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Channel gap effect on the passive cooling of the hybrid system PVT air collector

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***Abstract*—This work focused on the channel gap effect on the passive cooling of hybrid system PVT. The main objective of this numerical study is to improve passively the photovoltaic cells cooling. The resolution of the governing equations ensures by using the finite volume method. The numerical analysis has been made for fixed Rayleigh number and varied channel gap (S) between 6 to 36 cm. Validation of the numerical model is done with previous experimental work, such that the numerical results are in better agreement with that found in the literature. The numerical results show that the average Nusselt number increases proportionally with the channel gap. Correlations of mean Nusselt numbers were noted and commented.**

Keywords*— Finite volume method, Hybrid system PV/T, heat transfer enhancement, channel gap effect, passive cooling.

Introduction

The heat transfer by natural convection between vertical parallel plates is extremely important in many engineering problems. Particularly, this problem is of considerable interest to engineers because of its application to electronic equipment cooling and solar energetically system. They are often encountered in solar power systems. Elenbaas [1] is the first who studied the phenomenon of natural convection between the vertically parallel plates. Since then, extensive theoretical and experimental works have been carried out on this problem. Anand et al. [2] have studied a theoretical study of the gap height effect between hot parallel plates on free convection. A considerable attention has been given to natural convection between heated vertical parallel plates by the many scientific researchers in the past. The literary survey indicated that there are several theoretical, numerical and experimental works concentrated on the heat transfer rate by free convection along the slanted panel with an opposite wall [3-13]. Recently, many scientific researchers have been designing photoelectric/thermal technology (PV / T) to develop the performance of photovoltaic solar cells [14-19].

However, the information is very rare about the relationship between the channel gap and the heat transfer rate. Therefore,

Nomenclature

X,Y	Dimensionless Cartesian coordinates, [m]
U,V	Dimensionless velocity comments, [m / s]
g	Acceleration of gravity, [m / s2]
P	Dimensionless pressure, [-]
T	Temperature, [K]
θ	Dimensionless Temperature, [-]
Pr	Prandtl number, [-]
Ra	Rayleigh Number,
Ra*	Modified Rayleigh number, $Ra^* = (S/L)\cos(\alpha) Ra$
$Nu_{s\text{-}moy}$	Average Nusselt number
Greek symbols	
α	Channel inclination angle, α = 60 °
ρ	Density, [kg / m3]
λ	Thermal conductivity, [W /m.k]
β	Coefficient of thermal expansion, [1 / K]
μ	Dynamic viscosity, [Kg / m.s]
Indices and exhibitors	
abs	absorber
glass	Glass cover
moy	average
PV	pannel

This paper presents a numerical study of turbulent natural convection air-cooling of the photovoltaic panel. The main objective is to investigate numerically the channel gap effect on the air passive cooling of photovoltaic cells.

2. Formulations of the problem

2.1. Physical model

In this work, the aim is to enhance the heat transfer rate of one panel, the mass flow rate and the heat transfer rate of the single-pass thermal-photovoltaic module (PV/T-I) where panels cool down from below are considered for this study as shown in Figure 1.

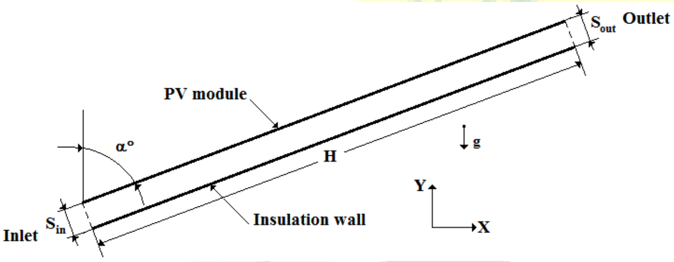


Fig. 1. Physical configurations of the studied problem

2.2. Dimensions of geometric parameters

All dimensions of three configurations are summarized in Table 1.

Table 1. Dimensions of the geometry

Geometric parameter	Symbol	Dimension (m)
Channel length	H	1.2
Channel depth ratio	H/S_{in}	3 to 20

2.3. Governing equations

The airflow through the hybrid system PV/T air collectors is prescribed by two-dimensional 2D laminar natural convection in Cartesian coordinates. The equations [18] that describe the flow are given by:

Continuity equation:

$$\frac{\partial \rho u}{\partial x}+\frac{\partial \rho v}{\partial y}=0 \quad (1)$$

X-momentum conservation equation:

$$\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left[(\mu+\mu_t)\frac{\partial u}{\partial x}\right]+\frac{\partial}{\partial y}\left[(\mu+\mu_t)\frac{\partial u}{\partial y}\right]-\frac{2}{3}\rho\frac{\partial k}{\partial x} \quad (2)$$

Y-momentum conservation equation:

$$\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}=-\frac{\partial p}{\partial y}+\frac{\partial}{\partial x}\left[(\mu+\mu_t)\frac{\partial v}{\partial x}\right]+\frac{\partial}{\partial y}\left[(\mu+\mu_t)\frac{\partial v}{\partial y}\right]-\frac{2}{3}\rho\frac{\partial k}{\partial x}+(\rho-\rho_0)g \quad (3)$$

Energy equation:

$$\frac{\partial(\rho uT)}{\partial x}+\frac{\partial(\rho vT)}{\partial y}=\frac{\partial}{\partial x}\left[\left(\frac{\lambda_c}{Cp}+\frac{\mu_t}{\mathrm{Pr}_t}\right)\frac{\partial T}{\partial x}\right]+\frac{\partial}{\partial y}\left[\left(\frac{\lambda_c}{Cp}+\frac{\mu_t}{\mathrm{Pr}_t}\right)\frac{\partial T}{\partial y}\right] \quad (4)$$

Turbulent kinetic Energy (k) equation:

$$\frac{\partial(\rho uk)}{\partial x}+\frac{\partial(\rho vk)}{\partial y}=\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x}\right]+\frac{\partial}{\partial y}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial y}\right]-\rho\varepsilon+\frac{g}{\rho}\left(\frac{\mu_t}{\mathrm{Pr}_t}\right)\frac{\partial \rho}{\partial y}+G \quad (5)$$

Kinetic energy of turbulence dissipation (ε) equation:

$$\frac{\partial(\rho u\varepsilon)}{\partial x}+\frac{\partial(\rho v\varepsilon)}{\partial y}=\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x}\right]+$$
$$\frac{\partial}{\partial y}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial y}\right]+C_1 f_1\left(\varepsilon/k\right)G-C_2 f_2\left(\varepsilon^2/k\right)$$

(6)

The average Nusselt number along the panels module is defined as:

$$Nu_{H\_moy}=\frac{\bar{h}\times H}{\lambda_c}$$

The average convective heat transfer coefficient is defined as:

$$\bar{h}=\int_{\substack{exposed\\surface}} h_{conv}\times dn$$

2.4. Boundary Conditions

Two-dimensional open channel asymmetrically heated by imposing constant temperature on the PV panels as noted in Table 2. All walls are no-slip and impermeable boundary conditions were used.

Table 2. Boundary conditions of the considered problem

Geometric parameter	Hydrodynamic	Thermal
Channel inlet	$P=P_{atm}$	$T=T_{amb}$
Channel outlet	$P=P_{atm}$	$T=T_{amb}$
Panels surface	$u=v=0$	$T=T_{pv}$
Insulation wall	$u=v=0$	$\partial T/\partial y=0$

3. Numerical approach

3.1. Numerical procedure

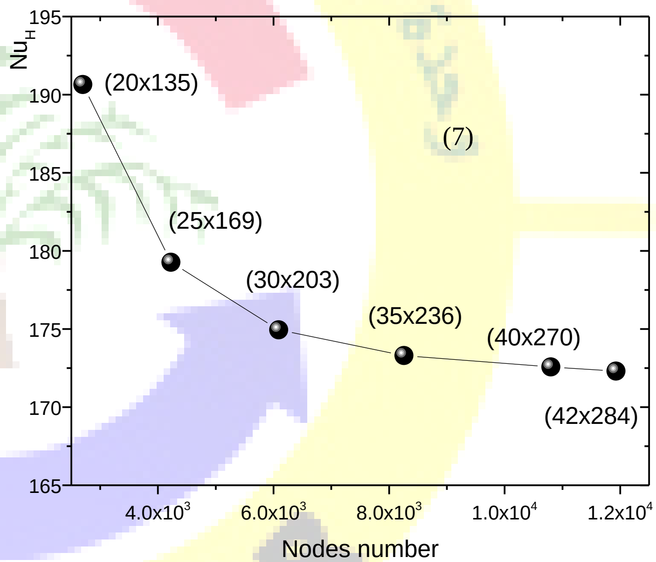
The governing equations presented above are discretized into algebraic equations by using the Finite Volume Method (FVM), which is widely used in computational fluid dynamics (CFD). The SIMPLE algorithm made coupling between the momentum and continuity equations.

(6)

3.2. Mesh independence study

Fig. 2 presents the average Nusselt number evolution as a function of the number of nodes for a modified Rayleigh, Ra*. The mesh test is used to ensure that the numerical solution is independent of the mesh. It is clear that the mesh (40x270) nodes, present a better solution, in order to optimize the CPU time and the cost of the calculations.

(7)



(7)

Fig. 2. Average Nusselt number evolution versus the nodes number for fixed Rayleigh number value of Ra=1.28E+10

3.3. Validations

The validation of the numerical code was carried out in under the same conditions of experimental studies that examined by of Onur et al. [19]. Fig. 3 presents the average Nusselt number evolution for fixed channel gap as a function of Rayleigh numbers, Ra. This numerical result is in a good agreement with the experimental value reported by the ref. [19] for the same geometrical conditions, in the present study, the average Nusselt number was estimated by following correlation equation:

$$Nu_{H\_moy} = 0.072(Ra)^{0.362} \quad (7)$$

The comparison of our correlation with the experimental evaluated by Onur et al. [19] has an error of about ε = 7%.

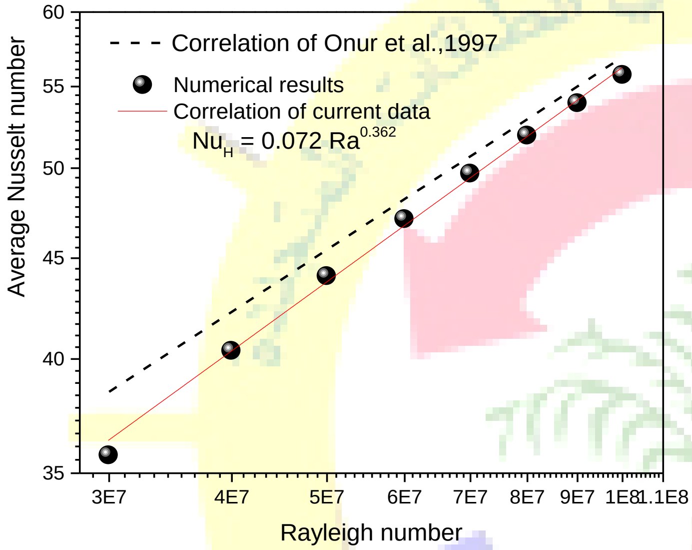


Fig. 3. Average Nusselt number evolution compared to the experiment examined by Onur et al. [19].for S=10mm.

4. Results and discussion

4.1. Influence of the channel gap on the temperature field

Figure 4 shows the temperature evolution along the inclined channel, for different channel air gaps.

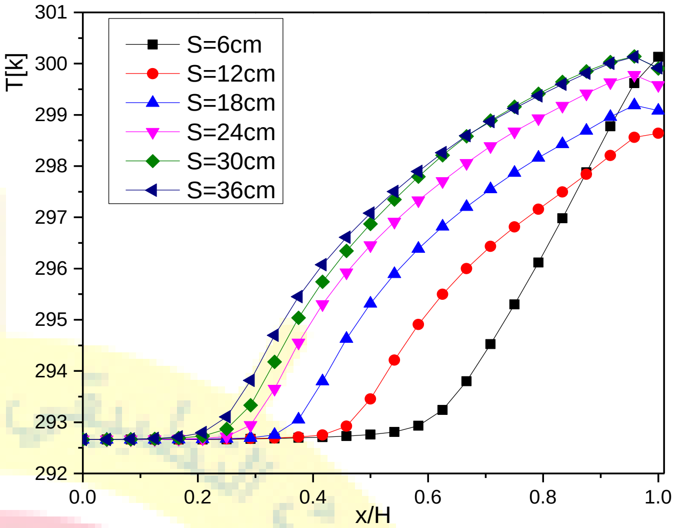


Fig. 4. Temperature evolution versus dimensionless channel for various channel gap at y=S/2cm and Ra = 1.28E+10

Everything is approaching in the center until the channel outlet of PVT air collector it is observed that the temperature is higher compared to the entrance, because of the thermal exchanges by the turbulent natural convection between the airflow and the panel wall. In addition, it is noted that the temperature increase with the increase of the channel gap. This is explained by the decreasing of the velocity suction of the air, which allows escaping a quantity of heat outside.

4.2. Influence of the channel gap on the velocity field

Fig. 5 illustrates the velocity evolution as a function of dimensionless channel height in the for the different channel air gap. As the channel gap of the hybrid system PVT increases, the velocity decreases. On the other hand, the velocity reaches its maximum at the channel outlet because of heat exchanges between the panel and the airflow.

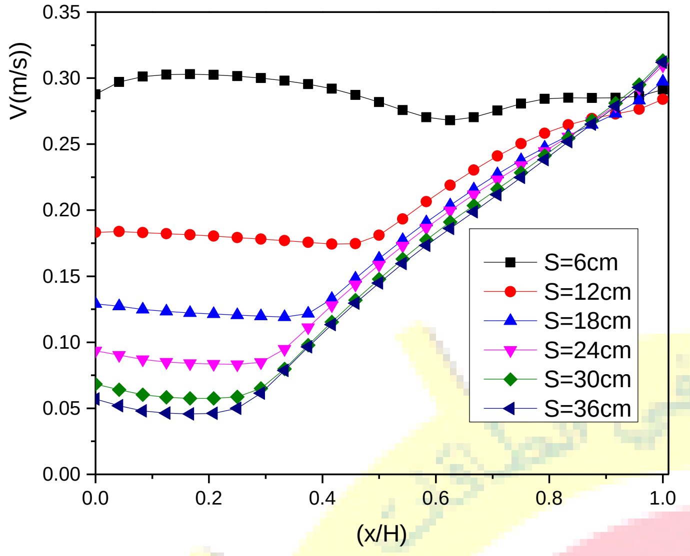


Fig. 5. Magnitude velocity evolution versus dimensioneless channel height for various channel gap is plotted at y=3cm and Ra_H= 1.28E+10

4.3. Influence of the channel gap on the mass flow rate

Figure 6 shows the evolution of the mass flow rate of the air as a function of the channel entrance. The mass flow rate of air rising with the increase in the aspect ratio of the channel length relative to its gap to a certain optimum value at Gamma = 7.

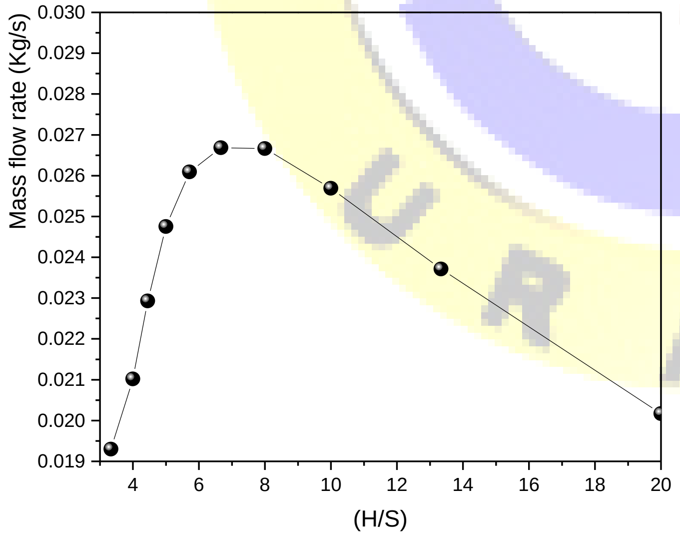


Fig. 6. Mass flow rate evolution versus aspect ratio plotted at y=3cm and RaH= 1.28E+10

4.3. Influence of the channel air gap on the Average Nusselt number

Fig. 7 illustrates the average Nusselt number evolution as a function of the aspect ratio. The heat transfer rate decreased with increasing the aspect ratio.

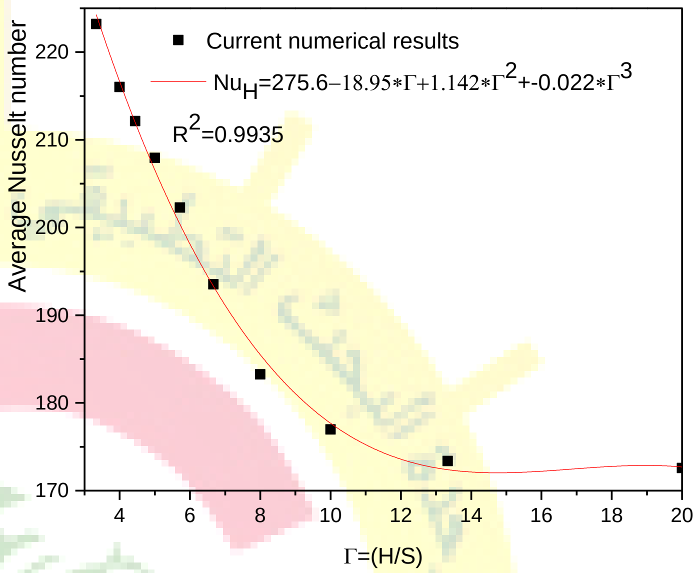


Fig. 7. Average Nusselt number evolution versus aspect ratio (Γ=H/S) RaH= 1.28E+10

It decreases of about 47% of the lower aspect ratio value compared to the large value Γ=3.33. The heat transfer rate as a function of aspect ratio is correlated by the following equation:

$$Nu_{H\_moy} = 275.6 - 18.95*\Gamma + 1.142*\Gamma^2 - 0.022*\Gamma^3 \qquad (8)$$

5. Conclusions

A bidimensionnelle numerical study on the channel gap effect on the air-cooling by turbulent natural convection was analysed. This work started with a comparison of current obtained numerical results and those obtained from the experiment of Onur [19], a good agreement has been found.

The main results are as follows:

- The velocity increases near the channel outlet of the hybrid system PVT due to the heat exchanged between the hot panels and the airflow.

- Correlations of average Nusselt numbers as a function of channel gap are analysed and discussed.
- The evolution of the average Nusselt number according to the air gap of the channel is increased proportionally.

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