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Thermal Behavior of a Horizontal Hollow Tile Submitted To a Sinusoidal Excitation

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ABSTRACT: *This work presents a numerical study, in a transient state, of the thermal behavior of a horizontal alveolar structure filled with air and vertically heated by a sinusoidal excitation applied to its superior surface. The lower surface is maintained to a constant temperature. On the vertical sides, a periodicity condition is imposed (repetitively condition). The coupled heat transfers by conduction, natural convection and by radiation is taken account. The Boussinesq approximation is valid. The system of equations is solved by the finite difference method based on the control volume approach and the SIMPLE algorithm. The response of the structure in term of the variation of the internal and external heat flux and the maximal stream function are compared for different considered parameters of the applied temperature. The average values of the frequency and the amplitude of these grandeurs are analyzed. The effect of the emissivity on the behavior of the nature flow and on the heat flux is also discussed.*

KEYWORDS *- Bénard Convection, Heat Flux, Hollow Structure, Phase Plane, Sinusoidal Thermal Excitation.*

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I. INTRODUCTION

The problem of the coupling between conduction, convection and /or radiation heat transfers, in closed cavities has been extensively studied using numerical simulations and experiments, owing to the practical importance of such configurations in many engineering applications (convective heat losses from solar collectors, thermal design of buildings, air conditioning and, recently, electronic cooling). The majority of the existing studies, which are of numerical nature, concerned with rectangular cavities where the temperature gradient is either horizontal or vertical, including different kinds of boundary conditions [1-5]. Results of these studies show that radiation affects the dynamical and thermal structures of the fluid, reduces natural convection heat transfer component, and contributes to increase the total amount of heat exchanged in the configurations considered. Most of the works conducted in the past on natural convection coupled with radiation inside rectangular enclosures have been substantially oriented to study unidirectional heat transfers resulting from imposed temperature gradients (due to heat fluxes or temperature differences) either parallel or normal to gravity. In some practical situations, much more complex boundary conditions may be encountered where horizontal and vertical temperature gradients could be simultaneously imposed across the cavity. This justifies the presence of published works where the rectangular cavities are heated from below and cooled from above and simultaneously submitted to various specified thermal boundary conditions at the sidewalls. In the absence of radiation effect, Corcione [6] and Cianfrini et al. [7] have investigated steady natural convection of air filled rectangular and square enclosures heated from below, cooled from above and submitted to various thermal boundary conditions at the sidewalls. The results obtained for the average Nusselt number of the whole cavity were expressed through a semi-empirical dimensionless correlation.

In the above mentioned studies, thermal boundary conditions were assumed to be either steady isothermal or constant heat flux wall conditions. However, in many engineering applications, the energy provided to the system is variable in time and gives rise to unsteady natural convection flow. Solar collectors and printed circuit boards are examples of such systems submitted to variable thermal boundary conditions. In addition, thermal and dynamical behaviors of a fluid subjected to time dependent thermal conditions are impossible to predict on the basis of the results obtained with constant temperature or heat flux conditions. This justifies the presence of some works in the literature in which the variable aspect of the thermal boundary conditions was considered. The numerical results obtained by Lakhal et al. in the case of a square cavity totally [8] or partially [9] heated from below with periodic variable temperatures showed that the resulting flow structure and heat transfer were strongly dependent on the amplitude and the period of the variable temperature.

On the other hand, the available studies in the literature which treated the coupling between conduction and convection are generally limited to simple configurations consisting in rectangular cavities with one or several conducting walls. In this sense, we can cite the investigations conducted by Balvanz and Kuehn [10], Kim and Viskanta [11] and Koutsoheras and Charters [12]. These studies in which the rectangular cavities are differentially heated, it has been shown that natural convection heat transfer is significantly reduced by conduction in the walls and/or radiation exchange between the cavity surfaces.

The problem of the coupling between the three modes of heat transfer was the subject of a set of investigations. An early study is a detailed numerical study which taken account, simultaneously, the two-dimensional conductive, convective and radiative heat transfers in concrete hollow blocks has been done by Abdelbaki and Zrikem [13]. Among the effects which were examined, there was the effect of cellular numbers in the two directions of heat transfer. The authors are concluded that the estimation of heat transfer through building walls consisting of hollow tiles can be reduced to a hollow tile with one air cells in the vertical direction. Also, based on the results of a numerical simulation in transient state of coupled heat transfers through a differentially heated hollow clay tile with two air cells deep, Abdelbaki et al. [14] have determined the overall thermal conductance’s from the empirical Transfer Function Coefficients (TFC) using an identification technique. The results have been presented for the hollow bricks which are widely used in the construction of the buildings vertical walls. More recently, the previous study has been extended to the case of the hollow tiles with one air cell in the vertical direction, which mostly used in the construction of building roofs [15-18]. In these studies, it required the resolution of the problem, both in permanent and in transient régime, of the coupling between the three heat transfer processes in alveolar structure vertically heated. Thus, the two situations have been considered: heating from below and heating from above. The considered hollow tiles had only one cell deep in the vertical direction. Thus, the overall thermal conductances are generated in the steady state regime and the transfer function coefficients in time varying regime for different considered hollow concrete blocks.

The importance of coupled heat transfers by natural convection conduction and thermal radiation in a hollow tile subjected to periodic boundary conditions in time is justified by the relevance of such a transitional process to many technological applications. The power supply of electronic circuits by an alternating current, the collectors of solar energy, rooms housing and building hollow blocks, in which recirculation is periodically driven by daily solar heating, are concrete examples.

In our knowledge, works dealing with time periodic combined natural convection conduction and radiation in rectangular cavities subjected to vertical thermal gradients are very few. This work is, therefore, a numerical contribution to the study the transient heat transfers coupled by conduction, natural convection and radiation within a horizontal hollow structure filled with air and heated from below by a constant temperature and submitted by the bottom to an excitation which varies sinusoidal in time. The main parameters governing the problem are the emissivity of the walls, the amplitude and the period of the exciting temperature. The effect of these parameters on heat transfer and fluid flow within the cavity is examined.

II. MATHEMATICAL MODEL

In the construction of buildings roofs, the used hollow structure has, in general, three cavities in the horizontal direction (Nx=3) and in the maximum two cavities in the vertical direction (Ny=2). The studied physical domain is the structure which is illustrated on Fig. 1. It is formed by two ranges of rectangular cavities of width l and height h surrounded by solid partitions of horizontal thickness ei for $1 \leq i \leq 4$ and vertical thickness e'j for $1 \leq j \leq 3$. The top horizontal side of the hollow structure is submitted to a sinusoidal thermal solicitation.

Where a is the amplitude of the excitation, τ its period and Ta its average value. The bottom horizontal side of the hollow structure is considered isothermal and is maintained at constant temperature Tin. On the vertical sides of the structure, we impose a periodicity condition (repetitively of the temperature profile on these faces). The inner surfaces, in contact with the fluid, are assumed to be gray, diffuse emitters and reflectors of radiation with an emissivity ε. The flow is conceived to be laminar, two-dimensional and incompressible with negligible viscous dissipation. The fluid is assumed to be no participating to radiation and the heat transfer is two-dimensional. All the thermophysical properties of the solid and the fluid are assumed constant except the density in the buoyancy term which is assumed to vary linearly with temperature (Boussinesq approximation), such a variation gives rise to the buoyancy forces.

Taking into account the above-mentioned assumptions, the dimensionless governing equations, translating the conservation of the mass, the quantity of movement and the energy in the cavities are as follows:

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

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

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+Pr\,(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2})+Ra.Pr.\theta_f \qquad (3)$$

$$\frac{\partial \theta_f}{\partial \tau}+U\frac{\partial \theta_f}{\partial X}+V\frac{\partial \theta_f}{\partial Y}=\frac{\partial^2 \theta_f}{\partial X^2}+\frac{\partial^2 \theta_f}{\partial Y^2} \qquad (4)$$

where U and V are the dimensionless velocity components in X and Y directions respectively, P is the pressure, θ_f is the dimensionless fluid temperature, Pr is the Prandtl number and Ra is the Rayleigh number given respectively by:

$$Pr=\frac{\upsilon}{\alpha_f} \qquad \text{and} \qquad Ra=\frac{g\,\beta\,H^3\,(\overline{T_{out}}-T_{in})}{\upsilon^2}Pr$$

The dimensionless equation of heat transfer by conduction in the solid walls is:

$$\frac{\alpha_f}{\alpha_S}\frac{\partial \theta_S}{\partial \tau}=\frac{\partial^2 \theta_S}{\partial X^2}+\frac{\partial^2 \theta_S}{\partial Y^2} \qquad (5)$$

Where θ_S is the dimensionless solid temperature and α_f and α_S are the fluid and the solid thermal diffusivities respectively. The hydrodynamic and thermal boundary conditions of the problem are:

* $U=V=0$ on the inner sides of each cavity.

* $\theta_s(X,0)=\theta_{in}=\dfrac{T-T_{in}}{T_{in}-\overline{T}_{out}}$ and $\theta_s(X,A)=\theta_{out}(t)=\dfrac{T-\overline{T}_{out}}{T_i-\overline{T}_{out}}$ for ($0\leq X\leq L/H$)

* $\theta_s(0,Y)=\theta_s(1,Y)$ for ($0\leq Y\leq 1$)

The continuity of the temperature and the heat flux at the fluid-wall interfaces gives:

$$\theta_S(X,Y)=\theta_f(X,Y) \qquad (6)$$

$$-\frac{\partial \theta_S}{\partial \eta}=-N_k\frac{\partial \theta_f}{\partial \eta}+N_r Qr \qquad (7)$$

Where η represents the dimensionless coordinate normal to the wall, N_k is the thermal conductivity ratio k_f/k_s, Q_r is the dimensionless radiative heat flux and N_r is the dimensionless radiation to conduction parameter defined by:

$$N_r=\frac{\sigma T_{out}^4 H}{k_s(\overline{T}_{out}-T_{in})}$$

The net radiative heat flux $q_{r,k}(r_k)$ exchanged by the finite area dS_k, located at a position r_k on the surface k, is given by the radiosity method [19] as:

$$q_{r,k}(r_k)=J_k(r_k)-E_k(r_k) \qquad (8)$$

Where $J_k(r_k)$ is the radiosity and $E_k(r_k)$ is the incident radiative heat flux on the surface dS_k given respectively by:

$$J_k(r_k)=\varepsilon_k\sigma(T_k(r_k))^4+(1-\varepsilon_k)E_k(r_k) \qquad (9)$$

$$E_k(r_k) = \sum_{j=1}^{4} \int_{A_j} J_j(r_j)\, dF_{dS_k - dS_j(r_k, r_j)} \tag{10}$$

Where ε_k is the emissivity of the surface k and $dF_{dS_k - dS_j}$ is the view factor between the finite surfaces dS_k and dS_j located at positions r_k and r_j respectively. Taking into account equations (8) to (10), the dimensionless radiative heat flux can be expressed as:

$$Q_{r,k}(r_k') = \varepsilon_k \left(\left| 1 - \frac{1}{G} \right| \theta_k(r_k') + \frac{1}{G} \right)^4 - \varepsilon_k \sum_{j=1}^{4} \int_{S_j} J_j'(r_j')\, dF_{dS_k - dS_j} \tag{11}$$

Where $G = \frac{\overline{T}_{out}}{T_{in}}$ is the temperature ratio, $J_j'(r_j')$ is the dimensionless radiosity at the position r_j on the surface S_j. By dividing the walls into finite isothermal surfaces, equation (11) leads to a set of linear equation where the unknowns are the dimensionless radiosities $J_j'(r_j')$.

The dimensionless stream function Ψ is defined as:

$$U = -\frac{\partial \Psi}{\partial Y} \quad \text{and} \quad V = \frac{\partial \Psi}{\partial X}$$

In transient state, the average heat fluxes at the outside and inside hollow structure surfaces are given, respectively, by:

$$Q_{out}(t) = -\frac{K_s}{L} \int_0^L \left(\frac{\partial T_s}{\partial y} \right)_{y=H} dx \tag{12}$$

$$Q_{in}(t) = -\frac{K_s}{L} \int_0^L \left(\frac{\partial T_s}{\partial y} \right)_{y=0} dx \tag{13}$$

The previous equations are discretized using the finite differences method based on the control volumes approach with a power law scheme and are solved by the SIMPLE Algorithm developed by Patankar [20]. The resulting system of algebraic equations is solved iteratively line by line by the Tri-Diagonal-Matrix-Algorithm. The numerical code has been validated by comparing its results with those reported in reliable works in the literature [15]. To realize a compromise between accuracy and computation time, a study on the effects of both grid spacing and time step on the simulation results has been conducted. It was found that a non- uniform grid size of 61 × 27 in both directions x and y, clamped near the partition walls and released within the cavities, is sufficient to model accurately the heat transfer and fluid flow inside the hollow structure. The dimensionless time used is 10^{-4}. The convergence criterion is 10^{-4}.

III. RESULTS AND DISCUSSION

Results presented in this study are obtained for a case of hollow structure with two air cells deep in the vertical direction (Ny=2), made in light concrete and characterized by the following geometrical parameters: cavities length l=13cm and height h=3.5cm. The thickness of the horizontal and vertical solid partitions are respectively ei=2.5cm for 1≤ i ≤4 and e'j=2cm for 1≤ j ≤2. The fluid that reigns in the cavities is the air with the Prandtl number Pr =0.71. The thermal conductivity of solid partitions is ks=0.5 W/mK and their thermal diffusivity is $\alpha s=4.25\times10^{-7} m^2/s$. The main parameters governing the problem are the amplitude of the exciting temperature 0≤ a ≤8°C it's period $0\le\tau\le24\times3600$s and the emissivity of the internal walls of the cavities, which are assumed to be gray diffuse reflection and transmission with range of $0 \le \varepsilon \le 0.9$.

3.1. Global behavior of heat transfer

Fig. 2 shows the responses of the upper surface flux, $Q_{out}(t)$ and lower surface $Q_{in}(t)$, obtained for a constant temperature T_{in} =20 °C and for sinusoidal excitation T_{out} of amplitude a = 10 and of a period $\tau = 24\times3600$s. This Figure shows that the hourly variations of heat flow (W/m²) are simple and regular oscillations which represent the curves perfectly sinusoidal. These functions of heat flux oscillate with a period identical to that of the excitation temperature, whereas its amplitudes are four times larger than that of the excitation. Also, it can be noted that the feedback of structure for this excitation is delayed about 10 hours. The amplitude of the heat flux evacuated by the bottom face is greater than that evacuated from the upper one with a shift in time about one hour. This discrepancy is due to the important thermal inertia of the system. Concerning, the hydrodynamic flow, which is described by the variations of the maximum and the minimum of the streamline function Ψmax and Ψmin in each cavity(not shown here) are also simples and regulars and have a mono-periodic behavior (this behavior be more detailed and discussed in the following paragraphs).

3.2. The excitation amplitude effect

The influence of the amplitude of the sinusoidal temperature on the heat flux at the external face $Q_{out}(t)$ and at the internal face $Q_{in}(t)$ and on the maximal dimensionless stream function in the six cells of the structure (Ψ_{max}) are illustrated respectively on the Fig. 3a, Fig. 3b and Fig 4. The obtained results are for emissivity $\varepsilon = 0.9$, the period excitation $\tau = 24 \times (3600s)$ and different amplitudes. These results show that the heat flux $Q_{out}(t)$ and $Q_{in}(t)$ and $\Psi_{max}(t)$ varies globally in the time as manner sinusoidal periodic of frequency $f=1/\tau =1/(24*3600)$. All functions oscillate with an identical period to the one of the exciting temperature. For the different considered amplitudes, the both of heat flux $Q_{out}(t)$ and $Q_{in}(t)$ oscillate around an appreciably identical average value to the one of the permanent regime (a=0).

3.3. The excitation frequency effect

For a constant amplitude a =5 of the excitation and different periods $\tau = 6, 12, 24$ ($\times$3600s) and $\tau = \infty$ (constant heating), Fig. 6a and Fig. 6b give the hourly variations of the heat flux crossing, respectively, the outside face Q_{out} and the inside face Q_{in} of the structure. The solutions which correspond to a constant heating are presented here as comparison. We can note that all resulting frequencies are identical to those of the temperature excitation but the form of the curves is visibly affected by the frequency of this temperature. Concerning the external heat flux Q_{out}, Fig 6a shows that the amplitude of the oscillations of heat transfer through the structure decreases considerably when the frequency increases (passing from frequency $\tau = 6$ to $\tau = 24$ the amplitude of heat flux decreases from 81.5 W/m^2 to 66.6 W/m^2). Whereas its average values, they stay very neighbors to the reference curve who represents the stationary régime 45 W/m^2. For the internal heat flux Q_{in}, Fig 6b shows that the amplitude of oscillations appears constants and unaffected by the temperature frequency.

For the temporal evolution of the maximal dimensionless streamline function (Ψ_{max}) is illustrated in Fig 7. Generally, the obtained oscillations are sinusoid for the small frequencies $\tau > 6(\times 3600s)$. Furthermore, in this range of frequency $\tau > 6(\times 3600s)$, as expected, the signals characterizing the temporal evolutions of heat flux Q which remain perfect periodic, and this periodicity is illustrated by the shape of the trajectory in the phase plane (Ψ_{max} , Q_{in}) who corresponds to a simple and closed curve (Figs 8b and 8c). But for the big frequency $\tau \geq 6$ ($\times$3600s), Ψ_{max} doesn't take a shape close to a sinusoid just after a sufficiently long time ($t \geq 80$ hours). The nature no-repetitive of these peaks is confirmed by the trajectory projected in the phase plane (Ψ_{max} , Q_{in}) who justifies the no-periodicity of the flow by the fact that the curve eventually closes after a multitude of distortions and withdrawals that characterizes its aspect (Fig. 8a).

3.4. The emissivity ε effect

In order to examine the effect of the radiation heat transfer, in transient state, on the global heat transfer and on the air flow in the cavities, the Fig. 9a and Fig .9b give the temporal variations of heat flux Q_{out} and Qin crossing the superior and inferior faces of the system for a constant amplitude a=5, and a constant frequency τ =24×3600s and different values of emissivity ε . The results show that the radiation increases, quantitatively, the amplitude and the average value of the global heat flux crossing the structure (see Figs. 9a, 9b). Indeed, in the case of emissive walls, the total heat transfer is more enhanced in comparison with the case of $\varepsilon = 0$. When the emissivity ε passes from 0 to 0.9 the enhancement of the average heat flux is about 47%.

While for the fluid flow, the maximal steam function in the six cavities of the structure $\Psi_{max}(t)$ which is shown in fig 10. We can see that the emissivity has an influence very limited on its intensity. But, generally, the form of all the functions remains regular sinusoids. Furthermore, The temporal evolutions of $Q_{out}(t)$, $Q_{in}(t)$ and $\Psi_{max}(t)$ are characterized by only one peak by period and the trajectory in the phase plan (Ψ_{max} , Q_{in}) is a simple closed curve (Figs.11a, 11b). As foreseen in the paragraph (§ 3.2), the fluid flow nature in the cells (1) and (3); (4) and (6) are anti-symmetrical: the air in the cavity (1) and (4) circulate in the clockwise sense whereas in the cavity (3) and (6) they circulate in the trigonometrically sense (positive and negative values of Ψ_{max} on the Fig. 10). This phenomenon is due to the fact that the two extreme cavities are submitted to the same conditions in its sides. In the central cells (2) and (5), the flow circulation is identical to the one of the two other near or neighbouring cells.

IV. FIGURES

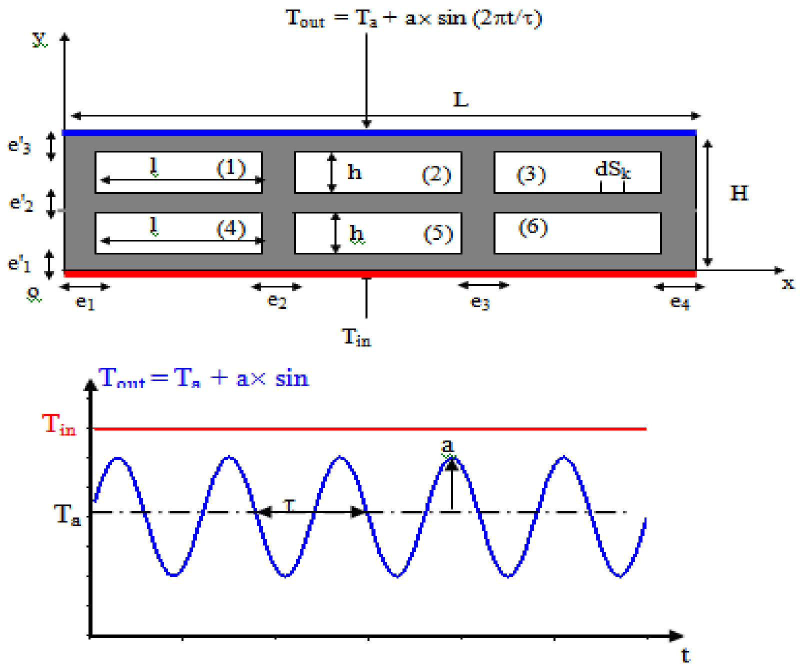


Fig 1. Studied configuration and imposed thermal excitation.

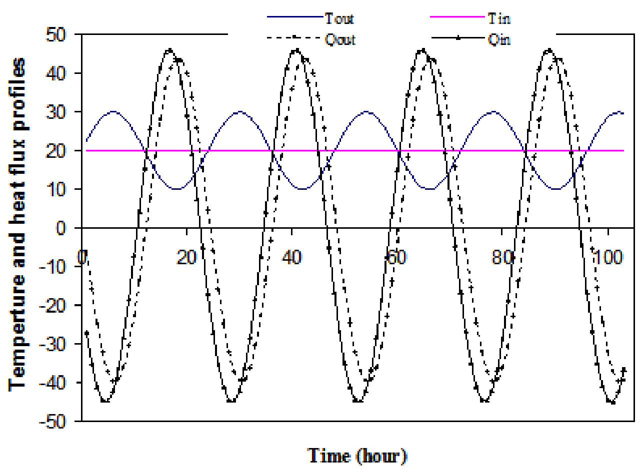
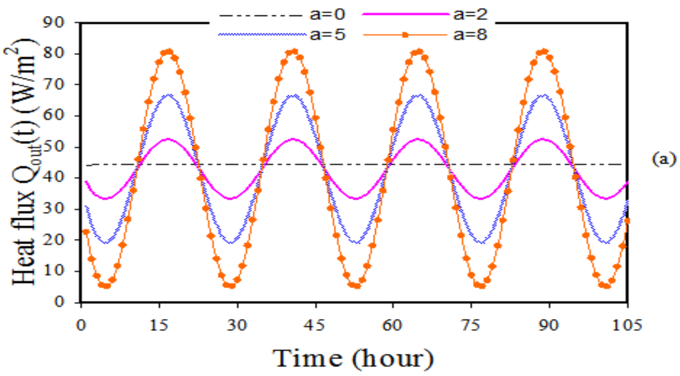


Fig. 2. Hourly variations of the global heat fluxes crossing the upper and lower surfaces of the structure.



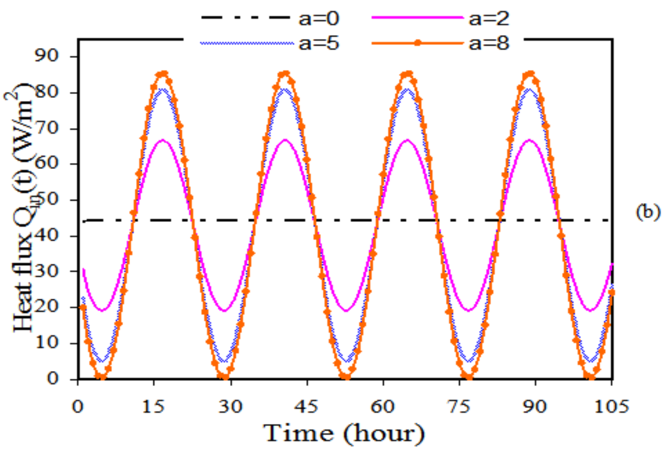


Fig. 3. Effect of the excitation amplitude a on heat flux: (a) $Q=Q_{out}(t)$; (b) $Q=Q_{in}(t)$ for the frequency $\tau = 24 \times (3600s)$ and emissivity $\varepsilon = 0.9$.

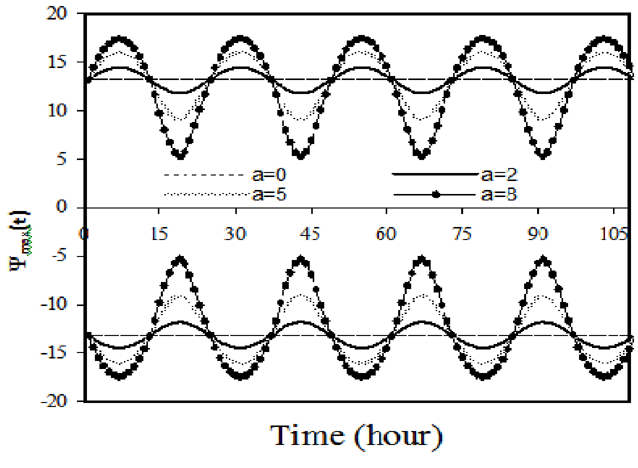


Fig. 4. Effect of the excitation amplitude *a* on streamline function $\Psi_{max}(t)$ for a frequency $\tau = 24 \times (3600s)$ and emissivity $\varepsilon = 0.9$.

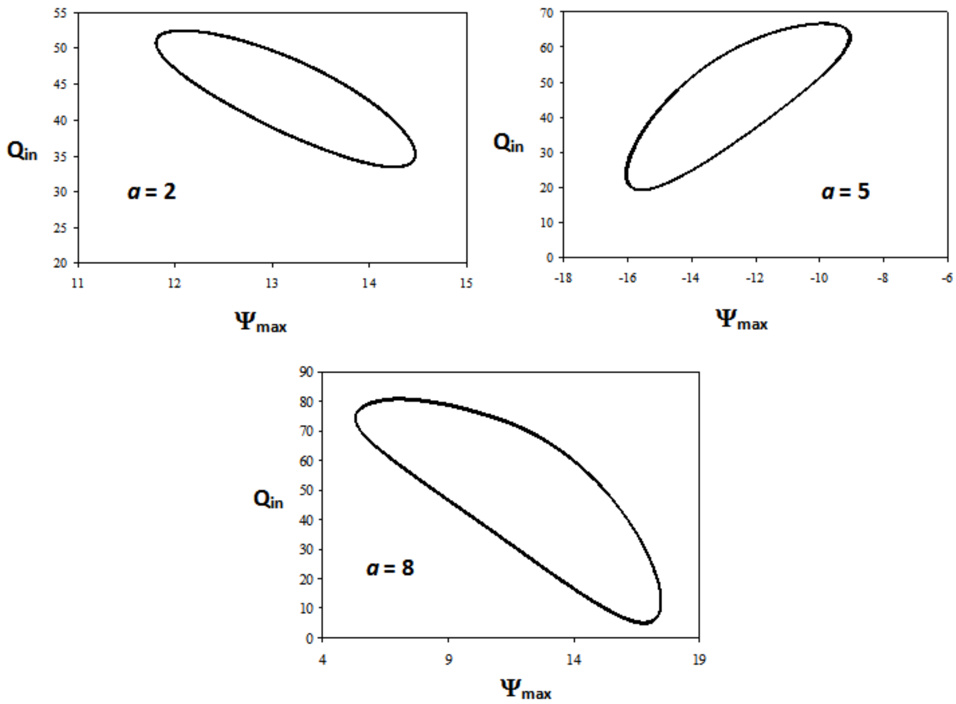


Fig. 5. Trajectory in the phase plane (Q_{in} , Ψ_{max}) for the amplitude respectively a =2, a=5 and a =8 for a frequency τ = 24 × (3600s) and emissivity ε = 0.9.

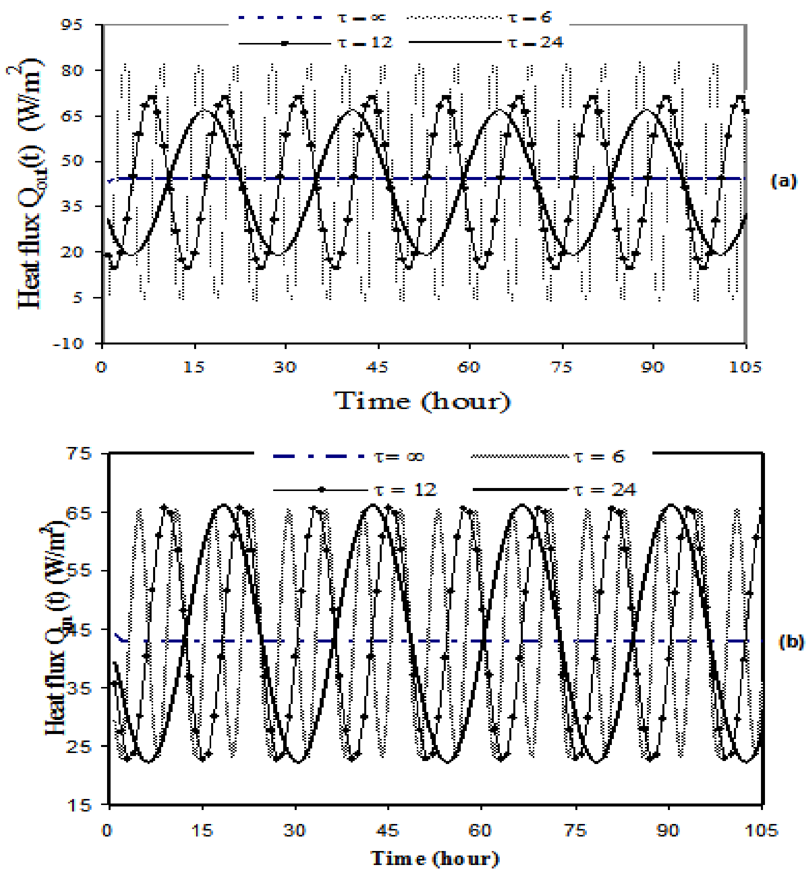


Fig. 6. Effect of the excitation frequency τ on heat flux: (a) $Q=Q_{out}(t)$; (b) $Q=Q_{in}(t)$ for the amplitude a =5 and emissivtiy ε = 0.9.

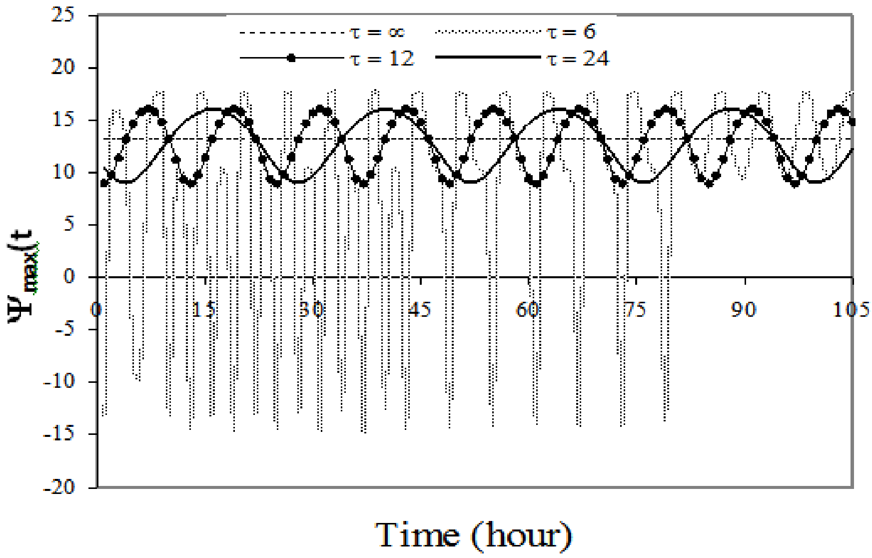


Fig. 7. Effect of the excitation frequency τ on streamline function $\Psi_{max}(t)$ for the amplitude a =5 and emissivity ε = 0.9.

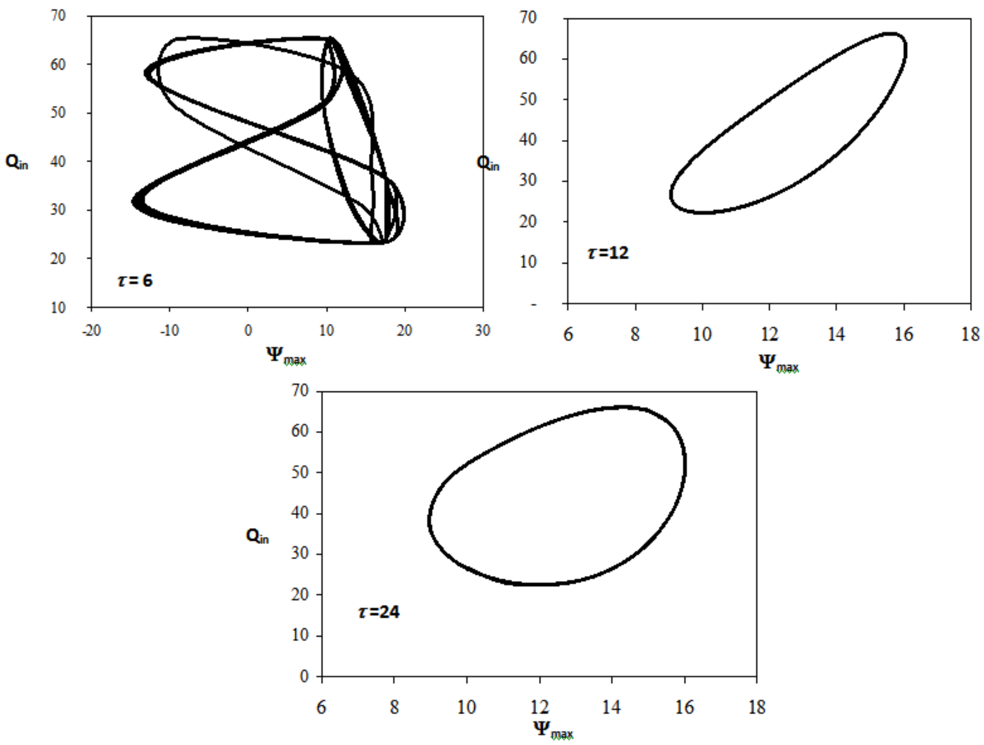


Fig. 8. Trajectory in the phase plane (Q_{in} , Ψ_{max}) for different frequencies τ= 6 , τ=12 and τ=24 for a constant amplitude a =5 and emissivity ε = 0.9

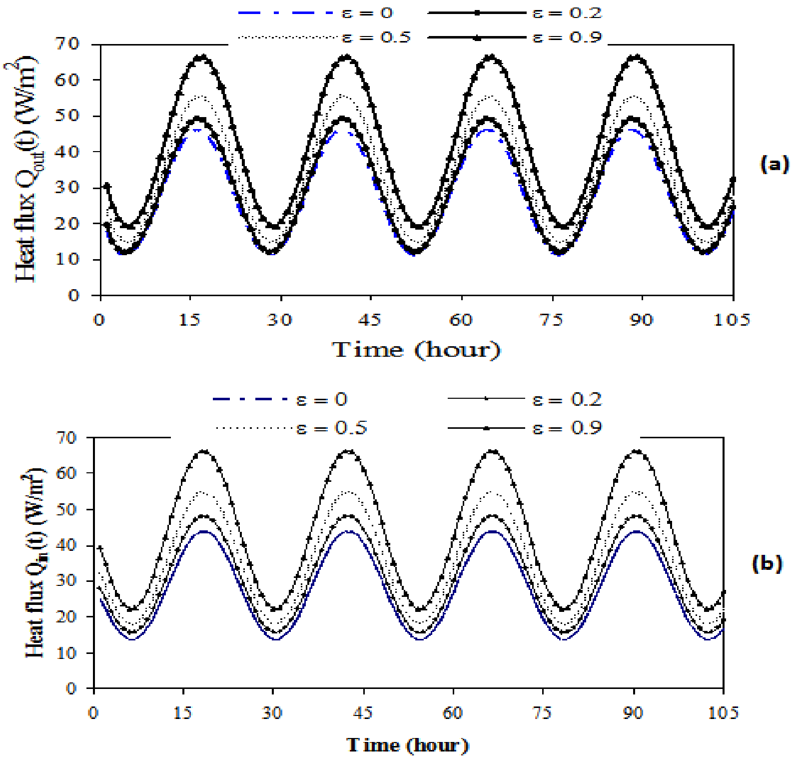


Fig. 9. Effect of the emissivity ε on heat flux: (a) $Q=Q_{out}(t)$; (b) $Q=Q_{in}(t)$ for a constant amplitude a =5 and a frequency τ=24.

V. CONCLUSION

The problem of coupled natural convection, conduction and radiation inside a horizontal hollow structure heated from below by a constant temperature and submitted by the bottom to a sinusoidal excitation temperature, is studied numerically. The main findings of this simulation can be summarized as follows:

a)- The obtained responses of the system in term of heat flux $Q_{out}(t)$, $Q_{in}(t)$ and streamline function and Ψmax(t) are, in general, periodic in time and their paces are similar to the one of the excitation temperature applied on the superior face.

b)- The heat flux evacuated by the bottom face is greater than that evacuated by the upper face with a shift in time about one hour and which is due to the thermal inertia of the system

c)- The variations of heat flux and the streamline function are considerably affected by the amplitude of the exciting temperature. The average values of heat flux $Q_{out}(t)$, $Q_{in}(t)$ are independent of this amplitude and they are around an appreciably identical average value to the one of the permanent regime. Whereas the amplitudes of $Q_{out}(t)$ and $\Psi_{max}(t)$ they are functions of the excitation amplitude.

d)- The frequencies of the responses resulting are identical to those of the temperature excitation but the paces of the curves are visibly affected by the frequency of the exciting temperature.

e)- The emissivity affects the global heat flux crossing the structure considerably: The enhancement of the average heat flux is about 47% when ε passes from 0 to 0.9. Bu its effect on the streamline function is almost negligible.

REFERENCES

[1]. C. Balaji and S.P. Venkateshan, Interaction of Surface Radiation with Free Convection in a Square Cavity, *International Journal of Heat Fluid Flow. 14(3)*, 1993, 260-267.

[2]. M. Akiyama and Q.P. Chong, Numerical Analysis of Natural Convection with surface radiation in a square cavity, *Numerical Heat Transfer, 31(4)*, 1997, 419-433.

[3]. G. Colomer, M. Costa, R. Consul and A. Oliva, Three Dimensional Numerical Simulation of Convection and Radiation in a Differentially Heated Cavity Using the Discrete Ordinates Method, *International Journal of Heat Mass Transfer. 47(2)*, 2004, 257-269.

[4]. C.Y. Han, and S.W. Beak, The Effects of Radiation on Natural Convection in a Rectangular Enclosure Divided by Tow Partitions, *Nnumerical Heat Transfer, Part A. 37(3)*, 2000, 249-270.

[5]. E.H. Ridouane, M. Hasnaoui, A. Amahmid and A. Raji, Interaction between natural convection and radiation in a square cavity heated from below. *Numerical Heat Transfer, Part A, 45,* 2004, 289-311.

[6]. M. Corcione, Effects of the thermal boundary conditions at the sidewalls upon natural convection in rectangular enclosures heated from below and cooled from above. *International Journal of thermal Sciences, 42, (2)* 2003, 199-208.

[7]. C., Cianfrini, M.Corc, and P.P. Dell'Omo, Natural Convection in Tilted square cavities with differentially heated opposite walls. *International Journal of thermal Sciences, 44(5),* 2005, 441-451.

[8]. E.K. Lakhal, M. Hasnaoui, P.Vasseur, and E. Bilgen, Convection Naturelle Dans une Cellule Carrée Chauffée Périodiquement par le Bas: Étude Numérique. *Revue Générale de Thermique, 392, (392-393),* 1994, 480-485.

[9]. E.K. Lakhal, M. Hasnaoui, P.Vasseur, and E. Bilgen, Natural Convection in a Square Enclosure Heated Periodically from Part of the Bottom Wall. *Numerical Heat Transfer Part A, 27(3),* 1995, 319-333.

[10]. J.L. Balvanz and T.H. Kuehn, Effect of wall conduction and radiation on natural convection in a vertical slot with uniform heat generation on the heated wall, *ASME HTD 8,* 1980, 55–62.

[11]. D.M. Kim and R.Viskanta, Study of the effects of wall conductance on natural convection in differentially oriented square cavities, *Journal of Fluid Mechanics, 144,* 1984, 153–176.

[12]. W. Koutsoheras, and W.W.S. Charters, Natural convection phenomena in inclined cells with finite side–walls: a numerical solution. *Energy, Vol. 19,* 1977. 433-438.

[13]. A.Abdelbaki and Z. Zrikem, Simulation numérique des transferts thermiques couplés à travers les parois alvéolaires des bâtiments. *International Journal of Thermal Sciences, 38,* 1999, 719-730.

[14]. A.Abdelbaki, Z. Zrikem, and F. Haghighat, Identification of empirical transfer function coefficients for a hollow tile based on detailed models of coupled heat transfers. *Building and Environment, 36,* 2000, 139-148.

[15]. T. Ait-Taleb, A. Abdelbaki and Z. Zrikem, Numerical simulation of coupled heat transfers by conduction, natural convection and radiation in hollow structures heated from below or above. *International Journal of Thermal Sciences, 47,* 2008, 378–387.

[16]. T. Ait-Taleb, A. Abdelbaki and Z. Zrikem, Transfer Function Coefficients for Time Varying Coupled Heat Transfers in Vertically Heated Hollow Concrete Bricks. *Buildings Simulation, 1,* 2008, 303–310.

[17]. T. Ait-Taleb, A. Abdelbaki and Z. Zrikem, Simulation of coupled heat transfers in a hollow tile with two vertical and three horizontal uniform rectangular cavities heated from below or above. *Energy and Buildings, 84,* 2014, 628–632.

[18]. T. Ait-Taleb, A. Abdelbaki and Z. Zrikem, Thermal behavior in a transient state of a case of hollow block: Transfer function coefficients. *Journal of Solar Energy Engineering: Including Wind Energy and Building Energy Conservation, ASME 137,* 4, 2015.

[19]. R. Siegel and J.R. Howell, Thermal Radiation Heat Transfer, (2^{nd} ed. McGraw-Hill, New York, 1981).

[20]. SV. Patankar, Numerical Heat Transfer and Fluid Flow. (Hemisphere, New York, 1980).

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