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Mathematical modelling and optimisation of a solar collector performances

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Abstract

Although a lot of research work has been performed in the field of renewable energy, this domain of research has received, with the recent technological developments, a significant turnover both on the theoretical and experimental sides. At low temperatures, the solar collector plays the role of a converter of solar radiation into heat. Our work focuses on the study of the dynamics and thermal transport phenomena in the air gap of a solar collector in order to optimize its performance. The evolution of thermal and dynamic profiles in the air gap is presented in this paper, which allows us to quantify the convective energy losses. The impact of the interposition of partitions in the air gap was analyzed as well. The influence of this new model on convection losses turns out to be significant

Keywords: solar energy, solar collector, mathematical modelling, performance optimisation;

1. Introduction

Solar water heating is a promising technology to produce hot water, it is a technology that is simple and easy to adopt. The growing importance of solar thermal collectors has led to significant progress in their design characteristics (design features), this was performed, on one hand by improving the absorption capacity of heat by the solar collector by adding new features to the absorber plate, AlShamaileh (2010),[1], improving the hydraulic and geometric design, Tanaka (2011)[2] and the use of alternative materials, Carlson et al (2014)[3], on the other hand, investigations are focusing on the minimization of heat losses to the outside, Vestlund et al (2006), Subiantoro et al (2013)[4 - 5]. A good solar thermal converter requires effective control of the heat loss at the absorber to the environment. Designers seek economical alternatives to minimize losses by thermal conduction and especially the coupling between convection and radiation. The most significant thermal losses happen through the glass cover, Benkhalifa A. (1998), Ghoneim (2005)[6 – 7] . Researches, in this axis, focus on reducing to the maximum their effect. This can be achieved by inserting baffles (obstacles) in the air gap between the absorber and glazing. Youcef-Ali (2005). H. Buchberg et al (2008) [8 – 9] showed that the installation of a rectangular cell positioned above a solar absorber is an effective device for lowering losses by natural convection. The obstacle should be very thin and transparent to avoid interference with different radiations. Finally, this study is equivalent to a study of natural convection in a partitioned cavity having dimensions of the solar collector. Natural convection, in partitioned cavities, has been extensively studied both numerically and experimentally, because of its substantial interest. We can cite the work of Frederick (1989) [10] which is a numerical study of natural convection in an inclined square cavity filled with air and heated differentially with a single partition attached to its cold wall, Ra = (10^3-10^5). It has been proved that the partition leads to the suppression of convection, and reduces the heat transfer by 47% compared to the cavity without partitions, in the same conditions. E. Bilgen (2004) [11] studied natural convection in cavities with a thin fin on the hot wall, he concluded that normalized Nusselt number, Nu is an increasing function of Rayleigh, Ra, and a decreasing function of fin length. Hua-Shu Dou et al (2016) [12] present a numerically investigation of natural convection flow in a differentially heated cavity. The effects of fin number, fin position, and fin length on heat transfer in natural convection are analyzed ,when Ra is relatively small, the fin(s) blocks the heat transfer. The effect of the fin number on heat transfer is negligible when the fin number is larger, than one ,when Ra is relatively large, the fin(s) enhances heat transfer ,the effect of fin number on heat transfer is also negligible when the fin number exceeds one. However, the heat transfer rate decreases when the fin number exceeds one comparing to that when there is only one fin in the cavity Mohammed Jami (2006) [13] presented a numerical study of heat transfer by laminar natural convection in an inclined enclosure differentially heated, with inclined partitions attached to the hot wall. He concludes that the reduction in heat transfer increases with increasing the partition length when the partition is tilted. Mezrab et al (2006)[14] studied the effect of the number of partitions on heat transfer phenomena in a tilted square cavity where the partitions are fixed to the cold wall. He showed that in the case of several partitions, the reduction of the heat transfer is considerably increased with the number of partitions up to a certain value it remains constant by the fact of the disappearance of the convection. Mr. Yousaf et al (2015) [15] A numerical algorithm was developed to analyze the role of the sinusoidal roughness elements on the thermal and hydrodynamic behavior of a fluid in laminar region. Two-dimensional studies were conducted in a square cavity ,the sinusoidal roughness elements were located on a hot, and both the hot and cold walls. He conclude that The maximum reduction in the average heat transfer was calculated to be 28% at Ra number equal to 10^5 when the sinusoidal roughness elements were located at both the hot and cold walls.. The remark drawn from this literature review is that for similar cavities with flat plate solar collectors, the conditions applied to the limits do not reflect the actual operating conditions of the solar collectors. The geometry dimensions in our study are real and are subject to a flow imposed on the absorber taking into account the losses by radiation and convection to the outer face of the solar flat plate collector. The partition conductivity is taken into account for a comprehensive study of the phenomenon. So the aim of this study is to identify the impacts of partitions on the dynamic and thermal phenomena in the air gap of a solar collector.

2. Generating of the problem

2.1. physical model

The studied geometry is a parallelepiped cavity with dimensions (L x D x e = 1m x 0.02m x em) with an aspect ratio A= L / D, formed by an absorber (lower face) of a glass cover (upper side) 4mm thickness and whose side walls are insulated. partitions are interposed in the interior of the cavity, attached to the inner face of the glass which is uniformly distributed; the study area is presented in Fig.1.

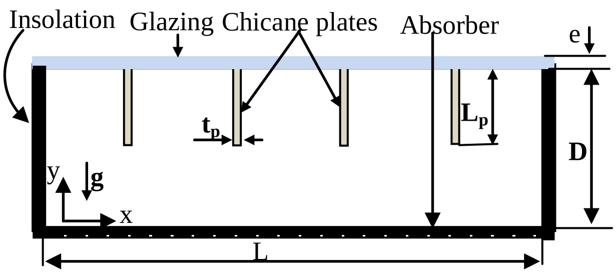


Fig. 1: Numerical model schema

A '2-D' simulation of the solar air gap spacing between plate and inner glasses equipped with obstacle is investigated. The following assumptions are imposed for the computational analysis: i- the flow is steady-state, laminar and two dimensional. ii- the thermal conductivity of the duct wall, absorber plate and roughness material are independent of temperature. iii- the duct wall, absorber plate and roughness material are homogeneous and isotropic, iv- In the cavity volume the Boussinesq approximation define the density as a constant, except for the buoyancy term in the momentum equation. The rest of fluid properties are also constant. In table 1 the air properties at the reference temperature, fixed at 300 k ,are defined, v- No-slip boundary conditions are assigned to the walls in contact with the fluid, vi- Negligible radiation heat transfer.

Table1: physical air properties: Boussinesq approximation

K (W/mK)	ρ (Kg/m^3)	C_p (J/Kg K)	μ (Kg/m.s)	β (1/K)
0.0263	Boussinesq Approximation	1007	$1.846.10^{-5}$	0.0033

2.2. boundary condition

The boundary conditions imposed in the numerical model are: i- the energy losses convection and radiation are taken into account at the outer side of the collector window. ii- an experimental correlation [16] is adopted to estimate the convective heat transfer coefficient for the flow above the glass h_{ext} = 5.7 + 3.8V where V is the wind speed, it is taken as 1.13m/s. iii- To model irradiative losses, according to [17], the black sky radiation temperature was considered equal to the air temperature. iv- The side walls are defined as adiabatic walls. v- A constant heat flux is imposed on the absorber to simulate solar radiation where q=50w/m^2. vi - The non-slipping condition is imposed on the walls.

2.3. mathematical modelling

In the case of natural convection, it is necessary to take into account the value of the number of Raleigh that can be used as a criterion for laminar and turbulent flow in the system. The problem is governed by two dimensionless parameters Ra and Pr, defined by:

$$Ra = \frac{g\beta L^4 q}{\nu\alpha k}, \ \Pr = \frac{\nu}{\alpha} \tag{1}$$

We present the system of conservation equations describing a stationary regime, 2D phenomenon in the air layer of the flat collector. These equations describe the transport of mass and heat inside the air gap, taking into account the above-mentioned assumptions.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{2}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \nu\left[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right] \tag{3}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial y} + \nu\left[\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right] - g\beta(T - T_0) \tag{4}$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right] \quad (5)$$

3. Numerical simulation

Calculations reported hereafter have been performed with the commercial code Fluent based on the finite volume method, the pressure based solver, implicit formulation, Green–Gauss node based and absolute velocity formulation were used, therefore, the double precision was selected. The governing equations of mass, momentum and energy are solved by the finite volume method in the steady-state regime, The SIMPLE algorithm (Semi-Implicit Method for Pressure Linked Equations), was adopted for coupled pressure and velocity. The Body Force Weighted discretisation scheme was used for pressure, however the second-order upwind is chosen for all other equations. The convergence criteria of 10^{-6} for the residuals of the continuity equation, the velocity components and 10^{-6} for the residuals of the energy are assumed. All simulations are started with under-relaxation factors for pressure, momentum, energy of 0.3, 0.7, and 0.9 respectively. The above mentioned values, was fixed to achieve the convergence criteria.

3.1. grid independence test

The developed model was used to study the obstacle effect on the performance of the absorber and on the convection, which causes the heat losses. The mesh tool, GAMBIT 2.2.30 was used to create the geometry and generate the mesh. The mesh independence was studied using three different meshes illustrated in Fig.2 (217728 as a coarse mesh cells, 528504 cells to a fine mesh and 709 120 cells for a finer mesh). This figure shows the temperature profiles in the middle of the cavity, between the obstacles, 2 and 3 . There is a perfect agreement in the temperature profiles corresponding to the two latest meshes. Therefore, for the calculations reported in this study, the fine mesh was chosen to optimize the relationship between accuracy and computation time.

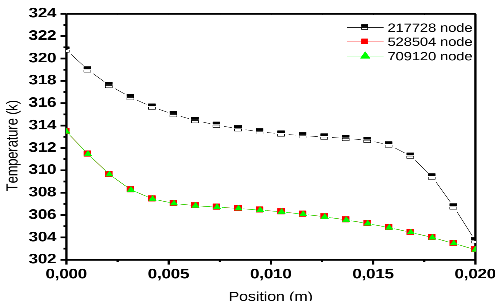


Fig.2 mesh independence for three different cases

3.2. heat transfert

4. The study of heat transfer requires a thorough knowledge of the local heat transfer coefficient (h) at the absorber. For the determining, choosing a distance between two obstacles that are divided into 100 to 200 elements to calculate their coefficient h. The desired distance reflects the phenomenon of periodicity observed in the sensor as illustrated in fig.3.

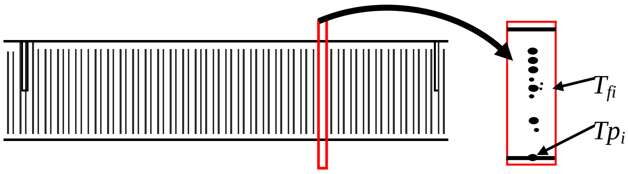


Fig.3 fragmentation pattern of an element of the absorber for calculating the local heat transfer coefficient.

Using the results of the simulations, for each elementary slice of the strip between two consecutives baffles, giving the wall temperature and the average temperature of the fluid above the point of the considered wall, one can calculate the local heat transfer coefficient by the expression.

$$h=\frac{q}{(T_p-T_{moy})} \quad (6)$$

Where T_p is the wall temperature and T_{moy} is the average temperature of fluid which is determinate by:

$$T_{moy}=\frac{\int_A \rho uTdA}{\int_A \rho udA} \quad (7)$$

The local Nusselt numbers Nu_L were calculated using custom field function CFD code for both cases (with and without partitions). $Nu_L=h\ .\ D\ /k$
where k is the air thermal conductivity.

4.1. validation

The model we have developed to solve the basic equations has been validated by comparison with results of studies [18], [19] and [20]. This comparison of results at different numbers of Ra, to those obtained by these different methods is illustrated in Table 2 and figs.4 and 5. Our model is applied to a cavity similar to that of [18], with the same conditions. The former table shows this good agreement for the Nusselt number to 1%. The figs.4 shows this good agreement through isotherms and the streamlines corresponding current lines at different Ra numbers in comparison with the above refs.. Concerning the computer code validation, the agreement obtained between our results and those available in the literature proved to be excellent.

Table 2. Comparison of Nusselt different values

Ra	Present study	Ref. [18]	Ref. [19]	Ref. [20]
10^3	**1.117**	1.118	1.118	1.115
10^4	**2.249**	2.252	2.243	2.250
10^5	**4.514**	4.545	4.523	4.569

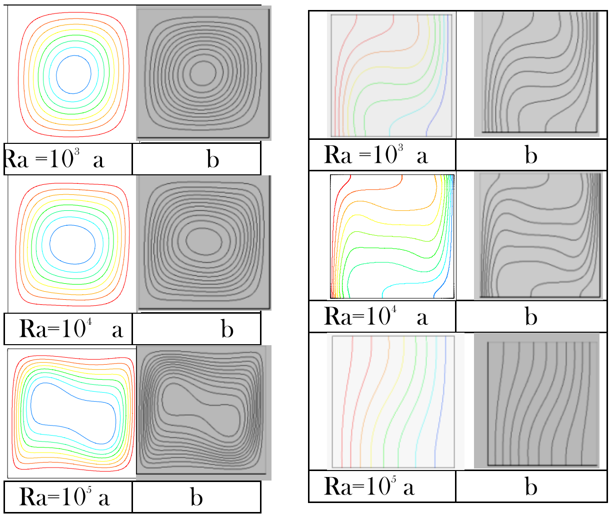
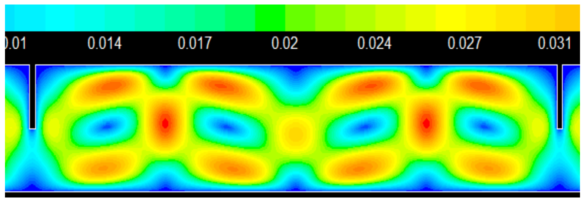


Fig.3 isothermes and streamlines for different Ra. a present study, b results of Ref.[18]

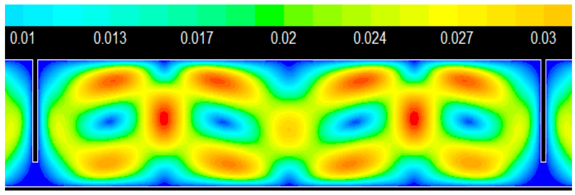
5. RESULTS

5.1. effect of the obstacles' length

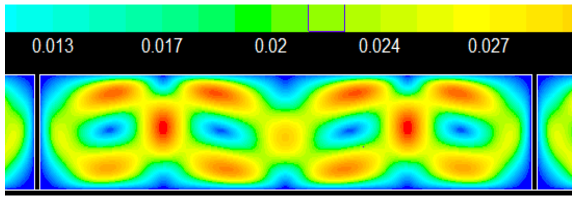
The purpose of the present study was to analyze the effects of the partitions elements in the air gap cavity of solar collector by Changing their length and number. Figure 4 shows the influence of the length of the partitions on the iso-contours speeds in the air space of the solar collector. It is observed in this figure a formation of Rayleigh Benard rolls. The effect of the partitions length does not affect the number of cells of air masses fig 5. However, the strength of circulation is influenced by this length fig4. Indeed, the increase in length leads to weaker flow intensity due to the expected increased resistance of motion of the rotating convection cells. which will curb the phenomenon of convection and therefore energy losses are also reduced because in the central zone transfer is done primarily by conduction. Then, the convection phenomenon affects only the ends of the cell. Figures, 5 and 6, show the distribution of the temperature of the surface of the absorber of the solar collector, The first remark shows that the temperature increases with partitioned cavity with respect to the cavity without partition. It is clear that the temperature of the absorber grows below the partition with the increase of the length Lp, the variation of the temperature is not distinctive beyond Lp = 1.2cm, for Lp = 2cm the temperature at the side of the partitions diminished because of the conduction where the partition joins the absorber. The temperature distribution is not uniform on the surface of the absorber, recording a temperature gap of 9.5° C between the extreme values, the influenced length is manifested near the end of the partitions.



a- Lp =1cm



b- Lp= 1.6cm



c- Lp =2cm

Fig.4 Velocity contours for different lengths of chicane

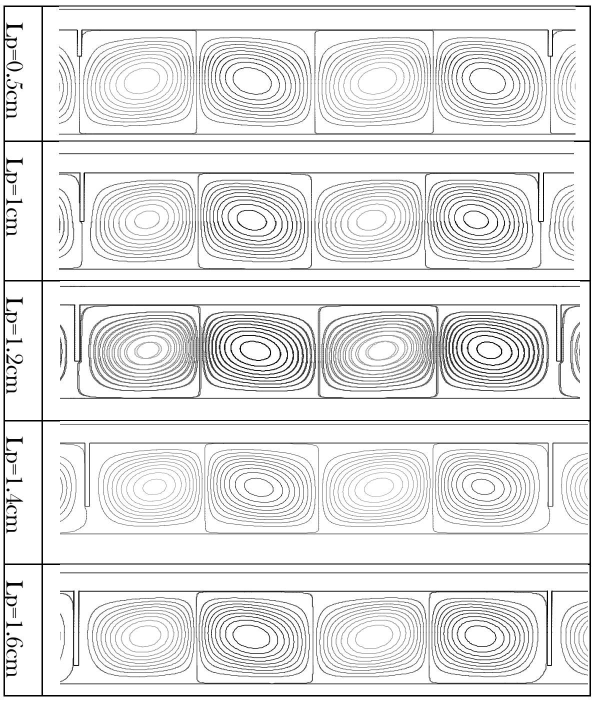


Fig.5: stream function for different value of Lp

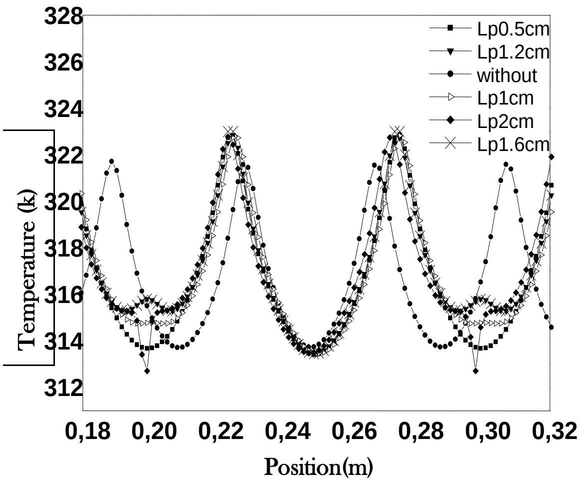


Fig.6: absorber temperature profile

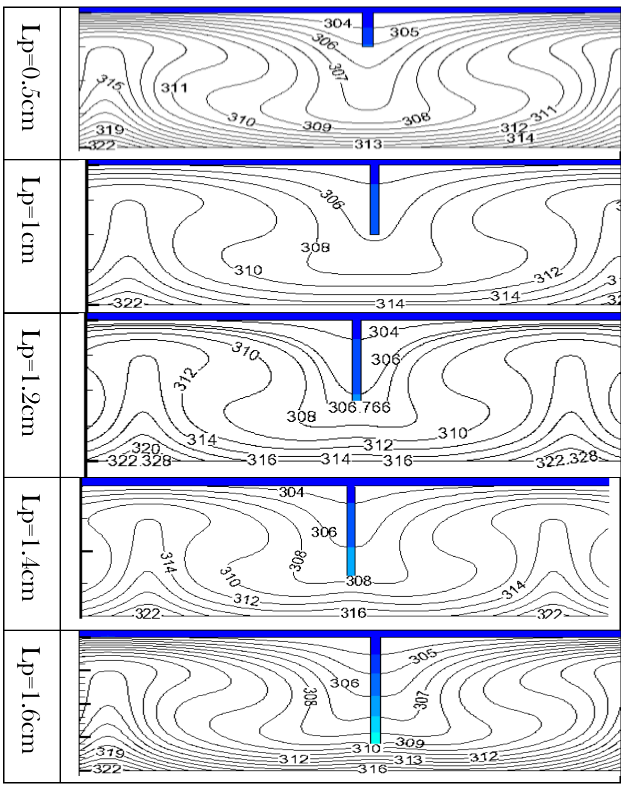


fig.7: isotherms for different value of Lp

fig.8 shows the variation of the average Nu with varying partitions length. It can be seen that the change in the average Nu was minimal for all cases, The maximum decrease in the average value of Nu as compared to the case without partitions, it was observed to be 7.56% at Ra number 10^4,

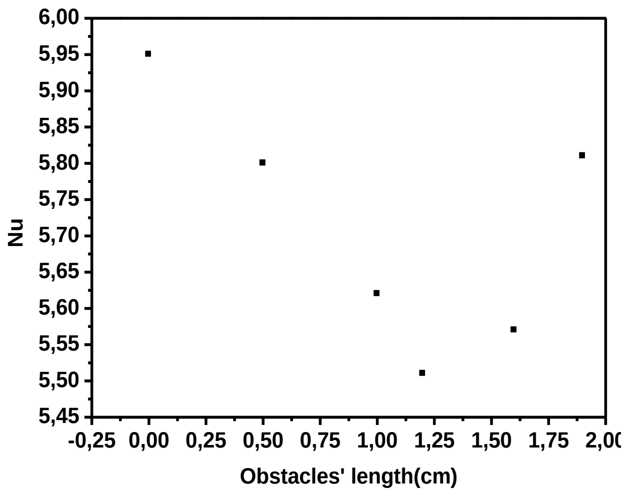
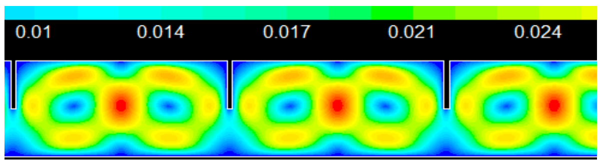


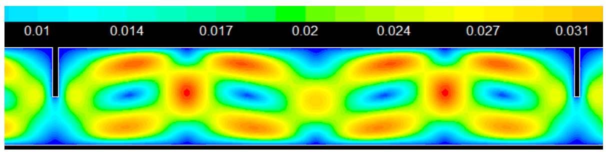
Fig. 8: Variation of the average Nusselt number on the hot wall with partitions length

5.2 EFFECT OF THE NUMBER OF PARTITIONS

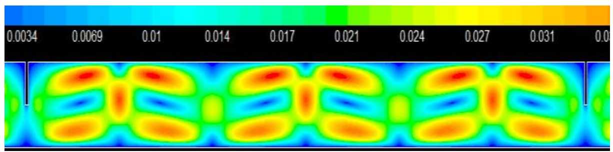
Figure 9,10 illustrates the existence of longitudinal cells that develop in the air layer having the same structure of Fig.4,5, but here, the structure varies according to the obstacle number. Indeed, when the number of obstacles is increased, the number of cells decreases fig.10. Its variation is significant both on the velocity field and on the temperature see fig. 11 and 12. It is noted that the maximum temperature gap between the case with the number of partitions 4 and 20, passes from 11.5°C to 6.6°C. This means that the high number of obstacles enhances significantly the absorber temperature and decrease the heat losses mainly by convection. The largest intensity of flow instability occurs when there is no partitions, the fin(s) blocks heat transfer, the heat transfer rate decrease when the partitions number increases.



a-24 partition



b-9 partition



c-4 partition

Fig.9: Velocity contours for different numbers of partitions

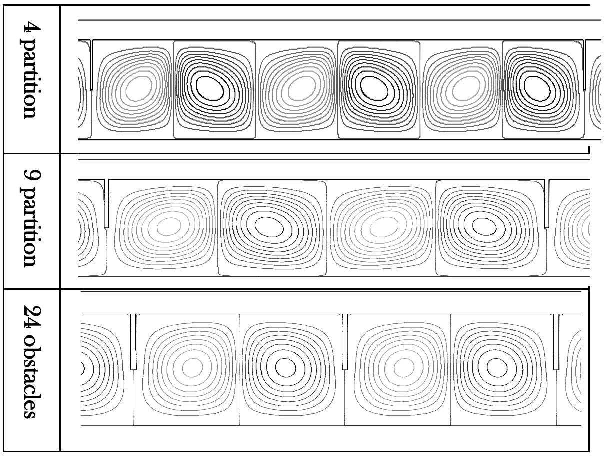


Fig.10: stream function for different partitions number

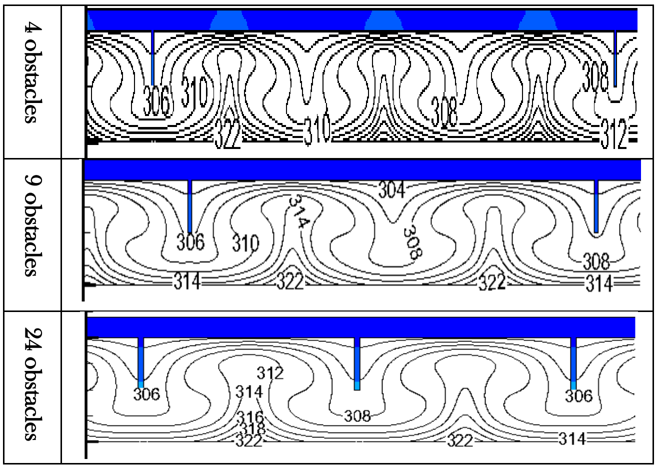


Fig.11: isotherms for different partitions number

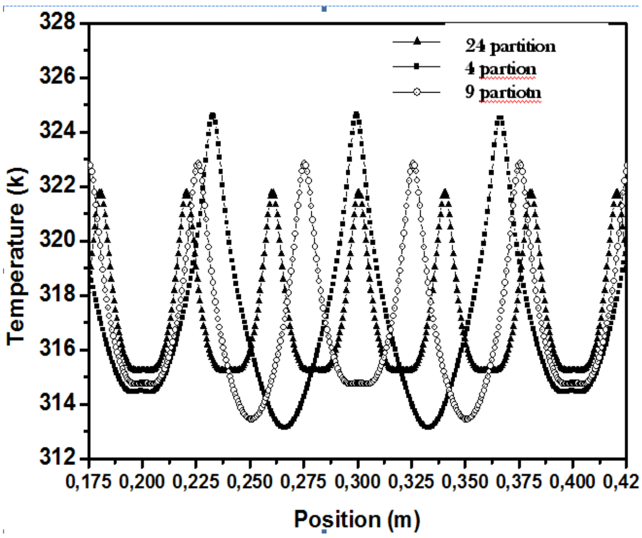


Fig.12: partitions number effect on the temperature

VI CONCLUSION

In this work, a 2D cavity, having the same dimensions of a solar collector has been studied, focusing on the presence of partitions and their effect on the collector inner flow when a heat flux is applied to the absorber. A comparative study was made between a cavity with and without partitions, whereas the length and the number of partitions were varied. In this study, the following conditions were assumed for Ra and Pr ($2.8 * 10^4$ and 0.71) respectively.

In general, increasing the number of partitions and their lengths contribute to the reduction of heat loss by convection and help to increase the temperature of the absorber. The optimal length of the partitions is close to 66.6% of the air gap.

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