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Effect of the Tilt Angle of Natural Convection in A Solar Collector with Internal Longitudinal Fins

Foued Chabane[1], Nourddine Moummi[2], Said Benramache[3]
[1,2] Mechanical Energy Department, Faculty of Technology, University of Biskra 07000, Algeria
[3] Material Science Departments, Faculty of Science, University of Biskra 07000, Algeria
([1]fouedchabane@live.fr, [2]n.moummi@hotmail.com, [3]saodzno@yahoo.fr)

Abstract- We present in this paper, a series of experimental tests carried out on a solar plan, with an area of 1.8236 m^2 capture, designed in the laboratory and tested in the region of Biskra. The complete experimental setup includes the measurement of global solar radiation, wind speed, airflow velocity and temperature and the ambient air at the inlet and outlet; optimize a tilt angle. The solar collector was oriented facing south Renewable and domestic energy source, and is essential components of a sustainable energy future. This paper deals with the influence of the tilt angle of solar collectors. The optimum angle is measured by searching for the values for which the total radiation on the collector surface is a maximum for a particular day or a specific period. An application of the model is done using the experimental data measured for Biskra in Algeria. For increasing the utilization efficiency of solar collectors, it is recommended that, if it is possible, the solar collector should be mounted at the monthly average tilt angle, and the slope adjusted once a month.

Keywords- *tilt angle; solar intensity; heat transfer.*

I. INTRODUCTION

Solar Panel from Environmental Solar Systems is the ideal choice for homeowners that want environmentally friendly and low-cost alternative heat. Solar panels harness the sun's energy in the form of light and convert the energy into heat or electricity. Sunlight – solar energy – can be used to generate electricity, provide hot water and to heat, cool and light buildings. Renewable solar energy is one of the fastest growing energy sources in the world. Although the average consumer might associate solar panels with residential rooftop assemblies, solar panels are available for a wide range of applications, including powering individual gadgets. Yasin Varol & al [1], reported that the laminar natural convection in inclined enclosures filled with different fluids was studied by a numerical method, and the heat transfer was lower in the air side of the enclosure than that of the water side. In all of these studies, the partition is not thermally conductive.

However, the thermal conductivity of the partition is an important factor from the control of heat transfer point of view. In this respect, Ho and Yih [2] made a numerical analysis on conjugate natural convection in air-filled rectangular cavities. Their results indicate that the heat transfer rate is considerably attenuated in the partitioned cavity in comparison with that for non-partitioned cavity. Recently, Kahveci [3] investigated natural convection in partitioned air-filled enclosure heated with a uniform heat flux using a differential quadrature method. He found that average Nusselt number increases with a decrease of thermal resistance of the partition and partition thickness has little effect on heat transfer. Then, he made analyses of different versions of this problem [4].

II. DEFINITION OF PHYSICAL MODEL

A. Collectors

In this study, an experimental setup, which consists of an air-type solar collector, a fan and air ducts, is build Fig. 1, an anemometer, we used a CMP 3 pyranometer is an instrument for measuring the solar irradiance and digital thermometer Model Number: DM6802B and dimensions of our experimental setup is given, Air-type solar collector has an inclination of 45°. The solar air heater constructed in this study is shown in Fig. 1-2 this air heater is used in space heating. It is a rectangular box of L = 2000 mm height, l = 1000 mm width and 100 mm thickness. There is one passage backup the absorber plate that air could flow within it. A black-painted of 0.5 mm thickness was used as absorber the plate absorption coefficient $\alpha = 0.95$. A galvanized flat plate of 0.6 mm thickness was used as back plate and an insulation plate (polystyrene) of 2000 mm height, 1000 mm width and 40 mm thickness with thermal conductivity 0.037 W/(m.K). The distance between two adjacent fins and fins height are 120 and 300 mm respectively and 6 mm thickness with longer 1880 mm. High transmittance Plexiglas with 3 mm thick, dimension size of 1940 × 940 mm, and which was to reduce the convective and long wave losses to the atmosphere, the transparent cover transmittance $\tau = 0.9$ and absorptive of the glass covers, $\alpha_g = 0.05$. We have the fins of semi-cylindrical longitudinal underside of the absorber. We have 16 Positions of thermocouples connecter to plates and two thermocouples to outlet and inlet flow.

Fig. 1 The photograph of experimental set-up.

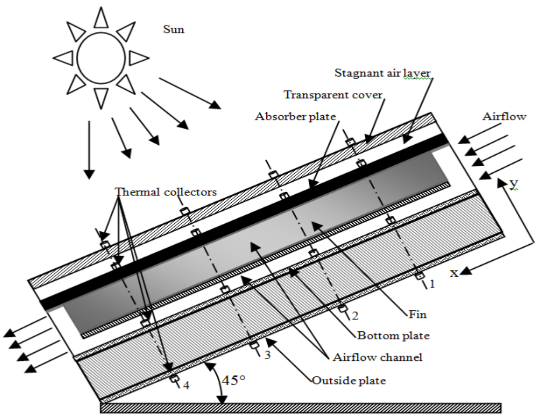


Fig. 2 Schematic showing the heat transfer components of the solar air collector.

B. *Heat transfer coefficients*

The convective heat transfer coefficient h_w for air flowing over the outside surface of the glass cover depends primarily on the wind velocity V_{wind}. McAdams [5] obtained experimental result as:

$$h_w = 5.7 + 3.8V_{wind} \tag{1}$$

Where the units of h_w and Vwind is W/m²K and m/s, respectively. An empirical equation for the loss coefficient from the top of the solar collector to the ambient was developed by Klein [6].The heat transfer coefficient between the absorber plate and the airstream is always low, resulting in the low thermal efficiency of the solar air heater. Increasing the absorber plate shape area will increase the heat transferred to the following air.

C. *Qualitative modelling research*

1. Heat transfer modelling

The heat transfer components of solar air collector in this work are schematically depicted in Fig.3.The heat transfer model is based on the following assumptions:

- ✓ The solar air collector is normally made of materials with small thermal capacity, such as iron, glass, expanded polystyrene board (EPS). Therefore, the thermal capacity of the collector components can be negligible (dT/dt = 0).
- ✓ The heated airflow inside the collector has a uniform velocity.
- ✓ The air temperature only varies along the airflow direction.
- ✓ The flow channel is hydraulically smooth.
- ✓ There is no air leakage to or from the collector, or between the stagnant air layer and airflow channel.
- ✓ The side plate is thermally insulated.

2. Collector Thermal Efficiency

The efficiency of a solar collector is defined as the ratio of the amount of useful heat collected to the total amount of solar radiation striking the collector surface during any period of time.

$$\eta = \frac{\textit{Solar Energy Collected}}{\textit{Total Solar Striking Collector Surface}} = \frac{Q_{useful}}{I \times A_s} \tag{2}$$

Useful heat collected for an air-type solar collector can be expressed as:

$$Q_u = \dot{m} C_p \left(T_{outlet} - T_{inlet}\right) \tag{3}$$

With $\dot{m} = V_f .S$

So, collector thermal efficiency becomes, On the basis of the energy distribution of the collector, as Fig. 2 presents, thermal efficiency of the collector, serving as the main indicator, is calculated by equation (4):

$$\eta = \dot{m} C_{pa} \frac{\left(T_{outlet} - T_{inlet}\right)}{I \;\; F_2} \tag{4}$$

Based on energy conservation, the internal energy change of the Collector is caused by thermal disturbance, expressed as:

$$Q_{ie} = Q_s - Q_u - Q_{tc,loss} - Q_{bp,loss} - Q_{re} \tag{5}$$

Where $Q_{tc,loss}$ and $Q_{bp,loss}$ are the accumulated heat losses of the transparent cover and back plate respectively, which can be calculated by the following equations:

$$Q_{tc,loss} = \int_0^t \left(h_{1ac,out,1} + h_{1r,sky} + h_{1r,g}\right)\left(T_1 - T_{a,out}\right)F.dt \quad (6)$$

$$Q_{tc,loss} = \int_0^t \left(h_{5ac,out,1} + h_{5ar,out}\right)\left(T_4 - T_{a,out}\right)F.dt \quad (7)$$

The convective heat transfer coefficient in equations (6) and (7) can be derived from the relationship drawn by Sparrow et al. [7], which is used for calculating the external heat transfer coefficient over the transparent cover and the back plate when ($Re=v_a D_h/\nu_a$) varies from 2×10^4 to 9×10^4:

$$Nu = 0.86\,Re^{1/2}\,Pr^{1/3} \quad (8)$$

Where $Nu = \dfrac{h_{ac,out} D_h}{\lambda_a}$, $Pr = \dfrac{C_{pa}\mu}{\lambda_a}$

And $D_h = \dfrac{2WL}{W+L}$

And the radiation heat transfer coefficient between the transparent cover and sky is written as [8]:

$$h_{1r,sky} = \frac{X_{sky}\varepsilon_1\sigma_b\left(T_{sky}^4 - T_1^4\right)}{T_{a,out} - T_1} \quad (9)$$

Where the equivalent sky temperature T_{sky} could be determined by Swinbank equation [9]:

$$T_{sky} = 0.0552\,T_{a,out}^{1.5} \quad (10)$$

The radiation heat transfer coefficient between the transparent cover and ground is expressed as [8]:

$$h_{1g,g} = \frac{X_g\varepsilon_1\sigma_b\left(T_g^4 - T_1^4\right)}{T_{a,out} - T_1} \quad (11)$$

For normal installation condition of a solar air collector (e.g., 40° installation angle form the horizontal plane), the view factor between the transparent cover and sky is 0.9, while the view factor between the transparent cover and ground is only 0.1 [10]. Therefore, it is reasonable to neglect the radiation between the transparent cover and ground.
Additionally, the view factor between the back plate and ground is nearly 1.0, so the radiation heat transfer coefficient between the back plate and ground is written as [8]:

$$h_{5ar,out} = \frac{\varepsilon_5\sigma_b\left(T_g^4 - T_4^4\right)}{T_{a,out} - T_4} \quad (12)$$

Where the ground temperature T_g is obtained from the climate database [11]. It is noticed that the reflection energy Q_{re} and the internal energy change Q_{ie} are difficult to be determined by test directly, so Q_{other} is defined to present the sum of them, and it can be calculated by equation (13):

$$Q_{other} = Q_s - Q_u - Q_{tc,loss} - Q_{bp,loss} \quad (13)$$

Thermal properties of air are considered to be variables according to the follow expressions (Tiwari, 2002), where the fluid temperature is evaluated in Celsius [12]:

$$C_p = 999.2 + 0.1434T_f + 1.101\times10^{-4}T_f^2 - 6.7581\times10^{-8}T_f^3 \quad (14)$$

$$\lambda = 0.0244 + 0.6773\times10^{-4}T_f \quad (15)$$

$$\nu = 0.0284\times10^{-4} + 0.00105\times10^{-4}T_f \quad (16)$$

The air density is calculated assuming fluid as an ideal gas by the expression

$$\rho = 353.44/T_f \quad (17)$$

Where T_f is the absolute air temperature.

III. Results and Discussion

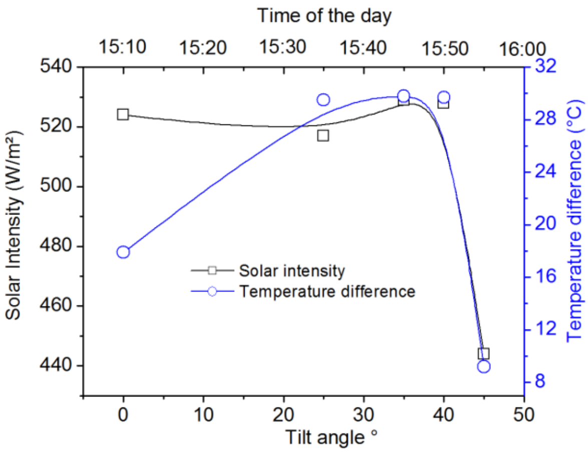


Fig. 3 solar intensity and temperature difference a function to tilt angle and the time of the day.

The effects of tilt angle and natural convection for solar air heaters are investigated experimentally under Algeria prevailing weather conditions during the month of May, with clear sky condition. Biskra is a city of Algeria located on 34°50'43.28"N latitude 5°44'49.11"E longitude. The solar collector was oriented facing south, subjected to environmental conditions.

Fig. 3 shows the solar intensity versus standard local time of the day in May; the experiment was carried out. The solar intensity increases from the early hours of day with about 528 W/m^2 at 15:10 h correspondent tilt angle $\beta = 0°$ and temperature difference $\Delta T = 28.7$ °C, and then, reduces later on during the day by $\Delta T = 8.6$ °C, The performance of the proposed single pass solar air heater with fin inferior the absorber plate has been studied and investigation the performance and heat exchange in the thickness of a solar air heater by the infect the tilt angle and a natural convection. During this work the results of solar radiation are varied by varied a tilt angle, the results in solar radiation attain the value of maximum equal to 528 w /m² at 15:50 with $\beta = 40°$ and $\Delta T = 29.3$ °C.

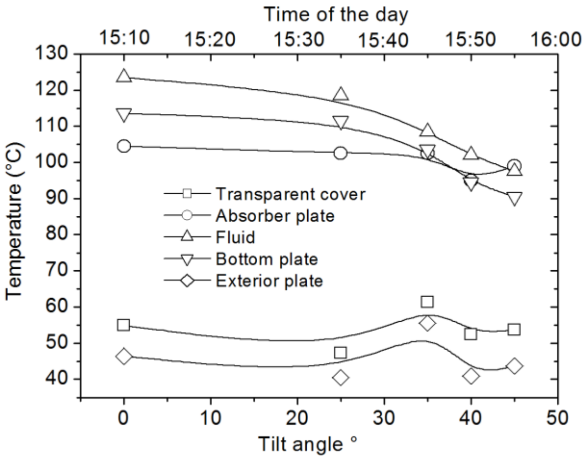


Fig. 4 Temperature areas of a solar collector a function to tilt angle and the time of the day.

Fig. 4, shows the hourly variations of the measured temperatures of the solar collectors as well as the measured temperatures of surface; such as transparent cover, absorber plate, fluid, bottom plate, exterior plate, it's a function to tilt angle, when the average wind speeds equal to 1.1 m/s. It is seen from the results of Fig. 4 that the maximum values of average temperatures (T_1, T_2, T_3, T_4, T_5) are 61.35, 104.5, 123.5, 113.5 and 55.55 °C at 15:10h, respectively, correspondent a tilt angle about β = 35, 25, 0, 0 and 35°, respectively. We can be seen in a curve the optimal tilt angle is 25° when a temperature of the surface becomes $T_1 = 47.3$, $T_2 = 102.5$, $T_3 = 118.5$, $T_4 = 111.5$, $T_5 = 40.55$; between during from 15:10h to 16:00h.

The wind speed is always changing its speed and its direction during the day and the month, measuring of the average wind speed is 1.9 m/s and 1.1 m/s Fig.5, respectively, that the wind speed be alternating in several directions; this affects the values of temperature of transparent cover and exterior plate, and these last is not infected directly on the elements internal of the solar panel but slight, the maximum values of the wind speed in May is 1.4 m/s about ambient temperature 36.3 °C Fig.6.

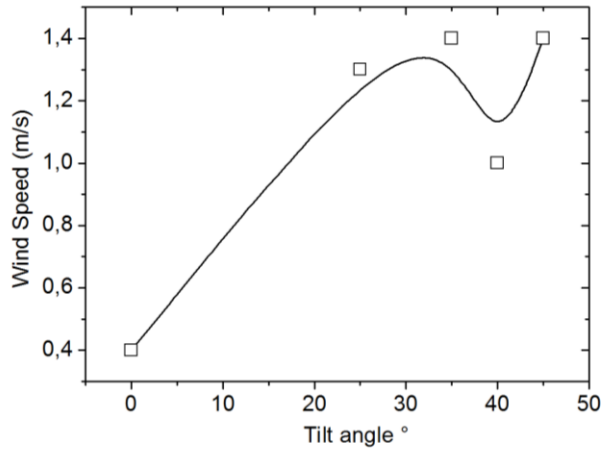


Fig.5 The weather condition on the test day of Wind velocity in May.

Fig.6 presents the ambient temperature it is function to tilt angle, this variation dependent of the weather conditions, the absolute ambient temperature correspondent $\beta = 0$ and 25° is 35.6 and 35.5 °C, about a mode with fins inferior the absorber plate. The ambient temperature it is help in a affect the values thermal of the solar collector and the natural convection, this effect shows in Fig. 7 it's estimated by inlet temperature function to time of the day and to tilt angle of a solar collector, this evolution remarkable; we can be seen the evolution of a inlet temperature Fig.7, therefore is prove we can be said the ambient temperature air as effect directly on solar panel, and in the thermal properties. The maximum temperature limited, equal to 36.3 °C, respectively a angle of inclination $\beta = 40°$.

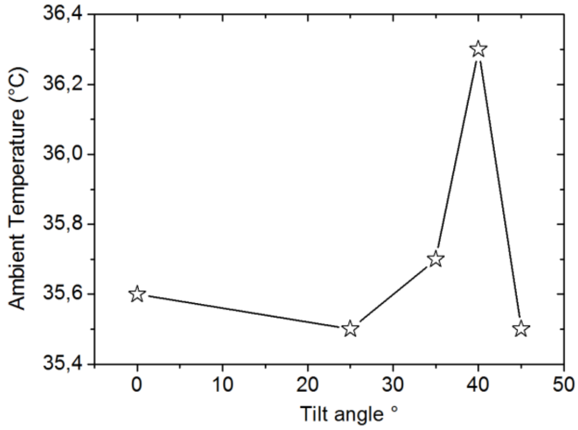


Fig. 6 Ambient temperature a function to tilt angle in May.

The effects of the air inlet temperature T_{in} by natural convection and incident solar radiation I on the collector efficiency as well as the air outlet temperature T_{out}, with fins attached inferior an absorber plate to have been investigated experimentally. As we can be seen from Fig. 7, variation in the inlet and a outlet temperature of a solar collector are a function to tilt angle and a time of the day, Although the aim for adding these fins in the inferior absorber plate for increase the heat exchange surface and uniform air flow through the entire solar air heater. Further tests are recommended to replace the fins in the lower absorber with an isolated partition to ensure a uniform distribution of the air flow rate through all the solar panel and to reduce the losses from the transparent cover et the exterior plate. The outlet temperatures presented in Fig. 7 are measurements for varied state values of solar irradiance. The outlet temperature is of maximum value is T_{out} = 74 °C for β = 35° when T_{in} = 46 °C about airflow speed out in a solar collector equal to 0.3 m/s, and decreased by T_{out} = 55 °C, when a tilt angle equal 45°, in this point we selected inlet temperature same to 46.5 °C. We can be seen the optimal various of the outlet temperature it's correspondent tilt angle equal to 35°, about solar intensity is 529 W/m², it is estimated a maximum solar intensity, the natural convection it's affect of a air flow by an influence of the difference heat temperature between the surface of a absorber plate and the bottom plate.

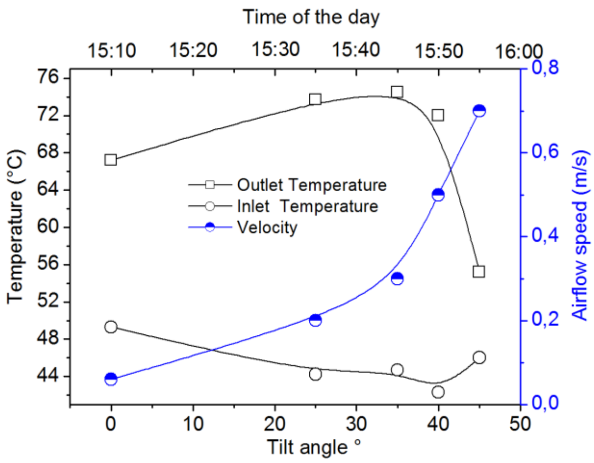
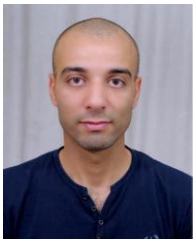


Fig. 7 Inlet and outlet temperature with airflow velocity a function to tilt angle in May.

IV. CONCLUSION

This study shows that for a single pass solar air heater using internal fins inferior an absorber plate, there is a significant increase in the thermal efficiency of the air heater. The outlet temperature increases when the angle of inclination β = 35°. The maximum temperature difference obtained from this study is 29.7 °C for tilt angles 25, 35 and 40°. The maximum outlet temperature obtained is T_{out} = 74 °C for β = 35° when T_{in} = 46 °C. The longitudinal fin used between the absorber plate and Bottom plate; for an increase the heat exchange and a help the air flow uniformed into channels of solar collector.



Foued Chabane was born in Algeria in 1980. He obtained his Engineer and his Magister degrees in Mechanical engineering option Energetic in 2004 and 2009 respectively. He works now towards PhD degree in Mechanical. His research interests are on solar air collector heater, I have designed new solar panel for my study, and He is currently an assistant Lecturer, in Physics and fluid mechanics Department, Biskra University, Algeria. E-mail: f.chabane@univ-biskra.dz fouedchabane@live.fr

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