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Research Paper

Determination of the heat transfer coefficient by convection, according to shape of the baffles (solar air collector)

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A B S T R A C T

This experimental study and mathematical modelling aim to improve the heat transfer by convection in different cases of transversal baffles inside a duct of a solar air collector. The study enabled us to formulate a mathematical equation of the heat transfer coefficient as a function of pressure drop and the mass flow rates. This mathematical model gave results that are close to the actual results in different inclination angles of the baffles (45°, 90°, 135°, and mixed between 135°and 45°). Through these results, it is found that the best heat transfer coefficient by convection was in the mode where $\beta = 90°$ and in the mixed mode. It was also noticed that the pressure drop in the mixed mode was smaller. With respect to the thermo-hydraulic performance factor (THPF), the case β=45° was retained as the best case.

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1. Introduction

Solar energy is free energy, doesn't disappear, and is very easy to use. One of the most important ways to benefit from it is to transform it into thermal energy by solar collectors which convert solar energy into a thermal and transfer it to the fluid passing through a channel. This energy is used in many fields like heating and drying. There are a lot of works that have been investigated in this field to improve the thermal efficiency of these collectors by taking into consideration the pressure drop (hydraulic performances), such as changing the shape and length of the channel [1-3], and adding baffles with different geometrical shapes [4-6]. In this context Chabane et al. [4] studied experimentally the thermal performance of solar air heaters with and without fins, the shape of these baffles was semi-cylindrical, and the inclination angle $\beta = 37°$. The results showed that the thermal efficiency of the solar air collector with semi-cylindrical-shaped baffles was larger by 19% than that without baffles. The highest value was recorded in the case of the collector with baffles for mass flow rate m=0.02Kg/s which was 75%, also Khanoknaiyakam et al. [7] carried out an experimental study of solar air collectors with V-shaped baffles, the study was on the thermal properties inside a rectangular duct. In this work it was shown that the addition of V-shaped baffles leads to an increase in pressure drop as well as an increase in heat exchange. Kumar et al [8] found that broken multiple e V-type baffle gives a great thermo-hydraulically results compared with the rest baffles through an experimental investigation performed with Reynolds numbers varying from Re=3000 to Re=8000. Sharma et al[9] performed experimental and CFD investigations of the heat transfer inside a duct of solar air collector with different shapes of baffles. They used six configurations to get the best shape through the thermohydraulic performances from Re=3000 to Re=18000. Khansari et al[10] presented an experimental and CFD study of solar air collectors with and without plus-shapes perforated baffles, and tested its effect on drying application. The study was divided into three baffles cases novel parallel-pass with double baffles (PPSCDB), parallel-pass with single baffles (PPSCB), and parallel-pass without baffles (PPSC). In the field of the thermohydraulic optimization of collectors, Pham et al. [11] performed a 2D CFD investigation of solar air collectors with three inclination angles of baffles by varying Reynolds number from Re=5000 to Re=20000. Hu et al.[12] studied experimentally and theoretically a new model of solar air collector, the channel was divided into five-chamber sections, in order to increase the length of the air channel. Through the experimental and theoretical results, it was found that the width of the first room has a significant effect on thermal efficiency and a low effect on the pressure drop for the width of 200mm, where the thermal efficiency has the greatest value, and it was greater than 16.90%, compared to the collector with rooms evenly distributed. In order to improve the heat exchange between the absorber plate and the air passes, Wang et al. [13] did an experimental study of a solar air heater with S-shape baffles; these baffles have an opening to facilitate the passage of air. In this study, the effect of some important factors on thermal efficiency and the difference in temperature between the inlet and outlet were studied. These results were compared to those of flat plate solar air collectors. A significant increase in thermal efficiency of this solar collector has been recorded compared to the same collector without baffles. An experimental study was performed for multiple obstacles geometrical designs and without baffles of solar air heater by Akpinar et al. [14], in order to improve the performance of this collector for three models of baffles (triangular baffles, leaf baffles, rectangular baffles), and without baffles. Their measurements were made with mass flow rate values m=0.0052 kg/s and m=0.0074 kg/s. Through the results of the study, it was found that the best model in terms of efficiency was model II (with leaf baffles), the lowest value was also recorded for the collector without baffles in all of conditions, where the efficiency has changed from 20% to 82%. This study showed that the thermal efficiency of the collector is closely related to the geometry of the obstacles, and it increases as the mass flow rate increases. A new model of solar air collector has been developed by Wijeysundera et al.[15], and the results were compared with a single-pass heat collector. Their study, that was performed for several operating conditions, confirmed that the efficiency of a double-pass solar air collector is better than that of a single-pass duct.

2. Experimental study

This present experimental study was carried out in the technological lobby of the department of mechanical engineering of the University of Biskra. The experiments are conducted for a constant inclination angle $\beta = 38°$. The solar air collector studied consists of a single channel through which the air passage is well insulated to reduce heat losses. All experiments were performed between 11:00 a.m. and 1:00 p.m., when the solar radiation was almost constant $I \approx 900 W/m^2$. Every quarter of an hour, the pressure and the temperature are measured at the inlet and outlet of the solar air collector, as well as the temperature of the absorber plate through five measurement points distributed over its surface (Figs. 1, 2).

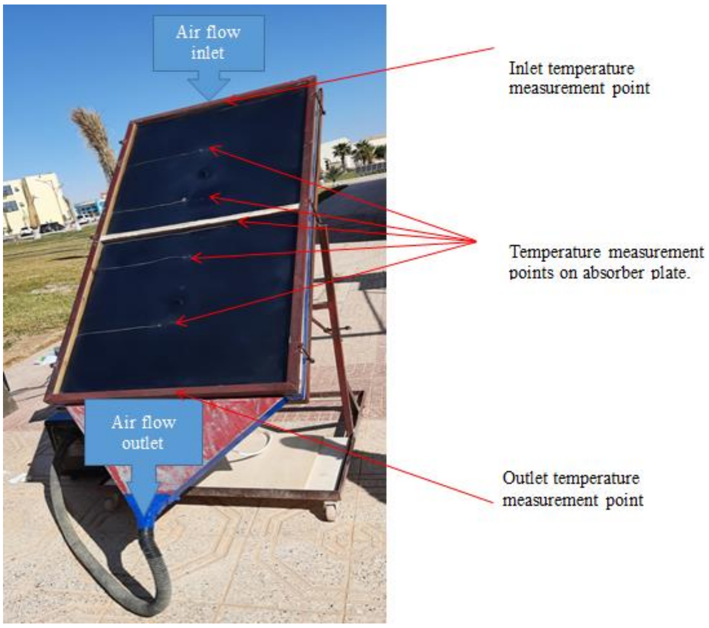

Fig. 1. Experimental setup of the solar collector.

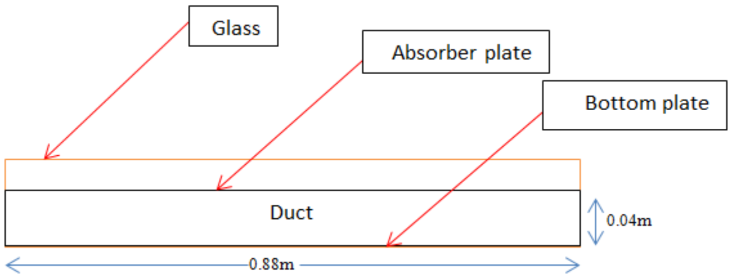

Fig. 2. Cross-section of the solar air collector without baffles.

2.1. Measuring devices

In this work, we use a four-socket Thermocouple (K-type -200~1372°C2501°F) to measure the temperature using thermal pickups (Fig 3) at five points on the absorber plate and two points in the level of inlet and outlet of the collector. Kimo 300 class was used to measure the difference in inlet and outlet pressure using pipes connected to it. The uncertainty of the measuring devices is $\omega_{\Delta T}=\pm0.2°C$ for the thermocouple and $\omega_{\Delta P}=\pm1Pa$ for Kimo 300 class.

We used the power regulation to control and modify the mass flow rates, with precise electronic control and reading from the digital screen directly (Fig. 4).

2.2. Baffles

To increase the heat transfer from the absorber to the air passing in the channel, and to break the dead layers in the areas close to the absorber, rectangular baffles have been added inside the channel perpendicular to the air stream (Fig. 5). Different inclination angles where considered and compared with works of Chabane et al. [16, 17]. In this work, three different inclination angles for obstacles of the same number and position are studied (18 Baffles), in addition to the smooth plate case.

a

b

c

Fig. 3. The devices used in the experiment:

a) Thermocouple to measure the temperature at various points, the reading is directly from the display screen, b) Kimo 300 class, c) Thermal pickups.

Fig. 4. Power regulation and aspirator respectively

Fig. 5. The rectangular shape of the baffles.

The baffles with different angles have been studied to get the best position that gives the highest value of the heat exchange coefficient, taking into consideration the pressure drop. The inclination angles considered were as follows: β = 90° [16, 17], β = 45°, β = 135° and β =135°-45° (Fig. 6).

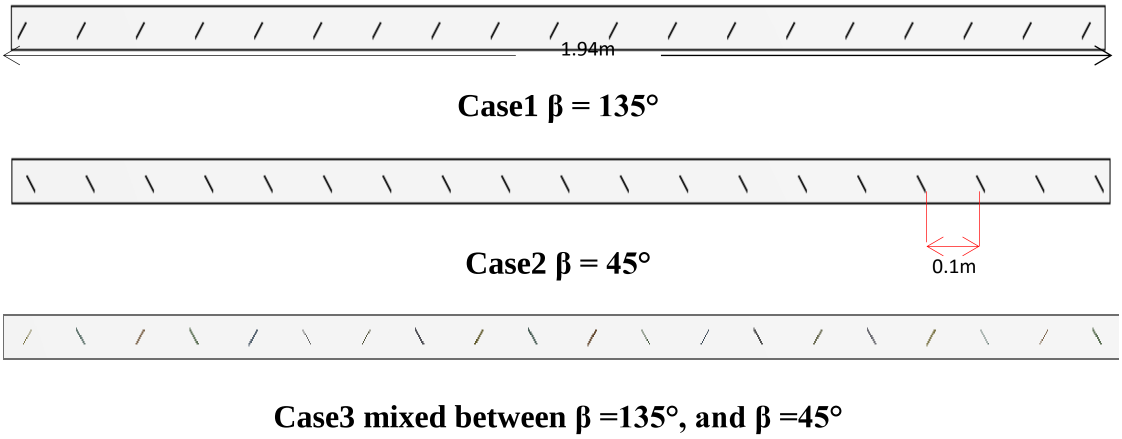

Fig. 6. The different angles of baffles.

3. Modelling

In these experiments, the temperature of the air at the inlet and outlet is measured at the same time; we measure the temperature at five points on the absorber plate, and that to calculate the heat flux of the solar air collector which is given by the following relation:

$$\Phi = m \times C_p \times (T_{out} - T_{in}) \quad (1)$$

Through this relation, the global heat transfer coefficient can be calculated by the temperature of the absorber plate and the temperature of the air passing through the duct, which is given by the following relation [18]:

$$h = \frac{\Phi}{S(T_{abs} - T_{air})} \quad (2)$$

Also during the experiment, the air pressure difference between the inlet and the outlet was measured. The pressure difference is given by the following relation:

$$\Delta p = \Lambda \frac{L}{D_h} \times \rho \times \frac{V^2}{2} \quad (3)$$

Where Λ is the pressure drop coefficient, D_h is the hydraulic diameter of the air channel and L is channel length. Also, the phenomenon is controlled by dimensionless properties. Reynolds number represents the relationship between inertial forces and viscous forces and characterizes the nature of the flow regime:

$$\mathrm{Re} = \frac{\rho \times V \times D_h}{\mu} \quad (4)$$

$$D_H = \frac{4A}{P} \quad (5)$$

To find the best thermal situation, taking into account the hydraulic side, a thermo-hydraulic performance factor parameter has been defined by R. L. WEBB [19]. This parameter is given by the following expression:

$$\mathrm{THPF}=(\mathrm{Nu}/\mathrm{Nu}_0)/(f/f_0)^{1/3} \quad (6)$$

Where Nu, f and Nu_0, f_0 are the Nusselt number and friction factor for duct with and without fins:

$$f = \frac{(\Delta P/L) D_H}{2\rho U^2} \quad (7)$$

$$Nu = \frac{h D_H}{\lambda} \quad (8)$$

4. Theorical study

The experimental study showed that the heat exchange coefficient is affected by two terms: the first one is the mass flow rate, and the second estimates the form of the baffle into the stream channel. Then we propose a new mathematical model for the coefficient of heat exchange by convection as a function of the mass flow rate and the pressure drop. Using the least squares method and the experimental results for different angles of inclination of the rectangular baffles, the heat exchange coefficient is expressed as follows:

$$h = a \times \left(m^b \times \Delta p^c\right) \quad (9)$$

The constants a, b, and c are function of the angle of the obstacle. They are given in Table 1.

Table 1. Change of coefficients as a function of the positions of baffles.

	45°	**135°**	**90°**	**45°-135°**
a	24.41	20.7	20.5	13.85
b	0.2	0.21	-0.045	0.07
c	0.423	0.46	0.58	0.46
R^2	0.9818	0.998	0.98	0.9665

5. Results and discussion

In this part of the study, we present the heat transfer coefficient and pressure drop in the field for Reynolds numbers varying from 624 to 2834 and different cases of the baffle inclination. These results are compared to those of Chabane [16, 17], whose results show the change of pressure drop and heat transfer coefficient as well.

Fig. 7 shows the variation of the heat transfer coefficient as a function of the mass flow rate. The first thing that can be noticed is the significant difference between the heat transfer coefficient of the solar collector with and without obstacles. The largest difference is recorded between the mixed-mode (135°and 45°) and the collector without baffles, with a maximal value of 62 W/m^{2}k for a mass flow rate m = 0.06 kg/s. Likewise, we can note that the heat transfer coefficient values corresponding to the two configurations with baffles, β = 90° and the mixed, are equal to h = 85 W/m^{2}k at mass flow rate m = 0.08 kg/s. We noticed also that the lowest values of the heat transfer coefficient are recorded at the lowest mass flow.

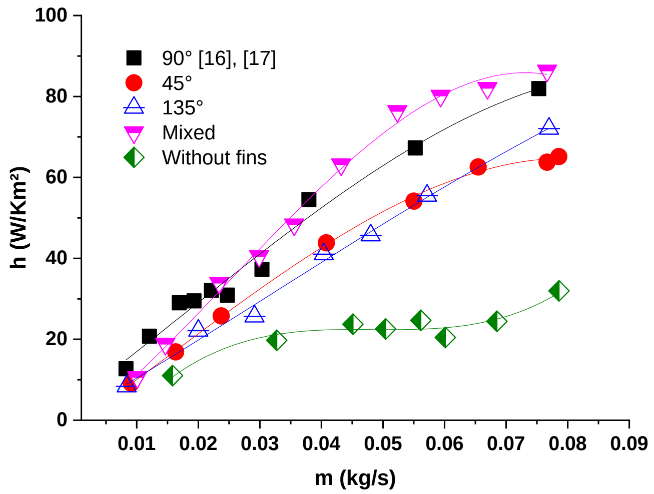

Fig. 7. Variation of heat exchange coefficient as a function of mass flow rate for different modes.

Fig. 8, shows the variation of the pressure drop as a function of the mass flow rate. We notice that the pressure drop increases with the increase of the mass flow rate. The lowest values are recorded for the baffle inclinations β = 45° then β = 135°, and the highest value of the pressure drop is recorded for the reference case where the inclination angle β = 90°.

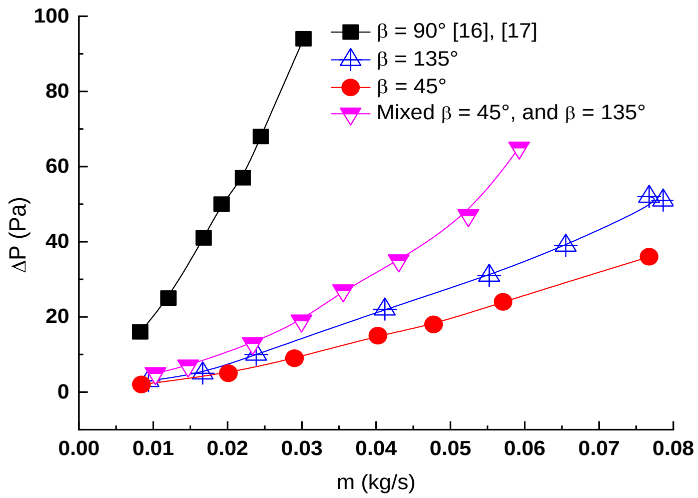

Fig. 8. Variation of pressure drop coefficient as a function of mass flow rate for different inclinations of the baffles.

The pressure drop coefficient is a parameter related to the hydraulic diameter of the duct and the density of the air passing through it, in addition to the air velocity and the length of the channel. There are two types of this parameter, the partial coefficient and the global coefficient: the first is in terms of the length of the channel, and the second is concerning one meter.

We notice that the change in this parameter is directly related to the pressure drop, as we notice that the lowest value corresponds to the case $\beta = 45°$, with the values ξ=0.2 and λ=0.1 for the global and the partial coefficient respectively (Fig 9). The highest values are found for the case corresponding to $\beta = 90°$, with the values ξ=1.65 for the global coefficient and λ=0.88 for the partial coefficient.

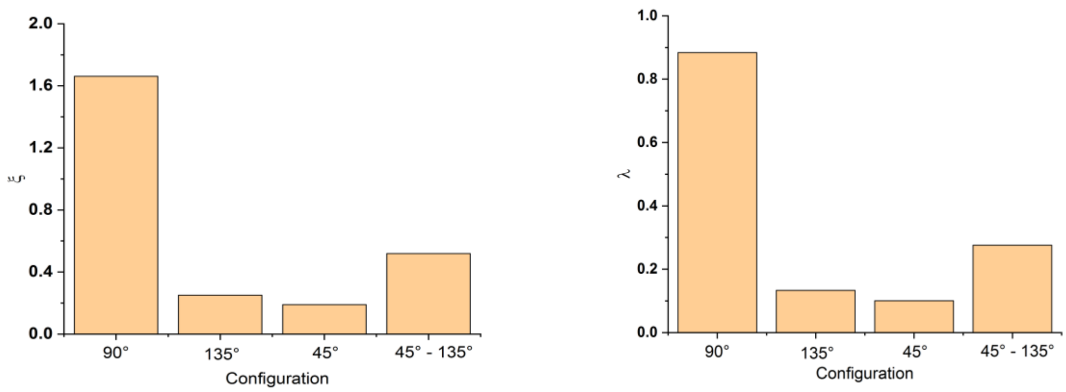

Fig. 9. Global and partial pressure drop coefficient.

One of the most important parameter to determine the best case in terms of thermal and hydraulic performances is the thermo-hydraulic performance factor (THPF). Its average values are calculated for the mass flow range m=0.01kg/s to m=0.03kg/s (Fig. 10). These results show that the highest value corresponds to the case β=45°, with THPF=0.540. This indicates its better efficiency compared to other cases. Also, the lowest THPF value corresponds to the reference case with a value of 0.336.

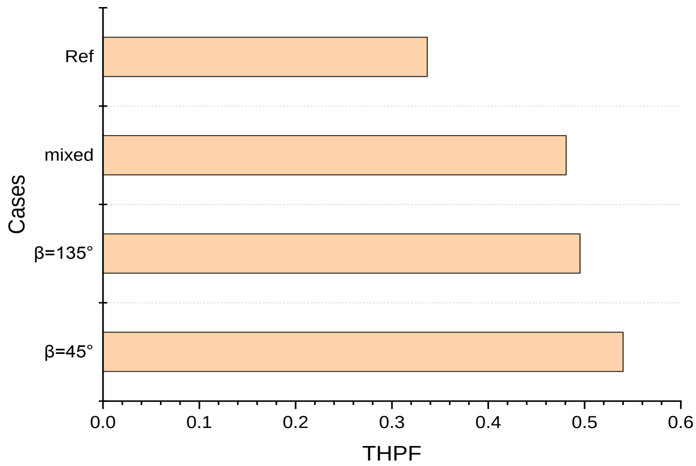

Fig. 10. Thermo-hydraulic performance factor for all cases.

6. Conclusion

The results obtained during these experiments show the effect of adding the baffles on the heat exchange between the absorber plate and the air and its effect on pressure drop. Three cases were studied in addition to the reference case and the case without baffles. The experimental results show that the addition of baffles increases the heat transfer between the air and the absorber plate, and that the addition of the baffles creates a turbulence that causes cracking of the dead layers near the absorber plate, which causes an improvement in the heat exchange because the dead layers decrease the thermal transfer through them. The best value of the heat transfer coefficient was recorded in the case with mixed inclination if the baffles, followed by the case with the inclination angle β = 90°. The results showed that the lowest pressure drop was recorded for the case corresponding to β = 45°. Regarding the results of the thermo-hydraulic performance factor (THPF), the best configuration corresponds to the case β=45°, the remaining cases are ranked from best to worst as follows: β=1355°, Mixed case, reference case.

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Nomenclature

Φ	Heat flux (W)
$\dot{m}$	Mass flow rate (kg/s)
Cp	Specific heat (J/Kg k)
Tabs	Average temperature of the absorber plate (k)
T_{air}	Average temperature of the air (k)
L	Length of the duct (m)
D	Hydraulic diameter (m)
ΔP	Pressure drop (Pa)
Re	Reynolds number
Λ	Pressure drop coefficient
ρ	Fluid density(kg/m^3)
V	Velocity (m/s)
μ	Kinematic viscosity (m^2/s)
h	Heat transfer coefficient (W/m^2 k)
S	Absorber plate area (m^2)