Heat and Mass Transfer (2020) 56:1833–1843
https://doi.org/10.1007/s00231-020-02815-y

ORIGINAL

Comparative study on the performances of solar air collectors with trapezoidal corrugated and flat absorber plates

Selçuk Darici[1] · Anil Kilic[1]

Received: 11 April 2019 / Accepted: 6 January 2020 / Published online: 24 January 2020

Abstract

Thermal performance of the solar air collectors which are mostly used for space heating and drying is generally low. Therefore there are different studies aimed at increasing the thermal performance of the solar air collectors. One of the technics used for this purpose is making changes in surface geometry of the absorber plate. In this research, the thermal performance of two solar air collectors constructed with trapezoidal corrugated and flat absorber plate is investigated experimentally under weather conditions of Konya/Turkey. The experiments were conducted for three different air mass flow rates of 0.022, 0.033 and 0.044 kg/s. The results obtained are compared to the ones of solar air collector with flat absorber plate. It has been observed that difference between inlet and outlet air temperatures of the solar air collectors increases as the mass flow rate decreases. For the air mass flow rate of 0.022 kg/s, the maximum temperature rise in solar air collector with trapezoidal corrugated plate is 9 °C compared to the flat plate solar air collector. It has been shown that thermal performance of the solar air collectors rises with the increase in mass flow rates. It is determined that the average daily thermal efficiency of the solar air collector with trapezoidal absorber plate is 63% for 0.044 kg/s.

Nomenclature

A_c	Duct cross-section area (m^2)
A_p	Surface area of the absorber plate (m^2)
C_p	Specific heat capacity of air (J/kgK)
D_h	Hydraulic diameter of the duct (m)
E_R	Relative error (%)
f	The average friction factor
h	The mean convective heat transfer coefficient (W/m^2K)
H	Height of the duct (m)
I	Global solar radiation on the glass cover (W/m^2)
k	Thermal conductivity of air (W/mK)
$\dot{m}$	Mass flow rate (kg/s)
Nu	The mean Nusselt number
P	Wetted perimeter of duct (m)
$\dot{Q}_d$	Daily total useful heat rate (W)
$\dot{Q}_u$	Useful heat rate (W)
Pr	Prandtl number
R	Any parameter
Re	Reynolds number
T_a	Ambient temperature (°C)
T_b	The average bulk temperature of air (°C)
T_i	Average inlet temperature of air (°C)
T_o	Average outlet temperature of air (°C)
T_p	Average surface temperature of the absorber plate (°C)
ΔT	Temperature difference of air ($=T_o - T_i$) °C
V	Average velocity of air in the duct (m/s)
W	Width of the duct (m)
w_R	Uncertainty for R value

Greek symbols

ρ	Density of air (kg/m^3)
μ	Dynamic viscosity of air (kg/ms)
η	Thermal efficiency
η_d	Daily average efficiency
w	Uncertainty

Highlights
• Two various types of solar air collectors with different absorber plate were investigated experimentally.
• The comparisons including temperature differences of air across the solar air collector and thermal efficiencies have been made.
• The thermal efficiency of the solar air collector with trapezoidal corrugated absorber plate is found to be higher.
• The thermal efficiency of the collectors increases with increase in mass flow rate.

✉ Selçuk Darici
sdarici@ktun.edu.tr

[1] Faculty of Engineering and Natural Sciences, Department of Mechanical Engineering, Konya Technical University, 42250 Konya, Turkey

1 Introduction

In Turkey solar energy is used mostly in hot water production systems. There hasn't been enough progress in terms of technology and application of solar collectors with air fluid compared with solar collectors having liquid fluid. Air heated in solar air collectors is used for an efficient drying in traditional convection type dryers under controlled conditions, as well as in space and greenhouse heating or paint spraying implementations. In addition to this, there has been an increase in use of the solar air collectors in ventilation applications for pre-heating of fresh air.

Light materials can be used in the design process of solar air collectors since they are not pressurized. Moreover they draw much attention on them in terms of being more inexpensive, having longer lifetime with easy installation and low operation costs. Also, they don't have any freezing and corrosion risks, and any air leakage problems. However, because of the air is weak compared with water in terms of thermodynamic properties, the efficiency values in solar air collectors are low. Different technics are being developed and implemented in order to decrease the heat loss into the environment and increase the heat transfer from absorber plate to the air. The most preferred ones can be mentioned as; increasing the number of transparent covers, making use of padding material, different fluid arrangements, adding wings to absorber plate and making use of absorber plates with different types of surfaces. The general purpose in these efforts is to increase the coefficient of heat transfer between the plate and air.

Priyam and Chand [1] have investigated the effects of mass flow rate, distance between fins and solar radiation on air temperature and thermal efficiency of wavy finned solar air heater experimentally under Jampshedbur weather conditions. The maximum efficiency was found to be 69.5% for mass flow rate of 0.0158 kg/s and distance between fins of 2 cm. Kumar et al. [2] have made an experimental study on defining the heat transfer and friction characteristics of a solar air heater roughened up artificially and having different w-formed protrusions on one side of the absorber plate. Saini and Verma [3] have studied experimentally in order to identify the effects of roughness and operating parameters of a solar air collector in a dimple formed roughened geometry channel on heat transfer and friction. They developed some correlations for Nusselt number and friction factor by using the data collected from the experiments. Ligrani et al. [4], by using flow visualization technique, presented the flow characteristics for a channel of which one wall with dimple surface and the opposite wall with and without protrusions. As a result local Nusselt numbers and local friction factors of the channel with protrusions on the upper wall have been found to be higher compared to the channel without any protrusions on the upper wall.

Chukin et al. [5] have investigated the effects of the converging and diverging channel geometry on the heat transfer for the flow on one hemisphere. Yadav et al. [6] have performed an experimental study on protrusions arranged in an angular form and roughened geometry. They have observed that heat transfer and friction factor increased respectively 2.89 and 2.93-fold compared to the smooth channel. Ravi and Saini [7] have made a study to investigate the effects of roughness parameters of a channel with discrete multi V formed and gradual ribs double pass on the thermo hydraulic performance. They concluded that roughness geometry used at the two sides of the plate in the double pass mode increased both the friction losses and the heat dissipation rate.

Khadraoui et al. [8] have made an experimental study by using the thermal heat storage in order to increase the efficiency of a simple fabricated solar air heater. In the study the temperature of the air outgoing from the collector during the night has increased when PCM was used. Daily thermal efficiency of the air heater reached 17% without PCM, while it reached 33% with PCM. Sridhar and Reddy [9] have aimed at reducing the heat loss occurring at night time to the possible minimum by using transparent isolation material in solar air collectors. Analyses have been made for different thicknesses of isolation material and for different collector tilt angles. Hatami and Bahadorinejad [10] have experimentally analyzed the heat transfer with natural convection at a solar air heater having a vertical flat absorber plate which is covered with one and two glass surface of 1 m width and 2.5 m height. The maximum thermal efficiency is achieved when the air heater has two glass covers on it.

Moummi et al. [11] have performed some experiments using solar air collectors having rectangular finned plate placed vertically to the flow. Outcomes of experiments performed with or without wings have been compared. In the experiments two absorber plate types namely 'selective (copper)' and 'non-selective (aluminum with black colored)' have been used. For north eastern India climate conditions effects of the different parameters on the performance of a solar air collector have been studied experimentally by Debnath et al. [12]. Parameters employed in the experiments are collector tilt angle, number of glass cover, air mass flow rate and structure of absorber plate surface. The total efficiency increase has been discovered to be within the range of 10.35% - 17.42% with the increase in number of glass cover and mass flow rate. By the use of corrugated absorber plate energy efficiency increased 14%. Sebaii et al. [13] have investigated solar air collectors with double pass flat and v-corrugated theoretically and experimentally. Effects of the air mass flow rate on pressure drop, thermal and thermo hydraulic efficiency of the solar air heaters have been studied. Thermal efficiency of the double pass v-corrugated solar air heater is found to be 11% to 14% higher than flat plate solar air heater.

Aldabbagh et al. [14] have used wire mesh as the fluid bed in single and double pass solar air collector and by this way they tested the performance of the collector experimentally. They discovered that with using wire mesh thermal efficiency has significantly increased compared to the solar air collectors having conventional flat absorber plate. Karim and Hawlader [15] have performed experimental investigation of three types of solar air collectors which are respectively having 'flat', 'with finned' and 'v-corrugated' absorber plates in order to design a solar collector suitable for solar drying. The collector with v-corrugated plates has been found to have the highest efficiency while flat plate collector has the lowest.

Ozgen et al. [16] have made an experimental analysis on a flat absorber plate double pass solar heater on and under which aluminum cans are placed. Within the study three different types of absorber plate have been tested and the highest efficiency has been achieved for the absorber plate with the aluminum cans placed staggered. Tyagi et al. [17] have investigated experimentally the energy and exergy analysis of two typical solar air heaters one with and the other without transient thermal energy storage material. Higher efficiency values have been achieved when the heat storage fluid have been employed. Bashria et al. [18] have developed a mathematical simulation in order to calculate the performances of solar air collectors with single or double glass cover, containing or not porous medium, with v-corrugated different forms of absorber plates. Both single and double flow types have been tested and as a result efficiency for double flow has been found to be 4%–5% higher than for single flow.

Dabra and Yadav [19] have designed a solar collector with coaxial glass tube in order to produce hot air and developed a mathematical model based on energy conservation equations for small control volumes along the axial direction of glass tube. In their study they investigated the effects of thirteen different parameters on the increase in the temperature of air outlet the collector. Aidinlou and Nikbakht [20] have designed a new air heating simulator with solar energy considering the comprehensive applications in Inagro industry. By using a mounting type so as to increase the heat transfer surface area, a smooth and effective heat diffusion have been obtained on the channel. Within the Reynolds number interval studied the highest thermo hydraulic performance has been achieved for 1000 W/m^2 heat flux. Kulkarni and Kim [21] have performed experiments on an air heater with solar energy in order to determine the effects of different blockage types and blockage arrangements on the performance of air heater. Results of the study have shown that performance factor is greater than one for all conditions tested and pentagonal blockage type has the best performance. Skullong et al. [22] have performed an experimental analysis to investigate the heat transfer and friction characteristics on a solar air collector with rib having different geometry and staggered thin rib. Experiments have been carried out in a range of 5000–24,000 Reynolds number. For inline rib array higher heat transfer and pressure loss have been achieved compared to staggered and single rib arrangement.

Although collectors with V-shaped, rectangular, square and triangular winglets and blockages were found to be more efficient compared to flat plate solar air collectors but an exact comparison could not be made due to differences in air mass flow rate, collector sizes, number of transparent covers, type of convective heat transfer, test facility, test procedure and climatic conditions. It is evident from the literature survey that no detailed experimental investigation on temperatures and energy performance for different air mass flow rates of single flow, double glazing, trapezoidal corrugated absorber solar air collector with forced convection is carried out. The main objective of this study is to carry out the thermal performance analysis of two solar air collectors with absorber plate having flat and trapezoidal corrugated surface by designing, constructing and analyzing experimentally under the weather conditions of Konya/Turkey with different air mass flow rates. The results of the thermal performance analysis are presented comparatively among solar air collectors having two different types of absorber plates. The results were compared with results available in the literature.

2 Experimental setup and procedure

In this study, two identical solar air collectors which of one with flat absorber plate and the other with trapezoidal absorber plate have been designed and implemented. Thermal performances of the both collectors have been investigated and the results achieved have been presented comparatively. The experimental set-up established for this purpose consists of glass covered solar collectors which contains flat and trapezoidal absorber plates and wooden frames, air inlet and outlet channels, radial fans, hot air drying cabins and measurement system. Schematic view of the experimental set-up is provided in Fig. 1.

Solar air collectors have been constructed of (196 × 98 × 20) cm dimensions. Two normal window glasses with a thickness of 4 mm were used as the transparent cover. Collector frame was made of wooden namely Medium Density Fiberboard (MDF). Air flow in both collectors is provided under the absorber plate. The height of the air flow duct is 8.5 cm. A radial fan with a maximum 90 W is used to supply the needed air into the solar collector. Absorber plates are made up of aluminum sheet with 0.5 mm thickness and the upside of them were painted with matte black color to absorb more solar radiation. The bottom and lateral sides of the collector frames were insulated by means of plate formed foams with 5 cm thickness in order to reduce heat losses to ambient. The air tightness of the glass that is used as the transparent cover was realized by using rubber cords and putty. The two hoods that regulate the air coming from the fan to collector inlet and exiting from collector outlet to inlet of the drying cabin were made up of galvanized sheet metal.

The experimental set-up can be moved on the ground by means of wheels under it. The experiments were conducted at the Konya Technical University campus in clear weather conditions in Konya (37:52°N latitude; 32:31°E longitude). In

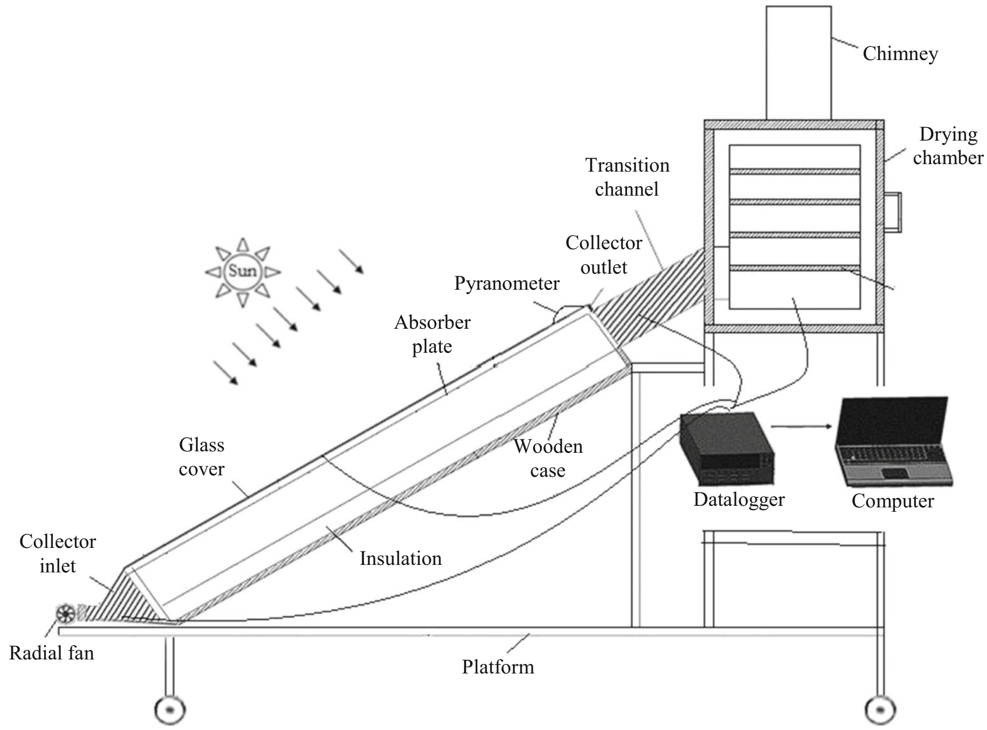


Fig. 1 Schematic view of the experimental set-up

summer months, the collectors were oriented facing south and tilted to 22.5 ° angle with respect to the horizontal plane (local latitude minus 15° value) in all experiments. All experiments were carried out daytime between 9 am and 5 pm with clear sky. The experiments have been carried out for three different mass flow rates (0.022, 0.033 and 0.044 kg/s) both for flat plate and trapezoidal corrugated plate solar air collectors. A constant mass flow rate was used during the day for any experiment. The ambient temperatures, temperatures of the air coming into/out of collector, surface temperatures of the upper glass and the absorber plate surface, velocity of the wind, and the amount of total solar radiation received by the collector surfaces have been measured hourly and written down during the experiments.

The absorber plate with trapezoidal corrugated profile is used at the top side in order to enlarge heat transfer surface and benefit more from the solar radiation. The section views of the collectors are given in Fig. 2. All dimensions in these figures are in mm. The technical specifications for the solar air collectors tested are given in Table 1.

Temperature measurements were done using K-type thermocouples. Thermocouple used for measuring the ambient temperature has been placed under the collector's case in order to protect it from direct sun light. The values of the average velocity of the air at the exit of the collectors were measured by a calibrated anemometer (Kestrel 4300). The air mass flow rate was calculated by eq. (5). The average flow velocity in the ducts was determined by means of the mass flow rate value. The revolution speed (rpm) of the air circulation fans were controlled by means of a dimmer (speed controller). All temperature rates used in collector experiments have been constantly monitored. Temperature values measured have been recorded with a 30 channel data acquisition system (COMET MS6D) which is controlled both manually and by computer. The incident hourly global solar radiation intensity on the inclined collector surfaces was measured with a pyranometer (Kipp & Zonen). In order to record the solar radiation levels different record equipment is also used.

Uncertainty of the experiment results depend on the errors made during the measurements. Let the R be a result achieved depending on the independent parameters of x_1, x_2, x_3, x_n. And let W_R be the uncertainty of the result R, and let w_1, w_2, w_3 …, w_n be the uncertainties of the parameters x_1, x_2, x_3, …, x_n respectively. By using W_R (Eq. (1)) [23, 24] can be calculated. Relative errors of the measured and calculated parameters can be determined by the expression (2).

$$W_R = \left[\left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \left(\frac{\partial R}{\partial x_2}w_2\right)^2 + \ldots + \left(\frac{\partial R}{\partial x_n}w_n\right)^2\right]^{1/2} \quad (1)$$

$$E_R = \frac{W_R}{R}*100 \quad (2)$$

Temperature difference of the air coming into and out of collector, total solar radiation incident to the tilted surface, the mass flow rate of the air, the useful heat gain of the air, uncertainties in thermal efficiency and the relative errors have been presented in Table 2.

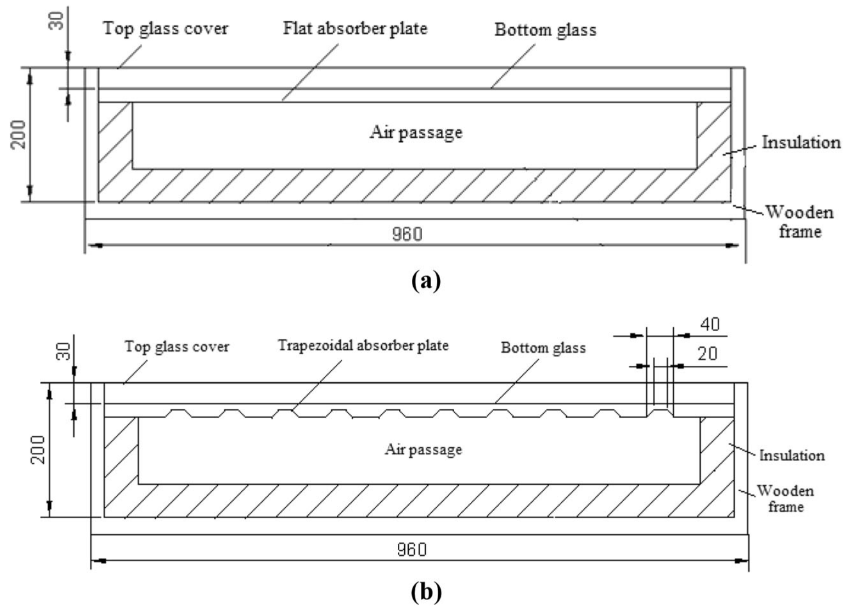


Fig. 2 Section views of the collectors (a) with flat plate (b) with trapezoidal corrugated plate

3 Data analysis

Instantaneous thermal efficiency of a flat plate solar air collector is the ratio of the useful thermal energy conveyed into the air (the working fluid of the collector), to the average thermal energy stemming by solar radiation on the collector surface from the sun at the same time interval. Under the steady state conditions, mathematically thermal efficiency of a solar collector is expressed by Hottel-Whillier-Bliss equation [25].

$$\eta = \frac{\dot{Q}_u}{IA_p} \tag{3}$$

Thermal efficiency of the solar air collector depends on many parameters such as; optical efficiency of the absorber surface and transparent cover, geometry of the absorber surface, type of the flow arrangement, solar radiation amount incident on collector surface, tilt angle of the collector from the horizontal surface, mass flow rate of the air, temperature of the ambient and the velocity of the wind.

Table 1 Technical specifications of the solar air collectors

Component	Technical specifications
Flat absorber plate	Aluminum; (850 × 1820 × 0.5) mm Heat transfer area: 1.547 m^2
Trapezoidal absorber plate	Aluminum; (850 × 1820 × 0.5) mm Heat transfer area: 1.978 m^2
Plate surfaces	Plain, matte black painted
Transparent cover	Normal window glass, thickness: 4 mm, 2 pieces
Insulation Air flow duct	XPS (extruded polystyrene), thickness: 50 mm (85 × 850 × 1820) mm
Collector frame	Wooden: Medium-density fibreboard; (940 × 1840 × 18) mm
Fan	Radial, 275 m^3/h, 90 W

The useful thermal energy for a solar collector is defined as the heat transferred from the absorber plate of the collector to the fluid exerted. Depending on the air temperatures at the inlet and outlet of the collector, the useful thermal energy gained by the collector under the steady-state condition can be calculated as below.

$$\dot{Q}_u = \dot{m}C_p(T_o - T_i) \tag{4}$$

Mass flow rate of air circulating in the collector is determined according to the following expression.

$$\dot{m} = \rho V A_c \tag{5}$$

Daily mean efficiency is more important practically for solar collectors. Daily mean thermal efficiency of the solar collector can be calculated as [26];

$$\eta_d = \frac{\dot{Q}_d}{A_p \sum_{t_1}^{t_2} I} \tag{6}$$

Total daily useful energy collected in the collector is determined as;

$$\dot{Q}_d = \sum_{j=1}^{n} \dot{Q}_j \tag{7}$$

Reynolds number (Re) for the flow within the air duct has been written as [27];

$$Re = \frac{\rho V D_h}{\mu} \tag{8}$$

Table 2 The uncertainties and relative errors for measured and calculated quantities

Parameter	Max. value	Uncertainty	Relative error (%)
Temperature difference of the air, ΔT (°C)	42	± 0.141	0.34
Solar radiation, I (W/m^2)	1026	± 10	0.97
Air mass flow rate, $\dot{m}$ (kg/s)	0.044	± 1.324*10^{-3}	3.01
Useful heat gain, $\dot{Q}_u$ (W)	1172	± 9.237	0.79
Thermal efficiency η (%)	74	± 0.015	2.03

The mean experimental Nusselt number in duct (Nu) is calculated as follows;

$$Nu = \frac{hD_h}{k} \tag{9}$$

The mean convective heat transfer coefficient (h) from the absorber plate to the working fluid in the channel is given by [27].

$$h = \frac{\dot{Q}_u}{A_p\left(T_p - T_b\right)} \tag{10}$$

The hydraulic mean diameter (D_h) of the rectangular duct is determined as;

$$D_h = \frac{4A_c}{P} \tag{11}$$

$$D_h = \frac{4(HW)}{2(H+W)} \tag{12}$$

The all thermo physical properties of the air in the duct are evaluated at the mean bulk temperature of T_b defined as;

$$T_b = \frac{T_i + T_o}{2} \tag{13}$$

In order to obtain the thermal properties of air such as density, thermal conductivity, specific heat at constant pressure and dynamic viscosity in the temperature range of 280–470 K the following equations were used respectively [28].

$$\rho = 3.9147 - 0.016082T_b + 2.9013x10^{-5}T_b^2 - 1.9407x10^{-8}T_b^3 \tag{14}$$

$$k = (0.0015215 + 0.097459T_b - 3.3322)x10^{-3} \tag{15}$$

$$c_p = 999.2 + 0.1434T_b + 1.101x10^{-4}T_b^2 - 6.7581x10^{-8}T_b^3 \tag{16}$$

$$\mu = \left(1.6157 + 0.06523T_b - 3.0297x10^{-5}\right)x10^{-6} \tag{17}$$

4 Results and discussion

Purpose of this study is to enhance the thermal efficiency of a solar air collector by making use of a passive method. Therefore two identical solar air collectors, which one with flat absorber plate and the other with trapezoidal corrugated plate, have been designed. After implementation of the collectors, experiments have been performed on them in August 2016 for the air mass flow rates of 0.022, 0.033 and 0.044 kg/s and thermal efficiency of the collectors have been investigated simultaneously. Results achieved have been presented comparatively for both collectors.

Both collectors have been subjected to experiments simultaneously for each air mass flow rate by four days. In order to compare the thermal efficiencies of the solar air collectors accurately, days of which the hourly radiation values are analogous have been picked. Hourly changes in global solar radiation amount incident to tilted collector surface have been presented in Fig. 3. For each of three mass flow rates, solar radiation values are very similar to each other. As seen from the figure, radiation values change between 474 and 486 W/m^2 at 9 am rise to 1013–1026 W/m^2 at 1 pm and decrease to 400–450 W/m^2 at 5 pm. Maximum solar radiation is measured as 1026 W/m^2 at 1 pm. In the tests performed for different mass flow rates there has been a maximum 10 W/m^2 difference observed between the daily average solar radiation values. The mean solar radiation intensity was about 787 W/m^2 for all days.

Hourly changes in the temperature of the ambient during different days of the experiments can be seen from the Fig. 4. As shown in the figure, the ambient temperatures increase with the solar radiation level up to 4 pm starting from 25 to 27 °C in the morning. After 4 pm, ambient temperature decreases with the decreasing solar radiation level. Along the day, ambient temperature changes between 26 °C and 35 °C. A temperature difference of 2.5 °C at max has been observed in the ambient temperatures measured hourly at different days. The maximum air temperature difference observed in daily average ambient temperature values is 2 °C. Since the experiments were carried out on consecutive days, there has been

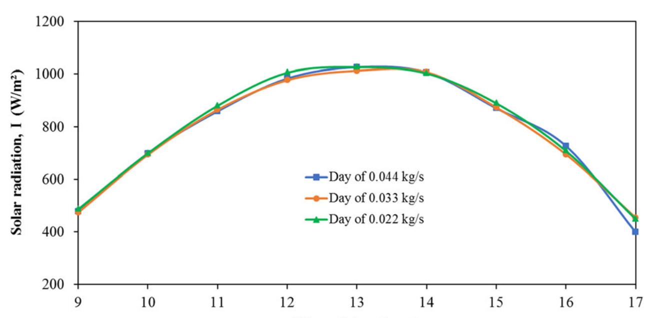


Fig. 3 Comparison of solar radiation intensity for different days

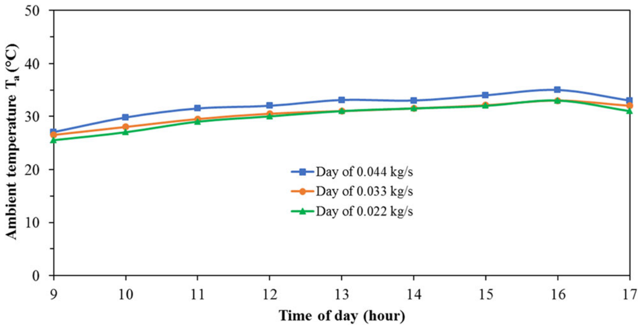


Fig. 4 Hourly variation of the ambient temperatures for different days of the experiments

very little change in the ambient temperatures and solar radiations.

Figure 5 is prepared in order to compare the solar collectors in terms of heat transfer performance and to determine the effects of the absorber plate surface geometry. Change of the average Nusselt number depending on Reynolds number is presented in this figure. In order to verify the experimental data, experimental mean Nusselt numbers calculated for the flat plate collector have been compared to the two different correlation results suitable for the problem. One of the correlations is the one which is suggested by Kays and Crawford [27] for the turbulent flow between two parallel plates of which one side is heated and the other side is insulated (Eq. (18)). The other correlation is the one which is suggested by Gnielinski for turbulent flow at low Reynolds numbers ($2300 < Re < 10^4$) [29]. The average friction factor through channel depends on the Reynolds number for turbulent flow can be determined by the relation (20).

$$Nu = 0.0158Re^{0.8} \tag{18}$$

$$Nu = \frac{(f/8)(Re-1000)Pr}{1+12.7(f/8)^{0.5}\left(Pr^{2/3}-1\right)} \tag{19}$$

$$f = [0.79\ \ln(Re)-1.64]^{-2} \tag{20}$$

As seen from the Fig. 5 for both collectors, Nusselt numbers increase depending on the increase in Reynolds numbers. For flat plate solar collector, experimental mean Nusselt

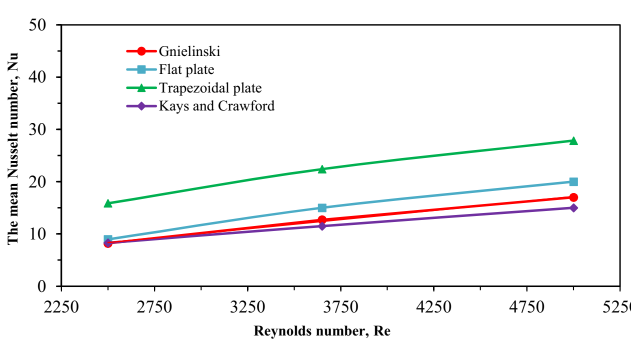


Fig. 5 The average Nusselt numbers versus Reynolds numbers

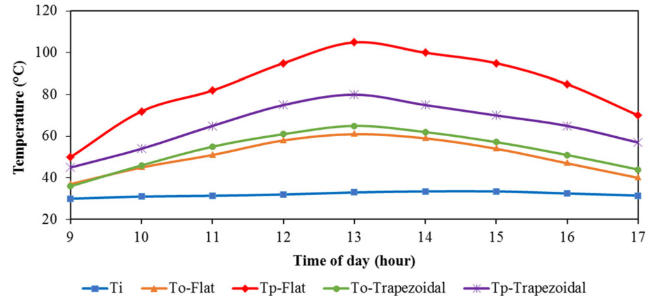


Fig. 6 Changes of temperatures for both solar air collectors in 0.033 kg/s

numbers have been found to be larger than the theoretical values. There have been 9–18% and 8–30% diversions between experimental Nusselt numbers and the Nu values calculated from the Gnielinski [29] and Kays and Crawford [27] correlations, respectively. It should be noted that for the flat plate collector, theoretical Nu values calculated from the Kays and Crawford correlation are lower 1% to 12% than the values calculated from the Gnielinski correlation. With the increasing Reynolds number, boundary layer thickness decreases at the plate surface and the useful heat amount transferred to air increases. Heat transfer at the trapezoidal absorber plate collector, depending on the Reynolds number, has reached to 1.4–1.77-fold of that of flat plate collector. As a result of the increasing heat transfer area in touch with the fluid although it is in the same projection area and the extra turbulence effects occurred in the flow, Nusselt number increases [30, 31].

Figure 6 depicts the hourly changes of temperatures of the air inlet and outlet of collectors and absorber plate surface for flat plate and trapezoidal corrugated plate solar air collector at 0.033 kg/s mass flow rate. The temperature of the air entering the collectors increases very slightly starting from morning up to 3 pm and then it decreases. The temperature of the air entering collectors gets around 30–34 °C daily. Temperatures of the air outgoing the collectors and the absorber plate surface temperatures increases up to noon time and reach its maximum value at 1 pm. Depending on the decreasing solar radiation, aforementioned temperatures decrease in the afternoon. According to the values measured hourly, maximum temperatures of the air entering to and going out from the flat plate collector and absorber plate surface are 34 °C, 61 °C and

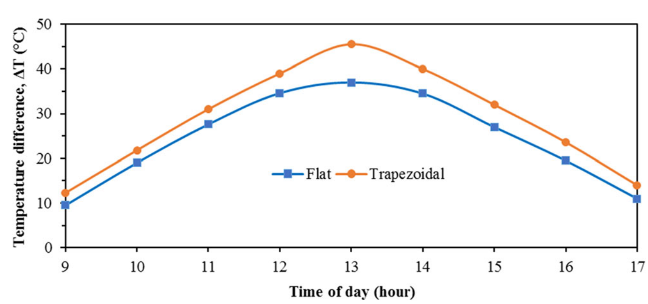


Fig. 7 Changes of the temperature difference for two types of collectors for $\dot{m} = 0.022\ kg/s$

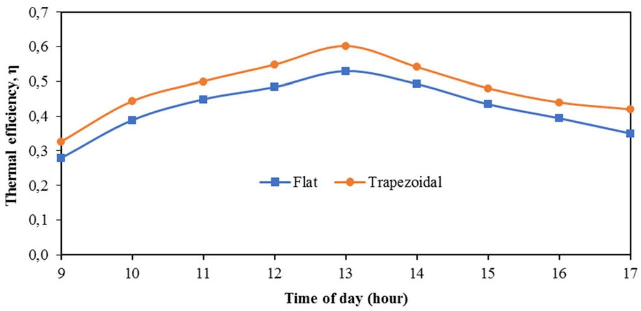


Fig. 8 Efficiency comparison for two types of collectors for $\dot{m} = 0.022\ kg/s$

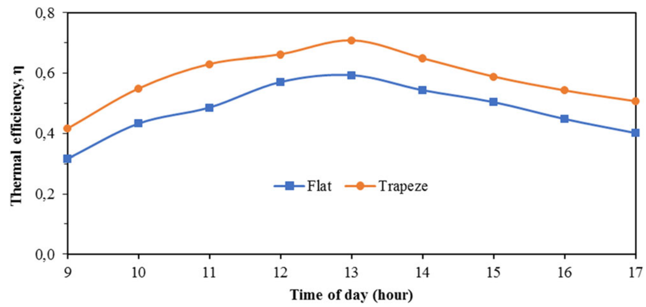


Fig. 10 Efficiency comparison for two types of collectors for $\dot{m} = 0.033\ kg/s$

105 °C, respectively. Similarly, the average values for those temperatures measured daily for flat plate collector is respectively 32 °C, 50 °C and 84 °C. For the trapezoidal collector, maximum hourly values of these temperatures measured from the experiments are 33 °C, 65 °C and 80 °C, respectively. The daily average values for trapezoidal collector observed for those temperatures are 31 °C, 53 °C and 65 °C, respectively.

By comparing the absorber plate surface temperatures seen in Fig. 6, temperatures of the trapezoidal corrugated plate surface is found to be lower than the flat plate surface temperatures during the experiments. Since trapezoidal corrugated absorber plate has more heat transfer surface area, air passing through the channel contacts more with the hot plate surface area. Thus, more heat transfer occurred in comparison with the flat absorber plate. This result obtained from the study is similar with the findings of Karim and Hawlader [15] and Kabeel et al. [32]. Another conclusion drawn from the figure is that the air outlet temperature for the trapezoidal collector during the day is greater than for the flat plate collector. This is because of the trapezoidal collector has more heat transfer surface area.

In Fig. 7, a comparison of difference between inlet and outlet air temperatures of the collectors have been depicted for flat and trapezoidal corrugated absorber plate at 0.022 kg/s mass flow rate. Temperature difference of the air increases from morning time similarly to the solar radiation for both collectors and reaches its peak value at noontime. In the afternoon, it constantly decreases up to the sunset. From the figure it can be concluded that for the trapezoidal absorber plate collector increase in the temperature of the air is larger by 2.8–5.5 °C than for the flat plate collector during all day except noon time. Air temperature difference is about 9 °C when the temperatures reach their peak values at 1 pm. The air temperature difference occurs between two collector models because of trapezoidal absorber plate collector has more heat transfer area when compared with the flat plate collector. Since the air passes with lower velocities through the channel at smaller mass flow rates, air contacts with the hot absorber plate surface for a longer period of time and as a result, air reaches higher temperatures until it gets out of the collector.

The instantaneous efficiency comparison for the two types of collectors at the mass flow rate of 0.022 kg/s is given in Fig. 8. Curves for thermal efficiency vary similarly with the curves of solar radiation and temperature increase in collectors. The maximum thermal efficiency rates at mass flow rate of 0.022 kg/s for the flat plate and trapezoidal plate collectors are 53% and 60% at noontime, respectively. Consequently, thermal efficiency of the trapezoidal plate collector is clearly higher than the flat plate collector's. The main reason for the difference in thermal efficiencies is that we can benefit more from solar radiation energy in the unit time with the increasing surface area of the absorber plate. In addition, useful heat gain from the plate to the air in the channel is larger for the trapezoidal corrugated plate in comparison with the flat plate collector.

Comparison of the air temperature differences between collector inlet and outlet is shown in Fig. 9 at the mass flow rate of 0.033 kg/s. This figure displays that air outlet temperature for the trapezoidal plate collector is larger than the flat plate collector's for the same reasons discussed earlier. Maximum

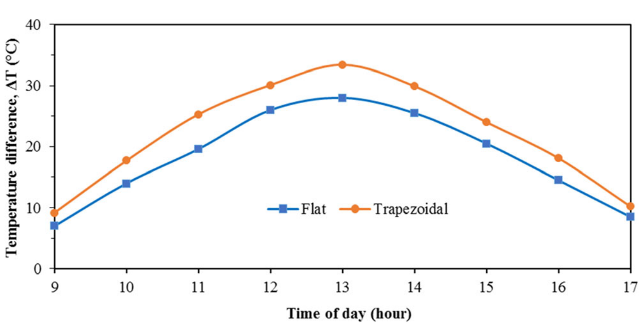


Fig. 9 Change of the temperature difference for two types of collectors for $\dot{m} = 0.033\ kg/s$

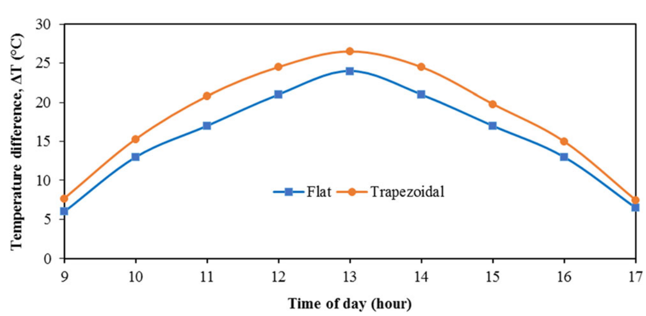


Fig. 11 Change of the temperature difference for two types of collectors for $\dot{m} = 0.044\ kg/s$

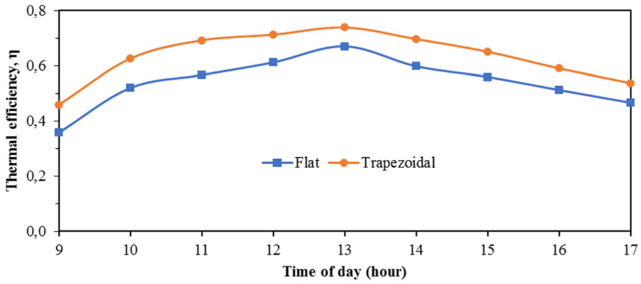


Fig. 12 Efficiency comparison for two types of collectors for $\dot{m} = 0.044\ kg/s$

level of air outlet temperature is found to be larger by 5.5 °C than the flat plate collector for trapezoidal corrugated plate collector at mass flow rate of 0.033 kg/s for. Increase in the outgoing air temperature for the trapezoidal corrugated plate collector is larger by 1.7–4.4 °C than flat plate collector during the whole day except noontime.

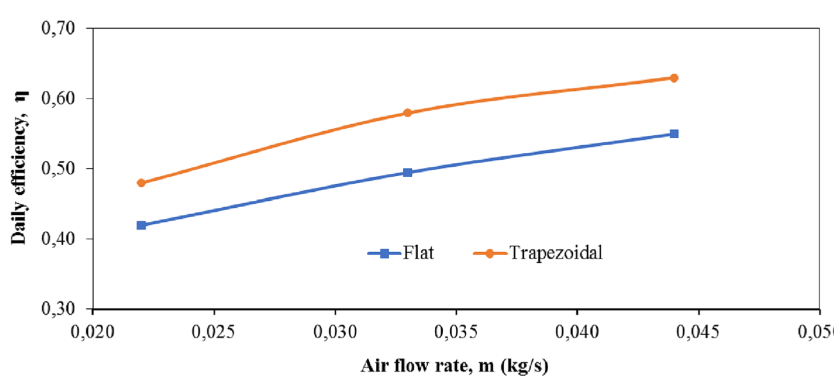
The change of instantaneous thermal efficiencies with respect to time of day for both solar collectors considered at the mass flow rate of 0.033 kg/s is depicted in Fig. 10. Figure clearly shows that thermal efficiency of the trapezoidal plate collector is larger than the flat plate collector's. For both collectors, peak values of thermal efficiency are achieved at 1 pm. Instantaneous maximum thermal efficiency observed for the flat plate and trapezoidal plate collectors are 59% and 71%, respectively. By examining the Fig. 8 and the Fig. 10 together it can be concluded that instantaneous thermal efficiencies of both collectors for the air mass flow rate of 0.033 kg/s are greater than for 0.022 kg/s.

Comparison of the flat and trapezoidal corrugated plate collectors in terms of the air temperature difference is given in Fig. 11 at highest air mass flow rate of 0.044 kg/s. Trapezoidal corrugated plate collector has the higher increase in temperature of the air compared to flat plate collector. It should be noted that this result is originated from the fact that trapezoidal corrugated plate collector has larger heat transfer area compared to the flat plate collector. Also extra turbulence effect occurred in the flow rises the heat transfer coefficient of air flowing across plate. It is inferred from the figure that when both collectors are compared for air mass flow rate of 0.044 kg/s increase in temperature of the air is smaller than the other air mass flow rates. The maximum level of the increase in the air temperature for trapezoidal corrugated plate collector is higher than the flat plate collector by 2.5 °C in average. The temperature of the outgoing air for trapezoidal corrugated plate collector is higher by 1.0–3.0 °C during the whole day except noon time compared to flat plate collector. Since the contact time of air with the absorber plate surface is shorter at higher air mass flow rates, temperature rise of the air in the collector is lower according to the smaller air mass flow rates.

Changes in the instantaneous thermal efficiencies of the flat and trapezoidal corrugated plate collectors during the day at 0.044 kg/s are shown in Fig. 12. Thermal efficiencies have its maximum levels for both collectors at 1 pm. The maximum thermal efficiencies for the flat and trapezoidal corrugated plate collectors are 67% and 74%, respectively. Thermal efficiency values for the trapezoidal corrugated plate collector are larger than flat plate collector along the day. Considering the all thermal efficiency values achieved for all air mass flow rates, the maximum level is reached for 0.044 kg/s. Heat transfer increased with the larger increasing absorber plate surface area and turbulence effect in the flow which resulted in to improve the thermal efficiency of the collector.

Daily average thermal efficiency values for collectors is more convenient in practically. In order to make a more accurate comparison between both collectors, effect of air mass flow rate on the daily average thermal efficiency of the collectors is portrayed in Fig. 13. As shown by the figure, daily average thermal efficiencies for flat plate collectors are 0.42, 0.48 and 0.54, while thermal efficiencies of trapezoidal corrugated plate collectors have 0.48, 0.58 and 0.63 for air mass flow rates of 0.022 kg/s, 0.033 kg/s and 0.044 kg/s, respectively. It is clearly determined that average thermal efficiency for trapezoidal corrugated plate collector is higher by 14–21% than the flat plate collector considering the all mass flow rates. Suman et al. [33], Karakaya and Durmus [34], Kabeel et al. [32] and Lakshmi et al. [35] found that the efficiency of solar air collector with corrugated absorber plate compared to the flat plate collector is higher by 14%, 15%, 10.5% and 23%,

Fig. 13 Daily efficiency for two types of collectors versus the mass flow rates



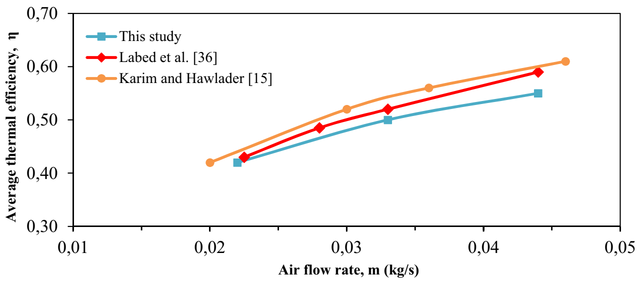


Fig. 14 Thermal efficiency comparison of flat plate collector with some solar air collectors available in literature

respectively. In this study, it is seen that for both collector daily average thermal efficiencies rises with the increasing air mass flow rates. Since more thermal energy is transferred to the air with the increasing turbulence effect in the channel at higher air mass flow rates, temperature of the absorber plate decreases. By this way heat losses from the collector to the ambient decrease which yields increasing thermal efficiency.

Figure 14 shows the thermal efficiency comparisons of flat plate solar air collector between our study and other relevant studies in the literature. The comparisons were performed according to the investigated range of the air mass flow rate. Here in both this study and others thermal efficiencies are increasing with increase in the mass flow rate of air. Also, average thermal efficiency values achieved in this study result show a maximum deviation 5.26% from Labed et al. [36] and 7.69% from Karim and Hawlader [15]. This result expresses the reliability of experimental data.

5 Conclusions

Thermal performances of solar air collectors of which one with conventional flat absorber plate and the other with trapezoidal corrugated plate have been investigated experimentally at three different air mass flow rates (0.022 kg/s, 0.033 kg/s and 0.044 kg/s) under climatic conditions of Konya/Turkey. Experimental set-up is built up to evaluate the effects of air mass flow rate and surface geometry of the absorber plates on the thermal efficiency. Results obtained from the experiments can be summarized as follows:

- Under configuration and operation conditions taken into consideration, daily average thermal efficiency of the trapezoidal corrugated absorber plate collector has been found to be higher by 14–21% than the flat absorber plate collector's. This resulted from the fact that heat transfer surface area of the absorber plate which is in contact with the air is larger and extra turbulence effects occurred in the flow.
- Trapezoidal corrugated absorber plate collector has greater performance in terms of temperature rise for the air in the collector. The maximum rise in temperature of the air at trapezoidal corrugated absorber plate collector has been 9 °C higher for the air mass flow rate of 0.022 kg/s and 3.5 °C higher for 0.044 kg/s compared with the flat plate collector.
- Useful heat transfer from the collectors to the air increased with higher air mass flow rates which resulted in a growth in thermal efficiency level of the collectors.
- Maximum daily average thermal efficiency of the trapezoidal corrugated plate solar air collector is obtained to be 63% for the air mass flow rate of 0.044 kg/s.
- Solar air collectors heat the air more at lower air mass flow rates due to the longer contact time of air in the collector to the absorber plate.
- Thermal efficiency of solar air collectors, compared to other parameters, changes significantly depending on the solar radiation incident to collector surface, air mass flow rate, and surface characteristics of the absorber plate.
- Solar air collectors having trapezoidal corrugated absorber plates can be used as an important heat source for space heating and drying of different agricultural products at lower temperatures in Turkey where there is a great solar energy potential in terms of renewable energy.

Acknowledgments The authors are grateful to the Scientific Research Project Coordination Unit of Selçuk University (BAP) for the financial supports they provided to this project (project No. 16201070).

Compliance with ethical standards This article does not contain any studies with human participants or animals performed by any of the authors.

Conflict of interest The authors declare that they have no conflict of interests.

Declaration of interests The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests:

Selçuk Darici

Anil Kilic

Informed consent None.

References

1. Priyam A, Chand P (2019) Experimental investigations on thermal performance of solar air heater with wavy fin absorbers. Heat Mass Transf:1–16. https://doi.org/10.1007/s00231-019-02605-1
2. Kumar A, Bhagoria JL, Sarviya RM (2009) Heat transfer and friction correlations for artificially roughened solar air heater duct with discrete W-shaped ribs. Energy Convers Manag 50:2106–2117
3. Saini RP, Verma J (2008) Heat transfer and friction factor correlations for a duct having dimple-shape artificial roughness for solar air heaters. Energy 33:1277–1287

4. Ligrani PM, Mahmood GI, Harrison JL, Clayton CM, Nelson DL (2001) Flow structure and local Nusselt number variations in a channel with dimples and protrusions on opposite walls. Int J Heat Mass Transf 44:4413–4425
5. Chukin AV, Koslov AP, Agachev RS (1995) Study and application of hemispherical cavities for surface heat transfer augmentation. ASME 40th international gas turbine and aero congress. No. 95-GT-59, Houston
6. Yadav S, Kaushal M, Goel V, Siddhartha V (2013) Nusselt number and friction factor correlations for solar air heater duct having protrusions as roughness elements on absorber plate. Exp Thermal Fluid Sci 44:34–41
7. Ravi RK, Saini RP (2016) Experimental investigation on performance of a double pass artificial roughened solar air heater duct having roughness elements of the combination of discrete multi V shaped and staggered ribs. Energy 116:507–516
8. Khadraoui A, Bouadila S, Kooli S, Guizani A, Farhat A (2016) Solar air heater with phase change material: an energy analysis and a comparative study. Appl Therm Eng 107:1057–1064
9. Sridhar A, Reddy KS (2007) Transient analysis of modified cuboid solar integrated- collector storage system. Appl Therm Eng 27:330–346
10. Hatami N, Bahadorinejad M (2008) Experimentally determination of natural convection heat transfer coefficient in a vertical flat-plate solar air heater. Sol Energy 82:903–910
11. Moummi N, Youcef-Ali SA, Moummi SA, Desmons JY (2004) Energy analysis of a solar air collector with rows of fins. Renew Energy 29:2053–2060
12. Debnath S, Das B, Randive PR, Pandey KM (2018) Performance analysis of solar air collector in the climatic condition of north eastern India. Energy 165:281–298
13. El-Sebaii AA, Aboul-Enein S, Ramadan MRI, Shalaby SM, Moharram BM (2011) Investigation of thermal performance of double pass-flat and v-corrugated plate solar heaters. Energy 36: 1076–1086
14. Aldabbagh LBY, Egelioglu F, Ilkan M (2010) Single and double pass solar air heaters with wire mesh as packing bed. Energy 35: 3783–3787
15. Karim MA, Hawlader MNA (2004) Development of solar air heater collectors for drying applications. Energy Convers Manag 45:329–344
16. Ozgen F, Esen M, Esen H (2009) Experimental investigation of thermal performance of a double-flow solar air heater having aluminum cans. Renew Energy 34:2391–2398
17. Tyagi VV, Pandey AK, Giridhar G, Bandyopadhyay B, Park SR, Tyagi SK (2012) Comparative study based on exergy analysis of solar air heater collector using thermal energy storage. Int J of Energy Research 36:724–736
18. Bashria AA, Adam NM, Sapuan SM, Daud M, Omar H, Megat HM, Abas F (2004) Prediction of the thermal performance of solar air heaters by internet-based simulation. Power Energy 218:579–587
19. Dabra V, Yadav A (2018) Parametric study of a concentric coaxial glass tube solar air collector: a theoretical approach. Heat Mass Transf 54:1613–1625
20. Aidinlou HR, Nikbakht AM (2017) Heat flux: thermohydraulic investigation of solar air heaters used inagro-industrial applications. Heat Mass Transf 53:917–928
21. Kulkarni K, Kim KY (2016) Comparative study of solar air heater performance with various shapes and configurations of obstacles. Heat Mass Transf 52(12):2795–2811
22. Skullong S, Thianpong C, Promvonge P (2015) Effects of rib size and arrangement on forced convective heat transfer in a solar air heater channel. Heat Mass Transf 51(10):1475–1485
23. Kline S, Mcclintock F (1953) Describing uncertainties in single-sample experiments. Mech Eng 75:3–8
24. Holman JP (1994) Experimental methods for engineers. McGraw-Hill Book Company, Singapore
25. Duffie JA, Beckman WA (1991) Solar engineering of thermal processes. John Wiley and Sons, New York
26. Ashrae, Ashrae Standard (1977) Methods of testing to determine thermal performance of solar collectors. ASHRAE 345 New York
27. Kays WM, Crawford ME (1980) Convective heat and mass transfer. Second ed. McGraw-Hill, New York
28. Hedayatizadeh M, Ajabshirchi Y, Sarhaddi F, Farahat S, Safavinejad A, Chaji H (2012) Analysis of exergy and parametric study of a v-corrugated solar air heater. Heat Mass Transf 48:1089–1101
29. Kakaç S, Liu H (2002) Heat exchangers: selection, rating, and thermal design, Second Edition. CRC Press, New York
30. Benli H (2013) Experimental derived efficiency and exergy analysis of a new solar air heater having different surfaces shapes. Renew Energy 50:58–67
31. Shuilian L, Hui W, Xiangrui M, Xinli W (2017) Comparative study on the performance of a new solar air collector with different surface shapes. Appl Thermal Eng 114:639–644
32. Kabeel AE, Khalil A, Shalaby SM, Zayed ME (2016) Investigation of the thermal performances of flat, finned and v-corrugated plate solar air heaters. J Sol Energy Eng 138:1–7
33. Suman S, Khan M, Pathak M (2015) Performance enhancement of solar collectors. A review renewable and Sust. Energy Reviews 49: 192–210
34. Karakaya H, Durmus A (2011) Efficiency and exergy analysis in solar air collectors having different types of surface geometries. Renew Sust Energ Rev 121:16–24 (in Turkish)
35. Lakshmi DVN, Layek A, Muthu PK (2016) Performance analysis of trapezoidal corrugated solar air heater with, sensible heat storage material. International conference on recent advancement in air conditioning and refrigeration, 10-12 November, India
36. Labed A, Moummi N, Benchabane A, Aoues K, Moummi A (2012) Performance investigation of single- and double-pass solar air heaters through the use of various fin geometries. Int J Sustain Energy 31(6):423–434