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THERMAL EFFICIENCY ANALYSIS OF A SINGLE-FLOW SOLAR AIR HEATER WITH DIFFERENT MASS FLOW RATES IN A SMOOTH PLATE

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ABSTRACT

This paper presents an experimental thermal efficiency analysis for a novel flat plate solar air heater with several mass flow rates. The aims are to review of designed and analyzed a thermal efficiency of flat-plate solar air heaters. The measured parameters were the inlet and outlet temperatures, the absorbing plate temperatures, the ambient temperature, and the solar radiation. Further, the measurements were performed at different values of mass flow rate of air in flow channel duct. After the analysis of the results, the optimal value of efficiency is higher level of mass flow rate equal to 0.0202 kg/s in flow channel duct for all operating conditions and the single-flow collector supplied with maximum mass flow rate appears significantly better than that another flow rate. At the end of this study, the thermal efficiency relations are delivered for different mass flow rates. Maximum efficiency obtained for the single pass air heater between the air mass flow rates from 0.0108 to 0.0184 kg/s; were 39.72% and 50.47 % respectively, with tilt angle equal 45° in location Biskra city of Algeria. The thermal efficiency correspondently the mass flow rates were 28.63, 39.69, 46.98, 55.70 and 63.61 %, respectively.

Keywords: *Solar air heater; Experimental; Exergy analysis; Single-flow; Thermal efficiency*.

1. INTRODUCTION

In this paper an attempt has been done to optimize the thermal performance of flat plate solar air heater by considering the different system and operating parameters to obtain maximum thermal performance. The report talks about thermal performance for different mass flow rates, emissivity of the plate, and tilt angle (Varun, 2010). In our design we can been found that the use of selected coatings on the absorbing plates of all the heaters considered can substantially enhance the thermal performances of the heaters, and the Plexiglas covers does not have such a significant effect on the thermal performances of the heaters (Wenfeng *et al*., 2007). There are different factors affecting the solar collector efficiency, e.g. collector length, collector depth, type of absorber plate, glass cover plate, wind speed, etc. Increasing the absorber area or fluid flow heat-transfer area will increase the heat transfer to the flowing air (Chabane *et al.,* 2013a-e). On the other hand, it will also increase the pressure drop in the collector, thereby increasing the required power consumption to pump the air flow crossing the collector (Akpinar and Koçyig˘it, 2010; Karsli, 2007). Kalogirou (2006) estimated the performance parameters of flat plate solar collectors using ANN and results obtained are compared with actual experimental values. A number of attempts (Charters, 2006; Hollands and Shewen, 1981; Bejan *et al*., 1981; Altfeld *et al*., 1988; Altfeld *et al*., 1988; Bhargava and Rizzi, 1990; Verma *et al*., 1992; Hegazy, 1996) have been made during the last 30 years in an effort to improve the thermal performance of flat plate SAHs by optimizing air channel depth with respect to its length or width. Work reported the effect the mass flow rate in range 0.0078 to 0.0166 kg/s on the solar collector with longitudinal fins (Chabane *et al*., 2012a).

2. EXPERIMENTAL SECTION

2.1. Collector analysis

The delivered energy output from the solar collector depends on the optical and thermal properties of the collector. This studied that contains an experimental background of the parameters used to characterize the collector thermally. The methods to characterize the collector thermally are presented. The first is to measure the interior temperature properties of the absorber plate and the bottom plate and then calculate the efficiency. The second is through outdoor measurements of temperature such as inlet, outlet and ambient temperature and the thermal characterization is also presented, indoor hot-box measurements. A typical flat-plate collector consists of an absorber in an insulated box together with transparent cover sheets (Plexiglas). The absorber is usually made of a metal sheet of high thermal conductivity, such as galvanized. Its surface is coated with a special selective material to maximize radiant energy absorption while minimizing radiant energy emission. The insulated box reduces heat losses from the back and sides of the collector (Duffie and Beckman, 1991). Plexiglas is a good material for glazing flat plate solar collectors as it transmits almost 90% of the received shortwave solar radiation. Types of plastics can also be used as covers as few of them can endure ultraviolet radiation for a long time. Polycarbonate rigid sheet, polycarbonate rigid film and corrugated sheets are plastic products available on the market. The benefit of using plastics is that they cannot be broken by hail of stones and they are flexible and light (Poulikakos, 1994).

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2.2. Thermal Analysis of Solar Air Collector

$$\eta = \frac{Q_u}{Q_s} \tag{1}$$

$$Q_u = \dot{m} C_p \left(T_{out} - T_{in}\right) \tag{2}$$

Where Q_u is the accumulated useful energy extracted from the collector during the working period, which is written as

$$Q_u = \int_0^t \dot{m} C_{pa} \left(T_{out} - T_{in}\right).dt \tag{3}$$

Qs is solar input energy, determined by equation (4):

$$Q_s = \int_0^t (IA).dt \tag{4}$$

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:

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

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

The convective heat transfer coefficient in equations (6) and (7) can be derived from the relationship drawn by Sparrow *et al*. (1979), which is used for calculating the external heat transfer coefficient over the transparent cover and the back plate when ($Re=V_a D_h/\nu_a$)

$$Nu = 0.86\, Re^{1/2}\, Pr^{1/3} \quad , \quad Nu = \frac{h_{ac,out} D_h}{\lambda_a} \quad , \quad D_h = \frac{2WL}{W+L} \quad ,$$

$$Pr = \frac{C_{pa}\mu}{\lambda_a} \tag{8}$$

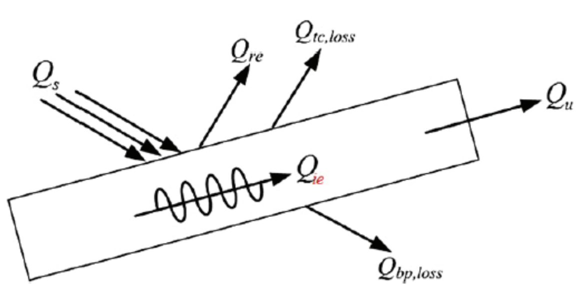

Fig. 1 Energy distributions of the solar air collector.

2.2.1. Heat transfer coefficients

The convective heat transfer coefficient $h_{1ac,\,out}$ for air flowing over the outside surface of the glass cover depends primarily on the wind velocity V_{wind}. McAdams (1954); obtained experimental result as:

$$h_{1ac,out} = 5.7 + 3.8\,V_{wind} \tag{9}$$

where the units of $h_{1ac,out}$ and V_{wind} are 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 (1975).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 flowing air.

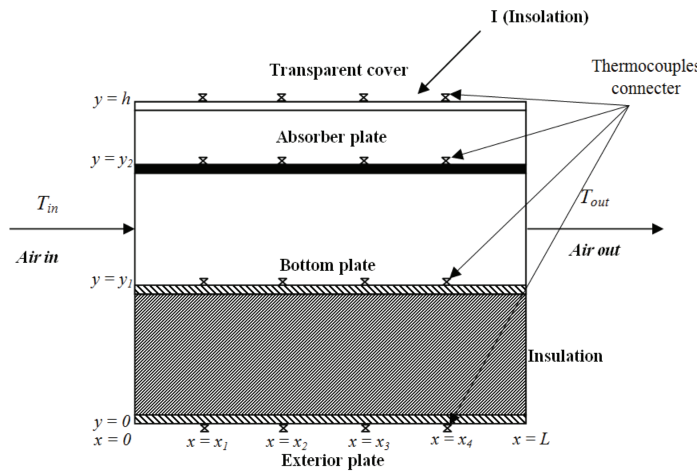

Fig. 2 Flat plate solar air heaters.

The flat-plate solar air heater (Chabane *et al*., 2012a, b, d; Close and Dunkle, 1976; Liu and Sparrow, 1980; Seluck, 1977; Tan, 1970; Whillier, 1963) are considered to be a simple device consisting of one (transparent) covers situated above an absorbing plate with the air flowing under absorber plate (Tan, 1970; Whillier, 1963; see Fig. 2). The conventional flat-plate solar air heater has been investigated for heat-transfer efficiency improvement by introducing forced convection (Duffie and Beckman, 1980; Tonui and Tripanagnostopoulos, 2007), extended heat-transfer area (Gao *et al*., 2007; Mohamad, 1997), and increase of air turbulence (Verma and Prasad, 2000; Yeh, 1992).

3. Description of solar air heater considered in this work

A schematic view of the constructed single flow under an absorber plate and in hollow of semi cylindrical fins which located under an absorber plate system of collector is shown in Fig. 1, the photographs of two different absorber plates of the collectors and the view of the absorber plate in the collector box are shown in Fig. 2, respectively. In this study, two modes of the absorber plates were used. The absorbers were made of galvanized iron sheet with black chrome selective coating. The plate thickness of two collectors was 0.5 mm. The cover window type the Plexiglas of 3 mm thickness was used as glazing. Single transparent cover was used of two collectors. Thermal losses through the collector backs are mainly; due to the conduction across the insulation (thickness 4 cm) (Chabane *et al.,* 2013a), those caused by the wind and the thermal radiation of the insulation are assumed negligible. After installation, the two collectors were left operating several days under normal weather conditions for weathering processes.

Thermocouples were positioned evenly, on the top surface of the absorber plates, at identical positions along the direction of flow, for both collectors. Inlet and outlet air temperature were measured by two well insulated thermocouples. The output from the thermocouples was recorded in degrees Celsius by using a digital thermocouple thermometer DM6802B: measurement range −50 to 1300 °C (−58 to 1999 °F), resolution: 1°C or 1°F, accuracy: ± 2.2 °C or ± 0.75 % of reading and Non-Contact digital infrared thermometer temperature laser gun model number: TM330: accuracy ±1.5 C/±1.5 %, measurement range −50 to 330 °C (−58 to 626 °F) resolution 0.1 °C or 0.1 °F,

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emissivity 0.95. A digital thermometer measured the ambient temperature with sensor in display LCD CCTV-PM0143 placed in a special container behind the collectors' body. The total solar radiation incident on the surface of the collector was measured with a Kipp and Zonen CMP 3 Pyranometer. This meter was placed adjacent to the glazing cover, at the same plane, facing due south. The measured variables were recorded at intervals of 15 min and include: insolation, inlet and outlet temperatures of the working fluid circulating through the collectors, ambient temperature, absorber plate temperatures at several selected locations and air flow rates (Lutron AM-4206M digital anemometer). All tests began at 9 AM and ended at 4 PM.

The layout of the solar air collector studied is shown in Figs. 1, 2. The collector A served as the baseline one, with the parameters as:

-The solar collecting area was 2 m (length) × 1 m (width);

-The installation angle of the collector was 45° from horizontal;

-Height of the stagnant air layer was 0.02 m;

-Thermal insulation board EPS (expanded polystyrene board), with thermal conductivity 0.037 W/(m K), was put on the exterior surfaces of the back, and side plates, with a thickness of 40 mm.

-The absorber was of a plate absorption coefficient α = 0.95, the transparent cover transmittance τ = 0.9 and absorption of the glass covers, $\alpha_g = 0.05$;

-16 positions of thermocouples connected to plates and two thermocouples to outlet and inlet flow, Fig.2.

4. RESULTS AND DISCUSSION

The single pass solar air heaters are investigated experimentally under Biskra prevailing weather conditions during the winter months, 24/01/2012, 25/01/2012, 01/02/2012, 19/02/2012, and 27/02/2012 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 performance of the solar air heater was studied and compared with the performance of a single pass solar air heater and an effect the mass flow rate of the air was varied from 0.0108 to 0.0201 kg/s.

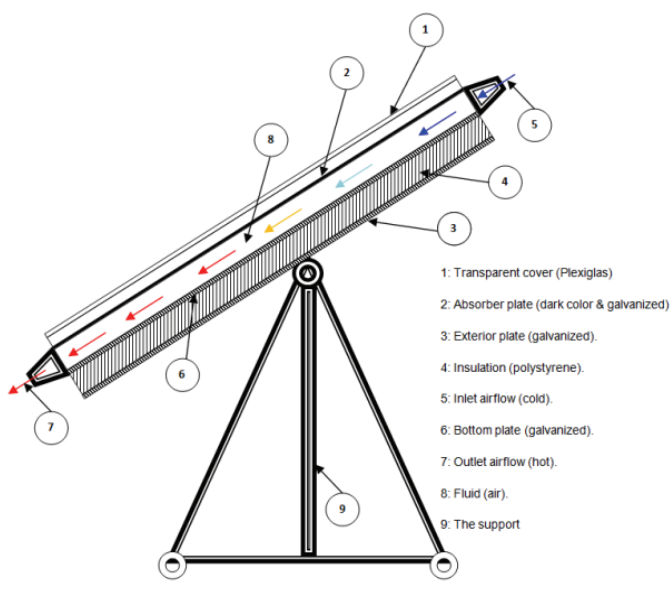

Fig. 3 Schematic view of the solar air collector

Figures 5 and 6 show the variation of the thermal efficiency and a solar intensity, respectively, with air mass flow rate. The thermal efficiency used to evaluate the performance of the solar air heater is calculated; from both figures that the thermal efficiency increases with increasing solar intensity and mass flow rate as a function of the time. The efficiencies of the rate 0.0202 kg/s are higher than inferior of 0.0202 kg/s. Figs. 5 and 6 shows the comparison of the thermal efficiency for the different mass flow rates from 0.0108 kg/s to 0.0202 kg/s. beside the results data of each value has been shown in Table 3a, 3b.

Fig. 4 The photograph of experimental set-up (Chabane *et al.,* 2013e, f)

Figures 5 and 6 show the variation of the thermal efficiency and a solar intensity, respectively, with air mass flow rate. The thermal efficiency used to evaluate the performance of the solar air heater is calculated; from both figures that the thermal efficiency increases with increasing solar intensity and mass flow rate as a function of the time. The efficiencies of the rate 0.0202 kg/s are higher than inferior of 0.0202 kg/s. Figs. 5 and 6 shows the comparison of the thermal efficiency for the different mass flow rates from 0.0108 kg/s to 0.0202 kg/s. beside the results data of each value has been shown in Table 3a, 3b.

Evidently the mean highest thermal efficiency (η = 58.02%) at solar intensity I = 898 $W.m^{-2}$ at air flow rate 0.0161 $kg.s^{-1}$ and 45° tilt angle (Chabane *et al.,* 2012c) at 13:10 h. The mean lowest thermal efficiency (η = 28 %) at solar intensity I = 883 $W.m^{-2}$ at 13:00 h was obtained with air flow rate 0.0108 $kg.s^{-1}$ and 45° tilt angle. Solar air heater were heated the air much more at the lower air rate, because the air had more time to get hot inside the collector.

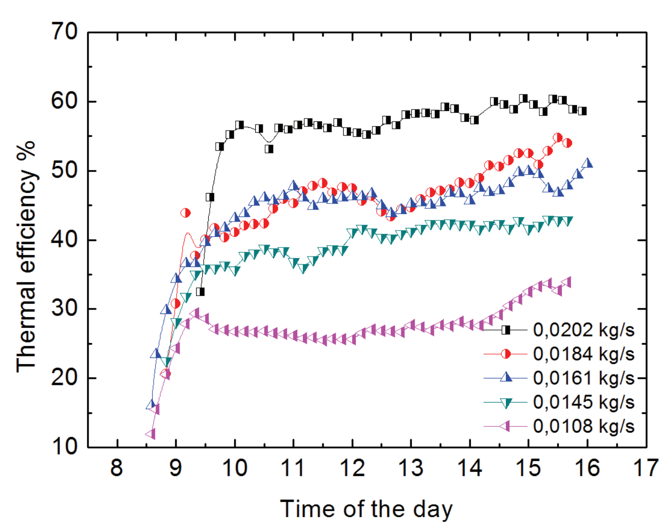

Fig. 5 Variation of collector efficiency at different mass flow rates.

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Efficiency versus time at various air rates for the single pass collector in this experiment is shown in Fig. 5. The efficiencies increase to a maximum value at 12:30–16:00 h, and then start to decrease later on in the afternoon. The efficiency of a mass flow rate m = 0.0202 kg/s is higher than the others mass flow rate by 7–35 % depending on the air mass flow rate. The efficiency of a single air pass solar collector notably depends on the air mass flow rate. The maximum efficiency obtained for these single pass air collector are 63.25 % for m = 0.0202 kg/s. For solar air collectors, it is clear that the efficiency increase with an increasing air mass flow rate of air as shown in Fig. 5. The curvature of the efficiency for mass flow air at 0.0202 kg/s is wider than that of air mass flow rates 0.0108 kg/s. Fig. 8 shows the plots of thermal efficiency of the single solar air heater collectors as a function of mass flow rate m, for the period of 8:30 h to 16:00 h of the day.

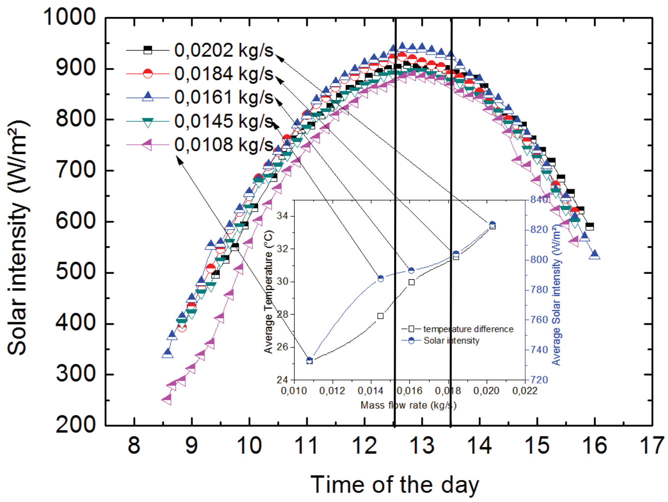

Fig. 6 Variation of solar radiation at different days.

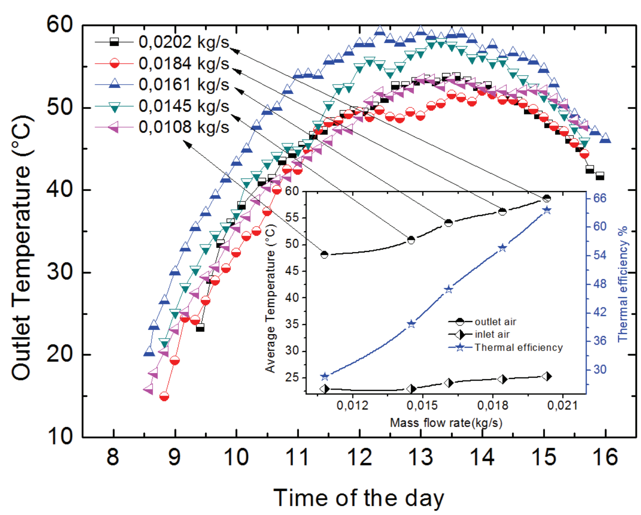

Fig. 7 Variation of outlet temperature at different mass flow rates.

The thermal efficiency of the heater improves with increasing air flow rates due to an enhanced heat transfer to the air flow while a temperature difference of fluid decreases at a constant tilt angle β = 45° (Chabane *et al.,* 2012c). Solar intensity is at their highest values at noon (at about 13:30) as is expected. The solar intensity decreases as the time passes through the afternoon. Fig. 7 it shows overall results of experiments, including the difference of air ambient and outlet temperature and daily instantaneous solar intensity levels. The ambient temperature was between 7 and 24 °C. The inlet temperatures of solar air collectors were measurement to ambient temperature. The temperature differences between the inlet and outlet temperatures can be compared directly when determining the performance of the collectors. The highest daily solar radiation is obtained as 881.38 and 943 W/m^2 for a Flat-plate.

Figure 6 shows the solar intensity versus standard local time of the day for all the days the experiment was carried out. The solar intensity increases from the early hours of day with about 250 W/m^2 at 8:30 h to a peak value at noon and then, reduces later on during the day (Fig. 6). The highest daily solar radiation obtained with single pass solar air collector, which was for day was 940 W/m^2 at 13:40 h and the average solar intensity through that particular day was about 793 W/m^2. Calculating the mean solar intensity for each day, there was stability in the solar radiation as all mean averages are within the same and close range. The mean average solar intensity for all the days of the experiment was 733 W/m^2 and 803.50 W/m^2 for single pass solar air collector. The obtained result shows a stable amount of solar radiation measured for each day of the experiment.

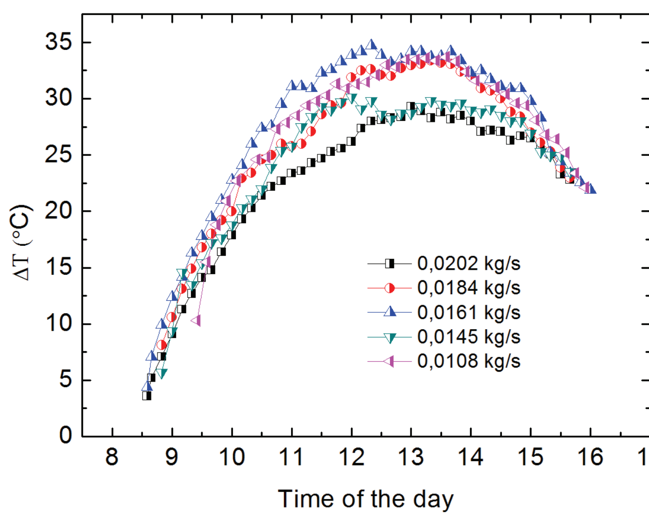

Fig. 8 Temperature difference versus standard local time of the day at different mass flow rates for double pass solar air heater.

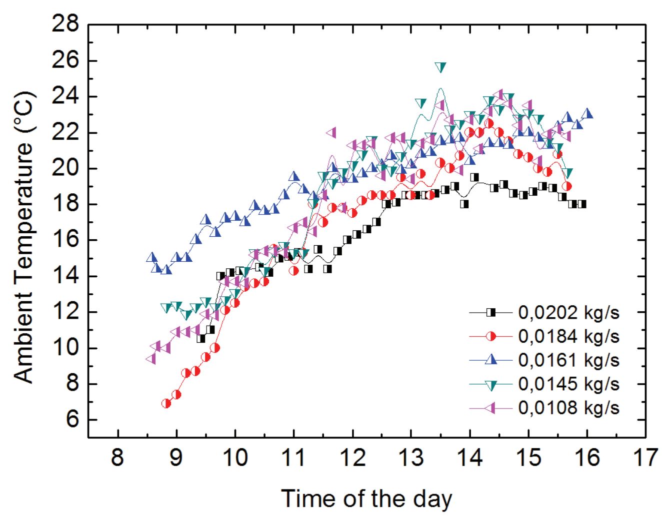

Fig. 9 Variation of ambient temperatures at different mass flow rates.

Figure 8 show the temperature differences, $\Delta T = T_{out} - T_{in}$, versus time of the day for different mass flow rates and for single pass solar air heater. The ambient temperature versus standard local time of the day for all the days the experiment was carried out is presented in Fig. 9. In general, the input temperature was found to be increasing exponential from the morning to evening with little fluctuation during some of the days. For the smooth plate of the solar air heater used in this work, ΔT was found to reduce with increasing air mass flow rate. Results show that for the same mass flow rate, the collector temperature differences increased with increasing solar radiation I, (Fig. 6) as expected. ΔT of air increases to a peak value of 34 °C for the single pass air heater of smooth plate. This peak temperature difference occurred between 12:00 h and 13:00 h for a minimum mass flow rate, m of 0.0108 kg/s. It then decreases as solar radiation drops to lower values later on during the day for the same mass flow rate of air. The changes in the peak value of the temperature difference between 12:00 h and 13:00 h is suitable to changing outdoor conditions like solar intensity (Fig. 6) and the wind speed.

5. CONCLUSION

This study shows that for a single pass solar air heater using smooth plate as absorber plate and we added the different mass flow rates there is a significant increase in the thermal efficiency of the air heater. The efficiency increases when the air mass flow increases from 0.0108 kg/s to 0.0202 kg/s. Also, the temperature difference between the outlet flow and the ambient, ΔT, reduce with an increase in the air mass flow rate. Further, results showed that for the same mass flow rate the collector temperature difference increase with increasing solar radiation, I, and decreases as solar radiation drops to lower values later on during the day. The maximum temperature difference obtained from this study is 34.70 °C for the single pass air heater about smooth plate for air mass flow rate of 0.0161 kg/s. The maximum thermal efficiency obtained is 60.40 % for single pass for air mass flow rate of 0.0202 kg/s.

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NOMENCLATURE

T_{ep} *temperature of exterior plat (°C)*
T_{ab} *temperature of absorber plat (°C)*
T_{pl} *temperature of transparent cover (°C)*
T_{bp} *temperature of bottom plat (°C)*
T_a *ambient temperature (°C)*
x_i *local direction longitudinal of points (m)*
y_i *local direction of thickness panel (m)*
T_{in} *temperature inlet (°C)*
T_{out} *outlet fluid temperature (°C)*
V_{wind} *wind velocity (m/s)*
$h_{ac,out}$ *convection heat transfer coefficient (W/m² K)*
C_p *specific heat of air (J/kg K)*
A_c *surface area of the collector = LW (m^2)*
i *position of thermocouple connected of 1 to 4.*
ΔT *temperature difference (°C)*
m *mass flow rate (kg/s)*
Q_u *: accumulated useful energy (W)*
Q_s *: solar input energy (W)*
$Q_{tc,loss}$ *: accumulated heat losses of the transparent cover (W)*
$Q_{bp,loss}$*: accumulated heat losses of the back plate (W)*

Greek symbols
η *collector efficiency (%)*
I *global irradiance incident on solar air heater collector (W/m2)*
m *air mass flow rate (kg/s)*
ε *emissivity of absorber plate*
α_a *absorber plate absorption coefficient*
τ *transparent cover transmittance*
α_g *absorptivity of the glass covers*

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