NUMERICAL STUDY OF HEAT TRANSFER IN A FLAT PLAT THERMAL SOLAR COLLECTOR WITH PARTITIONS ATTACHED TO ITS GLAZING

by

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Heat transfer in the air-gap of a horizontal flat plat thermal solar collector contain partitions attached to its glazing has been studied numerically. The absorber and the glazing are kept at constant and different temperatures, while the vertical walls (insulation) were kept adiabatically. A conjugate formulation was used for mathematical formulation of the problem and a computer program based on the control volume approach and the SIMPLER algorithm was used. The main aim of the current paper is to study numerically the effects of number of fins and their length on the air pattern and heat transfer. It was observed that the heat transfer rate through the air-gap is affected greatly and hence can be controlled by the number of attached fins to the glazing of the solar collector as well as the fin lengths, and the addition of partitions reduces the heat losses by convection by 90%.

Key words: *thermal solar collector, partitions, FLUENT, natural convection, thermal radiation, efficiency*

Introduction

In order to improve the performance of the flat plat thermal solar collector, the researchers offer several techniques. One of these techniques simple, less expensive, it is to add cells anti-loss attached to the glazing of the solar collector. The objective of these partitions is to prevent convective movements of the air in the air-gap of the collector. The purpose of this work is to study the influence of these partitions on the heat transfer in the collector, and to determine the number and the optimal length. Several studies have been made. The studies of Ahmed and Abid [1], indicate that the increase in the number of partitions reduces the heat transfer to the outside. The studies of Tong and Grener [2] show that the partitioning could produce a reduction of heat transfer in the cavities. They have studied the effect of a vertical wall fine on the natural convection in an enclosure filled with air. They have also studied the influence of the position of the baffle on the Nusselt number and have shown that its position in the middle of the enclosure produced the greater reduction of the thermal transfer, also they have shown that the partitioning could produce a reduction on the thermal transfer. King and Narayanaswamy [3] have studied the radiation effects on the natural convection in a rectangular cavity contain partitions, they noticed that the increase in the number of partitions cause a decrease in the value of the average Nusselt number in the presence of the radiation effects. El-Sherbiny [4] has studied the natural convection in a rectangular cavity partitioned into a function of the height and the thickness of the partition, the results obtained are in the form of

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the isotherms and streamlines. It has found that the Nusselt number decreases with the increase in the thickness and height of partitions, and increases with the increase in the Rayleigh number. Bahlaoui *et al.* [5] have studied numerically the natural convection coupled with the radiation in a rectangular cavity partitioned, they are interested in the influence of some parameters on the heat transfer as the value of the emissivity of surfaces, the location of the partition, and its height, its thickness is negligible. They reached that the presence of the radiation effects causes a good homogenization of the temperature and that the increase of parameters: height of partition, L_p, and the emissivity causes a decrease in the thermal transfer. Frederick [6] has studied numerically the natural convection in an inclined square enclosure with partitions attached to its cold wall with Rayleigh number 10^3-10^6, he reached that the partition causes convection suppression, and heat transfer reductions of up to 47% relative to the undivided cavity at the same Rayleigh number. Heat transfer reduction depends on Rayleigh number, partition length, and inclination. For long partitions, transition to bi-cellular flow occurs. At high Rayleigh numbers the heat transfer reduction is affected by secondary buoyancy forces, generated by the partition. Bilgen [7] has studied the natural convection in cavities with a thin fin on the hot wall, the horizontal walls are kept adiabatic, a thin fin is attached on the active wall, and he reached that the heat transfer may be suppressed up to 38% by choosing appropriate thermal and geometrical fin parameters. Bouali *et al.* [8] has studied radiation-natural convection heat transfer in inclined rectangular enclosures with multiple partitions, it was found that the total heat transfer in the enclosure is increased under thermal radiation heat flux and reduced significantly with increasing the number of partitions.

Khansila *et al.* [9] have studied numerically the natural convection in a square enclosure heated from bellow contained two partitions attached to its hot wall, they have studied the effect of high an location of the partition, The results are presented in forms of streamlines, isotherms and heat lines. It can be found that the flow field is single cell when the height of the partitions is less than 0.5. The flow field and heat transfer increase when the value of Darcy number is increased. Zemani *et al.* [10] have studied the effect of partial partitions on natural convection in air filled cubical enclosure with hot wavy surface, the geometry is a cube with wavy hot surface (three undulations) and three partitions the study showed that these partitions caused decrease of up 40% in the heat transfer with respect to the case at the same Rayleigh number. Also, the increase of the high of partitions causes a decrease in the average Nusselt number.

Yousefi *et al.* [11] have studied the effect of the partition inclination on the natural convection in a square enclosure, it found that the maximum and minimum heat transfer occurs at the partition angle of 45° and 15°, respectively. Koca *et al.* [12] have studied the effect of the thermal conductivity and the thickness of the partition in a square enclosure, It is found that both heat transfer and flow strength strongly depend on the thermal conductivity ratio of the solid material of partition and Rayleigh numbers. Also, thickness ratio is important for higher Rayleigh numbers. Wahebl *et al.* [13] have studied the natural convection in a rectangular enclosure containing two inclined partitions, the effect of the inclination of partition is studied, the results show that the maximum heat transfer rate occur at 45° for each partition. It is more than the non-partitioned case by 1.15%. The minimum heat transfer rate occurs at 90° for each partition. It is less than the non-partitioned case by 53%.

Haghighi and Vafai [14] have studied the effect of length and location of a vertical partition in a square cavite, the location is taken for aspect rations from 1 to 4, it found that the presence of the partition can more effectively reduce heat transfer when it is vertically fixed on the top or bottom wall. Except for a shorter horizontal partition, which can cause slightly more heat transfer reduction when located close to the mid-height of a square cavity as compared to an equivalent vertical case. Khatamifar *et al.* [15] they have studied the effect of the thickness and the thermal conductivity of the partition, the dimensionless partition thicknesses varied from (0.05, 0.1, and 0.2), and 3-D partition positions (0.25, 0.5, and 0.75), The results show that the average Nusselt number increases with the Rayleigh number but decreases with partition thickness. It is also found that the partition position has a negligible effect on the average Nusselt number for the whole range of Rayleigh number considered. Ghazian *et al.* [16] have studied experimentally the natural convection in an enclosure with partial partitions at different angles, the experimental data show that the average Nusselt number increases with increasing Rayleigh number for all the inclination angles and spacing. Rayleigh number there exists an optimum angle that maximizes the average Nusselt number from the hot wall. It is also observed that for acute angles, the local Nusselt number decreases along the hot wall at each Rayleigh number, while there is a location at which the Nusselt number reaches its maximum value and reduces afterwards for obtuse angles.

The objective of this work is to study the influence of the attachment of partitions on the performance of the thermal solar collector, As well as study the effect of the number and lengths of these fins and choose the ideal values as well as their impact on the thermal field and dynamic.

Mathematical formulation of the problem

Figure 1 explained the studied geometry is a rectangular cavity contains partitions attached to the glazing, in this study we are interested in the effect convective, which is happening within the air-gap. The height of the cavity is H and its length L, the dimensionless length of partitions L_p. The problem is being studied is 2-D.

Boundary conditions The horizontal walls (glazing and absorber) are isotherms at different temperatures, and the vertical walls are adiabatic(isolation):

– horizontal wall (absorber): $T = T_c$,
– horizontal wall (glazing): $T = T_h$,
– vertical walls (insulation): $\partial T/\partial x = 0$, and
– no-slip conditions in all walls: $u = v = 0$

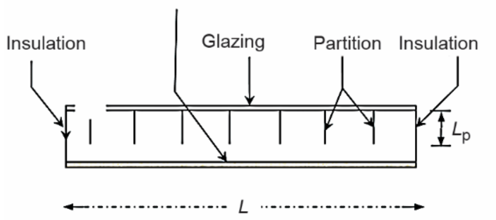

Figure 1. Field of study

After introduction of the following assumptions, we can establish the various equations necessary to the resolution of the problem considered in this study.

– The flow is stationary and 2-D.
– The fluid is Newtonian and incompressible.
– The flow generated is laminar.
– Work induced by the viscous forces and pressure, is negligible.
– The physical properties of the fluid are constant apart from the mass density which obeys the approximation of Boussinesq values in the term of the buoyancy.

Therefore:

$$\rho = \rho_0 \left[1 - \beta (T - T_0)\right] \tag{1}$$

– The power density dissipated is negligible.

After simplifications, the equations of the problem will be [17]:

The continuity: the continuity equation is written:

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

The equation of a quantity of movement:

– Following x: the equation of the quantity of movement is written:

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

– Following y: the equation of the quantity of movement is written:

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

The equation of energy: the energy equation is written:

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

– The average Nusselt number is given by the expression:

$$\overline{\mathrm{Nu}}=\frac{Q}{\lambda\Delta T} \tag{6}$$

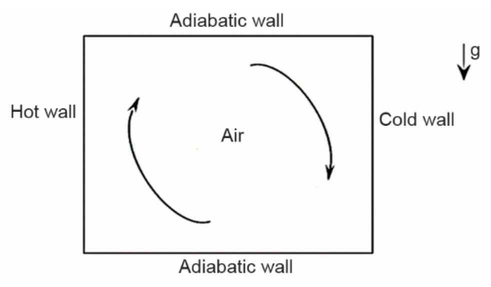

Figure 2. Geometry of a square cavity differentially heated [21]

Method of resolution

The previous equations are solved by the use of the software FLUENT [18] which based on the finite volume method which presented by Spalding and Patankar [19], this method is based on the discretization of the transport equation for volumes finished discreet. The coupling pressure-speed is treaty to the aid of the algorithm SIMPLER.

Validation and verification

In order to verify the accuracy of the numerical results obtained in the present work. A validation of the numerical code was made taking into account certain numerical studies available in the literature. The results of [20], obtained in the case of a square cavity in 2-D, containing air, have been used to test our simulation by the fluent software, fig. 2.

The comparison of the isotherms, streamlines and the value of the number of average Nusselt number were made in recital different values of the Rayleigh number: 10^3 and 10^5.

Isothermes: The comparison of isotherms is shown in the fig. 3.

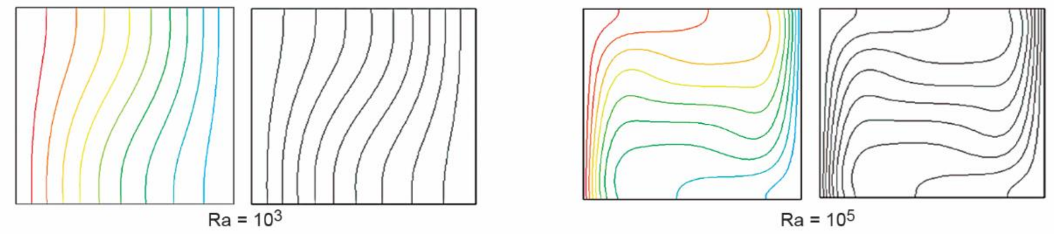

Figure 3. Comparison of the isotherms obtained in the present work and those obtained by [20]; (a) present work, (b) results of [20] (*for color image see journal web site*)

Streamlines: The comparison between streamlines is show in fig. 4.

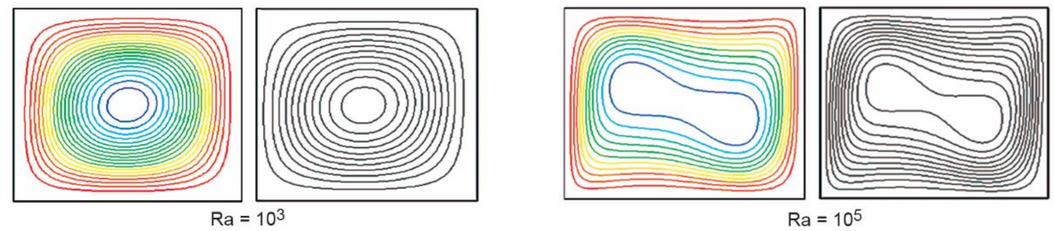

Figure 4. Comparison of the streamlines obtained in the present work and those obtained by [20]; (a) present work, (b) results of [20] (*for color image see journal web site*)

The average Nusselt number $\overline{Nu}$: Comparaison between somes values of the average Nusselt number is presented in the tab. 1. Based on the comparisons of the isotherms, streamlines, and the values of the average Nusselt number in the Nusselt obtained means with the code for the calculation fluent with the results of references, one finds that our results are similar and in agreement with those presented by different authors with a percentage of acceptable error. Thus, the comparison presents an excellent concordance, which allowed us to validate our procedure for numerical simulation.

Table 1. Comparison of the values of the average Nusselt number with the references values

Ra	Present works	[20]	[21]
10^3	1.116	1.118	1.118
10^5	4.549	4.545	4.523

Results and discussions

In this work the studied phenomenon is the natural convection in a rectangular cavity partitioned. The Inferior wall (the absorber) is aluminum, and the outer wall (the glazing) is glass. The cavity (the air-gap of the thermal solar collector), contain the air. The objective is to study the influence of partitions (their lengths and number) on the heat transfer in the cavity. In this study it was assumed a Rayleigh number is Ra = $2.51 \cdot 10^4$, and a Prandtl number Pr = 0.71. The results obtained are in the form of the isotherms, streamlines and the value of the average Nusselt number.

Table 2. Values of the average Nusselt number for different meshes

Mesh	20 × 400	30 × 600	40 × 800
Nu	73.16	74.87	75.34

To choose a good mesh, it has calculated the value of the Nusselt number means for different mesh sizes: 20 × 400, 30 × 600, and 40 × 800. The results are presented in tab. 2.

Given that the variation in the number of average Nusselt is not significant between the meshes 30 × 600 and 40 × 800, for this we chose the last mesh.

It considers that the solar collector is placed horizontally perpendicular to the field of gravity and the temperature gradient; the results obtained are presented below.

Effect of the length of the partition L_p:

– The thermal field: the isotherms are presented in the figures:

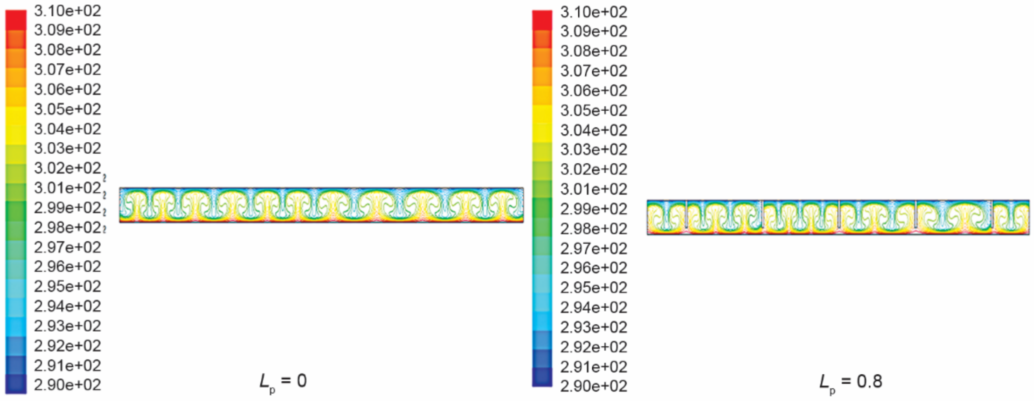

Figure 5. Isotherms for different values of L_p for Ra = $2.51 \cdot 10^4$, N = 5
(*for color image see journal web site*)

Figure 5 shows the isotherms for different values of L_p and for a number of partitions equal to 5. We note a variation in the distribution of the temperatures in the air-gap, when it increases the length of partitions, which means the influence of L_p on the thermal transfer by natural convection in the air-gap, therefore it is concluded that in the case horizontally the increase in the length of partitions cause a variation of thermal transfer by natural convection therefore a variation of thermal losses toward the outside. The movement of the fluid is forced to be close to the absorber.

– The dynamic field: the dynamic field in the form of streamlines is presented in fig. 6.

Figure 6 represents streamlines for different lengths of the partitions L_p which varies from 0 to 0.8, and a number of partition equal to 5, we note that the increase in L_p cause a variation of speeds, it follows that the presence of partitions has an influence on the convective movements of the air, so an influence on the performance of the solar thermal solar collector.

– The average number Nusselt: the variation in the average Nusselt number is presented in fig. 7.

Figure 7 shows the variation in the number of average Nusselt depending on the length of partitions, for a number of partitions equal to 5, we note a gradual decrease in the value of the Nusselt number when the length of partitions increases, which means a decrease of the thermal losses by natural convection to the outside Ra = $2.51 \cdot 10^4$, N = 5.

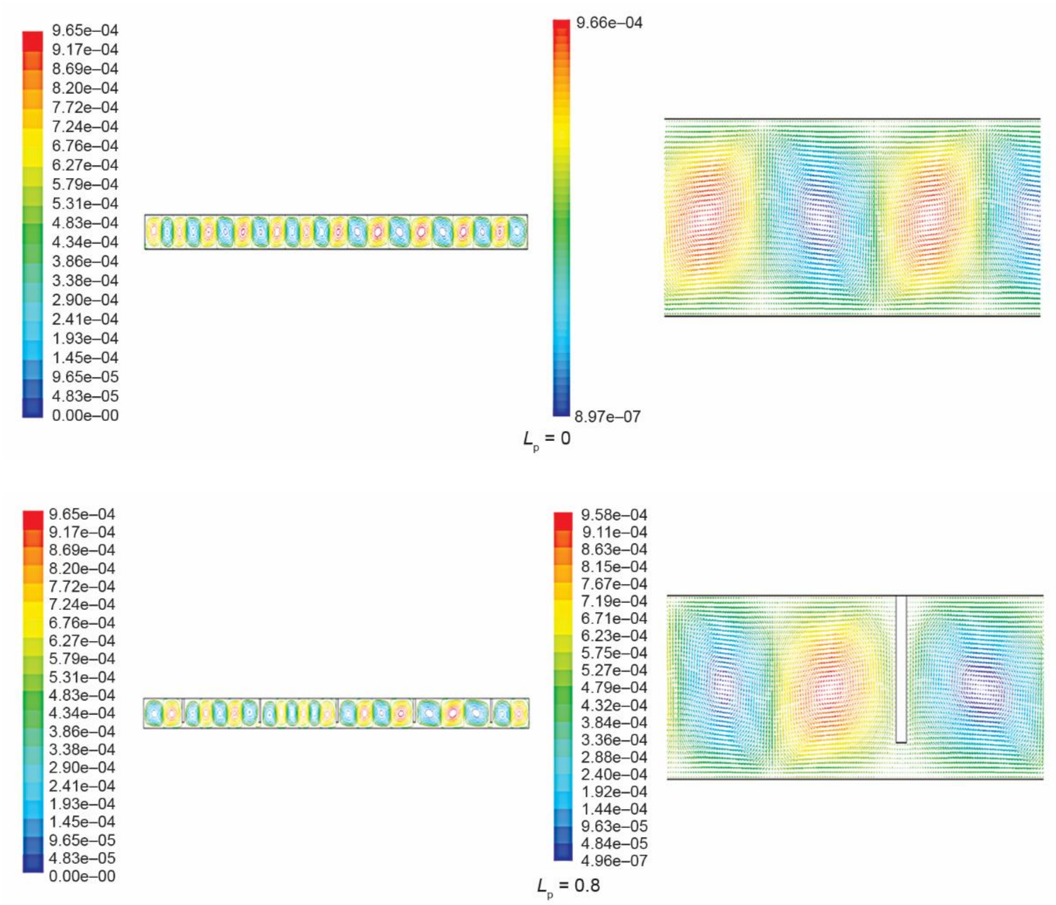

Figure 6. Streamlines and vectors velocity for different values of L_p, for Ra = $2.51\cdot10^4$, N = 5 (*for color image see journal web site*)

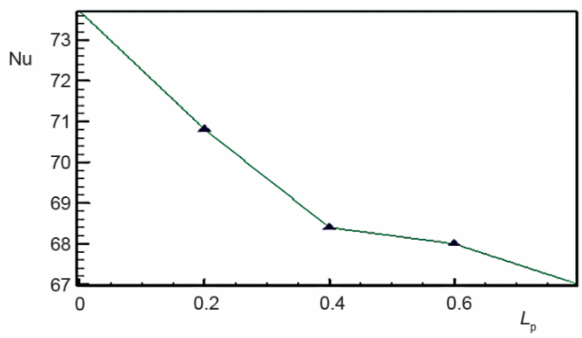

Figure 7. Variation in the number of average Nusselt for different values of L_p

Effect of the number of partitions:

– The thermal field: The thermal field is presented in the form of the isotherms:

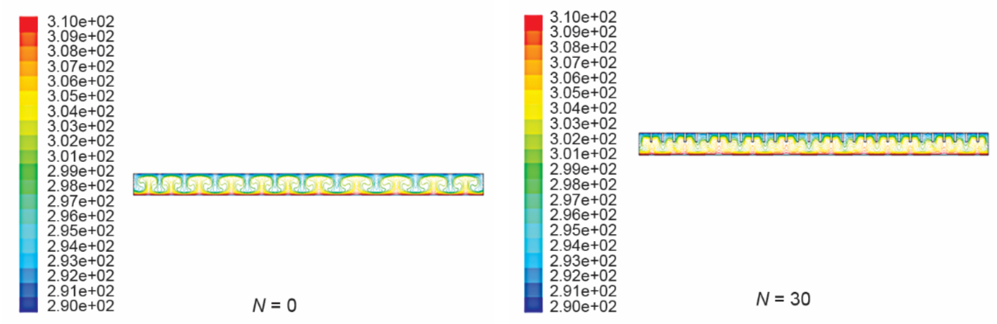

Figure 8. Isotherms for different values of the number of partitions *N* for Ra = 2.51·10^4, L_p = 0.4 (*for color image see journal web site*)

Figure 8 shows the isotherms for different numbers of partitions, in order to study the influence of the number of partitions. It has set their length to L_p = 0.4, and there has varied their number from 0 to 30. We note in this figure that the increase in the number of partitions cause a variation in the distribution of the temperatures in the air-gap of the solar collector, therefore an influence on the performance of the solar collector.

– The dynamic field: the dynamic field in the form of streamlines and velocity vectors:

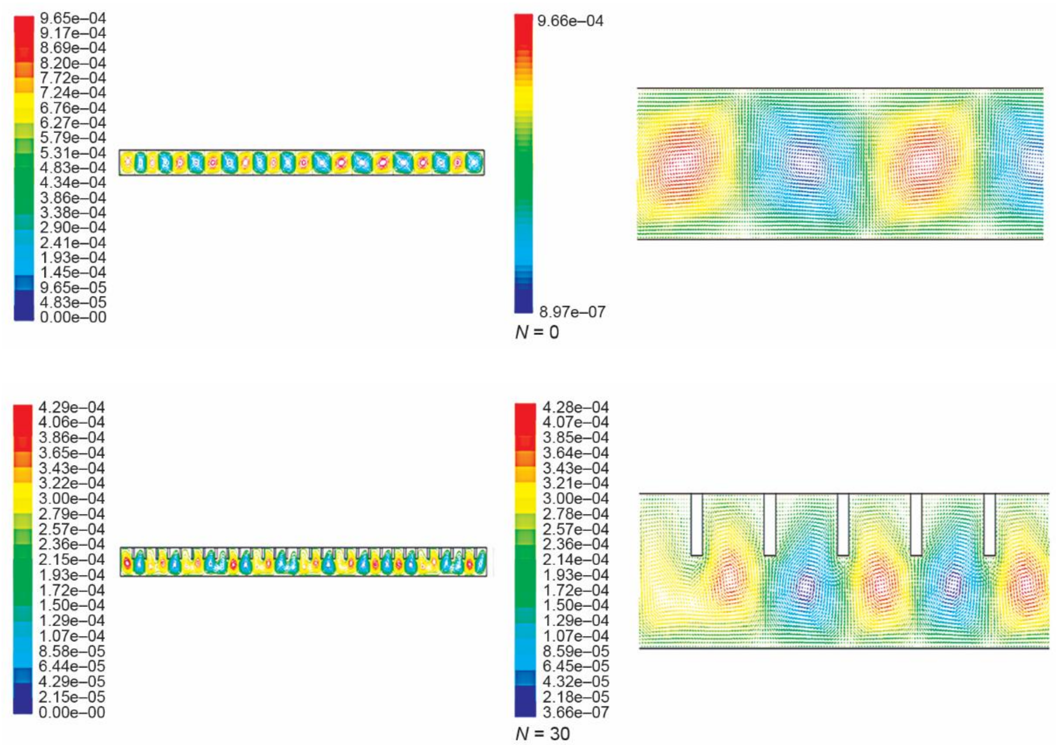

Figure 9. Streamlines and velocity vectors for different values of the number of partitions *N*, for Ra = 2.51·10^4, L_p = 0.4 (*for color image see journal web site*)

The average Nusselt number:

Figure 9 shows the streamlines for different numbers of partitions to a length of 0.4. We note that the movement of the air is close to the absorber below the partitions and form of waves when there is an increase in the number of partitions, which means an impediment of convective movements of the air in the air-gap, this impediment causes a reduction of losses convective. It is observed that the reduction is at a maximum when $N = 30$.

– Variation in the values of the average Nusselt number is represented in fig. 10.

Figure 10 represents the variation in the number of average Nusselt in function of the number of partitions, we note a decrease in the values of this last when the number of partitions increases, it is concluded that the increase in N causes a minimization of convective losses, but it should not exceed a certain number of partitions for do not influence the transmission of the glass, so it must take an optimal number.

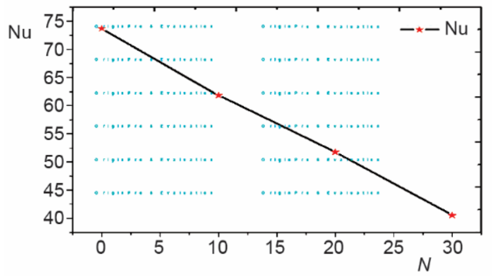

Figure 10. Variation in the number of average Nusselt for different numbers of partitions does for Ra = 2.51·10^4

Conclusion

In this study it indicate that the addition of partitions to the glazing wall can enhance the efficiency of the flat plat thermal solar collector with reduce of thermal losses by natural convection. The effect of number and length of partitions is studied numerically. According to the previous results, it can be concluded that:

- The presence of partitions causes a decrease in the value of average Nusselt, therefore a minimization of thermal losses to the ambient.
- The increase in the length of the partitions cause a diminution the flow exchanged by convection.
- The increase in the number of partitions causes a decrease in the value of average Nusselt, and therefore a minimization of thermal losses toward the outside, by the impediment of convective movements of the air.
- The increase in the number of partitions causes a lowering of the rollers to the bottom near the absorber, and makes on obstruction of convective movements of the air.

Nomenclature

g – gravity, [m^{2}s^{-1}]
H – height of the air-gap, [m]
L – length of the thermal solar thermal, [m]
L_p – dimensionless length of partitions
l – partitions length, [m]
N – number of partitions
Nu – Nusselt number
P – pressure, [Pa]
Pr – Prandtl number
Q – heat flow density, [Wm^{-2}]
R_a – Reynolds number
T_h – the absorber temperature, [K]
T_c – the glazing temperature, [K]
T_0 – reference temperature, [K]
x, y : x, y – Cartesian co-ordinates, [ms^{-1}]
v – speed following y, [ms^{-1}]
u – speed following x, [ms^{-1}]

Greek symbol

α – thermal diffusivity, [m^{2}s^{-1}]
β – coefficient of volume dilatation, [K^{-1}]
λ – thermal conductivity, [Wm^{-1}K^{-1}]
υ – kinematic viscosity, [m^{2}s^{-1}]
ρ_0 – reference density, [kgm^{-3}]

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