Indonesian Journal of Science & Technology 3 (2) (2018) 95-104

Indonesian Journal of Science & Technology

Journal homepage: http://ejournal.upi.edu/index.php/ijost/

Numerical Simulation of Natural Convection in the Air Gap of a Vertical Flat Plat Thermal Solar Collector with Partitions Attached to Its Glazing

Adel Laaraba [1]*, *Abderrahmane Khechekhouche*[2]

[1]Department of physics. University of Batna (05000) Batna, Algeria

[2]Renewable Energy Research unite in Arid Zones, University of El-Oued (3900) , Algeria

*Correspondence: E-mail: laaraba_adel_83@yahoo.fr

ABSTRACT

Heat transfer in the air gap of a vertical flat plat thermal solar collector containing 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. Experimental results showed that interesting phenomena happened, especially in the heat transfer process. It was observed that the heat transfer rate through the air gap is affected greatly and 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 60%. This study will brings good advantages for further uses, especially related to the heat transfer phenomena in the solar applications.

ARTICLE INFO

Article History:

Submitted/ Received 01 May 2018

First Revised 05 Jul 2018

Accepted 20 Aug 2018

First available online 28 Aug 2018

Publication Date 01 Sep 2018

Keywords:

Thermal solar collector,

partitions,

fluent,

natural convection,

thermal efficiency.

1. INTRODUCTION

In order to improve the performance of the flat plat thermal solar collector, many researchers offer several techniques. One of these techniques that must be simple and less expensive is when adding system by 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 purposes of this work are 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 (Amraqui *et al.,* 2011), indicate that the increase in the number of partitions reduces the heat transfer to the outside. (Bahlaoui *et al.,* 2008), 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 Lp and the emissivity causes a decrease in the thermal transfer Ramon. Experimental and numerical studies in heat transfer (Sacadura, 1980; Spalding *et al.*, 1972; Amraqui, 2009, Davis, 1983), have been done by several researchers. (Frederick, 1989) has studied numerically the natural convection in aninclined 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 bicellulaire flow occurs. At high Rayleigh numbers the heat transfer reduction is affected by secondary buoyancy forces, generated by the partition. (Khansila *et al.,* 2014), 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 a high 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.,* 2014) 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.,* 2011), have studied the effect of the partition inclination on the natural convection in a square enclosure, it was found that the maximum and minimum heat transfer occurs at the partition angle of 45° and 15° respectively. (Koca *et al.,* 2013), 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. Jamal Hameed Waheb et al [18], 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 occur at (90°) for each partition. It is less than the non-partitioned case by 53%. (Haghighi *et al.,* 2014), 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 is 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

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

close to the mid-height of a square cavity as compared to an equivalent vertical case. (Khatamifara *et al.,* 2016), 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 three dimensionless 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. The effect of the seasons, solar radiation and the thickness of the glazing influence any solar system, that was proved on a solar device in the experiment made by (Khechekhouche *et al.,2017*; Khechekhouche *et al.,* 2017). The works of (Benabderrahmane *et al.,* 2017) shows that the use of dimpled absorber can enhance the performance of the parabolic trough collector. (Ghodbane *et al.,* 2017), studied the use of the parabolic trough collector as a solar system for heating water, the study based on numerical simulation.

The objective of this study is to see the improvement of the flat solar efficiency by adding the partitions on the inside face of the glazing and to see the effect of the partitions on the thermal losses by natural convection.

2. MATERIALS AND METHODS

2.1. Mathematical formulation of the problem.

The studied geometry is a rectangular cavity contains partitions attached to the glazing. In this study we focused on the effect convective, which is happening within the air gap. We tested the height of the cavity (H) and its length (L) and the dimensionless length of partitions (Lp). The problem is being studied is two-dimensional.

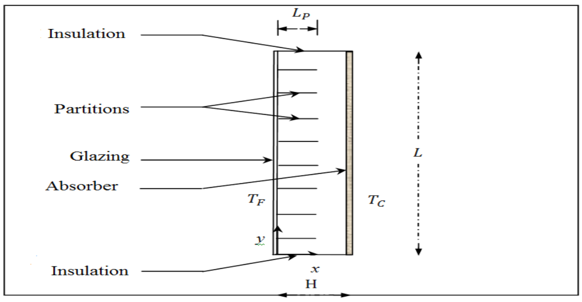

Figure 1. Field of study

1. **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): $\frac{\partial T}{\partial x} = 0$

No-slip conditions in all walls: u=v=0

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

a. The flow is stationary and two-dimensional.
b. The fluid is Newtonian and incompressible.
c. The flow generated is laminar.
d. Work induced by the viscous forces and pressure, is negligible.
e. 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.
f. Therefore:

$$\rho = \rho_0 \left(1 - \beta \left(T - T_0\right)\right) \quad (1)$$

g. The power density dissipated is negligible.

After simplifications, the equations of the problem will be Sacadura (1980).

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

2. **The continuity:**
   The continuity equation is written:

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

3. **The equation of a quantity of movement:**

Following equation of the quantity of movement is written:

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

Following equation of the quantity of movement is written:

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

4. **The equation of energy:**
   the energy equation is written:

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

The average Nusselt number is given by the expression:

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

2.2 Method of resolution:

The previous equations are solved by the use of the software Fluent based on the Finite Volume Method, which presented by Spalding *et al.*, (1972). 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.

2.3 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 by Amraqui (2009) obtained a square cavity in 2D, (containing air), has been used to test our simulation by the fluent software.

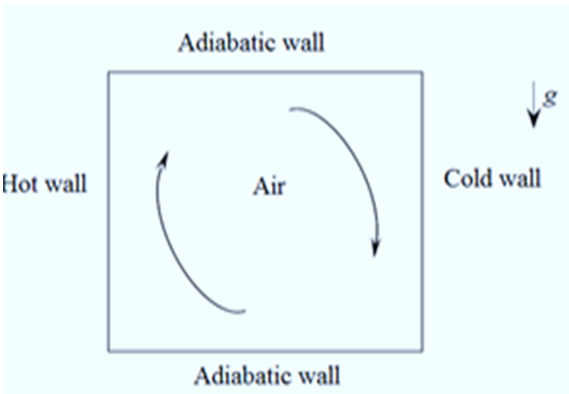

Figure 2. Geometry of a square cavity differentially heated Vahl (1983)

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

1. Isothermes

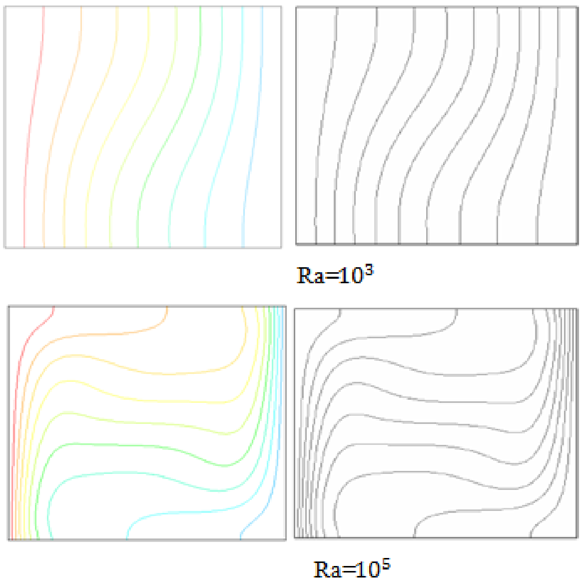

Figure 3. Comparison of the isotherms obtained in the present work and those obtained by Amraqui (2009)

Figure 3. Shows the analysis of isoterm condition. We compared Reynold number of 10^3 and 10^5 .

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

2. Streamlines

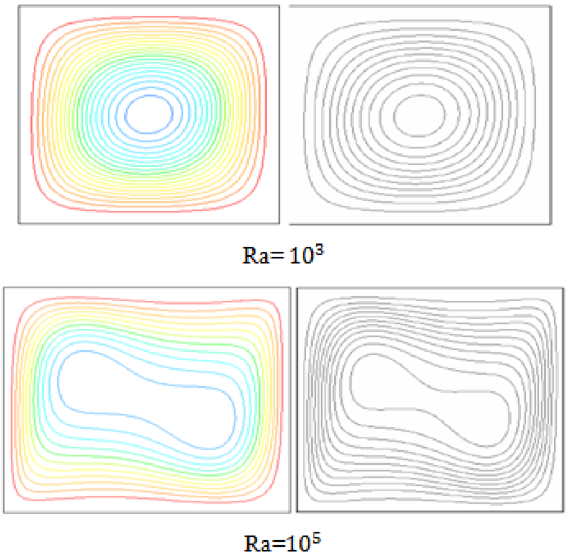

Figure 4. Comparison of the streamlines obtained in the present work and those obtained by Amraqui (2009)

Figure 4 shows comparison of streamlines. We compared Reynold number of 10^3 and 10^5.

3. The average Nusselt number $\overline{Nu}$

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

Ra	Present Works	Amraqui (2009)	Davis (1983)
10^3	*1.116*	1.118	1.118
10^5	*4.549*	4.545	4.523

Based on the comparisons of the isotherms, see **Table 1.** 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. 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.

3. 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=$2.51.10^4$, 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. The thermal proprieties of the fluid (the air) are:

Density: ρ=1.204 Kg/m^3.

Specific heat Cp=1006 $J/Kg.K$.

Thermal conductivity : λ=0.0257 $W.m^{-1}.K^{-1}$.

Dynamic viscosity: μ=1.81. $10^{-5}\ Kg.m^{-1}.s^{-1}$.

The length of the collector is L=1m and its high is H=0.025m. The temperature of the absorber is Tc=310K, and the temperature of the glazing is Tf =290 K. The thickness of the partitions is e=0.0025m. The number of Rayleigh is calculated by the expression:

Ra= Gr. Pr

The number of grashof is calculated by the expression:

$$Gr = \frac{g\beta\Delta TH^3}{\nu^2}$$

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 the **Table 2**:

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

Mesh	20×400	30×600	40×800
$\overline{Nu}$	73.16	74.87	75.34

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

Given that the variation in the number of average Nusselt is not significant between the meshes 30×600 and 40×800. Thus, we chosed 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:

1. **Effect of the length of the partition Lp**

Figure 5 shows the isotherms for different values of Lp and for a number of partitions equal to 30. We note a variation in the distribution of the temperatures in the air gap. It increases the length of partitions, which means the influence of Lp on the thermal transfer by natural convection in the air gap, therefore it is concluded that in the vertical case 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.

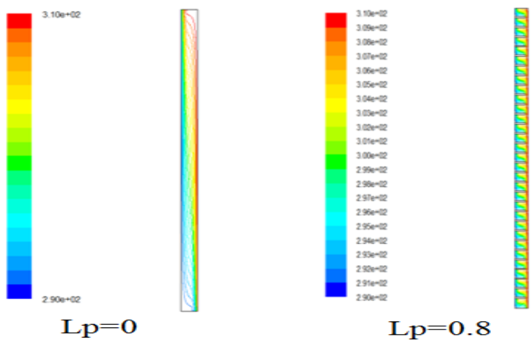

Figure 5. Isotherms for different values of Lp for *Ra*=2.51. 10^4, *N=30.*

Figure 6-9 represents streamlines for different lengths of the partitions (Lp), Varlying from 0 to 0.8 and a number of partition equal to 30. We noted that the increase in Lp cause a variation of velocity. It follows that the presence of partitions has an influence on the convective movements of the air. Thus, so an influence on the performance of the solar thermal solar collector.

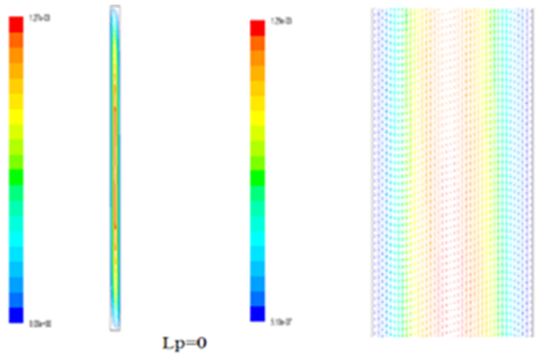

Figure 6. Streamlines and vectors velocity at *Lp* = 0, for *Ra*=$2.51.10^4$, *N=30*

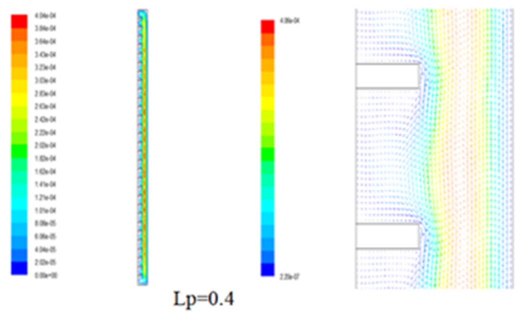

Figure 7. Streamlines and vectors velocity at *Lp* = 0.40, for *Ra*=$2.51.10^4$, *N=30*

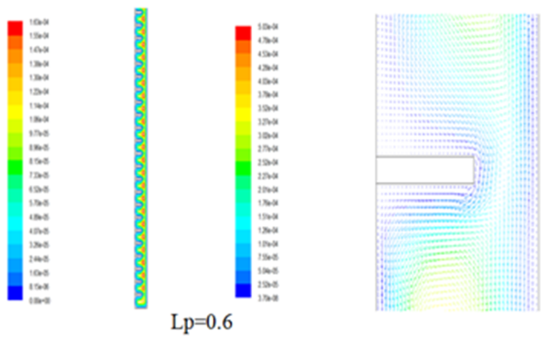

Figure 8. Streamlines and vectors velocity at *Lp* = 0.60, for *Ra*=$2.51.10^4$, *N=30*

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

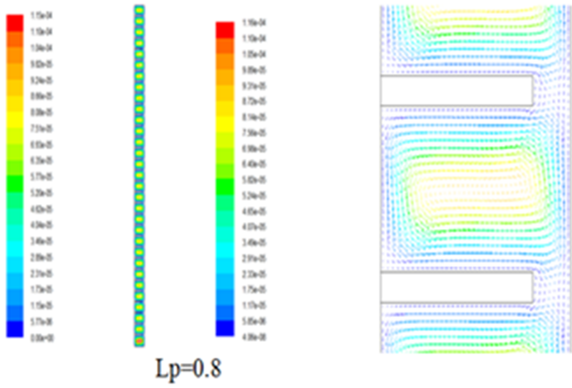

Figure 9. Streamlines and vectors velocity for different values of Lp, for *Ra*=$2.51.10^4$, *N=30*

Figure 7 shows the variation in the number of average Nusselt depending on the length of partitions, for a number of partitions equal to 30, 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=$2.51.10^4$, N=30.

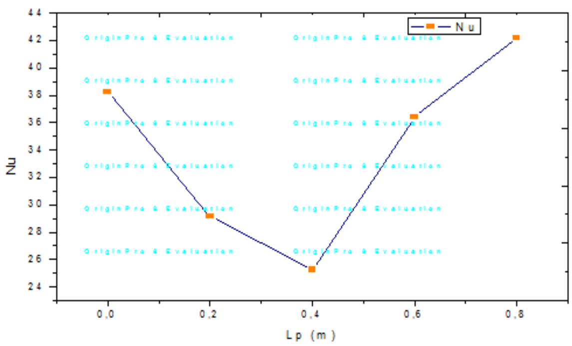

Figure 10. Variation in the number of average Nusselt for different values of Lp.

2. Effect of the number of partitions

Figure 11 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 Lp =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** represents streamlines for different numbers of partitions N, varying from 0 to 30, and lenght of Lp=0.4. We noted that the increase in N cause a variation of velocity values. It follows that the presence of partitions has an influence on the convective movements of the air. Thus, an influence on the performance of the solar thermal solar collector.

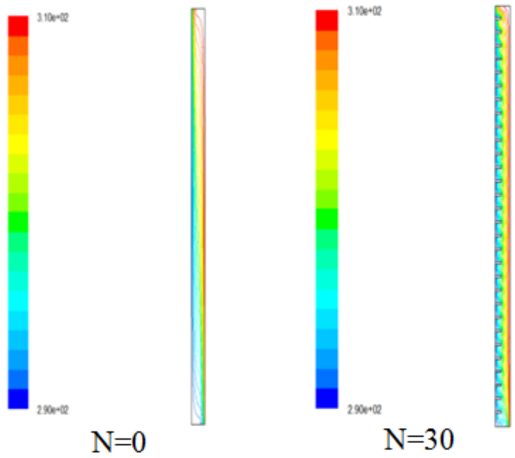

Figure 11. Isotherms for different values of the number of partitions N for *Ra*=$2.51.10^4$, *Lp=0.4*

3. The average Nusselt number

Figure 12 shows the streamlines for different numbers of partitions to a length of 0.4. We noted 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. This means an impediment of convective movements of the air in the air gap, this impediment cause a reduction of losses convective. It is observed that the reduction is at a maximum when N=30.

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

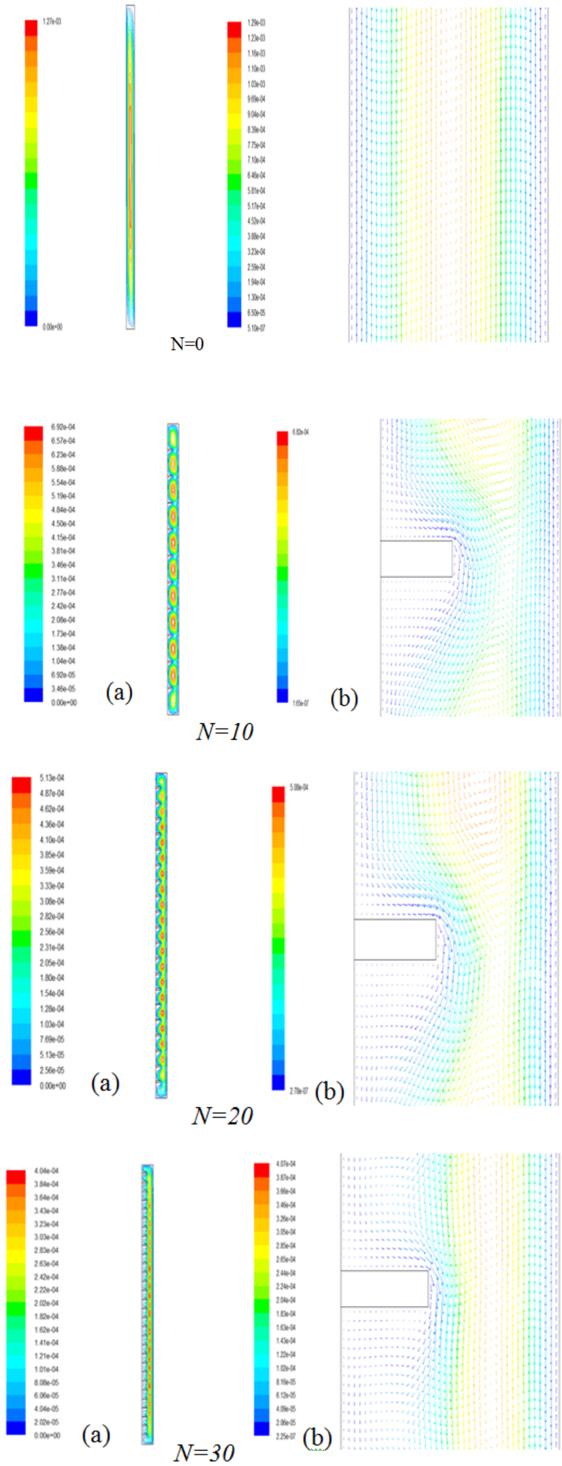

Figure 12. Streamlines and velocity vectors for different values of the number of partitions N, for Ra=$2.51.10^4$, Lp=0.4

Figure 13 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 cause a minimization of convective losses, but it should not exceed a certain number of partitions for do not influence the transmission of the glass. Thus, it must take an optimal number.

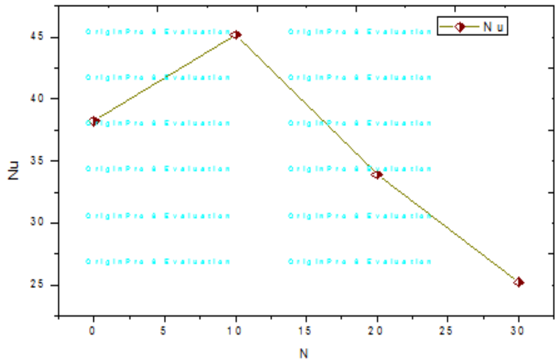

Figure 13. Variation in the number of average Nusselt for different numbers of partitions does for *Ra* = $2.51.10^4$

6. CONCLUSION

The effect of number and length of partitions is studied numerically. The result 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 value Lp=0.4 is the optimal value of the length of partitions.
- The number of partitions must be superior than 10.
- 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.

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

- The increase in the number of partitions causes a lowering of the rollers to the bottom near the absorber, and dons an

5. ACKNOWLEDGEMENTS

The authors are thankfull to University of Batna and University of El-Oued facilities for the research.

6. AUTHORS' NOTE

The author(s) declare(s) that there is no conflict of interest regarding the publication of this article. Authors confirmed that the data and the paper are free of plagiarism.

7. REFERENCES

Amraqui, A. M. S., et Abid, C. (2011). Combined Natural Convection and Surface Radiation in Solar Collector Equipped with Partitions. *Applied Solar Energy*, 47, 36-47.

Amraqui. S,. (2009). Modeling of thermal transfers paired in a thermal solar collector cells with anti-losses, doctorat thesis.

Bahlaoui, A., Raji A., Hasnaoui M., El Ayachi, R., Naimi, M., Makayssi , T., Lamsaadi, M. (2008). Numerical study of mixed convection coupled with radiation in a vented partitioned enclosure. *International Scientific Journal for alternative energy and ecology*, 6 (62).

Benabderrahmane, A., et Benazza, A. (2017). Numerical investigation of heat transfer enhancement inside a parabolic trough solar collector using dimpled absorbe. *International Journal of Energetica* (IJECA), 2(2). 01-05.

Haghighi, A. et Vafai K., (2014). Optimal positioning of strips for heat transfer reduction within an enclosure. *Numerical Heat Transfer*, A66, 17-40.

Frederick, R., (1989).Natural convection in an inclined square enclosurwith a partition attached to its cold wall. *Inl. J. Heat Mass Transfer*, 32, 87-94.

Ghodbane, M., et Boumeddane B. (2017). A parabolic trough solar collector as a solar system for heating water: a study based on numerical simulation. International Journal of Energetica (IJECA), 2(2). 29-37.

Khatamifara, M., et al (2016) .conjugate natural convection heat transfer in a partitioneddifferentially-heated square cavity. *International Communications in Heat and Mass Transfer*, 12.

Khechekhouche, A., Boukhari, A., Driss, Z., Benhissen, N., (2017). Seasonal effect on solar distillation in the El-Oued region of south-east Algeria. International Journal of Energetica (IJECA), 2 (1), 42-45.

Khechekhouche, A., Driss, Z., Benhaoua B., (2017). Solar distillation between a simple and double-glazing, *Revue de mécanique*. 2(2).

Koca, A., Hakan, O., Varol, Y., Mobedi, M., (2013). using of Bejan's heat line techniquefor analysis of natural convection in a divided cavity with differentially changing conductive partition. *Numerical Heat Transfer*, 64 (4), 339–359.

Khansila, P., Witayangkurn, S., (2014). Numerical study of natural convection in porous square enclosure non-uniformly heated from the partitions. *International Journal of Pure and Applied Mathematics*, 96(2), 2013-228.

Sacadura J. F., (1980). Initiation to the thermal transfers.

Spalding D., Patankar, S., (1972). A calculation procedure for heat, mass and momentum transfer in three-dimensional parabolic flows. *International Journal of heat and mass transfer*, 15, 1787-1806.

DOI: http://dx.doi.org/10.17509/ijost.v3i2.12753 |
p- ISSN 2528-1410 e- ISSN 2527-8045 |

Davis, G., (1983). Natural convection of air in a square cavity: a bench mark numerical solution. *Int. J. for Numerical Methods in fluids*, 3, 249-264.

Yousefi, T., Karami, A., Mahmoudinezhad, S., Mahdipour Jalilian, M., Ahmadi M., (2011) .Modeling the free convection heat transfer in apartitioned cavity by the use of fuzzy logic. *International Journal of Modeling and Optimization*, 1(4), 284-288.

Zemani F., Sabeur-Bendehina A., Boussoufi, M., (2014). Effect of partial partitions on natural convection in air filled cubical enclosure with hot wavy surface. Chemical Engineering &Process Technology, 5(191).