 FDMP, vol.8, no.3, pp.339-351, 2012

Influence of the Air Gap Layer Thickness on Heat Transfer Between the Glass Cover and the Absorber of a Solar Collector

F.Z. Ferahta[1,2], **S. Bougoul**[1], **M. Médale**[2] **and C. Abid**[2]

Abstract: A numerical study is carried out to evaluate the thermal performances of a solar collector. As it is well known, that the thermal losses of such systems are mainly of a convective nature, the study is concentrated in particular on the features of natural convection that is activated in the air domain delimited by the upper glass and the lower absorber of the solar collector. The efficiency of such a system depends essentially on both the temperature difference and the distance between the absorber and the glass. Since the temperature difference remains an uncontrolled variable (because it depends on the incident heat flux), while the distance between the glass and the absorber can be "controlled" at the design stage, we focus specifically on the latter, i.e. the effect of the thickness of the air gap on the resulting properties of buoyancy convection (essentially in terms of emerging structure and heat transfer coefficient).

Keywords: Solar collector, heat transfer, convection, numerical simulation.

Nomenclature

The designation of the various parameters used in the different equations is given in the table below.

u, v, w	Velocity components
P	Pressure
e	Air gap thickness
g	Gravity
q	Heat flux
T	Temperature
t_0	Reference temperature

[1] Département de Physique, Faculté des Sciences, Université de Batna, Algérie
[2] Aix-Marseille université, Polytech' Marseille, IUSTI CNRS UMR 7343, Marseille France

S	Absorber Surface
Nu	Nusselt number
Pr	Prandtl number
Ra	Rayleigh number

Greek Symbols

ρ	Density
ν	Kinematic viscosity
λ	Thermal conductivity
β	Expansion coefficient
ϕ	Collector inclination

1 Introduction

Numerical and experimental studies of natural convection in confined cavities have received considerable attention these last years. The success is due to the large range of potential applications. Indeed, these flows are encountered in many industrial applications such cooling of electronic components, designing building, solar collectors, etc. For example, see Karayiannis, Ciofalo and Barbaro (1992); Khan and Yao (1993) and Nag, Sarkar and Sastri (1994).

In a passive solar system, the heat balance of a plate solar collector shows that thermal losses by the front face are about 20%. Coating the absorber with a selective surface could reduce the absorber thermal radiative losses. Once radiation losses have been restricted, natural convection dominates heat losses. As it is described by Hollands (1978), the reduction of convective losses requires the investigation of some parameters as: the temperature difference between the absorber and the glass cover, the angle of inclination of the collector, the thickness of air gap confined between the absorber and the glass cover, the fluid properties and finally the enclosure geometry.

To reduce the convective heat transfer some authors investigated the partition of the enclosure in order to increase the viscous forces and thus to decrease the convective flow. Francia (1961) proposed the use of very high aspect ratio cylindrical honeycombs made of glass and carbonized paper to eliminate radiative losses. Wu-Shung, Jyi-Ching and Wen-Jiann (1989) investigated numerical transient laminar natural convection in a two dimensional enclosure partitioned by an adiabatic partition. They noted that the partition effect is more pronounced when Rayleigh number is small. Recently, Amraqui, Mezhrab and Abid (2011) conducted a numerical investigation to examine the radiation-natural convection interactions in solar collector equipped partitions to its glass cover. Although this study is interesting, however, as it is two-dimensional it doesn't correctly take into account the complex three-dimensional flow.

Another way allowing the reduction of convective heat transfer consists of the optimization of the heat transfer in the air gap confined between the glass cover and the absorber. The control of this complex heat transfer requires the use of experimental approaches but also suitable numerical tools to predict such phenomena often associated with the set up of thermo-convective instabilities see Choukairy, Bennacer, Beji, El Ganaoui and Jaballah (2006). These instabilities lead to large changes in fluid flow. Whatever the configuration, the flow behavior is often non-intuitive when the Rayleigh number reaches values large enough. To examine the influence of the slope and the aspect ratio on the rate of heat transfer, El Sherbiny (1996) analyzed experimentally the natural convection in an inclined rectangular cavity with aspect ratios varying from 5 to 110 and Rayleigh numbers from 100 to 2.10^7. They developed a correlation of heat transfer for a slope of 60° and proposed a linear interpolation for slopes between 60° and 90°. Ozoe, Fujii, Lior and Churchill (1983) conducted a three dimensional study in a slight sloping cavity, they observed a series of aligned rolls. By further increasing the slope they found that the rolls become oblique and beyond a certain inclination, a quasi two-dimensional cell is observed where flow goes up the heated plate and down the cooled one. They also measure the heat transfer rate, they found that the average Nusselt number begins to increase with tilt angle and then decreases when the oblique rolls appear and finally increases again when the cell becomes almost two-dimensional. Manz (2003) conducted a numerical study to find the critical Rayleigh number for transition to turbulence. Small differences were found between his results and those of Batchelor (1954). Most work on the Rayleigh-Bénard convection like Dubois and Bergé (1978) focus on low Rayleigh numbers. The studies are usually two-dimensional and do not include the effects of the Three-dimensional instabilities.

This study aims a three-dimensional numerical simulation of natural convection in an inclined cavity with a large aspect ratio in order to model the natural convection between the glass cover and the absorber in a solar collector. The modelisation of the fluid flow and the heat transfer in the space between the glass and the absorber allows us to determine the flow structure and the dominating heat transfer mode. For that, we studied numerically the three-dimensional natural convection for various thicknesses of the air gap using a reliable numerical model and without changing the geometry of the flat plate solar collector, in order to have an optimal thickness, which reduces thermal losses at the front of the glass.

2 Problem statement and mathematical formulation

2.1 Configuration of the Problem

The studied geometry is an inclined rectangular cavity, composed of an absorber (lower heated face), insulated lateral sides and a glass cover (upper face). The inclination angle is φ=45°, the thickness of the glass cover is fixed at 4 mm, the width l is of 0,10 m and the total length L is of 1 m. This geometry is represented on figure 1. The thickness of the air gap varies according to the studied case.

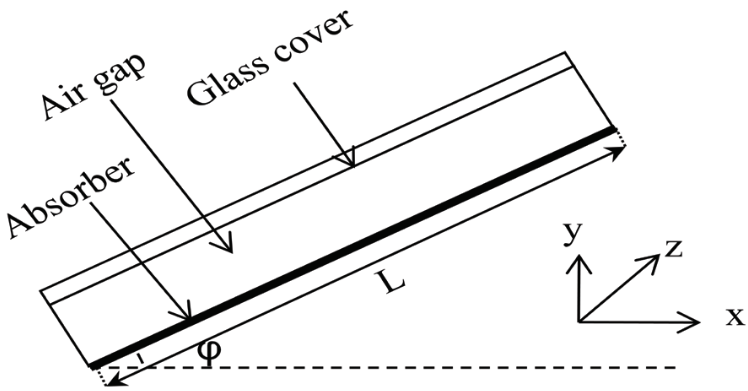


Figure 1: Geometry of the problem

2.2 Mathematical Model

For an unsteady flow of an incompressible and Newtonian fluid, the equations of mass conservation, balance momentum and energy conservation can be written as:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial P}{\partial x}+v\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)-g\beta(T-T_0)\cos\phi \tag{2}$$

$$\frac{\partial v}{\partial t}u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho}\frac{\partial P}{\partial y}-g+v\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)-g\beta(T-T_0)\sin\phi \tag{3}$$

$$\frac{\partial w}{\partial t} u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial y} + v \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \tag{4}$$

$$\frac{\partial T}{\partial t} u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = a \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \tag{5}$$

To write the equations above, viscous dissipation was neglected, gravity has a vertical effect and the properties of the fluid are supposed to be constant. The Boussinesq approximation is applied (see Gray and Giorgini 1976). The modified Rayleigh number can be given by the following formula:

$$Ra = \frac{g \beta q e^4 Pr}{\lambda S v^2} \tag{6}$$

q corresponds to the heat flux supplied to the absorber and S the heated surface (absorber).

The above equations depend on the boundary conditions. On the walls, the velocity satisfies the nonslip condition except for the wall located at the middle vertical plane (symmetry axis) where a null velocity gradient is applied. Condition of constant heat flux is applied to the horizontal lower wall (absorber), for the following results q is equal to 400 W/m^2. The lateral walls are well insulated such that one considers them as adiabatic. For the upper wall (glass), a mixed condition (convective and radiative) heat transfer with the surroundings is applied.

3 Results and discussion

The convection in the air gap of the solar collector is studied numerically by a CFD analysis using Fluent package. The thermal losses from the glass cover are in relation with the climatic conditions surrounding the collector, such as the outside and the sky temperatures. This is confirmed by the boundary conditions imposed on the outside of the glass. Simulations are carried out in instationary state with the Boussinesq approximation; the SIMPLE algorithm is used to couple the speed and the pressure. The construction of the geometry and the mesh generation were performed using the Gambit 2.0.4. The mesh was tight near the walls to take account of the strong velocity and temperature gradients at the boundary layer and slighty wider elsewhere. The second order scheme is used to discretize the convective terms. Piso algorithm is used for the velocity-pressure correction. In the case of natural convection in closed cavities,the convergence criterion should not be based on the residue, so in addition to the value of the residues which took less than 10^{-6} for all variables, we tested the convergence by monitoring the evolution of the average temperature of a selected section and also the average temperature of the air volume of the air layer; when we see it no longer varies, we stop the iterations.

After several tests, the choice of the relaxation factor has also an effect on the various results obtained and its value must be around of 0.9.

To see the effect of the grid on the results, two grids were selected for each configuration (each thickness of the air gap). After superposition, we see clearly that there was no difference between both of the grids and so the larger grid was considered to be sufficient to conduct the various simulations. The considered grids are shown on table 1 for various thicknesses.

Table 1: Grids for various thicknesses

Thickness e (cm)	0.5	0.7	1
Nodes (N_xxN_yxN_z)	1440000	2224800	2000000

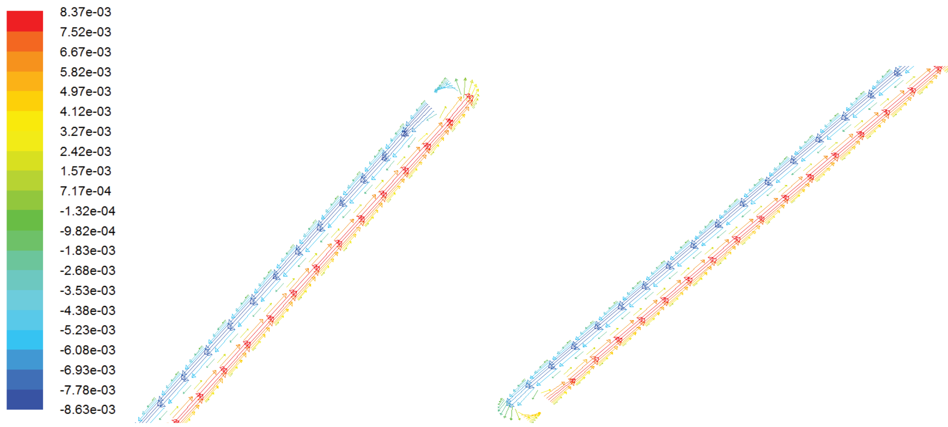
Many simulations were performed for a fixed heat flux q supplied to the wall. The external temperature was taken equal to 293 K and the sky temperature equal to 263K.

First, we compute thermal and velocity fields for the smallest thickness (e=0.5 cm). In order to illustrate, we plot on figure 2 the velocity fields and on figure 3 the thermal fields in a vertical middle plane.

It is well known that when a vertical thermal gradient is applied to a horizontal cavity (the bottom is heated while the top is cooled), fluid remains quiescent till the Rayleigh number based on this thermal gradient exceeds a threshold value which is in the case of infinite extension Ra_c=1708. When the cavity is inclined, we have the coexistence of the horizontal and the vertical thermal gradients; in this case natural convection sets up instantaneously, there is no need to a threshold in Rayleigh number. Indeed, fluid near the heated wall heats and so becomes lighter, it grows up along the heated wall then it cools near the cold wall and goes down. Thus we have the formation of a large cell in the cavity; that is what we observe on figures 2 and 3.

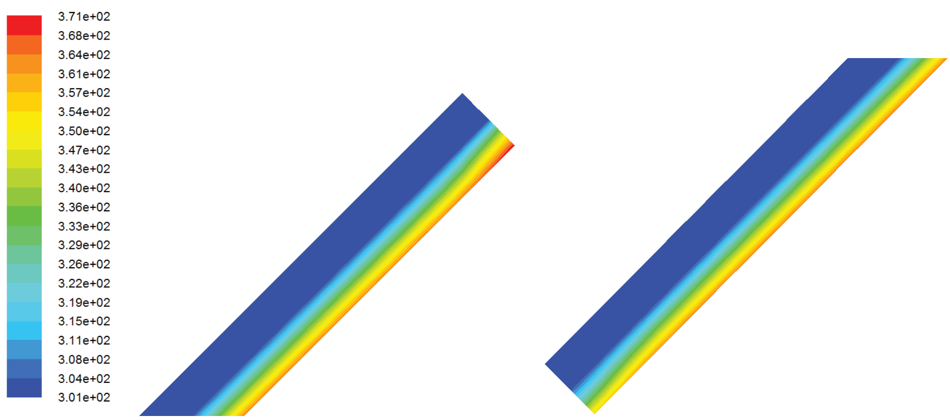
The velocity profile according to the air thickness is plotted on the figure 4 where it is clearly shown the fluid recirculation: positive velocity corresponds to the fluid ascension along the heated wall and negative part corresponds to its return back along the cold glass cover. However, it is noteworthy that this induced flux is very week such as heat transfer is mainly conductive.

It appears that for small thicknesses thus for small Rayleigh numbers, the flow structure consists of one transversal cell, hot fluid goes up the heated wall (absorber) and then down along the cold wall (glass cover), flow is strictly bi-dimensional.



(a) (b)

Figure 2: Longitudinal velocity fields of the collector in (xoy) plane (q=400 W/m^2; e=0.5 cm): (a) longitudinal upper part, (b) longitudinal lower part



(a) (b)

Figure 3: Longitudinal thermal fields of the collector in (xoy) plane (q=400 W/m^2; e=0.5 cm): (a) longitudinal upper part, (b) longitudinal lower part

When we increase the air thickness (e=0.7 cm), the velocity component along (z) is triggered; there is a structure that corresponds to the Rayleigh-Bénard thermal instability. We notice that progressively, longitudinal rolls set up, spread gradually to invade the whole section. On figures 5 and 6 are visualized the contour of temperature and velocity on a cross section (yoz) (sections perpendicular to x) located at mid-x.

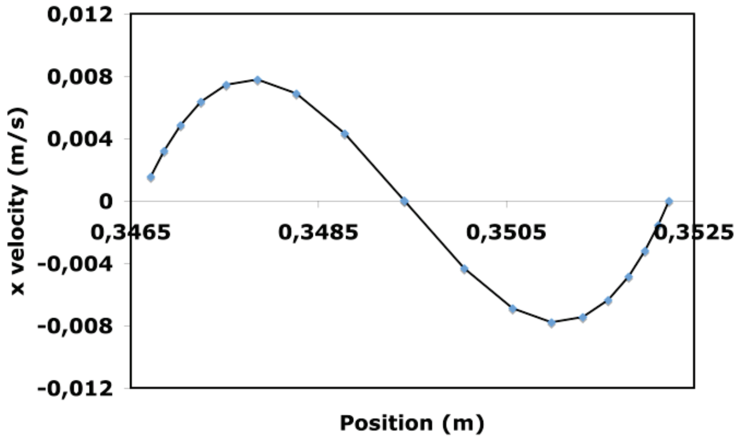


Figure 4: Velocity profile in a transversal cross section at mid length of the collector (q=400 W/m^2; e=0.5 cm)

One can notice the formation of rolls with a roll number almost equal to the aspect ratio. The zoom of the velocity vector (figure 6c) shows the ascending of hot fluid in the middle of two pairs of rolls and the descending of cold fluid along the both sides. These rolls remain stable and stationary.

A larger value of the thickness (e=1 cm) leads to a chaotic flow (absence of periodicity). Figure 7 shows the temperature field at various times for a cross section (yoz plane) at mid-x.

We notice that the flow is highly unsteady and looks like a turbulent state. This unsteadiness as it is shown on figure 8 where is visualized the thermal field on the absorber (heated wall) affects heat transfer. Indeed, the fact that rolls are unsteady contributes to the agitation of the thermal boundary layer so that the heated wall (absorber) is continuously refreshed by cold fluid while the warmed one is advected to the cold wall (the glass cover).

In order to assess this effect, heat transfer coefficient has been determined for various axial coordinates and air layer thicknesses. On figure 9 is plotted the heat transfer coefficient for a fixed air layer thickness (e=1 cm) at 3 various axial coordinates corresponding to the inlet, the mid and the outlet of the cavity.

As fluid structure is tri-dimensional, the corresponding heat transfer is inhomogeneous in a cross section. In order to investigate the thickness influence on heat transfer, figure 10 displays heat the longitudinal heat transfer coefficient averaged on a cross section for the three considered thicknesses.

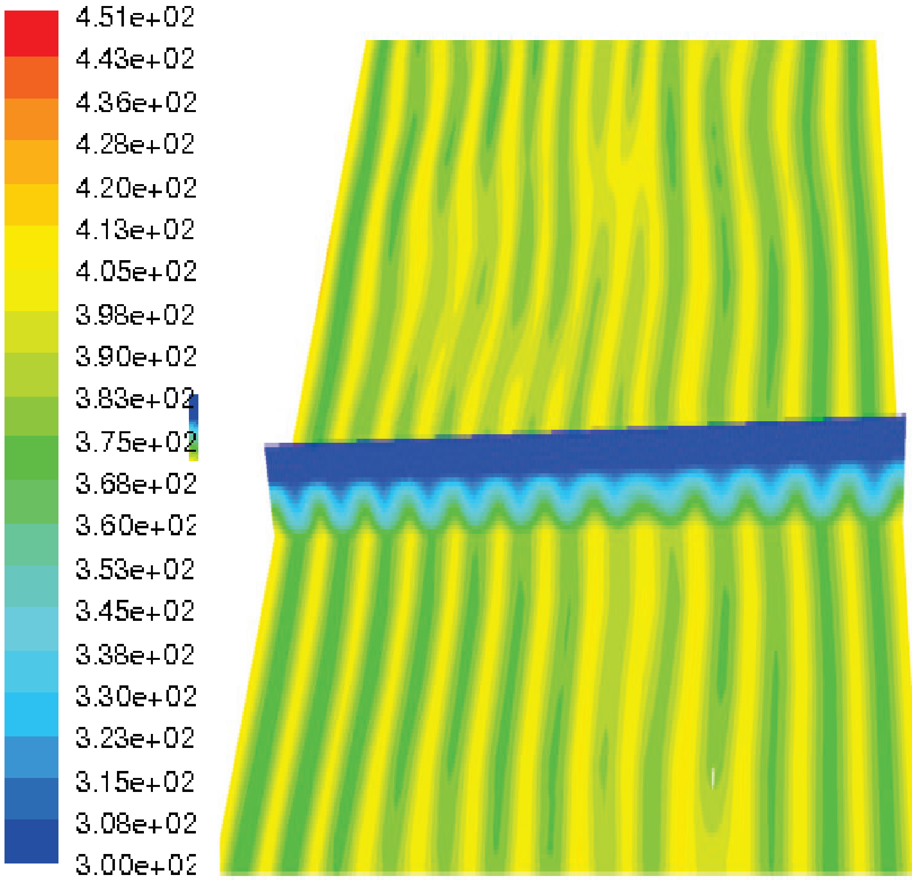


Figure 5: Temperature field (q=400 W/m^2; e=0.7 cm)

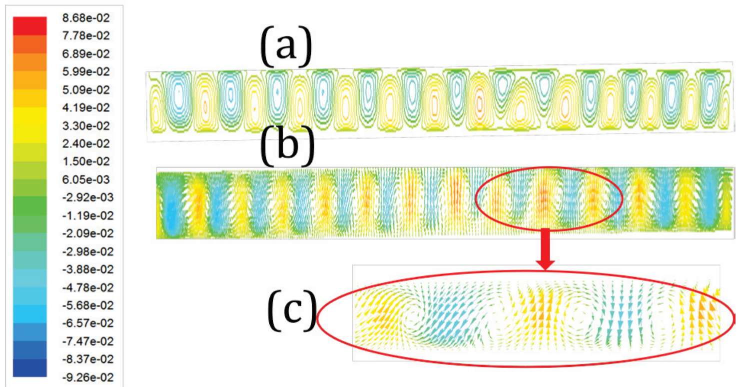


Figure 6: Contour of velocity (a), vector velocity field (b), zoom of vector velocity (c) in a cross section at mid-x: (q=400 W/m^2; e=0.7 cm)

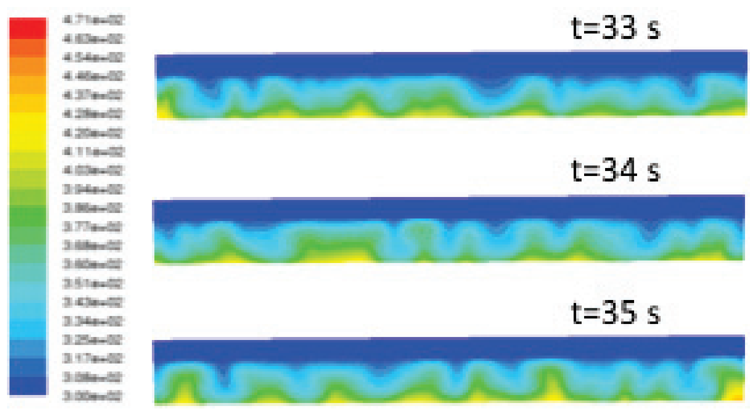


Figure 7: Temperature field in a cross section at mid-x: (q=400 W/m^2; e=1 cm)

This figure highlights the fact that heat transfer is reduced for the intermediate thickness. Indeed, the smallest one induces the most important heat transfer due the mainly conductive heat transfer mode while the highest thickness leads to unsteady flow where heat transfer is rather enhanced.

4 Conclusion

A three-dimensional thermal convection study with various thicknesses of the air gap of a thermal solar collector was carried out by CFD analysis using Fluent.

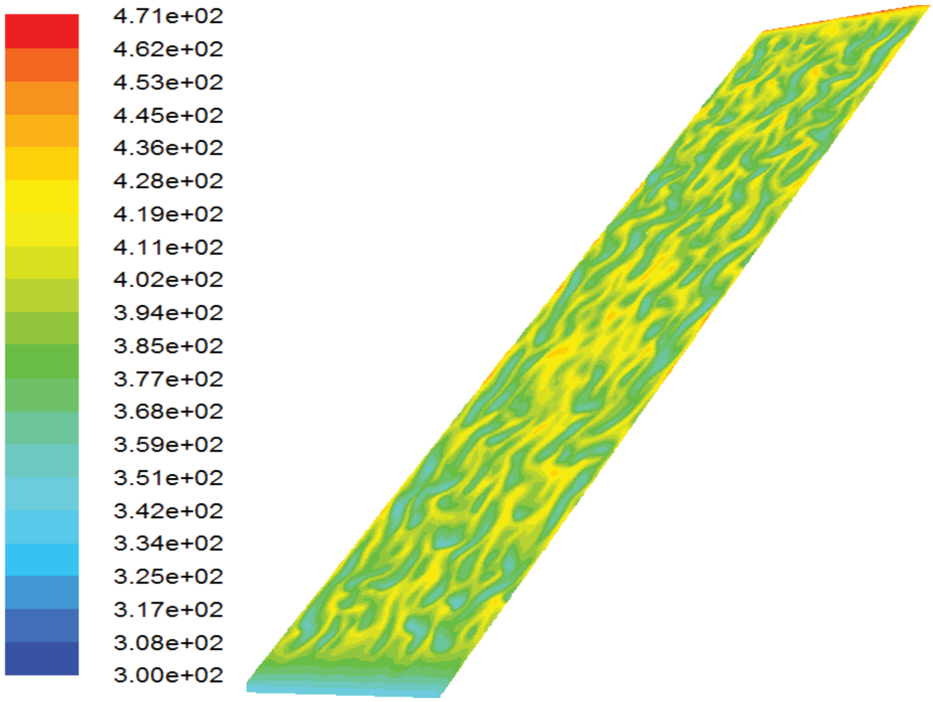


Figure 8: Thermal field at the absorber: (q=400 W/m^2; e=1 cm)

Various simulations carried out showed us the importance of certain parameters to arrive at a good results convergence, in particular the form and the size of the grid as well as the iteration number. For a small thickness of the air gap confined between the glass cover and the absorber, conduction dominates; for large thicknesses, the convection dominates. The increasing of the thickness leads to the development of the flow structure from a well-organized flow with steady longitudinal rolls to a chaotic one where rolls are no longer identified. This situation affects heat transfer where we observe a significant intensification and thus enhancement of thermal losses from the front of the solar collector.

References

Amraqui, S.; Mezhrab, A.; Abid, C. (2011): Combined natural convection and surface radiation in solar collector equipped with partitions. *Applied Solar Energy*, vol. 47, no. 1, pp. 36-47.

Batchelor, G.K. (1954): Heat transfer by free convection across a closed cavity

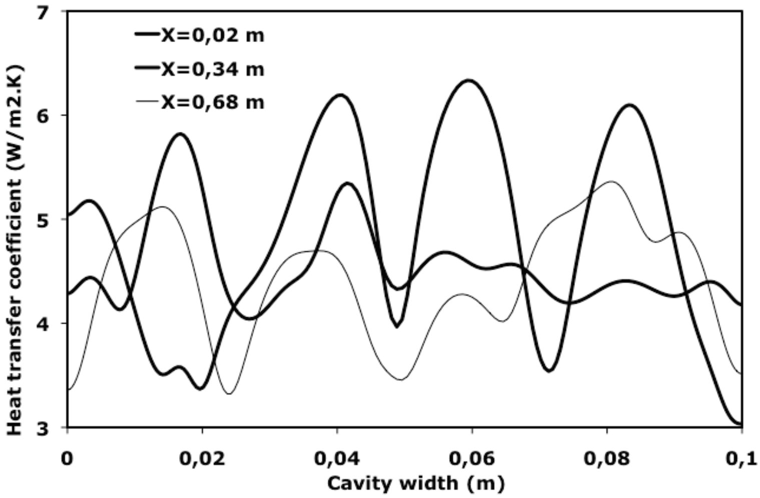


Figure 9: Heat transfer coefficient for various axial coordinates (q=400 W/m^2; e=1 cm)

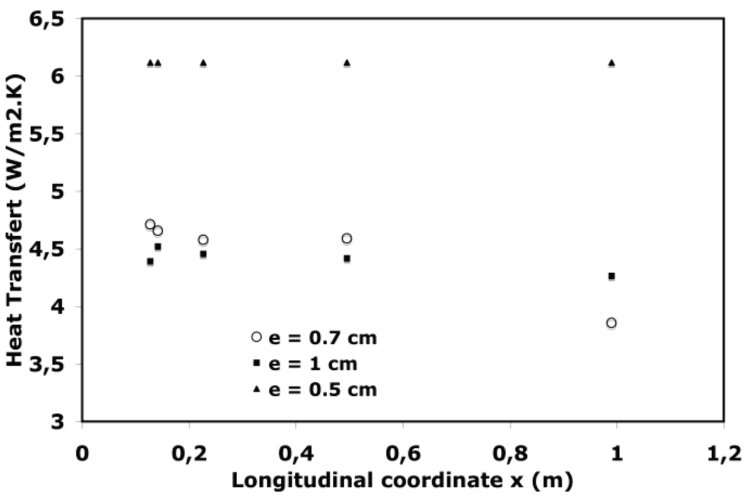


Figure 10: Longitudinal heat transfer coefficient for various thicknesses

between vertical boundaries at different temperatures, *Quart. Appl. Math. XII*, pp. 209-233.

Choukairy, K.; Bennacer, R.; Beji, H.; El Ganaoui, M.; Jaballah, S. (2006): Transient behaviors inside a vertical cylindrical enclosure heated from the side walls. *Num. Heat Transfer part A*, vol. 50, pp. 1-13.

Dubois, M.; Bergé, P. (1978): Experimental study of the velocity field in Rayleigh-Bénard convection, *J. Fluid Mech.*, vol. 85, pp. 641-653.

Elsherbiny, S.M. (1996): Free convection in inclined air layers heated from above. *Int. Journal of Heat and Mass Transfer*, pp. 3925-3930.

Francia, G.A. (1961): *A new collector of solar radiant energy theory and experimental verification*. Proceedings of United nations Conference on New Sources of Energy, Rome, Italy.

Gray, D.D.; Giorgini, A. (1976): The validity of the Boussinesq approximation for liquids and gases. *Int. Journal of Heat and Mass Transfer*, vol. 19, pp. 545-551.

Hollands K.G.T. (1978): *Free convection in solar collectors, Mar. Energy conversion, An Introductory course, selected lectures from the 5 Course on Solar Energy Conversion*. Dixon A.E., Leslie J.D. (ed), Pergamon Press.

Karayiannis, T.G.; Ciofalo, M.; Barbaro, G. (1992): On natural convection in a single and two zone rectangular enclosure. *Int. J. Heat Mass Transfer*, vol. 105, pp. 89-95.

Khan, J.A.; Yao, G.F. (1993): Comparison of natural convection of water and air in a partitioned rectangular enclosure. *Int. J. Heat Mass Transfer*, vol. 36, pp. 3107-3117.

Manz, H (2003): Numerical simulation of heat transfer by natural convection in cavities of façade elements. *Energy and building*, vol. 35, pp. 305-311.

Nag, A.; Sarkar, A.; Sastri, V.M.K. (1994): Effect of a thick horizontal partial partition attached to one of the active walls of a differentially heated square cavity. *Num. Heat Transfer A*, vol. 25, pp. 611-625.

Ozoe, H.; Fujii, K.; Lior, N.; Churchill, S.W. (1983): Long rolls generated by natural convection in an Inclined, rectangular enclosure. *Int. Journal of Heat and Mass Transfer*, vol. 26, pp. 1427-1438.

Wu-Shung, Fu; Jyi-Ching, P.; Wen-Jiann, S. (1989): Transient laminar natural convection in an enclosure partitioned by an adiabatic baffle. *Num. Heat Transfer A*1, vol. 16, pp. 325–350.