Experimental study of heat transfer and thermal performance with longitudinal fins of solar air heater

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ABSTRACT

The thermal performance of a single pass solar air heater with five fins attached was investigated experimentally. Longitudinal fins were used inferior the absorber plate to increase the heat exchange and render the flow fluid in the channel uniform. The effect of mass flow rate of air on the outlet temperature, the heat transfer in the thickness of the solar collector, and the thermal efficiency were studied. Experiments were performed for two air mass flow rates of 0.012 and 0.016 kg s\(^{-1}\). Moreover, the maximum efficiency values obtained for the 0.012 and 0.016 kg s\(^{-1}\) with and without fins were 40.02%, 51.50% and 34.92%, 43.94%, respectively. A comparison of the results of the mass flow rates by solar collector with and without fins shows a substantial enhancement in the thermal efficiency.

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Introduction

Solar air heaters are effective devices to harness solar radiation for space heating and other purposes, and the efficiency of solar air collector can be improved by producing new designs of fins. Because of their simple construction and low cost, solar air collectors are extensively used in the world for heating purposes. In this study, a test of solar air collector was performed based on the heating of air by longitudinal fins (semi-cylindrical form) and the surface area for heat exchange. Our study seeks to increase the thermal efficiency of the solar collector, by using a single pass counter flow solar air collector with longitudinal fins. To this end, a semi-cylindrical form is one of the important and attractive design improvements that has been proposed to improve the thermal performance. This paper presents an experimental analysis of a single pass solar air collector with and without fins.

Comparison of the results reveals that the thermal efficiency of a single pass solar air collector increases with the increase of mass flow rate. Increasing the absorber area or fluid flow heat transfer area will increase the heat transfer to the...
flowing air; on the other hand, it will increase the pressure drop in the collector, thereby increasing the required power consumption to pump the air flow to cross the collector [1,2]. On the other hand, several configurations of absorber plates have been designed to improve the heat transfer coefficient. Artificial roughness obstacles and baffles in various shapes and arrangements were employed to increase the area of the absorber plate. As a result, the heat transfer coefficient between the absorber plate and the air pass is improved [3]. Reports are available on experimental investigation of the thermal performance of a single- and double-pass solar air heater with fins attached and a steel wire mesh as absorber plate [4]. The bed heights were 7 cm and 3 cm for the lower and upper channels, respectively. The result of a single or double solar air heater, when compared with conventional solar air heater, shows much more substantial enhancement in the thermal efficiency.

Few studies were carried out on numerical of the performance and entropy generation of the double-pass flat-plate solar air heater with longitudinal fins [5]. The predictions are done at air mass flow rate ranging between 0.02 and 0.1 kg s⁻¹. Fins serve as heat transfer augmentation features in solar air heaters; however, they increase pressure drop in flow channels. Results show that high efficiency of the optimized fin improves the heat absorption and dissipation potential of a solar air heater [6]. A double flow solar air heater with fins attached over and under the absorbing plate was designed. This resulted in a considerable improvement in collector efficiency of double flow solar air heaters with fins compared to single flowing, operating at the same flow rate [7]. An experimental investigation was carried out on the thermal performance of the offset rectangular plate fin absorber plates with various glazing [8]; in this work, the offset rectangular plate fins, which are used in heat exchangers, are experimentally studied. As the offset rectangular plate fins are mounted in staggered pattern and oriented parallel to the fluid flow, high thermal performances are obtained with low-pressure losses. [9]. A few experiments were carried out to study the performance of three types of solar air heater, namely flat-plate, finned, and V-corrugated solar air heaters. The V-corrugated collector was found to be most efficient, while the flat-plate collector was the least efficient. Another work used the cross-corrugated absorbing plate and bottom plate to enhance the turbulence and the heat transfer rate inside the air flow channel and tested its thermal performance [10,11]. The work title of the studies on a novel solar air collector of pin–fin integrated absorber was designed to increase the thermal efficiency [12]. In the performance analysis of varying flow rate on PZ7-11.25 pin–fin arrays collector, the correlation equation for the heat transfer coefficient is obtained, and the efficiency variation versus air flow rate is determined in this work. Another work compared results to those obtained with a solar air collector without fins using two types of absorbers: selective (in copper sun) and non-selective (black-painted aluminum) [13]. The report presents a solar water heater designed with a local vegetable material as insulating material. The study focuses on the comparative thermal performance of this collector and another collector, identical in design, fabrication, and operating under the same conditions, using glass wool as heat insulation [14]. Some studies reported the effect of the mass flow rate in range 0.0078–0.0166 kg s⁻¹ on the solar collector with longitudinal fins [15,16]. The flat-plate solar air heater [17–21] is considered to be a simple device consisting of one (transparent) cover situated above an absorbing plate with the air flowing under absorber plate [20,21] Fig. 2. The conventional flat-plate solar air heater has been investigated for heat transfer efficiency improvement by introducing forced convection [22,23], extending heat transfer area [24,25] and increasing air turbulence [26,27].

### Experimental

**Thermal analysis and uncertainty**

**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 $V_{\text{wind}}$. McAdams [28] obtained the experimental result as:
\[ h_w = 5.7 + 3.8V_{\text{wind}} \]  
where the units of \( h_w \) and \( V_{\text{wind}} \) are \( \text{W/m}^2 \text{K} \) and \( \text{m/s} \), respectively. An empirical equation for the loss coefficient from the top of the solar collector to the ambient was developed by Klein [29]. The heat transfer coefficient between the absorber plate and the airstream is always low, resulting in low thermal efficiency of the solar air heater. Increasing the area of the absorber plate shape will increase the heat transferred to the following air.

**Collector thermal efficiency**

The efficiency of a solar collector is 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 [30–32]:

\[ \eta = \frac{\text{Solar Energy Collected}}{\text{Total Solar Striking Collector Surface}} = \frac{Q_u}{I_0 \times A_c} \]  

The equation for mass flow rate (\( m \)) is

\[ m = \rho \times Q \]

where \( \rho \) is the density of air, which depends on the air temperature, and \( Q \) is the volume flow rate, which depends on the pressure difference at the orifice, which is measured from the inclined tube manometer and temperature.

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

\[ Q_u = \dot{m} C_p (T_{\text{out}} - T_{\text{in}}) \]  

where \( C_p \) is the specific heat of the air and \( A_c \) is the area of the collector. The fractional uncertainty about the efficiency from Eq. (3) is a function of \( \Delta T, \dot{m}, \) and \( I_0 \), considering \( C_p \) and \( A_c \) as constants.

With \( \dot{m} = V_f \cdot S \)
So, collector thermal efficiency becomes:

$$\eta = n t C_p \frac{(T_{\text{out}} - T_w)}{IA_s} \quad (4)$$

**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 that located under an absorber plate system of collector is shown in Fig. 1, and 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. 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 for two collectors. Thermal losses through the collector backs are mainly due to the conduction across the insulation (thickness 4 cm), and 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 temperatures 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; 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 insulation, 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 and 2. The collector A served as the baseline one, with the following parameters:

- 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 $\alpha = 0.95$, the transparent cover transmittance $\tau = 0.9$ and absorption of the glass covers, $\varepsilon_g = 0.05$.
- 16 positions of thermocouples connected to plates and two thermocouples to outlet and inlet flow.
- Five fins under the absorber plate with a semi-cylindrical longitudinal form was 1.84 m (length) × 0.03 m (Radius); the distance between two adjacent fins and fins $t$ are 120 mm and 5 mm thickness, respectively (Fig. 2).

**Results**

Here, the results of the experimental study on thermal performance of the solar collector with and without fins have been presented. In particular, hollow longitudinal fins for an absorber plate have to be created to in order to increase; heat exchange surface, outlet temperature, and thermal efficiency. It can be seen in Tables 1a and 1b that increases in the mass flow rates affect the temperature of the bottom plate and the temperature of an absorber plate by rates between 4 and 6 °C, for the solar air collector without using fins and with using fins. The efficiency of the type with fins is found to be higher than the type without using fins by rates of 5.1% and 5.83%, respectively; the mass flow rates of 0.012 and 0.016 kg s$^{-1}$; see Tables 3a and 3b and a lesser solar intensity by rates 142 and 157 W m$^{-2}$, respectively; we can recover this loose solar intensity by adding the fins back the absorber plate for moving the heat energy into channel and keep the heat energy on an absorber plate for transport of fluid with the mass flow rates of 0.012 and 0.016 kg s$^{-1}$; see Tables 3a and 3b. The maximum thermal efficiency values obtained were 34.4% and 50.33%, respectively.

**Discussion**

Figs. 3a and 3b shows the average temperature distribution in the thickness of a solar collector and shows the variation of the average temperature corresponding to the transparent cover, absorber plate, bottom plate, and exterior plate. The difference can be seen in Figs. 3a and 3b; at the mass flow rates of 0.012 and 0.016 kg s$^{-1}$, the change in curves is remarkable, and the role of the fins is to allow cooled absorber and ensure a better heat exchange. It can be seen in Tables 1a and 1b that increases in the mass flow rates affect the temperature of the bottom plate and the temperature of an absorber plate by rates between 4 and 6 °C, for the solar air collector without using fins and with using fins. The temperature values of the bottom plate and the absorber plate corresponding to 0.012 and 0.016 kg s$^{-1}$ were ($T_{bp} = 75.02$ and 78.75 °C) and ($T_{ab} = 87.50$ and 93.03 °C) (see Table 1a), and ($T_{ab} = 70.25$ and 74.50 °C) and ($T_{bp} = 91.25$ and 94.02 °C) (see Table 1b), respectively. The collectors are mounted on a galvanized metal frame. In the field, the solar energy passing through the cover glass is absorbed by the absorber plate. The heat generated is then transferred to the collector fluid [33].

Figs. 4a and 4b shows the average temperature of a solar collector in the absence of fins for lengths ranging from 0.388 to 1.552 m, and corresponding to mass flow rates of 0.012 and 0.016 kg s$^{-1}$. The average temperature of the bottom plate in a length of $x_2 = 0.776$ m at $m = 0.012$ and 0.016 kg s$^{-1}$ was ($T_{bp} = 86$ and 84.50 °C), and the average temperature of an absorber plate was ($T_{ab} = 88$ and 89.50 °C); see Table 2a; the average temperature of the bottom...
plate takes more heat from an absorber plate, which means the fluid that is between the bottom plate and an absorber plate takes heat from the absorber plate. The temperature of an absorber plate at the point $x_2$ is decreased, which causes the air flow in channel, and becomes stable for all points.

The difference in average temperatures of both $x_1$ and $x_2$ can be seen in Figs. 4a and 4b, which means $T_{ab}(x_2) < (T_{ab}(x_3)$ and $T_{ab}(x_4)) < T_{ab}(x_1)$; this makes clear that the fluid takes less heat energy for each location in a length of a solar collector except in point $x_2$. Increase in the mass flow rate has effects on the average temperature of an absorber plate and decreases it slightly, except in $x_2$: the average temperature of the bottom plate approaching to the average temperature of an absorber plate is not prospective, because the fluid takes more heat energy from the absorber plate and at the same time the bottom plate takes this energy too, resulting in poor air distribution.

Figs. 5a and 5b shows the average temperature of a solar collector as a function of length, from 0.388 to 1.552 m, corresponding to modes with using fins at mass flow rates of 0.012 and 0.016 kg s$^{-1}$. As can be seen, the evolution of the curves

| Table 1a | Experimental data of average temperature for flat-plate, corresponding to the mass flow rate 0.012 and 0.016 kg s$^{-1}$, on 24 and 25/01/2012, for solar collector thickness from 0 to 0.1 m with tilt angle $\beta = 45^\circ$. |
|---|---|---|---|---|---|---|
| Mode of solar collector | $y$ (m) | Average temperature (°C) |
| $m$ (kg s$^{-1}$) | 0.012 | 0.016 |
| Flat-plate | | |
| 0 | 34.80 | 51.73 |
| 0.02 | 87.50 | 93.03 |
| 0.06 | 75.02 | 78.75 |
| 0.10 | 21.10 | 22.95 |

| Table 1b | Experimental data of average temperature for with using fins corresponding to the mass flow rate 0.012 and 0.016 kg s$^{-1}$, on 13 and 15/05/2012, for solar collector thickness from 0 to 0.1 m with tilt angle $\beta = 45^\circ$. |
|---|---|---|---|---|---|---|
| Mode of solar collector | $y$ (m) | Average temperature (°C) |
| $m$ (kg s$^{-1}$) | 0.012 | 0.016 |
| Longitudinal fins ($n = 5$) | | |
| 0 | 51.28 | 53.25 |
| 0.02 | 91.25 | 94.02 |
| 0.06 | 70.25 | 74.50 |
| 0.10 | 35.13 | 36.28 |

| Table 2a | Experimental data for flat-plate corresponding to the mass flow rate 0.012 and 0.016 kg s$^{-1}$; on 24 and 25/01/2012 for length of solar collector from 0.388 to 1.552 m. |
|---|---|---|---|---|---|---|
| Mode of solar collector | $m$ (kg s$^{-1}$) | $x$ (m) | $T_{pl}$ (°C) | $T_{ab}$ (°C) | $T_{bp}$ (°C) | $T_{ep}$ (°C) |
| Flat-plate | 0.012 | 0.388 | 34.05 | 87.00 | 67.50 | 22.05 |
| 0.776 | 34.05 | 88.00 | 96.00 | 21.65 |
| 1.164 | 35.55 | 87.00 | 79.00 | 20.55 |
| 1.552 | 36.20 | 95.00 | 71.00 | 20.25 |
| 0.016 | 0.388 | 38.85 | 88.00 | 60.00 | 21.65 |
| 0.776 | 37.85 | 89.50 | 74.50 | 21.10 |
| 1.164 | 41.55 | 86.50 | 76.50 | 21.00 |
| 1.552 | 42.70 | 99.00 | 69.00 | 20.40 |

| Table 2b | Experimental data for solar collector with using fins corresponding to the mass flow rate 0.012 and 0.016 kg s$^{-1}$; on 13 and 15/05/2012, for length of solar collector from 0.388 to 1.552 m. |
|---|---|---|---|---|---|---|
| Mode of solar collector | $m$ (kg s$^{-1}$) | $x$ (m) | $T_{pl}$ (°C) | $T_{ab}$ (°C) | $T_{bp}$ (°C) | $T_{ep}$ (°C) |
| Longitudinal fins ($n = 5$) | 0.012 | 0.388 | 47.75 | 83.00 | 53.00 | 35.50 |
| 0.776 | 54.20 | 88.00 | 66.50 | 35.20 |
| 1.164 | 58.35 | 94.50 | 74.00 | 35.00 |
| 1.552 | 60.05 | 94.50 | 78.00 | 35.05 |
| 0.016 | 0.388 | 46.55 | 84.00 | 59.00 | 34.85 |
| 0.776 | 51.00 | 87.50 | 71.00 | 34.45 |
| 1.164 | 55.50 | 92.50 | 78.00 | 33.95 |
| 1.552 | 60.25 | 94.50 | 81.50 | 34.10 |
takes a regular form and the temperature values of the absorber plate and the bottom plate automatically increase in a regular fashion: ($T_{ap} = 88$ and $78.50$ °C) and ($T_{bp} = 66.50$ and $71$ °C) at $x_2 = 0.776$ m, see Table 2b. It can be explicated that the fluid takes more heat energy from the absorber plate and the bottom plate, which is working as another surface of heat exchange with fluid from first point to finally as a function to length of the solar collector; and when there is an increase in the mass flow rate, the temperature of the bottom plate decreases, means that the process of bringing the fluid takes more heat from the bottom plate and cooling it. It should be pointed out that for curves corresponding to mass flow rates, the average temperature $T_{ap}(x_1, m) < T_{ap}(x_2, m) < T_{ap}(x_3, m) < - T_{ap}(x_4, m)$. A bottom plate helps us in steady the temperature of fluid to kept, means work as the storage or alimentation the air by heat energy.

The reason for the difference between the Figs. 4a and 4b and Figs. 5a and 5b was supplemented to add the fins back an absorber plate to solar collector, for the best thermal performance of solar air heater; the air is distribution very well and takes more heat energy from the bottom plate and the absorber plate.

Using fins with the absorber plate, the values of temperature vary (increase), because fins obtain more heat due to an

| Table 3a | Experimental data for flat-plate, corresponding to the mass flow rate 0.012 and 0.016 kg s$^{-1}$ on 24 and 25/01/2012, according to the time of day, between 9:00 and 16:00, with tilt angle $\beta = 45^\circ$. |
| Flat-plate | |
| $t$ (h) | $T_m$ (°C) | $T_{ap}$ (°C) | $T_{bp}$ (°C) | $I$ (W m$^{-2}$) | $\eta$ (%) | $\Delta T$ (°C) | $h_w$ (W K$^{-1}$ m$^{-2}$) | |
| $m = 0.012$ kg s$^{-1}$ | |
| 9:00 | 14.60 | 25.20 | 12.40 | 421 | 22.94 | 10.60 | 09.12 | |
| 10:00 | 17.20 | 37.20 | 13.10 | 627 | 29.10 | 20.00 | 14.82 | |
| 11:00 | 19.10 | 44.90 | 15.30 | 783 | 33.52 | 31.90 | 06.46 | |
| 12:00 | 23.10 | 56.80 | 21.40 | 895 | 33.59 | 33.00 | 09.88 | |
| 13:00 | 23.80 | 56.30 | 23.00 | 847 | 34.42 | 32.00 | 09.88 | |
| 14:00 | 24.30 | 51.40 | 19.80 | 485 | 34.92 | 23.00 | 15.96 | |
| 15:00 | 23.00 | 46.00 | 19.00 | 485 | 34.92 | 23.00 | 15.96 | |
| $m = 0.016$ kg s$^{-1}$ | |
| 9:00 | 14.90 | 19.30 | 07.40 | 433 | 25.09 | 09.40 | 05.70 | |
| 10:00 | 13.60 | 19.30 | 12.50 | 649 | 33.48 | 18.80 | 07.98 | |
| 11:00 | 16.60 | 24.20 | 14.30 | 809 | 36.85 | 25.80 | 08.74 | |
| 12:00 | 19.40 | 49.50 | 17.50 | 900 | 38.65 | 30.10 | 08.74 | |
| 13:00 | 20.30 | 49.01 | 18.50 | 912 | 36.37 | 28.70 | 09.50 | |
| 14:00 | 23.01 | 52.02 | 22.00 | 854 | 39.24 | 29.00 | 09.50 | |
| 15:00 | 21.90 | 48.80 | 20.60 | 727 | 42.76 | 26.90 | 07.98 | |
| 16:00 | 20.90 | 44.40 | 19.00 | 618 | 43.94 | 23.50 | 10.26 | |

| Table 3b | Experimental data for solar collector with using fins, corresponding to the mass flow rate 0.012 and 0.016 kg s$^{-1}$ on 13 and 15/05/2012, according to the time of day, between 9:00 and 16:00, with tilt angle $\beta = 45^\circ$. |
| Longitudinal fins $n = 5$ | |
| $t$ (h) | $T_m$ (°C) | $T_{ap}$ (°C) | $T_{bp}$ (°C) | $I$ (W m$^{-2}$) | $\eta$ (%) | $\Delta T$ (°C) | $h_w$ (W K$^{-1}$ m$^{-2}$) | |
| $m = 0.012$ kg s$^{-1}$ | |
| 9:00 | 30.20 | 43.10 | 25.00 | 417 | 27.47 | 12.90 | 07.22 | |
| 10:00 | 33.10 | 53.70 | 29.40 | 570 | 32.09 | 20.60 | 10.64 | |
| 11:00 | 35.00 | 63.70 | 30.50 | 675 | 37.75 | 28.70 | 08.74 | |
| 12:00 | 37.00 | 67.50 | 32.50 | 740 | 36.60 | 30.50 | 14.06 | |
| 13:00 | 38.50 | 70.10 | 30.60 | 753 | 37.26 | 31.60 | 11.40 | |
| 14:00 | 39.20 | 69.20 | 34.50 | 684 | 38.94 | 30.00 | 10.64 | |
| 15:00 | 39.10 | 66.70 | 31.80 | 617 | 39.72 | 27.60 | 10.26 | |
| 16:00 | 38.50 | 64.70 | 30.80 | 580 | 40.02 | 26.20 | 08.74 | |
| $m = 0.016$ kg s$^{-1}$ | |
| 9:00 | 30.30 | 41.10 | 26.30 | 437 | 29.54 | 10.80 | 15.20 | |
| 10:00 | 31.80 | 53.30 | 28.60 | 566 | 45.40 | 21.50 | 13.30 | |
| 11:00 | 34.70 | 61.80 | 29.40 | 683 | 47.42 | 27.10 | 09.50 | |
| 12:00 | 36.60 | 65.90 | 31.80 | 744 | 47.07 | 29.30 | 08.74 | |
| 13:00 | 38.00 | 68.10 | 33.40 | 755 | 47.65 | 30.10 | 10.64 | |
| 14:00 | 38.10 | 67.30 | 34.20 | 701 | 49.77 | 29.20 | 08.74 | |
| 15:00 | 39.20 | 64.00 | 36.50 | 589 | 50.33 | 24.80 | 09.12 | |
| 16:00 | 38.00 | 63.00 | 35.50 | 480 | 51.50 | 25.00 | 09.52 | |
**Fig. 3a** Average temperature in the thickness of a solar collector versus the whole area of the solar collector plates for a single pass solar air heater, with flow rates of 0.012 and 0.016 kg s\(^{-1}\), for the solar collectors without using fins.

**Fig. 3b** Average temperature in the thickness of a solar collector versus the whole area of the solar collector plates for a single pass solar air heater, with flow rates of 0.012 and 0.016 kg s\(^{-1}\), for the solar collectors with using fins.

**Fig. 4a** Average temperature along the length of solar collectors versus thickness of panel of between 0 and 0.1 m for single pass solar air heater, at flow rates of 0.012 kg s\(^{-1}\), corresponding to the flat-plate solar collector.

**Fig. 4b** Average temperature along the length of solar collectors versus thickness of panel of between 0 and 0.1 m for single pass solar air heater, at flow rates of 0.016 kg s\(^{-1}\), corresponding to the flat-plate solar collector.

**Fig. 5a** Average temperature along the length of solar collectors versus thickness of panel of between 0 and 0.1 m for single pass solar air heater, at flow rates of 0.012 kg s\(^{-1}\), corresponding to solar collectors with using fins.

**Fig. 5b** Average temperature along the length of solar collectors versus thickness of panel of between 0 and 0.1 m for single pass solar air heater, at flow rates of 0.016 kg s\(^{-1}\), corresponding to solar collectors with using fins.
Fig. 6a  Solar intensity and thermal efficiency versus time of day for a single pass solar air heater, with flow rates at 0.012 kg s\(^{-1}\), corresponding to solar collectors without using fins.

Fig. 6b  Solar intensity and thermal efficiency versus time of day for a single pass solar air heater, with flow rates at 0.016 kg s\(^{-1}\), corresponding to solar collectors with using fins.

Fig. 7a  Solar intensity and thermal efficiency versus time of day for a single pass solar air heater, with flow rates at 0.012 kg s\(^{-1}\), corresponding to solar collectors with using fins.

Fig. 7b  Solar intensity and thermal efficiency versus time of day for a single pass solar air heater, with flow rates at 0.016 kg s\(^{-1}\), corresponding to solar collectors without using fins.

Fig. 8a  Temperature versus different standard local time during days for single pass solar air heater, of the flow rate at 0.012 kg s\(^{-1}\), corresponding to the outlet, inlet, and ambient temperature of a solar collector without using fins.

Fig. 8b  Temperature versus different standard local time during days for single pass solar air heater, of the flow rate at 0.016 kg s\(^{-1}\), corresponding to the outlet, inlet, and ambient temperature of a solar collector without using fins.
increase in heating time through circulating the air inside, and a transparent cover helps to decrease convection heat losses. In the presence of fins, this exchange is effective along the entire length of the channel.

Figs. 6a, 6b, 7a and 7b show the variation of the thermal efficiency and a solar intensity with air mass flow rate. The thermal efficiency used to evaluate the performance of the solar air heater is calculated; from both figures, it can be said that the thermal efficiency increases with increasing solar intensity and mass flow rate as a function of time. The efficiencies of the finned collectors are higher than those of the collector without using fins. Figs. 6a, 6b, 7a and 7b show the comparison of the thermal efficiency for two different mass flow rates between solar collector with and without using fins. Beside the results, data of each solar air heater have been shown in Tables 3a and 3b.

Evidently, the mean highest thermal efficiency ($\eta = 51.50\%$) at solar intensity $I = 480 \text{ W m}^{-2}$ by type with fins was obtained at an air flow rate of $0.016 \text{ kg s}^{-1}$ and $45^\circ$ tilt angle at 16:00 h.

The performance curves of two modes of the solar air collectors tested for this study are shown in Figs. 6a, 6b, 7a and 7b, based on the performance curves at the tilt angle of $45^\circ$ [34]. The thermal efficiency of the solar air collector with fins was higher than the one without using fins, corresponding to tow air flow rates. The solar collector of flat-plate had a higher solar intensity than the type with using fins dependent on the air flow rates $0.012$ and $0.016 \text{ kg s}^{-1}$. It can be seen that the lowest solar intensity conversely can be the highest thermal efficiency, and this helps to add fins back the absorber plate. Solar air heater heated the air much more at the lower air rate, because the air had more time to get hot inside the collector.

Figs. 8a, 8b, 9a and 9b show the variation of the ambient outlet and inlet temperatures as a function of air mass flow rates and time during day (please refer to Tables 3a and 3b). The temperature was measured experimentally, and it can be seen from Figs. 8a, 8b and 9a, 9b that the curves of outlet temperature tend to increase with decreasing air mass flow rate. For a specific air mass flow rate at a constant ambient temperature, the outlet and inlet temperatures increase with increasing solar intensity. Again, it can be clearly explained that the longitudinal fins came back to an absorber plate; it helps for increasing the outlet air temperature. In general, the inlet temperature was found to be increasing exponentially from the morning for mass flow rates $m = 0.012$ and $0.016 \text{ kg s}^{-1}$. In particular, $T_{\text{in}} = 30.2$ and $30.3 \text{ C}$ at 9:00 h, for ambient temperatures $T_{\text{amb}} = 25$ and $26.3 \text{ C}$, respectively.

The thermal efficiency of the heater improves with increasing air flow rates due to an enhanced heat transfer to the air flow, and the temperature difference decreases at a constant tilt angle of $45^\circ$. Solar intensity is at their highest values at noon about 13:30 as is expected. The solar intensity decreases as the time passes through the afternoon. Figs. 6a, 6b, 7a and 7b shows overall results of experiments, including the difference of air inlet and outlet temperature and daily instantaneous solar intensity levels. The ambient temperature was between 20 and 33.4 °C. The inlet temperatures to the two types of solar air collectors were measurement to ambient temperature. The temperature differences between the inlet and the outlet temperatures can be compared directly when determining the performance of the collectors. The highest daily solar radiation is obtained as 895 and 900 W m$^{-2}$ for a flat-plate and 753 and 755 W m$^{-2}$ at solar collector with fins. As expected, it increases during the morning to some peak value and starts to decrease in the afternoon for all the days in which experiments were conducted.

Conclusions

The present study aims to review designs and analyze a thermal efficiency of solar air heater. This experimental study compared a solar collector without using fins and with using fins attached back the absorber plate. The efficiency of the solar air collectors depends significantly on the solar radiation, mass flow rate, and surface geometry of the collectors and with using fins back the absorber plate. The efficiency of the collector improves with increasing solar intensity at mass flow rate of $0.012$ and $0.016 \text{ kg s}^{-1}$, due to enhanced heat transfer to the air flow. The efficiency of the solar air collector is proven to be higher. The highest collector efficiency and air temperature rise were achieved by the finned collector with a tilt angle of $45^\circ$, whereas the lowest values were obtained from the collector without using fins.
Optimum values of air mass flow rates are suggested to maximize the performance of the solar collector. The reason for the significant increase in efficiency from 0.012 to 0.016 kg s$^{-1}$ can be attributed to changes in flow condition from laminar to turbulent. It could also be seen that slope of the efficiency curves decreases, meaning decrease in loss coefficient, with increase in mass flow rates. Experimental results show better agreement when the inlet temperature is close to the ambient temperature. The following conclusions can be derived:

– The efficiency of the solar air collectors depends significantly on the solar radiation and surface geometry of the collectors.

– The efficiency increases as the mass flow rate increases from 0.012 to 0.016 kg s$^{-1}$.

– The efficiency of solar air collector is proven to be higher. The highest collector efficiency and air temperature rise were achieved by the finned collector with angle of 45°, whereas the lowest values were obtained from the collector without fins.

– The values of thermal efficiency at the mass flow rate of 0.012 and 0.016 kg s$^{-1}$ with and without using fins varied from 40.02% to 51.50% and from 34.92% to 43.94%, respectively.

Conflict of interest

The authors have declared no conflict of interest.

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References

[1] Akpinar EK, Koçyigit F. Experimental investigation of thermal performance of solar air heater having different obstacles on absorber plates. Int Commun Heat Mass Transfer 2010;37(4):416–21.

[2] Karsli S. Performance analysis of new-design solar air collectors for drying applications. Renew Energy 2007;32(10):1645–60.

[3] Romdhane BS. The air solar collectors: comparative study, introduction of baffles to favor the heat transfer. Solar Energy 2007;81(1):139–49.

[4] Omojaro AP, Aldabbagh LBY. Experimental performance of single and double pass solar air heater with fins and steel wire mesh as absorber. Appl Energy 2010;87(12):3759–65.

[5] Paisarn N. On the performance and entropy generation of the double-pass solar air heater with longitudinal fins. Renew Energy 2005;30(9):1345–57.

[6] Nwachukwu PN. Employing exergy-optimized pin fins in the design of an absorber in a solar air heater. Energy 2010;35(2):571–5.

[7] El-Sebaii AA, Aboul-Enein S, Ramadan MRI, Shalaby SM, Moharram BM. Thermal performance investigation of double-pass finned plate solar air heater. Appl Energy 2011;88(5):1727–39.

[8] Hachemi A. Experimental study of thermal performance of offset rectangular plate fin absorber-plates. Renew Energy 1999;17(3):371–84.

[9] Karim MA, Hawlder MNA. Development of solar air collectors for drying applications. Energy Convers Manage 2004;45(3):329–44.

[10] Lin W, Gao W, Liu T. A parametric study on the thermal performance of cross-corrugated solar air collectors. Appl Therm Eng 2006;26(10):1043–53.

[11] Gao W, Lin W, Liu T, Xiu C. Analytical and experimental studies on the thermal performance of cross-corrugated and flat-plate solar air heaters. Appl Energy 2007;84(4):425–41.

[12] Donggen P, Xiaosong Z, Hua D, Kun L. Performance study of a novel solar air collector. Appl Therm Eng 2010;30(16):2594–601.

[13] Moumni N, Youcef-Ali S, Moumni A, Desmons JY. Energy analysis of a solar air collector with rows of fins. Renew Energy 2004;29(13):2053–64.

[14] Andoh HY, Gibaha P, Koua BK, Kofli PME, Touré S. Thermal performance study of a solar collector using a natural vegetable fiber, coconut coir, as heat insulation. Energy Sustain Dev 2010;14(4):297–301.

[15] Chabane F, Moumni N, Benramache S, Tolba AS. Experimental study of heat transfer and an effect the tilt angle with variation of the mass flow rate on the solar air heater. Int J Sci Eng Invent 2012;1(9):61–5.

[16] Chabane F, Moumni N, Benramache S. Experimental performance of solar air heater with internal fins inferior an absorber plate: in the region of Biskra. Int J Energy Technol 2012;4(33):1–6.

[17] Close DJ, Dunkle RV. Behaviour of adsorbent energy storage beds. Solar Energy 1976;18(4):287–92.

[18] Liu CH, Sparrow EM. Convective-radiative interaction a parallel plate channel-application to air-operated solar collectors. Int J Heat Mass Transfer 1980;23(8):1137–46.

[19] Seluck MK, Sayigh AAM. Solar air heaters and their application. New York: Academic Press; 1977.

[20] Tan HM, Charters WWS. Experimental investigation of forced-convective heat transfer for fully developed turbulent flow in a rectangular duct with asymmetric heating. Solar Energy 1970;13(1):121–5.

[21] Whillier A. Plastic covers for solar collectors. Solar Energy 1963;7(3):148–51.

[22] Duffie JA, Beckman WA. Solar engineering of thermal processes. 3rd ed. New York: Wiley; 1980.

[23] Tonui JK, Tripanagnostopoulos Y. Improved PV/T solar collectors with heat extraction by forced or natural air circulation. Renew Energy 2007;32(4):623–37.

[24] Gao W, Lin W, Liu T, Xiu C. Analytical and experimental studies on the thermal performance of cross-corrugated and flat-plate solar air heaters. Appl Energy 2007;84(4):425–41.

[25] Mohamad AA. High efficiency solar air heater. Solar Energy 1997;60(2):71–6.

[26] Verma SK, Prasad BN. Investigation for the optimal thermohydraulic performance of artificially roughened solar air heaters. Renew Energy 2000;20(1):19–36.

[27] Yeh HM. Theory of baffled solar air heaters. Energy 1992;17(7):697–702.

[28] McAdams WH. Heat transmission. McGraw-Hill; 1954.

[29] Klein SA. Calculation of flat-plate loss coefficients. Solar Energy 1975;17(1):79–80.

[30] Karsli S. Performance analysis of new-design solar air collectors for drying applications. Renew Energy 2007;32(10):1645–60.

[31] Kurtbas I, Durmus A. Efficiency and exergy analysis of a new solar air heater. Renew Energy 2004;29(9):1489–501.

[32] Esen H. Experimental energy and exergy analysis of a double-flow solar air heater having different obstacles on absorber plates. Build Environ 2008;43(6):1046–54.

[33] Azad E. Design installation and operation of a solar thermal public bath in eastern Iran. Energy Sustain Dev 2012;16(1):68–73.

[34] Chabane F, Moumni N, Benramache S. Effect of the tilt angle of natural convection in a solar collector with internal longitudinal fins. Int J Sci Eng Invent 2012;1(7):13–7.