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Research Article

The performance of a Heat Exchanger with Wings associated to Solar Photovoltaic Collector

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Abstract

The current investigation deals with the numerical study of heat transfer in a heat exchanger, with wings, associated to a sensor using solar photovoltaic cells. The heat exchanger is composed of a support cavity. The upper wall of the cavity is in contact with PV cells, while the bottom wall is a good thermal conducting metal. Besides, the bottom wall is in contact with the room (local cavity) to be heated. This problem is governed by the conservation equations of mass, momentum and energy when the solid and fluid phases are not in thermal equilibrium. The parameters characteristic of this system such as the number of wings, N_{fi}, their length, L_{fi} and the number of Rayleigh, Ra, were optimized by analyzing their effects on the heat transfer, in order to cool the PV cells and heating the local cavity. Results proved that in case the heat flux increases the fluid flow will be convective progressively; the dimension of fins will be affecting the behaviours of flow. The longer fin has more remarkable effects on the flow fields. Moreover, it is seen that temperature in the local cavity is enhanced with increasing the number of fins. In contrast, in the support cavity, the temperature is observed to be lowered by augmenting the number of fins.

Keywords: *Natural Convection, Fin-Heat Exchanger, Solar Energy, PV Cells, Heat Transfer*

1. Introduction

In industrialized countries, buildings transport and industrial sectors are great energy consumers. The integration of renewable energy systems will be one of the solutions to the key issues of economizing fossil resources and reducing the effects of global warming. Solar energy represents the best renewable, environmentally friendly source of energy that can be used for the heating and cooling of houses. The response of fins is important in a wide range of engineering devices including heat exchangers, clutches, motors and so on.

Enhancement in heat transfer coefficient improves the performance of heat exchanger and also reduces the size of the heat exchanger. In general, the heat transfer enhancement techniques are classified in two groups: active and passive techniques. Active techniques require external forces like fluid vibration, electric field, and surface vibration whereas passive techniques require special surface geometries or fluid additives and various tube inserts. Both techniques have been widely used to improve heat transfer performance of heat exchangers. There exist basically two types of coils; helical coils and spiral ones. The several studies have indicated that helically coiled tubes are superior to straight tubes when employed in heat transfer applications . The secondary developed vortices, due to the fluid flow in curve tube of the helical coil, results in increased turbulence which increases heat transfer rate. This phenomenon can be beneficial especially in laminar flow regime. The effects of geometric and operating parameters on the performance of helical coiled tube are presented by Pramod *et al*. The result shows that the Nusselt number (Nu) increases with increase in curvature ratio (δ) for the same Re. The analysis also reveals that in laminar region the secondary developed vortex in the fluid flow gains volume, as Re increases, which strengthens the turbulence in the fluid flow.

In addition, the increase in turbulence allows proper mixing of the fluid, which enhances the Nu and hi. Beigzadeh *et al*. investigated the effects of curvature ratio and torsion on friction factor (f) and Nusselt number (Nu) from coiled tubes. Hot water was passed through coiled tubes placed in a cold bath. At various Reynolds numbers, Nu, f and thermal hydraulic performance were obtained.

Targui *et al*. analyzed numerically the fluid flow and heat transfer in a double pipe heat exchanger with porous structures inserted in two configurations. They noted that the highest average Nusselt number ratio is obtained at a low permeability and a high thickness of the porous structures in the case of a low thermal

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conductivity ratio. Kasayapanand, proposed a numerical modelling of the electric field effect on natural convection in the square enclosures with a single fin and with multiple fins. They noted that the heat transfer coefficient is substantially increased with the number of fins and with the fin length.

Gongnan Xie *et al*. proposed both a parametric study and multiple correlations on the heat transfer characteristics and friction of the air-side heat exchanger with wings and tube and with a great number of lines having a broad diameter of tube. They noted that the Nusselt number and friction factor decrease with the increase of the number of tube rows. The heat transfer for a fin surface material with a large thermal conductivity can be enhanced due to the decrease of the thermal resistance.

Wu *et al*. proposed a new approach for optimizing the RC of fin-and-tube heat exchanger based on the genetic algorithm. This approach extends the GA application range to the RC optimization of fin-and-tube heat exchanger, and introduces a simpler and more effective method for practical RC optimization.

Zhang *et al*. proposed an experimental and numerical study of heat transfer in a helically baffled heat exchanger combined with one three dimensional finned tube. They noted that the Nusselt number increases when the Reynolds number augments. Chu *et al*. proposed three-dimensional numerical simulations to be employed to investigate the heat transfer characteristics and flow structure in full-scale fin-and-tube heat exchangers with LVGs. They noted that the average Nusselt number decreases with the increasing angle of attack and tube row number.

Kundu *et al*. presented an analytical approach for the execution of performance of the triangular fins on the simultaneous heat and mass transfer. They noted that the fin performances vary insignificantly with the relative humidity. The temperature on the fin surface increases with the increase of the relative humidity.

Wen *et al*. proposed an experimental heat transfer study for forced convection using the airflow over the heat exchanger with three different fin configurations (plate fin, wavy fin and compounded fin). They strongly suggested the use of the compounded fin for the heat exchanger. Banerjee *et al*. studied numerically the natural convection heat transfer in a horizontal, two-dimensional square cavity with two flush-mounted heat sources on the bottom wall. The ratio of flux strengths of the heat sources and their lengths are taken as variable, while the total heat input to the cavity is kept as constant. Rees and Pop studied the natural convection in a square porous cavity by using the model outside the thermal balance on a microscopic scale between the liquid and solid phases.

One of the key shortcomings of photovoltaics, however, is their relatively low efficiency. Typically, commercially available PV modules are only able to convert 6-18% of the incident radiation falling on them to electrical energy, with the remainder being lost by reflection or as heat (Bazilian *et al.*). However, a small portion of the heat is "sunk" into the cells resulting in a reduction in their efficiency. Helden *et al*. noted that PV collectors absorb 80% of the incident solar radiation but convert only a small part of it into electrical energy. Koyunbaba *et al*. presented a comparison of Trombe wall system between a PV panel with a single glass and double glass. After validation of the numerical models with the experimental results, these systems will be used in a building in different climatic conditions, glass types and thermal masses. Furthermore, they noted that the temperature reached by PV cells is higher than the ambient temperature and that the efficiency of PVTs is greater than the combined sum of separate PV and thermal collectors. In the light of this, they suggest that PVT systems offer a cost effective solution for applications where roof area is limited. Kamthania *et al*. evaluated the performance of a hybrid PVT double pass facade for space heating in the composite climate of New Delhi by using a semi transparent PV module. Thermal modeling has been carried out based on the first and second law of thermodynamics in order to estimate the electrical and thermal energy along with the exergy for a "semi transparent" hybrid PVT double pass facade instead of an opaque PV panel.

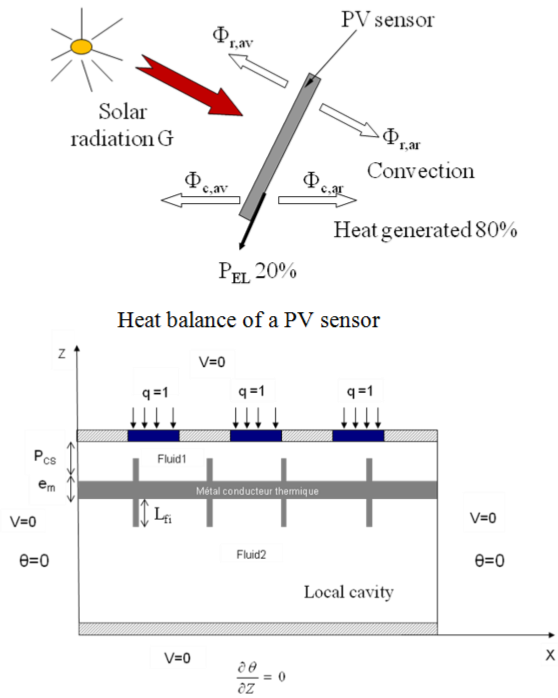

Figure 1 Geometrical configuration

In the light of this, they suggested that the semi transparent PV module has more electrical efficiency than the opaque PV module. Sarhaddi *et al*. evaluated the exergetic performance of a solar PV/T air collector and carried out the detailed energy and exergy analysis to calculate the thermal and electrical parameters, exergy components and exergy efficiency of a typical PV/T air collector. Caluianu *et al*.developed and validated, using experimental data, a BP 585 F photovoltaic panel thermal model. They made a simulation by applying the galerkin finite element method to the flow and energy equations, incorporating an implicit convective boundary condition. The values of the root mean square error and the correlation coefficient between the experimental and the simulated values of the panel temperature show a good accuracy of the model. Afterwards, this was used to simulate the convective flow between the photovoltaic module and the roof wall. The influence of this channel width was then studied. Velocity and temperature variations were calculated at the beginning, the middle and the exit of the channel for 10 mm, 20 mm and 30 mm width. We propose in this work to recover the heat released by PV cells through a fluid (fluid 1) imprisoned between PV cells and a metal fin exchanger forming the roof of a room to be heated filled with air (fluid 2) as presented in Figure 1. The characteristics of flow and heat transfer are discussed in conjunction with Rayleigh number, number of fins and fin length.

2. Governing Equations

The movement of the fluids and their temperature are governed by continuity, momentum and energy equations in the solid and fluid phases. The heat flux imposed on the up boundary is considered only on the level of PV cells, with density q. The flow is considered to be steady, laminar and in two dimensions via a computational fluid dynamics technique. The thermo physical properties of the fluid are assumed to be constant, except for the density in the buoyancy term, which depends linearly on the temperature, i.e., the Boussinesq approximation is assumed to be valid. Thus, the reference variables for length, velocity and temperature are chosen respectively as:

$$X = \frac{x}{H},\ Z = \frac{z}{H}, V_{ref} = \frac{\vartheta}{H} \text{ and } \Delta T = \frac{q}{\lambda} H \qquad (1)$$

Where: x and z are the dimensional spatial coordinates

The dimensionless variables are: $\theta = \frac{T-T_a}{\Delta T}$ and $\vec{V}(\mathrm{U,W})$

The dimensionless conservation equations of mass and momentum for the dimensionless natural convection flow of fluid1 and fluid2 are:

Navier-Stokes equations:

$$\vec{\nabla}.\vec{V} = 0 \qquad (1)$$

$$(\vec{V}.\vec{\nabla}).\vec{V} = -\vec{\nabla}P + Gr\theta\overrightarrow{e_g} + \Delta\overrightarrow{V} \qquad (2)$$

where $\overrightarrow{e_g} = -cos\beta\vec{\imath} - sin\beta\vec{\jmath}$

In this case, the sensor is positioned vertically so: $\beta = 90$

Energy equation at each fluid phase:

$$\overrightarrow{V}.\vec{\nabla}\theta_f = \frac{1}{Pr}\vec{\nabla}.(\vec{\nabla}\theta_f) + \frac{Bi}{Pr}(\theta_f - \theta_s) \qquad (3)$$

Energy equation at solid phase:

$$\overrightarrow{\nabla}.(\vec{\nabla}\theta_s) + \frac{Bi}{R_\lambda}(\theta_s - \theta_f) = 0 \qquad (4)$$

The problem is characterized by the classical Prandtl number, Pr, the Rayleigh number, Ra, the Biot number of each fluid, $Bi = \frac{h_{fsi}H}{\lambda_{fi}}$, which takes the value zero away from interfaces and the ratio of thermal conductivity $R_{\lambda_i} = \frac{\lambda_s}{\lambda_{fi}}$ with i=1 or 2, corresponding to fluid1 and fluid2.

3. Boundary Conditions

The boundary conditions are:
at X = 0 and 0 ≤ Z ≤ 1, U=0, W=0 and θ=0
at X = A and 0 ≤ Z ≤ 1, U=0, W=0 and θ=0
at Z = 1 : U=0, W=0 and $\frac{\partial\theta}{\partial Z} = 1$, on the PV cells region and U=0, W=0 et $\frac{\partial\theta}{\partial Z} = 0$, elsewhere
at Z = 0 and 0 ≤ X ≤ A, U=0, W=0 and $\frac{\partial\theta}{\partial Z} = 0$

In other words, at Z=1, the heat flux imposed is considered only on the PV cells regions. However, the regions separating two cells are taken adiabatic.

4. Results and Discussions

4.1. Validation of Numerical Code

The governing Eqs. 1-4 with the boundary conditions were solved using the finite volume method. The computational domain is divided into rectangular control volumes with one grid point located at the centre of the control volume that forms a basic cell. The conservation equations are integrated in the control volumes, leading to a balance equation for the fluxes at the interfaces. A second order scheme is used to distinguish the equations; a false transient procedure is used in order to obtain a permanent solution. To accelerate the convergence, the SIMPLER algorithm, originally developed by Patankar, is coupled with the SIMPLEC algorithm of van Doormaal and Raithby. Non uniform grids are used in the program, allowing fine grid spacing near the boundaries. Trial calculations were necessary to optimize the computation time and accuracy. Convergence with mesh size was verified by using coarser and finer grids for selected test problems. The convergence criterion is based on both the maximum error of continuity equation and the average quadratic residue over the whole domain for each equation being less than a prescribed value ζ (generally less than 10^{-6}).

The main objective of the present study is to investigate, numerically, the natural convection in a horizontal cavity heated on the top boundary by an irregular flux, q, while the bottom plate is adiabatic. The heat quantity transferred, at position z, is calculated from the average Nusselt number, Nu (z), a long x direction, defined as:

$$Nu(z) = \frac{1}{A}\int_0^A Nu(x,z)dx$$

And the average Nusselt number,

$$Nu = \int_0^1 Nu(z)dz$$

Where

$Nu(x,z) = -\left(\frac{\partial\theta(x,z)}{\partial x}\right) + W(x,z)\theta(x,z)$: the local heat transfer at any point.

Where W is the velocity component along Z axis.
In order to obtain a grid independent solution, a grid refinement study is performed for a rectangular horizontal cavity. Figure 2 shows the convergence of the Nusselt number, at the heated surface with grid refinement. It is observed that grid independence is achieved with a 121*41 grid for which there is an insignificant change in *Nu*.

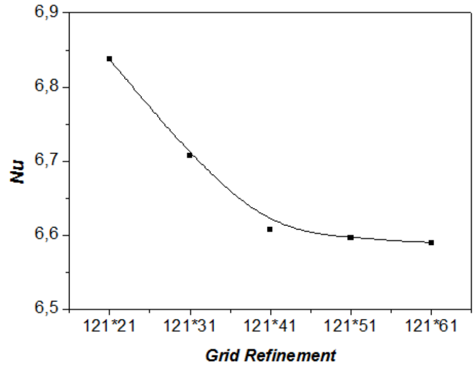

Figure 2 Influence of the grid on the heat transfer

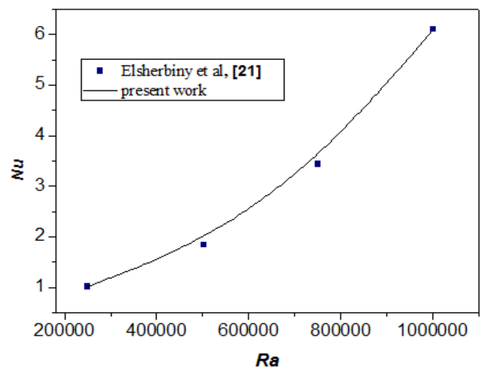

Figure 3 Comparison of our results with the experimental correlation of Elsherbiny *et al.*

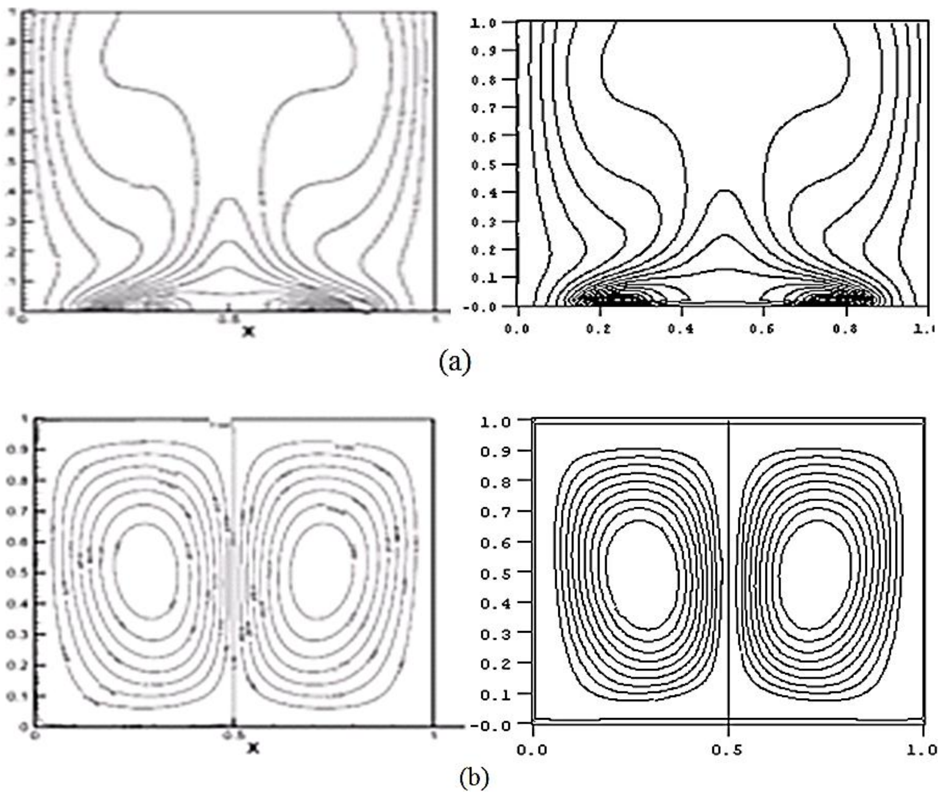

Figure 4. Comparison of our results with those S. Banerjee *et al :* (a) Isotherms and (b) streamlines

Published experimental data are not available for the cavity configuration and boundary conditions similar to those undertaken in the present study. Thus, direct validation of the computations against suitable experimental data could not be performed. However, in order to validate the predictive capability and accuracy of the present code, computations are performed using the configuration and boundary conditions of the experiment carried out by Elsherbiny *et al*. They measured the natural convection heat transfer in vertical and inclined rectangular cavities where the isothermal sidewalls were at different temperatures and the end walls were perfectly conductors having a linearly varying temperature bounded by the temperature of the sidewalls.

The Nusselt number computed by the present code for values of *Ra* ranging from 10^3 to 10^6 are compared with the correlation of Elsherbiny *et al*. given in Figure 3. The agreement is found to be excellent.

The present numerical results are evaluated for accuracy against numerical results published in previous works. Comparisons of our results (Isotherms and Streamlines) with those of Banerjee *et al*. are shown in Figure 4. The agreement is found to be excellent which demonstrates the validity of the formulation and the computer code.

4.2. Parametric Analysis

In this work, we will focus our study on the analysis of the effect of the following parameters: the heat flow collected, *Ra*, the fin length, L_{fi} and the number of fins, N_{fi}.

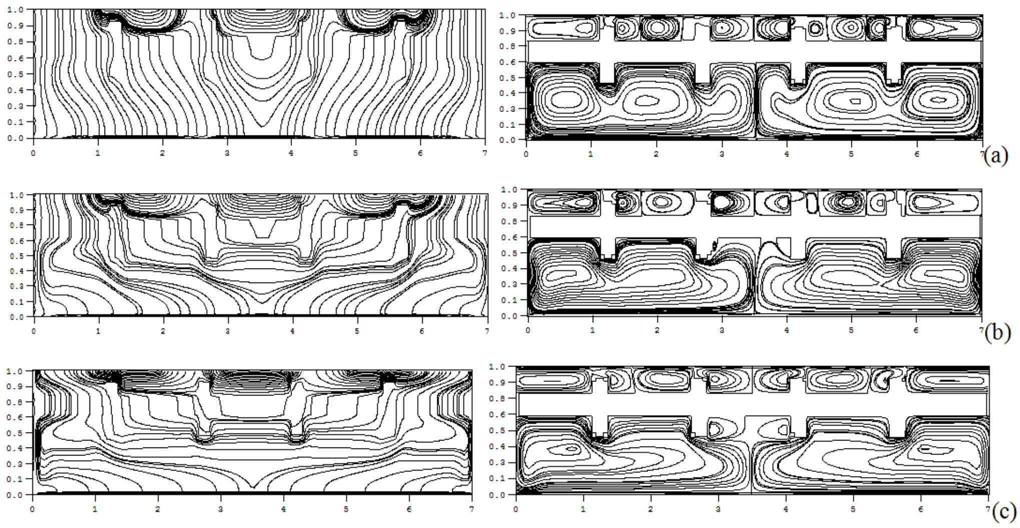

Figure 5 Isotherms (left) and streamlines (right) for different Ra number: e_m = 1/8; Pr=0.71; A=7; L_{fi}=0,125; N_{fi}=4 and P_{cs}=1/8 with (a): Ra=10^5, (b): Ra=10^6 and (c): Ra=10^7

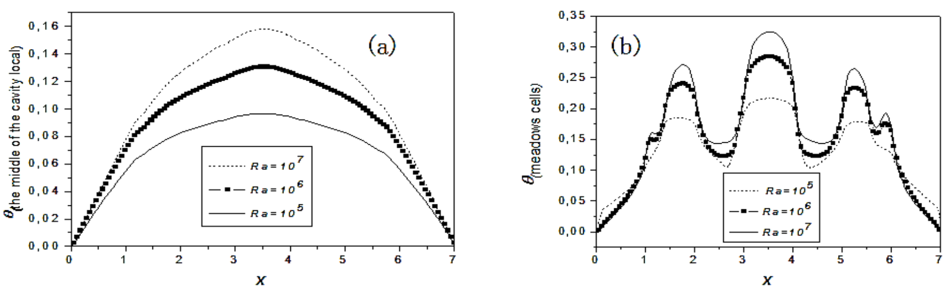

Figure 6 Rayleigh number effect on temperature profiles at the two monitoring positions: (a) in the median plane of the local cavity and (b) in the higher part of the support cavity for: e_m = 1/8; Pr=0.71; A=7; L_{fi}=0.125; N_{fi}=4 and P_{cs}=1/8

4.2.1. Effect of Heat flux

The aim of this work is to increase the heat transfer surface to recover the maximum of heat released by PV cells. Thus to investigate the nature of flow in the two spaces with the presence of multiple fins attached uniformly on the metal, we illustrate on Figure 5 the effects of the increase of heat flux (Rayleigh number, *Ra*) on the behaviors of flow and temperature field, with and without fins.

Through the variation of the Rayleigh number values, *Ra*=10^5, 10^6, 10^7, for e_m = 1/8; *Pr*=0.71; *A*=7; L_{fi}=0,125; N_{fi}=4 and P_{cs}=1/8, we note from Figure 5, that the increase of heat flux, the fluid flow will be convective progressively and creates a clockwise rotating cellular motion on the two cavity. The consequence effect is showed through the isotherms where the heat transfer is accelerated and the conductive mode is dominated by the natural convective one. This can be explained by the fact that the metal interface does not transmit heat to the local cavity with the same proportion as that transmitted from the PV cells to the fluid present in the support cavity. Moreover, the thermal boundary layers appear gradually by further augmenting the Rayleigh number. To better illustrate this, we depict in Figure 6 the temperature profiles at the two monitoring positions for different values of heat flux: in the higher part of the support cavity and in the median plane of the local cavity. It is noticed that the effect of the Rayleigh number appears in both planes. For the higher part of the support cavity, we note that the value of the temperature decreases when *Ra* increases whereas in the median plane of the local cavity, we may observe that the temperature is enhanced with the increasing of *Ra*. This explains the strengthening of the heat transfer towards the local cavity when augmenting the Rayleigh number, *Ra*.

4.2.2. Effect of the Fins Length

In the second step, as know, the dimension of fins will be affecting the behaviours of flow.

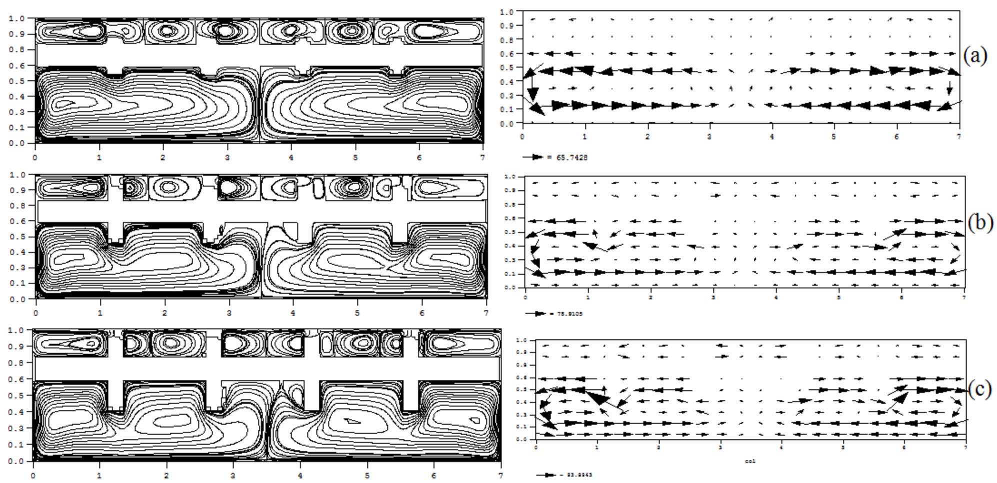

Figure 7 Effect of the variation of the fins length on the streamlines and the velocity field for: e_m = 1/8; Pr=0.71; A=7; Ra=10^6; N_{fi}=4 and P_{cs}=2/8 for (a): L_{fi}=0.06, (b): L_{fi}=0.125 and (c): L_{fi}=0.18.

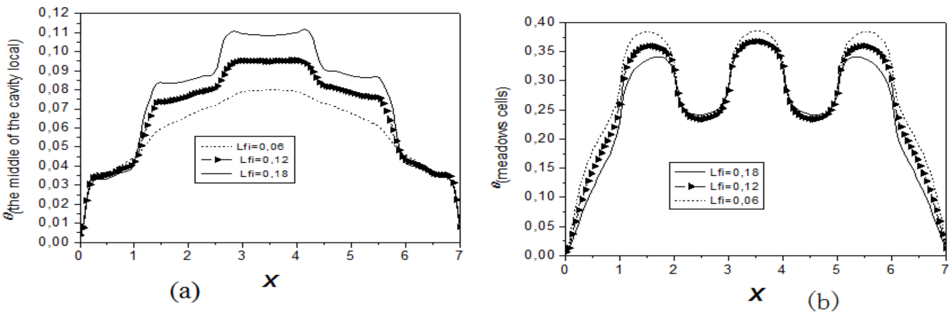

Figure 8 Fin length effect on temperature profiles to the two positions: (a) in the median plane of the local cavity and (b) in the higher part of the support cavity for: e_m = 1/8; P_{cs}=2/8; Pr=0.71; A=7; Ra=10^6 and N_{fi}=4.

Thus, to explore this, we illustrate in Figure 7, isotherms and stream lines by the variation of the fin length symmetrically relatively the solid exchanger: L_{fi}=0.06, L_{fi}=0.125, L_{fi}=0.18, for e_m = 1/8; Pr=0.71; A=7; Ra=10^6, N_{fi}=4 and P_{cs}=2/8. According to this figure, we note that with a longer fin bringing about more changes to the flow compared to a shorter fin. It is noticed that for $L_{fi} \geq 0.125$ (half the width of the support cavity), many recirculation vortices are found in the local cavity. The longer fin has more important effects on the flow fields and affects, to a significant degree, the velocity field. Indeed, the increase in L_{fi} facilitates the heat transfer towards the room.

This is confirmed in Figure 8 depicting the temperature profiles along the upper part of the support cavity and along the horizontal median plane of the local cavity for various values of fin lengths. We note that the increase of fin lengths, L_{fi} leads to a drop in the temperature at the upper part of the support cavity and an increase in the temperature in the horizontal median plane of the local cavity. This is due to a longer fin brings about more changes to the flow compared to a shorter fin.

4.2.3. Number of Fins Effect

In this section we attempt to study the influence of the number of fins, N_{fi} on the flow pattern and the heat transfer rate. According to Figure 9, we notice that the increase of the number of fins causes different flow patterns to occur in the cavity. It can be observed that the separation over the fin appears especially at the top zone of the enclosure. As it is seen, the temperature in the local cavity is enhanced with increasing the number of fins. However, in the support cavity, temperature decreases with increasing the number of fins. That may results in maximum heat transfer coefficient.

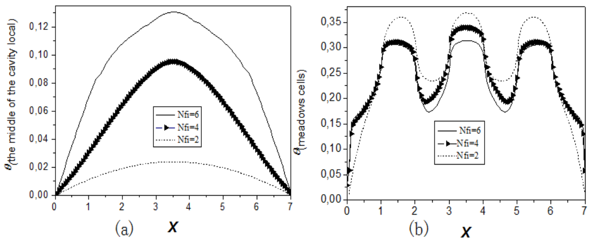

Figure 9 Number of fins effect on temperature profiles to the two positions: (a) in the median plane of the local cavity and (b) in the higher part of the support cavity for: e_m =1/8; P_{cs}=2/8; Pr=0.71; A=7; Ra=10^6 and L_{fi}= 0.125.

Conclusion

In the current investigation, we carried out a study of a fin-heat exchanger from three coolants (two fluids and one solid) associated with a solar sensor of photovoltaic cells, used to produce a continuous current. In order to cool the cells and to recover the maximum of heat dissipated by joule effect from PV cells, we explored the effects of fin heat exchanger to the improvement of the effectiveness of such a system. We proved when the heat flux increase the fluid flow will be convective progressively and creates a clockwise rotating cellular motion on the two cavities and the thermal boundary layers appear gradually with the increase of the Rayleigh number. Moreover, for the higher part of the support cavity, we note that the value of the temperature decreases when Ra increases whereas in the median plane of the local cavity, we note that the temperature increases with the increasing of Ra. As know, the dimension of fins will be affecting the behaviours of flow. The longer fin has more important effects on the flow fields and affects, to a significant degree, the velocity field. Indeed, the increase in L_{fi} facilitates the heat transfer towards the room. Besides, we note that the augmenting of fin lengths, L_{fi} introduces a drop in the temperature at the upper part of the support cavity and an increase in the temperature in the horizontal median plane of the local cavity. Moreover, as seen that temperature in the local cavity strengthens with increasing the number of fins, N_{fi}. Nevertheless, in the support cavity, temperature decreases with augmenting the number of fins, N_{fi}.

Nomenclature

A Aspect ratio: A=L/H
Bi Biot number
e_m Thickness of metal plate
g Gravitational acceleration ($m.s^{-2}$)
Gr Grashof number
H Height of the cavity (m)
L Length of the cavity (m)
L_{fi} fin length (m)
N_{fi} Number of fins (m)
P Pressure (Pa)
Pr Prandtl number
P_{cs} Depth of the cavity support
Ra Rayleigh number
T Dimensional temperature (K)
V Non-dimensional velocity
X, Z Non-dimensional coordinates

Greek Symbols

θ Non-dimensional Temperature

Subscripts

fi Fin
f Fluid
s Solid
cs Cavity support
m Metal

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