Numerical simulation of natural convection in the air gap of an inclined flat plat thermal solar collector with partitions attached to its glazing

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Abstract:

In this work, the natural convection in the air gap of an inclined solar collector contains partitions attached to its glazing has been studied numerically. The temperatures of the two horizontal walls are constants and different. The two vertical walls are supposed adiabatic. The equations of the problem are solved with the finite volume method, using of the Fluent software. The necessary objective is to study the influence of the partitions (length and number) on the natural convection in the air gap of the solar collector. The obtained results indicate that the presence of the partitions has important influence on the heat transfer with the decreasing of heat losses with natural convection, so improving of the solar collector efficiency. In this study, it reached that the number of partitions must be higher than 10, and their optimal length is Lp=0.4. The presence of the partitions with the optimal values reduces the heat losses by natural convection with 46 %.

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

1. 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 **S. Amraqui [1]**, indicates that the increase in the number of partitions reduces the heat transfer to the outside. The studies of **T.W. Tong** and **F.M.Grener (1986) [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, also they have studied the influence of the position of the baffle on the 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. **J. C. King and R. Narayanaswamy (1987) [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. **Samy. El-Sherbiny (2004) [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 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 (2007) [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 Lp and the emissivity causes a decrease in the thermal transfer. **Ramon. L. Frederick [11]** 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 bicellular flow occurs. At high Rayleigh numbers the heat transfer reduction is affected by secondary buoyancy forces, generated by the partition. **E. Bilgen [12]**, 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. **Mohammed Rabhi et al [13]**, 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. The works of **Adel Laaraba** and **al [14]**, **[15]** indicates the inclusion of partitions to the glazing can enhance the thermal performance of a flat plat thermal solar collector in both cases vertical and horizontal, also they studied the effect of partitions length and number, and choosing optimal values.

The objective of this work is to study the influence of the attachment of partitions (length and number), and the choice of their optimal values on the performance of the thermal solar collector.

Nomenclature:

L: Length of the thermal Solar collector(m)

L_p : dimensionless length of partitions

T: the temperature(K)

N: number of partitions

Tc: The absorber temperature (K)

Tf: The glazing temperature

x, y : x, y Cartesian coordinates

v : Following Speed y

u : Following Speed x

X, Y: Cartesian coordinates no dimensional

U : Speed following dimensionless x

V : dimensionless Speed following y

Greek Symbols

θ : Angle of tilt of the collector in relation to the ground in the angle unit

λ : Thermal conductivity(W/m.K)

ν : the kinematic viscositym^2/s

β : coefficient of volume dilatation(K^{-1})

The dimensionless numbers

Nu **:** the Nusselt number

R_a : Rayleigh number

Re: Reynolds number

$\overline{N_U}$: the average Nusselt number

Pr: Prandtl number

Gr: number of Grashof

1. Mathematical formulation of the problem:

In this application we are interested in the effects caused by the angle of tilt, on the fluid flow and heat transfer in the thermal solar collector of height H and length L , filled with air (Pr = 0.71).

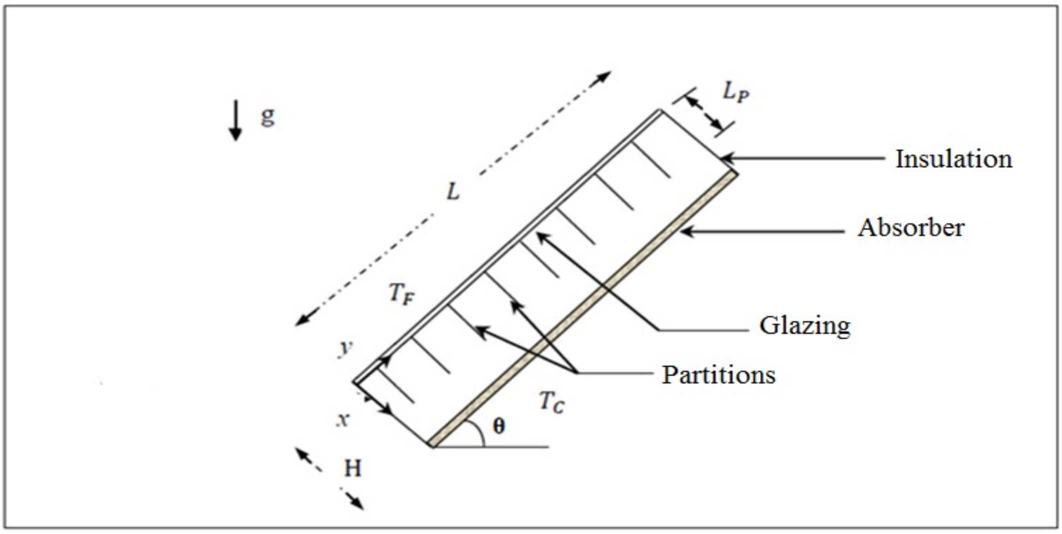


Figure1. *Field of study*.

The boundary conditions of the problem are:

-Horizontal Wall (absorber): $T = T_c$ and. $u = v = 0$

-Horizontal Wall (Glazing): $T = T_f$ and $u = v = 0$

- Vertical walls (insulation): $\frac{\partial T}{\partial x} = 0$ and $u = v = 0$

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

- ✓ The flow is stationary and two-dimensional.
- ✓ 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:

- ✓ The power density dissipated is negligible.

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

After simplifications, the equations of the problem will be **[6]**:

- **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:

$$u\,\frac{\partial u}{\partial x}+v\,\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+v\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right]+g\,\beta\left(T_c-T_f\right)\cos\varphi$$

(3)

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

$$u\,\frac{\partial v}{\partial x}+v\,\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+v\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right]-g\,\beta\left(T_c-T_f\right)\sin\varphi \tag{4}$$

- **The equation of energy**: the energy equation is written:

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

(5)

The average Nusselt number is given by the expression:

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

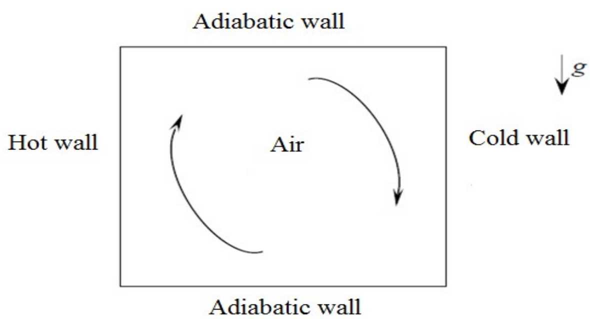
3. Method of resolution:

The previous equations are solved by the use of the software FLUENT **[8]** which based on the Finite Volume Method which presented by **Patankar (1980) [7]**, this method is based on the discretization of the transport equation for volumes finished discreet. The coupling pressure-Speed is treaty to the help of the algorithm **Simpler.**

4. 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 **[9]**, obtained in the case of a square cavity in 2D, containing air, have been used to test our simulation by the fluent software(**figure2**).

Figure 2.



*Geometry of a square cavity differentially heated **[10].***

The comparison the value of the number of average Nusselt number was made in different values of the Rayleigh number:

- **The average Nusselt number** **:** $\overline{Nu}$ $\overline{Nu} = \frac{Q}{\lambda \Delta T}$

The values of the average Nusselt number are calculated by the expression:

Table1 Comparison of the values of the average Nusselt number with the reference values

Ra	Present Work	[9]	[10]
10^4	2.261	2.252	2.243
10^5	*4.549*	4.545	4.523
10^6	8.87	8.853	8.826

Based on the comparisons of the isotherms, stream lines and the values of the average Nusselt number in the Nusselt obtained means with the code for the calculation fluent with the results of reference, 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. The solar collector is inclined with angle of 45°.

It is assumed a solar collector plan of length of L= 1m and high of H=0.025m, contain partitions with thickness of e=0.0025m.

In this work the studied phenomenon is the natural convection in a rectangular cavity partitioned. The Inferior wall (the absorber) is aluminum at the temperature of Tc=310 k, and the outer wall (the glazing) is glass at the temperature of Tf=290 k. The cavity (the air gap of the thermal solar collector), contain the air. The objective is to study the influence of partitions (lengths and number) on the heat transfer in the cavity.

The thermal proprieties of the air, glass and the aluminum used in the present work are presented in the table below:

Table 2 thermal proprieties of: air, aluminum and glass.

	Densité (kg/m^3)	Specific heat j/kg. k	Thermal conductivity (w/m. k)
Air	2500	840	0.81
Glass	1.204	1006	0.0257

Aluminium	2700	900	237

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, stream lines and the value of the the average Nusselt number.

To choose a good mesh, it has calculated the value of the Nusselt number means for different mesh sizes:

20×400, 30×600, 40×800. The results are presented in the following table:

Table3 Values of the average Nusselt number for different meshes.

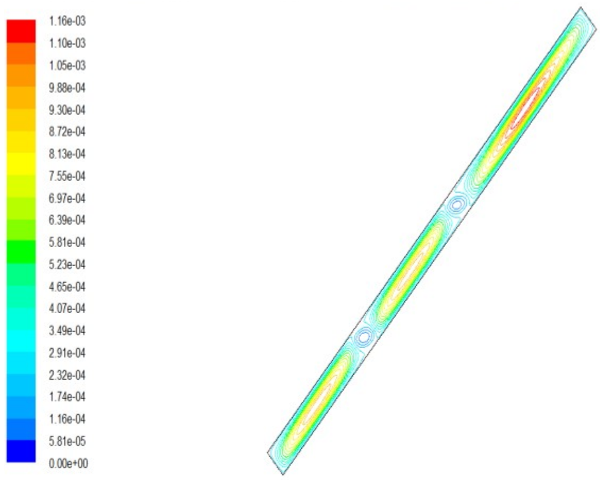
Mesh	20×400	30×600	40×800
$\overline{Nu}$	73.16	74.87	75.34

Given that the variation in the number of average Nusselt is not significant between the meshes 30×600 and

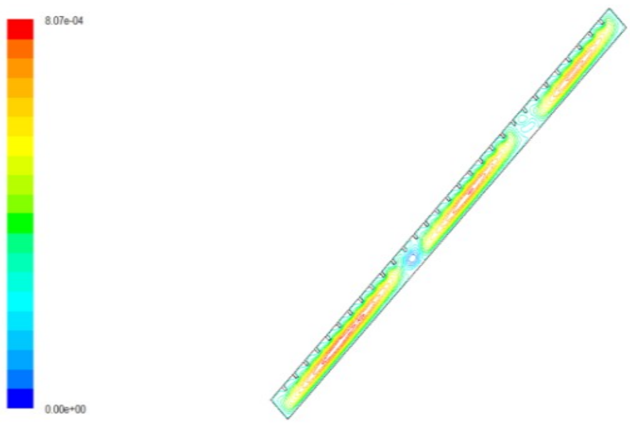
40×800, for this we chose the last mesh.

- **Effect of the length of the partition Lp:**
- **The dynamic field:** The dynamic field in the form of streamlines is presented below:

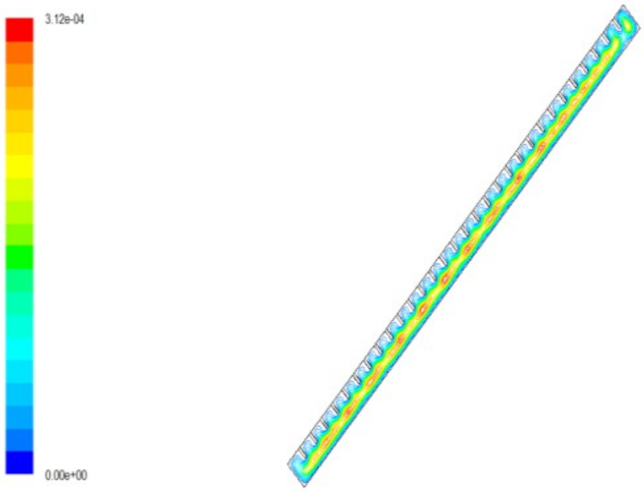
Figure 3 shows the streamlines in the case tilted. The presence of partitions with a number of 30 and a length of Lp=0.4 cause a maximum reduction of convective losses, because in this case it is observed that the movement of the air is done close to the absorber below of the partitions in the form of waves , which means a large impediment of convective movements, beyond this value (Lp=0.4) the movement of the fluid rises to the top and form of the rollers between each two partitions , which means the increase of the convective movement of the Air , so the value Lp=0.4 is optimal for the case tilted.



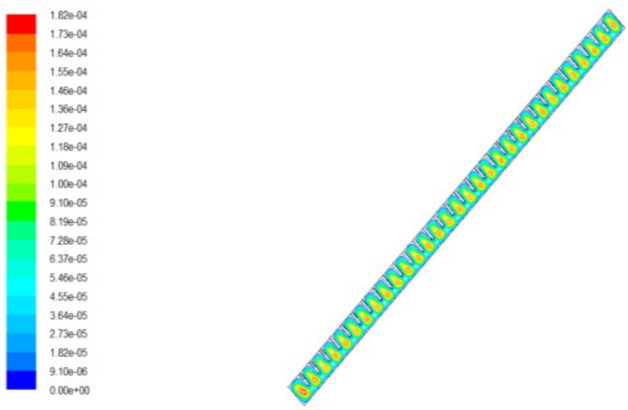
Lp=0



Lp=0.2



Lp=0.4



Lp=0.6

Figure 3. *Stream lines for different values of Lp pour Ra*=$2.51.10^4$, *N=30, θ*=$45.^0$

- **The average number Nusselt** : the variation in the average Nusselt number is presented figure below:

The variation in the number of average Nusselt depending on the length of the partitions is presented in the **Figure 4**. We note a maximum decrease in the values of the average Nusselt number when Lp=0.4, which means a maximum reduction of convective transfers to this value, and this is compatible with the approaching results reached by **[1]**.

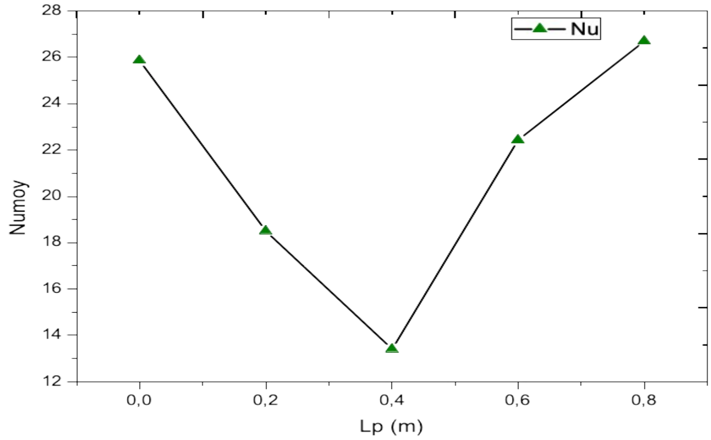
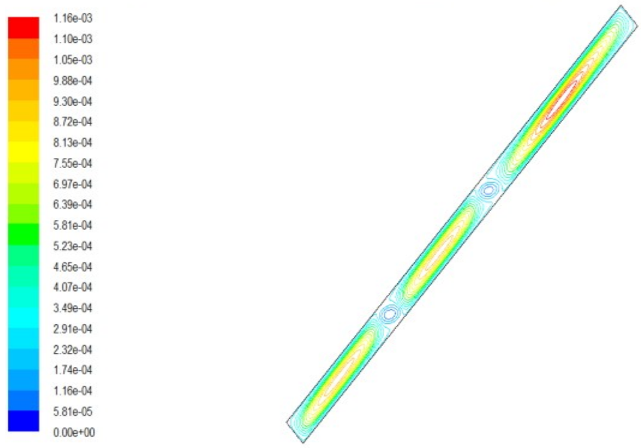


Figure 4. *Variation in the number of average Nusselt for different values of* ***Lp*** *for*

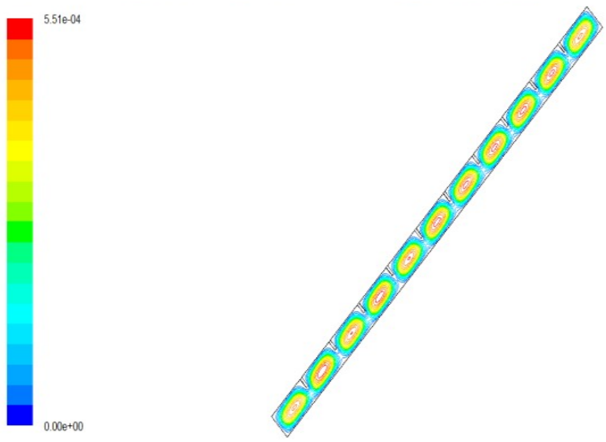
Ra=2.51.10^4, N=30, φ=45^0.

Effect of the number of partitions N:

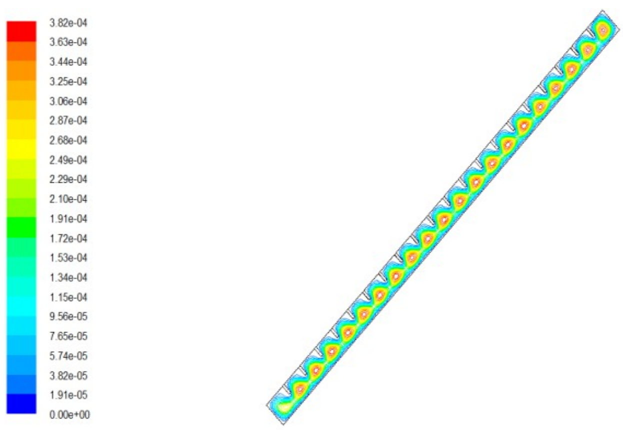
- **The dynamic field:** The dynamic field in the form of streamlines is presented below:
- **Figure 5** shows the influence of the number of partitions on the convective heat transfer between the absorber and the ambient air. The movement of the Air takes the form of the rollers between each two partitions when N=10, which means an increase of convective movements. When we increase the number of partitions (N=20, 30…), the movement of the air descends to the bottom below the partitions forming of waves which means an impediment of convective movements. The increases in the number of partitions cause an improvement of the performance of solar sensor thermal.


1.16e-03
1.10e-03
1.05e-03
9.88e-04
9.30e-04
8.72e-04
8.13e-04
7.55e-04
6.97e-04
6.39e-04
5.81e-04
5.23e-04
4.65e-04
4.07e-04
3.49e-04
2.91e-04
2.32e-04
1.74e-04
1.16e-04
5.81e-05
0.00e+00

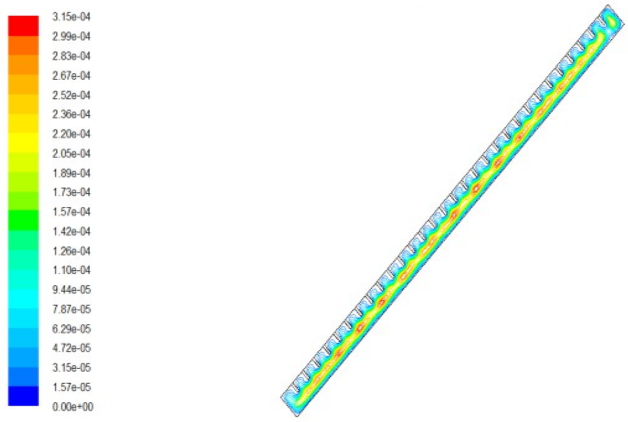
N=0


5.51e-04
0.00e+00

N=10


3.82e-04
3.63e-04
3.44e-04
3.25e-04
3.06e-04
2.87e-04
2.68e-04
2.49e-04
2.29e-04
2.10e-04
1.91e-04
1.72e-04
1.53e-04
1.34e-04
1.15e-04
9.56e-05
7.65e-05
5.74e-05
3.82e-05
1.91e-05
0.00e+00

N=20



N=30

Figure5. *Streamlines for different values of the number of partitions N for*

Ra=2.51.10^4, *Lp=0.4,* φ=45.^{0}

- **The average number Nusselt** : the variation in the average Nusselt number is presented figure below:

Figure 6 shows the variation in the number of average Nusselt in function of the number of partitions. There is a gradual decrease in the number of average Nusselt with the increase of N except for the value N=10, there is an increase in the number of average Nusselt. Therefore this decrease begins from a certain value of N greater than 10, And this is compatible with the approaching results reached by **[1].** For this it is necessary to master a number of optimal partitions to ensure the minimization of convective losses, but it should not exceed a number for do not influence the transmission of the glass.

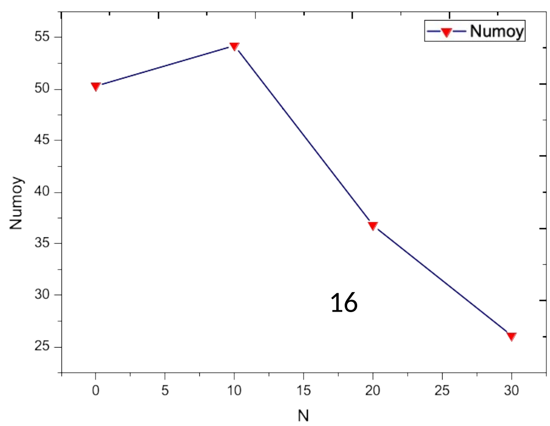


Figure 6 *. Variation in the values of the average Nusselt number for different values of the number of partitions for* Ra=$2.51 . 10^4$, Lp=0.4, φ=45^0.

6. Conclusion:

According to the previous results, it can be concluded that:

- ✓ The presence of partitions for the tilted cases cause a decrease in the value of average Nusselt, therefore a minimization of thermal losses to the ambient.
- ✓ The length of partitions has a great influence on the thermal transfer. In the case tilted the optimal length is Lp= 0.4.
- ✓ For case tilted, it must choose an optimal number of partitions to ensure a minimization of thermal losses, (it must be that n>10).
- ✓ The presence of partitions with an optimal length and number can improve the flat plat collector with the percentage of 46 %.

References:

[1] Ahmed. M. S. Amraqui, and C. Abid, "Combined Natural Convection and Surface Radiation in Solar Collector Equipped with Partitions " *Applied Solar Energy,* vol. 47, pp. 36-47, 2011.

[2] T. W. Tong, F.M.Grener, "natural convection in partitioned air-filled rectangular enclosure," *Int, J. Heat and Mass Transfer,* vol. 13, pp. 99-108, 1986.

[3] J. C. King, R. Narayanaswamy, "radiative effects on natural convection heat transfer in enclosure with multiple partitions," *Department of Mechanical Engineering Curtin University of Technology* 1987.

[4] Samy- El-Sherbiny, "natural convection in partitioned rectangular enclosures Air," *Alexandria Engineering Journal,* vol. 43, pp. 593-602, 2004.

[5] Bahlaoui and al, "numerical study of mixed convection coupled with radiation in a vented partitioned enclosure," *International Scientific Journal for alternative energy and ecology № 6 (62)* 2008.

[6] J. F. Sacadura, "initiation to the thermal transfers," 1980.

[7] D. Spalding, S. Patankar, "a calculation procedure for heat, mass and momentum transfer in three-dimensional parabolic flows," *International Journal of heat and mass transfer,* vol. 15, pp. 1787-1806, 1972.

[8] "Heat Transfer Fluent user guide (Chapter11).".

[9] Amraqui. S, "modeling of thermal transfers paired in a thermal solar collector cells with anti-losses," doctorat thesis ,2009.

[10] G. De. Vahl. Davis, "natural convection of air in a square cavity: a bench mark numerical solution," *Int. J. for Numerical Methods in fluids,* vol. 3, pp. 249-264, 1983.

[11] Ramon.L.Frederick, "Natural convection in an inclined square enclosur with a partition attached to its cold wall," *InI. J. Heat Mass Transfer,* vol. 32, pp. 87-94, 1989.

[12] E. Bilgen, "Natural convection in cavities with a thin fin on the hot wall," *InI. J. Heat Mass Transfer,* vol. 48, pp. 3493–3505, 2005.

[13] H. Bouali, Mohammed Rabhi, Ahmed Mezrhab, "Radiation–natural convection heat transfer in inclined rectangular enclosures with multiple partitions," *Energy Conversion and Management,* vol. 49, pp. 1228–1236, 2008.

[14] Adel. Laaraba, Abderrahmane khechekhouche," Numerical simulation of natural convection in the air gap of a vertical flat plat thermal solar collector with partitions attached to its glazing", I*ndonesian Journal of Science & Technology*, vol.3 (2), pp95-104, 2018.

[15] Adel. Laaraba," CFD simulation of natural convection and heat transfer in a flat solar

thermal collector with fins on the glazing - horizontal case", *International Journal of Energetica* (IJECA),vol3(2),pp 01-05,2018.