also contained part of the source terms.  $b_\phi$  contains all the terms those are treated as known (source terms, differed corrections and part of the unsteady term).

The momentum, pressure correction and temperature are solved sequentially using an ILU-preconditioned GMRES procedure implemented in the IML++ library [14]. All of the computational meshes were generated using the open-source software Gmsh [15].

#### IV. RESULT AND DISCUSSION

A numerical simulation is made to study the effect of wavy inclined wall on natural convection flow and heat transfer inside the rectangular cavity. Various studies investigated [16] flow and heat transfer inside inclined cavities for different Rayleigh number  $Ra$ . In this study the effect of the Rayleigh number and the inclination angles on the flow and inside the cavity is presented.

The effective Parameters on natural convection are the Rayleigh number which changes between  $10^3$  and  $6 \times 10^6$  and the inclination angle which changes from  $30^\circ$  to  $90^\circ$ . Prandtl number is taken as fixed as 0.71 which corresponds to air.

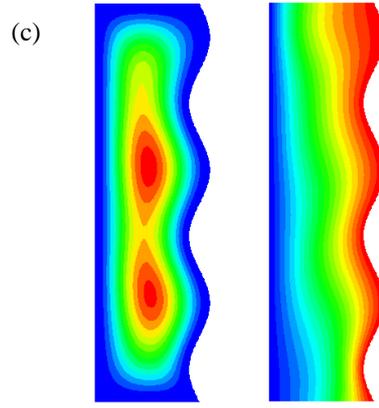
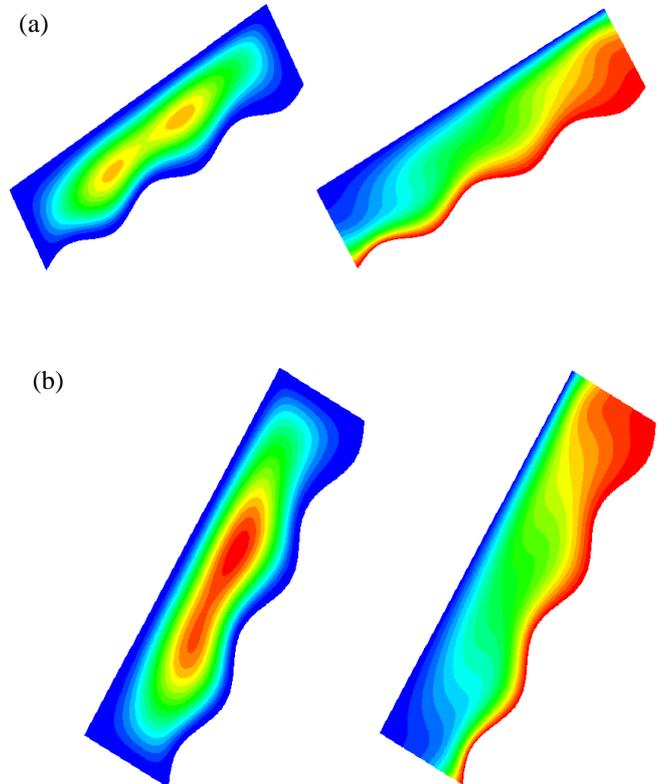
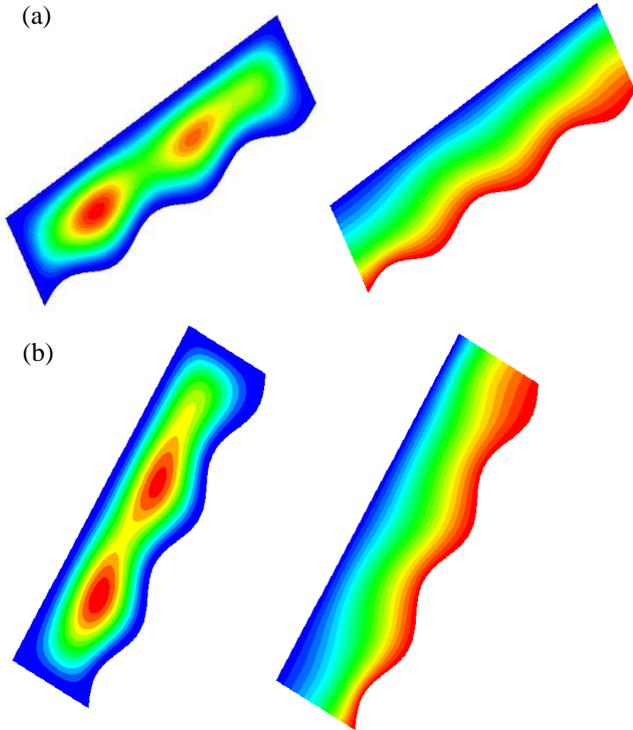


Figure.2 The stream function and isotherms at  $Ra= 10^3$ : (a)  $\Theta=30^\circ$ , (b)  $\Theta=60^\circ$ , (c)  $\Theta=90^\circ$ .

The effects of inclination angles are presented in Fig. 2 with streamlines and isotherms. It is clearly seen that the main two cells are obtained in the middle of the cavity.

When the inclination angle is increased the two cells are moved to each other. This movement is due to value of the gravitation component affected by the inclination angles. Isotherms show plume-like shape from bottom to top. They acted as differentially heated cavity.

While increasing Rayleigh number from  $10^6$  to  $6 \times 10^6$  respectively in Fig. 3 and Fig. 4. It is obviously noted that movement of the two cells is accelerated to develop a single main cell at  $\Theta=90^\circ$ .



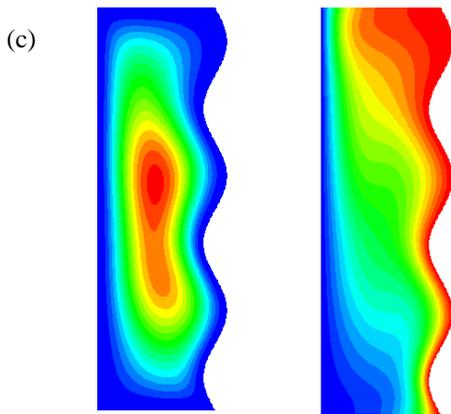


Figure.3 The stream function and isotherms at  $Ra= 10^6$  : (a)  $\Theta=30^\circ$ , (b)  $\Theta=60^\circ$ , (c)  $\Theta=90^\circ$ .

The change from multiple cells to a single cell expand the heat transfer inside the enclosure. The comparison between the Fig.2, Fig.3 and Fig.4 show isotherms that are highly concentrated on the hot wall. Heated fluid moves up to the top of the cavity and the flow which encroached on to the upper surface turns to the left and right.

The revise of this isotherms from a thermal point of view exhibit an important enhancement on the distribution of the temperature, which occurs inside the wavy enclosure. It is evident that both flow and temperature differences are influenced by the inclination angles of the enclosure at the same Rayleigh number.

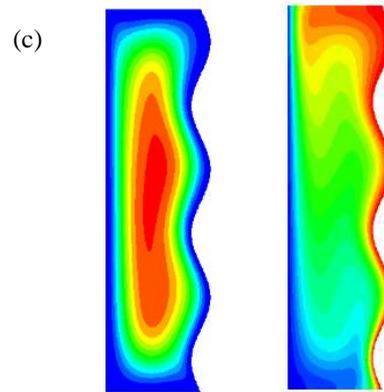


Figure.4 The stream function and isotherms at  $Ra= 6 \times 10^6$  : (a)  $\Theta=30^\circ$ , (b)  $\Theta=60^\circ$ , (c)  $\Theta=90^\circ$ .

Fig.5, Fig.6 and Fig.7 define the variation of local Nusselt number along the wavy wall with different inclination angles at different value of Rayleigh number. The mode of the local Nusselt number is wavy as indicated by Dalal and Das [17].

It is clearly seen that nusselt number increases with an increase in Rayleigh number for all the inclination angles.

Fig.5 represents the value of nusselt number at  $Ra=10^3$  for  $\Theta=30^\circ$ ,  $\Theta=60^\circ$ ,  $\Theta=90^\circ$  which proves the patterns of the isotherms and the stream function as shown in Fig.1.

It has peak value on the top of each wave and their value is almost the same for different inclination angles except the right hill of the wavy inclined bottom surface, it dues to the adiabatic boundaries that influences the temperature distribution in the cavity.

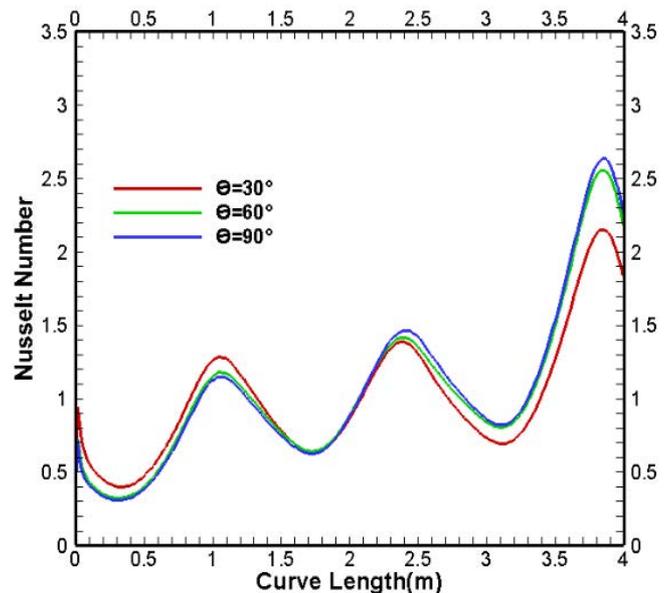
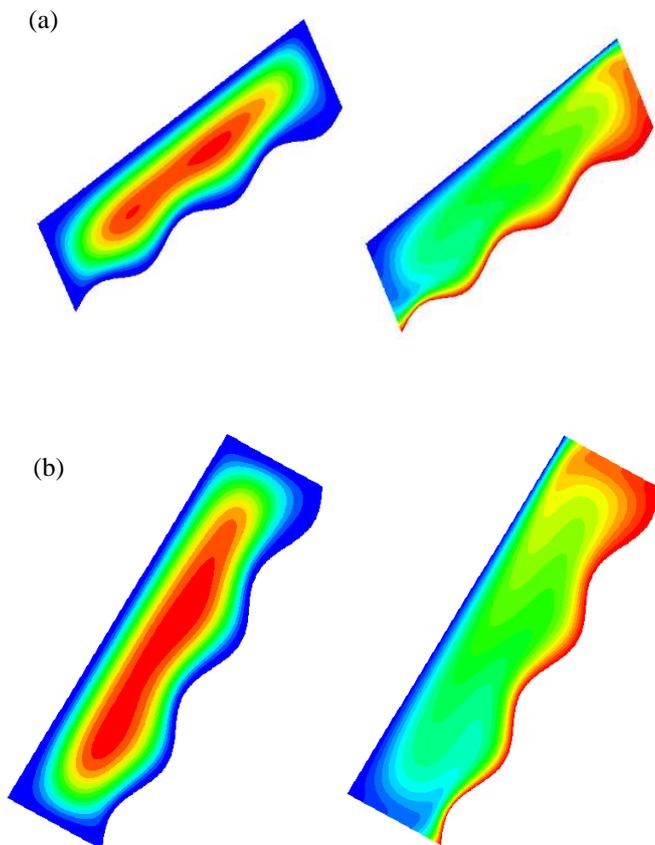


Figure.5 The nusselt number of the hot wavy surface at  $Ra= 10^3$

## V. CONCLUSION

A numerical investigation is carried out on natural convection heat transfer inside a rectangular wavy inclined cavity. This study shows the influence of the governing parameters (Rayleigh number and the inclination angle) at same amplitude and wave length of the sinusoidal bottom wall. A thermal analysis is performed to determine the optimal angle that present a higher nusselt number (heat transfer) in order to give designers, manufacturers and researchers an effective configuration for different application such as a shape of solar collector, thermal exchangers.

The main results can be cited as follows:

1. Nusselt number increases with Rayleigh number at same angle of inclination.
2. At same Rayleigh number, there is an enhancement on nusselt number when the angle of inclination augment.
3. The importance of the gravitational component on the distribution of the temperature inside the cavity.

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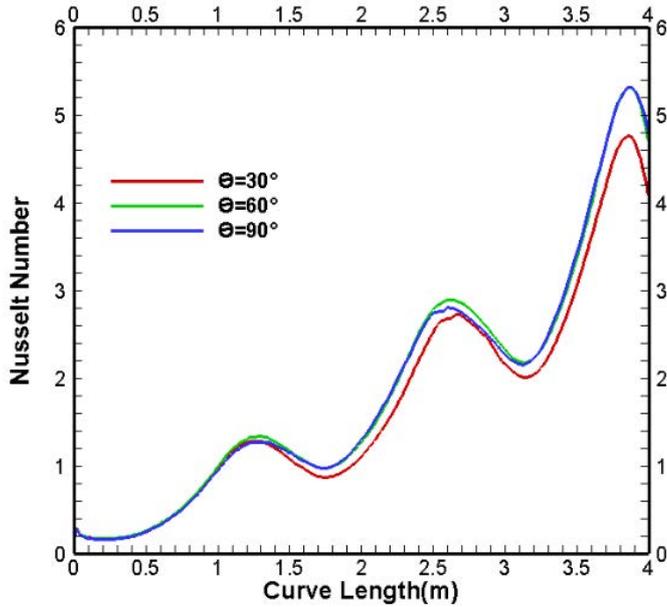


Figure.6 The nusselt number of the hot wavy surface at  $Ra= 10^6$

As expected, Rayleigh number enhances the Nusselt number. The highest Nusselt number is obtained for  $\Theta=90^\circ$ , however, the smallest Nusselt number is formed at  $\Theta=30^\circ$ , interestingly. When the values of  $Ra=6 \times 10^6$ , the value of nusselt number decreases for the higher angles.

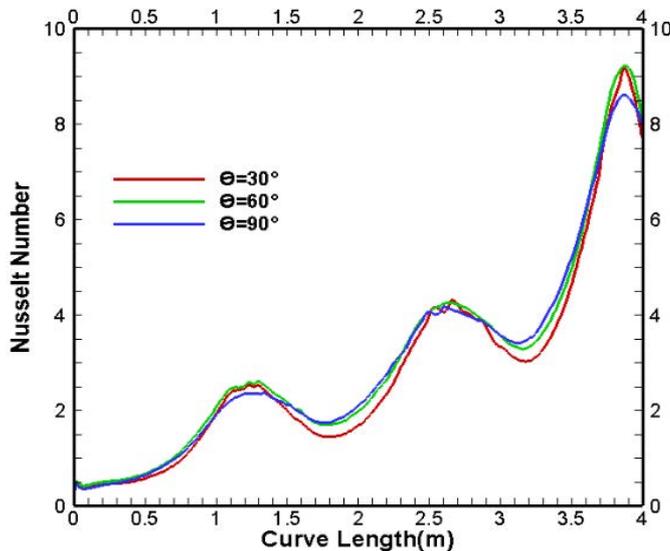


Figure.7 The nusselt number of the hot wavy surface at  $Ra= 6 \times 10^6$

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