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EXPERIMENTAL ANALYSIS OF THERMAL PERFORMANCE OF SOLAR AIR COLLECTORS WITH ALUMINUM FOAM OBSTACLES

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Abstract: An experimental study has been performed to investigate the energy analysis of flat-plate solar air collectors (SACs) with porous obstacles at two different thicknesses and without obstacles (flat plate). As a porous material aluminum foams are used. They are placed sequentially and in a staggered manner onto the cover of the SACs. The measurements are performed at two different values of air mass flowrate as 0.016 kg/s and 0.025 kg/s and with thickness of 6 mm and 10 mm. Energy efficiencies of SACs are calculated based on first law of thermodynamics. In the experiments, five types of solar air collectors are tested and a comparison is made among them from the point of energy efficiency. The obtained results showed that the highest collector efficiency and air temperature rise are achieved by SACs at 6 mm thickness and 0.025 kg/s air mass flowrate.
Keywords: Solar air heaters, thermal efficiency, energy, heat transfer, porous obstacles

KAPALI HÜCRELİ ALÜMİNYUM KÖPÜK ENGELLER SAHİP HAVA ISITMALI GÜNEŞ KOLLEKTÖRLERİNİN ISIL PERFORMANSININ DENEYSEL ANALİZİ

Özet: Bu deneysel çalışmada, farklı iki kalınlığa sahip gözenekli engeller ile engelsiz yüzeye sahip hava ısıtmalı güneş kollektörünün enerji analizi incelenmiştir. Gözenekli malzeme olarak kapalı hücreli alüminyum köpük kullanılmış ve kollektörlerin emici yüzeyine sıralı ve şaşırtmalı olarak yerleştirilmiştir. Ölçümler, 6 ve 10 mm kalınlıklarındaki engeller ile 0.016 kg/s ve 0.025 kg/s debilerinde çalışılarak gerçekleştirilmiştir. Kollektörlerin enerji verimleri Termodinamiğin I. kanunundan yararlanılarak hesaplanmıştır. Deneylerde beş tip hava ısıtmalı güneş kollektörü kullanarak birbirleri ile karşılaştırılarak test edilmiştir. Elde edilen sonuçlara göre, en yüksek verim ve sıcaklık değerleri 0.025 kg/s debisindeki 6 mm kalınlığındaki şaşırtmalı dizilimde gerçekleşmiştir.
Anahtar Kelimeler: Hava ısıtmalı güneş kollektörleri, ısıl verim, enerji, ısı transferi, gözenekli engeller

NOMENCLATURE

A_c	Surface area of collector [m^2]
A	Section area [m^2]
C_p	Specific heat of air [kJ/kgK]
e	Empirical factor
f	Empirical factor
F_R	Collector heat removal factor depending air inlet temperature
F_o	Collector heat removal factor depending air outlet temperature
F'	Collector efficiency factor
h	Heat transfer coefficient [W/m^{2}K]
h_w	Wind heat transfer coefficient [W/m^{2}K]
I	Solar radiation [W/m^2]
k	Thermal conductivity of insulation [W/mK]
L	Thickness of the insulation [m]
L_{ct}	Collector length [m]
$\dot{m}$	Mass flowrate [kg/s]
N	Number of glass cover
R	Resistance
$\dot{Q}_u$	Useful energy gain [kW]
T	Temperature [^{o}C]
U_b	The bottom heat loss coefficient [W/m^{2}K]
U_e	The edge heat loss coefficient [W/m^{2}K]
U_L	The collector overall heat loss coefficient [W/m^{2}K]
U_t	The top heat loss coefficient [W/m^{2}K]
V	Velocity of fluid (air) [m/s]
V_w	Wind velocity [m/s]
η	Thermal efficiency
ε_g	Emissivity of the glass cover

ε_p	Emissivity of the absorbing plate
β	Collector tilt [^{o}]
$\tau\alpha$	Effective transmission
σ	Stefan-Boltzman constant [W/m^{2}K^4]

Subscripts

a	Ambient
c	Cover
e	Edge
i	Inlet
g	Glass
o	Outlet
p	Plate

Abbreviation

Case I	flat plate collector of without obstacles
Case II	collctor of 6 mm sequent
Case III	collector of 6 mm staggered
Case IV	collector of 10 mm sequent
Case V	collector of 10 mm staggered
SAC	Solar air collector

INTRODUCTION

Solar collectors are the main device to collect energy from the sun and they are considered a kind of heat exchanger that transmits energy from the sun to flow materials (e.g. water or air). Solar air collectors/heaters are simple in design and maintenance and they require low investment cost but their efficiency is lower than that of water heater collectors. In solar air heaters, corrosion and leakage problems are less when compared with water heater collectors. Low thermal capacity of air is a disadvantage for these collectors. However, different modifications are applied to improve the heat transfer coefficient between the absorber plate and air flow. One of the effective ways to increase the convective heat transfer rate is to extend the heat transfer surface or to increase turbulence inside the flowing channel by using fins or a corrugated surface.

SACs may be classified into four main groups according to their application areas as follows: drying, pre-heating, space heating and cooling. The researchers have tried to develop thermal performance of collectors with passive methods by making different designs to enhance the heat transfer and efficiency. In this context, the performance of single and double-pass solar air heater with fins and steel wire mesh as absorber is investigated experimentally by Omojaro and Aldabbagh (2010). They found that the thermal efficiency is increased with the increase of air mass flowrate. The efficiency of the double-pass collector is found to be higher than the single pass by 7–9%. Nwosu (2010) investigated the pin fins attached to the absorber of a solar air heater.Karwa and Chauhan (2010) and Karmare and Tikekar (2010), studied the effect of rib roughness over the absorber plate on thermal performance of solar air heaters. They indicated that the artificial roughness breaks the viscous sub-layer and increases the heat transfer with the penalty of the undesirable increase in the pressure drop due to the increased friction. A novel solar air collector of pin-fin integrated absorber was designed to increase the thermal efficiency by Peng et al. (2010). Through analyzing the experimental results, the heat transfer coefficients on pin-fin arrays collectors and flat-plate collector were obtained under an air volume flow rate of 19m^3/h and a heat transfer coefficient of PZ3-11.25 pin-fins collector can reach three times than that of the flat-plate collector. Studies on heat transfer enhancement techniques can be found in Mittal and Varshney (2006) who investigated the performance of SACs theoretically. As a similar work, Hans et al. (2010) analyzed the solar air collectors which include the v-shaped bars for 8 different types. They also developed a correlation between heat transfer and friction factor. Özgen et al. (2009) experimentally investigated of the thermal performance of a double-flow solar air heater having aluminum cans. They have achieved the highest collector efficiency for type I at 0.05 kg/s air mass flowrate the by inserting an absorbing plate made of aluminum cans into the double-pass channel. Akpinar and Kocyigit (2010) worked on first and second law of the new flat-plate SAC. Pakdaman et al. (2011) experimentally investigated the different thermal characteristics of a natural convection flat plate solar air heater with longitudinal rectangular fins array. Correlation for Nusselt number was developed and exergy analysis has been carried out to determine the optimum conditions in which the system has the highest performance. Thermal performance of a double pass solar air heater is studied by El-khawajah et al. (2011). They attached 2, 4 and 6 fins onto collector. They illustrated that the efficiency increases with the increase of mass flowrate. Yeh (2012) analyzed the effect of internal recycle on the collector efficiency in upward-type flat-plate solar air heaters attached with fins. He found that more than 100% of improvement in collector efficiency is obtained by recycling operation. Other related work with solar air heaters which different plate inserted are studied by Gupta and Kaushik (2009), Hachemi (1999) and Gill et al. (2012). Chamoli et al. (2012) have made a new research reviewing the literature relevant with double-pass solar air heaters. These studies include the design of double pass solar air heater, heat transfer enhancement, flow phenomenon and pressure drop in duct. Oztop et al. (2013), a review has been performed on experimental and numerical studies of solar air heaters. Bayrak et al. (2013) calculated energy and exergy values of porous obstacles inserted solar air collectors for building applications. Also, they have found relative CO_2 reduction potential of solar air collectors using the rational exergy management model.

The main aim of this work is to examine the thermal efficiency of solar air collector with porous obstacles at different thickness rates.

EXPERIMENTAL SET-UP AND MEASUREMENT PROCEDURE

The constructed experimental set-up for SAC with obstacles is located in Elazıg, Turkey with 36^o and 42^o North latitudes. The main solar radiation intensity is measured about 3.67 kWh/m^2 day. The amount of solar radiation received overall of the Turkey, in other words, the gross solar energy potential is 3517 EJ/year. The schematic of an experimental set-up is shown in Fig. 1. The setup was designed and constructed in order to obtain data for the investigation. Dimensions (height - width) and plate thicknesses of collectors were 120mm x 70mm x 12mm, respectively. Absorber plates of collectors were formed by a dull black painted galvanized sheet with 0.8 mm thickness. Normal window glass of 4 mm thickness was used as cover. The glass wool was used for isolation of the bottom and sides of collectors. Closed-cell aluminum foams are used as a obstacle (passive element). These foams have porosity rate of 85%, dimensions (height - width) 50mm x 100mm and for each collector forty-four pieces are to used. Fig. 2 shows detailed view for aluminum foams. This case investigates the collector efficiency effect with changing of foams thicknesses.

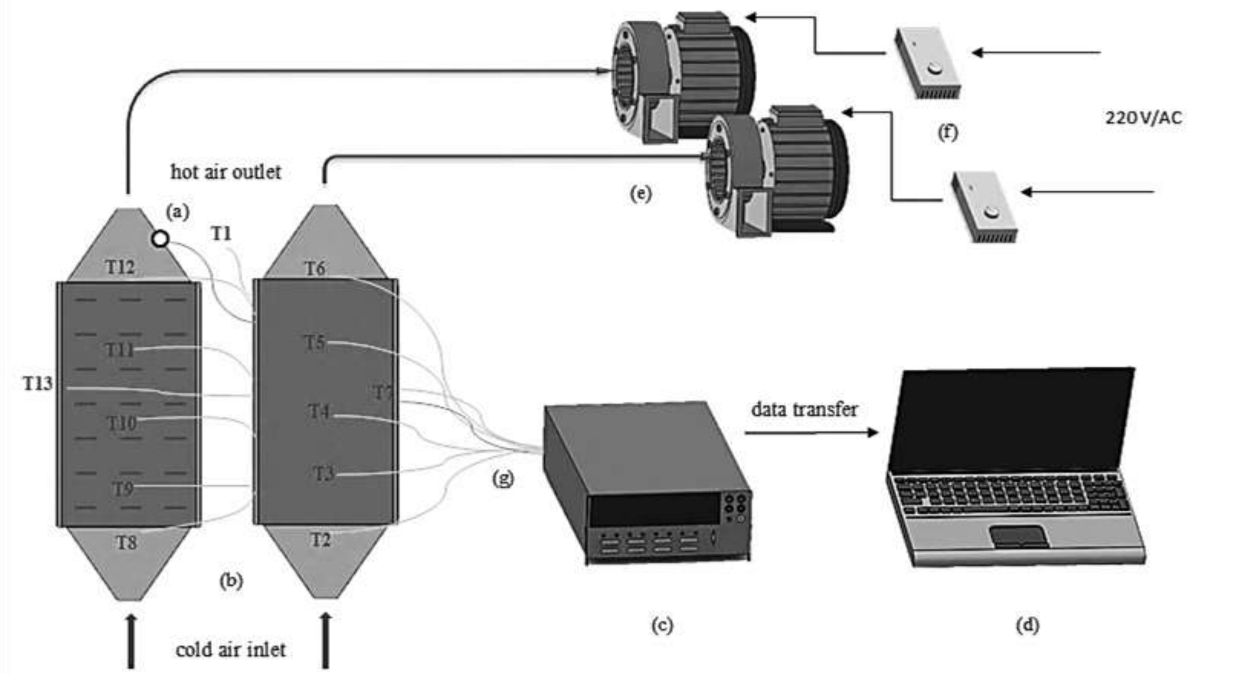


Fig. 1. Schematic view of experimental set-up. (a) pyranometer, (b) solar air collectors, (c) data logger, (d) computer, (e) radial fans, (f) velocity control devices, (g) thermocouples

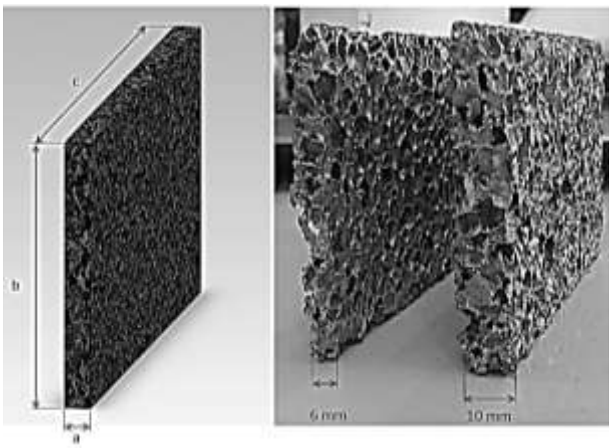


Fig. 2. Detailed view for porous materials

The air is supplied by a radial fan with a maximum of 0.55 kW. The radial fan placed at the outlet of the collectors sucked in the air. If the radial fan is placed at the inlet of collectors, the turbulence flow could have occurred because of blowing. The experiments were repeated for different air mass flowrate (0.016 and 0.025 kg/s). The ambient temperature, collector input output and surface temperatures are measured by using T- type thermocouples. The total solar radiation incident on the surface the collector was measured with a Kipp&Zonen CMP3 Pyranometer. The pyranometer was fixed beside the glass cover of the collector. Table 1 present the properties and sensitivity of the measuring devices such as Fan, Velocity controller, anemometer, thermocouples, data logger, pyranometer which are used in the experimental set-up. The solar air heater was oriented facing south and tilted to an angle of 38^o with respect to the horizontal to maximize the solar radiation incident on the glass cover. Air is circulated for 30 min prior to the period in which data are taken. The measured variables; ambient, inlet, outlet, surface temperatures of the working fluid circulating through the collector, wind velocity and relative humidity ratio were recorded.

Table 1. Properties and sensitivity of the measuring devices

FAN	
Voltage (V)	230
Frequency (Hz)	50
Power (W)	570
Mass flow rate (m^3/h)	1580
Rev (min^{-1})	2800
Current (A)	2.5
VELOCITY CONTROLLER	
Power (W)	1000
Current (A)	5
ANEMOMETER	
Air velocity measurement range	0.3 – 45 m/s
Air velocity stability	% ±3 - % ±0.20 m/sn
THERMOCOUPLES	
Temperature range	-200 ^oC / +300 ^oC
T-type	Cu-Co
Accuracy	% ±0.5
Diameter	0.51 mm
DATA LOGGER	
Multiplexer module	40 Channel
Accuracy	% ±1
PYRANOMETER	
Accuracy	% ±5

ANALYSIS OF TEMPERATURE DISTRIBUTIONS IN FLAT PLATE COLLECTORS

The collector performance is determined by three parameters as U_L, F_R and $(\tau\alpha)$. These parameters depend on construction materials, flow conduction and design type of the collector. Measuring the inlet and outlet fluid temperatures and the mass flow rate of air, the useful energy can be evaluated using Eq. (1). The efficiency of solar collectors is the ratio of useful energy obtained by the collector from the solar radiation incoming to collector. Following equations are used

$$\dot{Q}_u = \dot{m} C_p \left(T_o - T_i\right) \quad (1)$$

$$\eta = \frac{\dot{Q}_u}{A_c I} \quad (2)$$

where C_p is the specific heat of the air, $\dot{m}$ is a function of ρ, A and V, A_c is the area of the collector. If Eq. (1) is substituted in Eq. (2), it is arranged as below:

$$\eta = \frac{\dot{m} C_p \left(T_{out} - T_{in}\right)}{A_c I} \quad (3)$$

The efficiency of solar collectors can be formulated as the above. The calculation of heat loss from the collector to the surroundings is required for the design or the simulation of the performance of solar collectors. The major heat loss is realized from the top through the glass cover. The heat losses through the bottom and the sides are very small in comparison to the top heat loss and are easier to calculate. The top heat loss coefficient (U_t) is evaluated by considering convection and radiation losses from the absorber plate in the upward direction. There are two approaches for calculating U_t : (i) by numerical /iterative procedure / spread sheet; and (ii) approximate method (Akhtar and Mullick, 2007). The well-known empirical relation for calculating U_t proposed by Klein (Duffie and Beckman, 1991),

$$U_t = \left\{ \frac{N}{\dfrac{C}{T_p}\left[\dfrac{\left(T_p - T_a\right)}{N+f}\right]^e} + \frac{1}{h_w} \right\}^{-1} + \frac{\sigma\left(T_p + T_a\right)\left(T_p^{\,2} + T_a^{\,2}\right)}{\left(\varepsilon_p + 0.0059 N h_w\right)^{-1} + \left[\dfrac{2N + f - 1 + 0.133\varepsilon_p}{\varepsilon_g}\right] - N} \quad (4)$$

Where

$$f = \left(1 + 0.089 h_w - 0.1166 h_w \varepsilon_p\right)\left(1 + 0.07866 N\right) \quad (5)$$

$$C = 520\left(1 - 0.000051\beta^2\right) \text{ for } 0^o < \beta < 70^o \quad (6)$$

for $70^o < \beta < 90^o$ use $\beta = 70^o$

$$e = 0.430\left(\frac{1-100}{T_p}\right) \quad (7)$$

In its most general form, the Nusselt–Jurges correlation between the wind convection coefficient, h_w, and the parallel to the surface component of the wind velocity, V_w, which drives the convection can be written as

$$h_w = 5.678\left\{ a + b\left[\left(\frac{294.26}{273.16 + T_a}\right) V_w \middle/ 0.3048\right]^n \right\} \quad (8)$$

where *a*, *b*, *n* are empirical constants, and T_a the ambient temperature in ^oC. The correction in the innermost parenthesis is dictated by the fact that the original correlation was developed for a reference temperature of 294.26 K (21.1 ^{o}C). Actually, the original 1922 global correlation of Nusselt–Jürges based on their copper plate data, if written for SI units would have the form (Palyvos, 2008),

$$h_w = 7.13V_w^{0.78} + 5.35e^{-0.6V_m} \tag{9}$$

The following empirical equation to calculation the h_w, (Duffie and Beckman, 1991) has still been widely employed for flat plate solar collectors.

$$h_w = 5.7 + 3.8V_w \tag{10}$$

According to (Duffie and Beckman, 1991) , it is not reasonable to adopt Eq. (10) for plate lengths higher than 0.5 m. Even with such a recommendation, this equation has been applied to flat plate solar collectors greater than 0.5 m, due to the lack of a corresponding reliable equation. The back loss coefficient U_b is approximately;

$$U_b = \frac{1}{R} = \frac{k}{L} \tag{11}$$

where k and L are the insulation thermal conductivity and thickness, respectively. The edge heat loss coefficient (U_e), (Ucar and Inallı, 2006),

$$U_e = \frac{(UA)_{edge}}{A_c} \tag{12}$$

Where

$$(UA)_{edge} = \frac{k}{L_e} A_e L_{ct} \tag{13}$$

It is assumed that all losses occur to a common sink temperature T_a, the collector overall loss coefficient U_L is the sum of the top, bottom and edge loss coefficients;

$$U_L = U_t + U_b + U_e \tag{14}$$

The collector heat removal factor F_R, is the heat-transfer capacity of collector. It is measurement of the heat transferred to the fluid flowing though the absorber.

$$\frac{\dot{Q}_u}{A_c I} = F_R(\tau\alpha) - F_R U_L \frac{(T_i - T_a)}{I} \tag{15}$$

The instantaneous collector efficiency relates the useful energy to the total radiation incident on the collector surface by Eq. (15) (Duffie and Beckman, 1991). This equation indicates that a plot of instantaneous efficiency value $(T_i - T_a)/I$ will result in a straight line whose slope and intercept are $F_R U_L$ and $F_R(\tau\alpha)$, respectively. Thus, if the optical properties of the system are known F_R and U_L can be determined. But, Eq. (15) cannot be used for conventional air heaters where T_i is usually equal to T_a. Instead of this, instantaneous efficiency values are plotted against $(T_i - T_a)/I$ values. The slope and intercept of the straight line give $F_o U_L$ and $F_o(\tau\alpha)$ parameters, respectively. Here, F_o is collector heat gain factor expressed on the basis of air exit temperature. The modification converts Eq. (15) into the following form (Akpinar and Kocyigit, 2010).

$$\frac{\dot{Q}_u}{A_c I} = F_o(\tau\alpha) - F_o U_L \frac{(T_o - T_a)}{I} \tag{16}$$

Another important parameter used to explain collector performance is collector efficiency factor (F') (Duffie and Beckman, 1991)

$$F'U_L = -\frac{\dot{m}C_p}{A_c} \ln\left(1 - \frac{F_R U_L A_c}{\dot{m}C_p}\right) \tag{17}$$

The thermal resistance of glass cover is a small fraction of total thermal resistance to upward heat flow and can be approximated. The wind heat transfer coefficient has been considered as independent variable. For estimation of glass cover temperature following empirical equation has been given by (Kumar and Mullick, 2012),

$$T_g = T_a + h_w^{-0.38}\left[0.567\varepsilon_p - 0.403 + T_p/429\right]\left(T_p - T_a\right) \tag{18}$$

The analysis of uncertainties in experimental measurement and results is a powerful tool, particularly when it is used in the planning and design of experiments. In the experiment study, the temperatures, velocity of air, solar radiation and pressure loss were measured with appropriate instruments. During the measurements of the parameters, the uncertainties occurred were presented in Table 2. Considering relative uncertainties in the individual factors denoted by X_n, an uncertainty analysis was made using the following equation [Akpinar and Koçyiğit, 2010]:

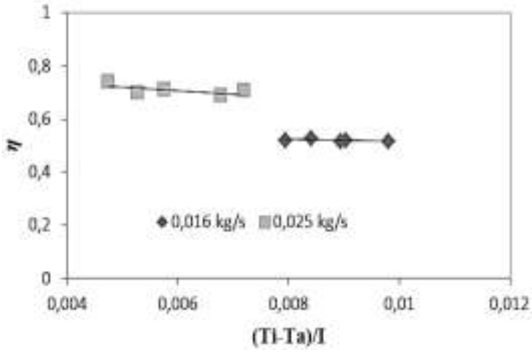
$$W_t = [(x_1)^2 + (x_2)^2 + \cdots (x_n)^2]^{1/2} \tag{19}$$

Table 2. Uncertainty parameters associated with the measured values.

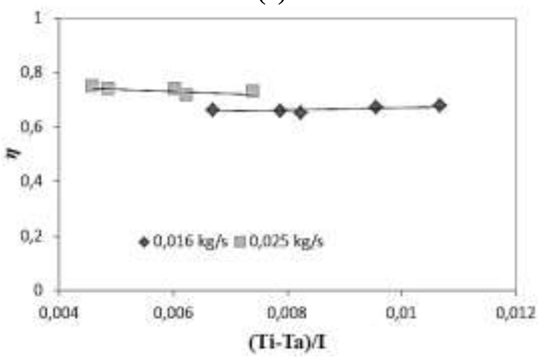
	The error occurred parameters	Totally error (%)
Temperatures (^oC)	Collector inlet temperature	± 0.547
	Surface temperature of absorber plate	± 0.547
	Collector outlet temperature	± 0.547
	Environment temperature	± 0.547
Time (min.)	Temperature values	± 0.05
Velocity (m/s)	Uncertainty in the air velocity measurement	± 0.244
Pressure (psi)	Uncertainty in the measurement of pressure loss	± 0.3
Others (%)	Uncertainty in reading values of table	± 0.1-0.2

RESULTS AND DISCUSSION

The closed-celled aluminum foams are inserted into solar air collecors to investigate the efficiency of collector in Elazig, Turkey prevailing weather conditions during the summer months of 2011. The collector slope was adjusted to 38^o which is considered suitable for the geographical location of Elazig. This study includes the investigation of five types of solar air collectors at two different air mass flowrate. The peak ΔT was achieved between 12:00 h and 13:00 h of the local time at which the solar intensity the outside condition solar intensity.



(a)



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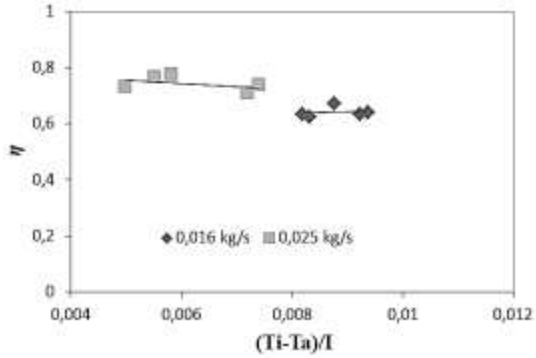

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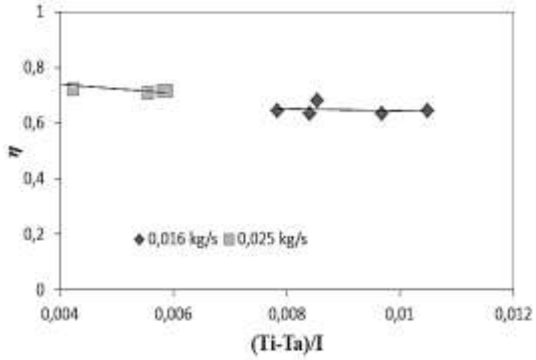

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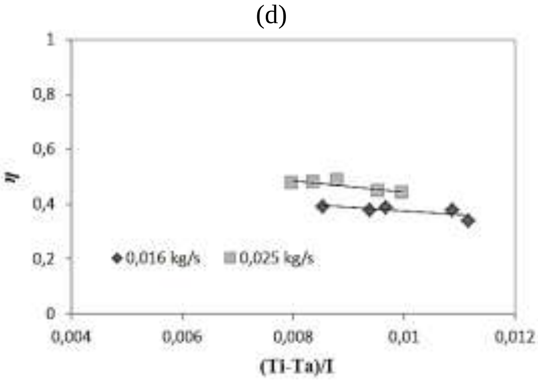


(e)

Fig. 3. Variation of collector efficiency with the temperature parameters $\left(T_i - T_a\right)/I$ at different mass flow rates (a) Case I, (b) Case II, (c) Case III, (d) Case IV and (e) Case V

The general test procedure is to operate in the test facility under nearly steady conditions, measure the data to determine Q_u from Eq. (1), and measure I, T_i and T_a which are need for analysis based on Eq. (15). Collector tests are performed with a range of inlet temperature conditions. To minimize the effects of heat capacity of collectors, the tests are usually made in nearly symmetrical pairs, one before and one after solar noon, with results of the pairs averaged (Duffie and Beckman, 1991). The values of conversion efficiency into heat energy of incident solar radiation on the collector were computed by using data from the daytime measurements between 11:30 and 13:30. The different results are obtained from the studied configurations which can be compared and the configuration giving a high thermal performance can be selected. Fig. 3 presents the collector efficiency versus the temperature parameter $\left(T_i - T_a\right)/I$ for the five SAC at air mass flowrate of 0.016and 0.025

kg/s. The maximum efficiencies for Type I, Type II, Type III, and Type IV are determined as 0.37, 0.53, 0.63, 0.60 and 0.62 for air mass flow rate 0.016 kg/s and 0.48, 0.73, 0.77, 0.68 and 0.75 for 0.025 kg/s, respectively. It is evident from Fig. 3 that the efficiency decreases with the increasing of temperature due to absorbing of heat inside the collector. At higher temperature parameters, the overall loss becomes lower. It can be seen from the figure that the collector efficiency increases with the increase of air mass flowrate considerably for all cases due to changing of kinetic energy of the flow. Thus, the higher temperature parameter is formed for Cases I and V. Because changing of type is also an effective parameter on kinetic energy of the flow. Thus, partitions play a role on control of heat and fluid flow. The key parameter here is that the temperature of absorber must be reduced to lower the forward heat loss by radiation as indicated by Akpinar and Kocyigit (2010).

The values of F_R, F_o, F' and U_L were calculated from the slopes and intercepts of the best lines and shown in Fig. 3. The heat gain factors values are shown in Fig. 4. The highest F_R, F_o and F' values are obtained by SACs with 6 mm staggered obstacles (Case III), whereas the lowest values are determined for the SAC without obstacles (Case I), i.e. flat plate collector. The F_R, F_o and F' values increased with increasing air mass flowrate and the lowest value is formed for Case I. This is again directly related with the control of kinetic energy of the flow with location of the partitions.

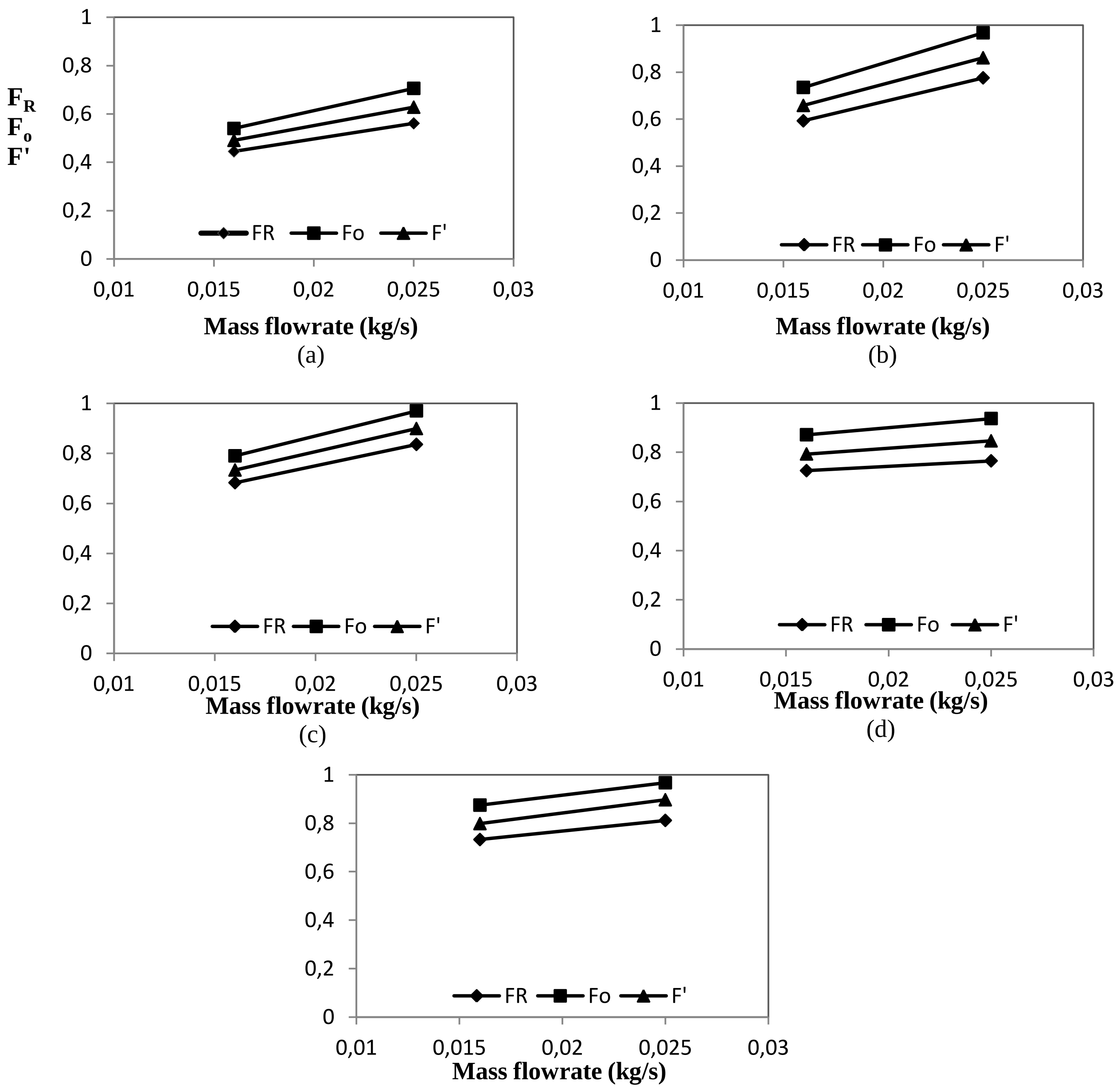
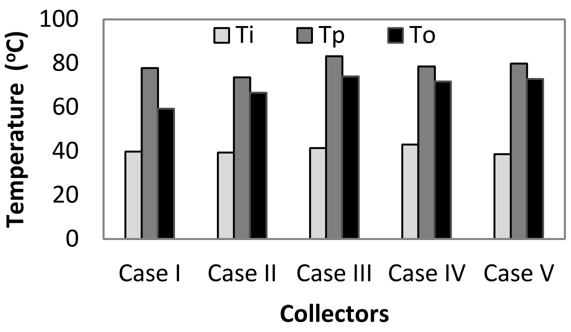
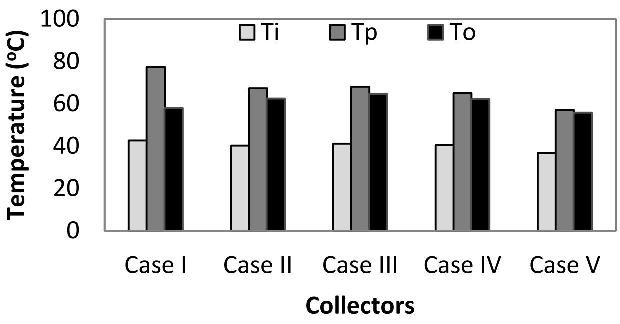


Fig. 4. Heat gain factors of solar air collectors (a) Case I, (b) Case II, (c) Case III, (d) Case IV and (e) Case V



(a)



(b)

Fig. 5. Maximum temperature values of the collectors (a) 0.016 kg/s, (b) 0.025 kg/s mass flowrates

Fig. 5 show that maximum temperature values for the five different type of solar air collectors (Case I-V) were 19.58, 27.21, 33.52, 28.64 and 33.29 ^{o}C for 0.016 kg/s, 15.31, 22.21, 23.36, 21.63 and 20.04 ^{o}C for 0.025 kg/s, respectively. The highest temperature difference occurred through Case III, while the lowest through Case I. As result of this study, the maximum temperature difference was obtained as 33.52 ^{o}C for the solar intensity of 986.233 W/m^2 with air mass flow rate of 0.016 kg/s by using Case III. As a comparison study with the literature that, the maximum temperature difference recorded by (Ozgen et al. 2009) were about 23 ^{o}C at solar radiation of 880 W/m^2 and mass flow rate of 0.02 kg/s. In this context, El-Sebaii et al., 2007 reported that the maximum value of ΔT was 48 ^{o}C when iron scraps were used as packed bed above the absorber plate and 39 ^{o}C for gravel used as a packed bed where the air mass flow rate was 0.0105 kg/s and solar intensity was 850 W/m^2. These graphs are approved the effectiveness of the location of partitions inside the collectors which are used as passive control element for heat and fluid flow.

The thermal efficiency values have been fluctuations while the solar radiation variation is parabolic distribution. The reason of these fluctuations is external factors such as the wind, clouds, dust etc. Fig. 6 shows the effects of mass flowrate on thermal efficiencies of the collectors. It is observed that the collector performance improves with increasing air mass flowrate due to incoming on more energy. Fig. 6 answers the question of how thickness of the obstacle affects the thermal efficiency and location of obstacles onto collector cover. It means that aluminum foams are behaved as thermal energy storage materials for the collector. The maximum values of the thermal efficiency are obtained for 10 mm as 0.72 and 0.71 for staggered and non-staggered at the mid-time of the day for 0.025 kg/s, respectively. However, the efficiency values for 6 mm are 0.77 for staggered and 0.73 for non-staggered locations of at the same flowrate, respectively. Finally, location of porous partition becomes more important for the higher mass flowrate than that of low flowrate due to increasing of kinetic energy and transporting of more energy from the sun to collector.

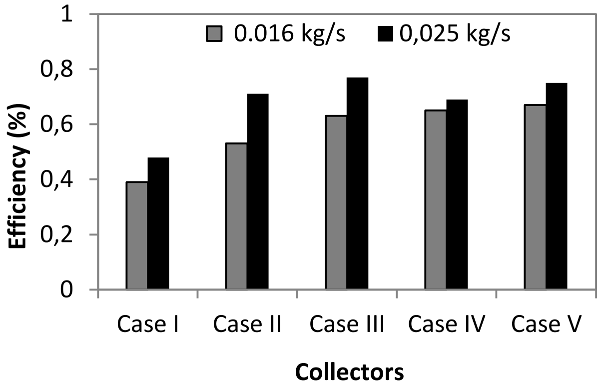


Fig. 6. Thermal efficiencies of the five collectors for different flowrates

The heat loss coefficients are increased with increasing of air mass flowrate as given in Fig. 7. In this figure, the heat loss coefficients are presented for five different collector types. The heat loss coefficients for Case I-V are determined as 7.82, 6.23, 3.80, 4.40, 4.20 W/m^{2}K and 10.86, 7.41, 4.95, 7.30, 6.80 W/m^{2}K for 0.016 and 0.025 kg/s air mass flowrate, respectively. However, the heat loss factors are found lower for Case III at two air mass flowrate. This result stems from the enhancements in heat gain have resulted efficient absorption of solar radiation and heat transfer between working fluid and absorbing surfaces inside the Case III. In this case, the radiation heat transfer becomes reduced and heat losses. This result is supported with literature of Akpinar and Kocyigit, 2011.

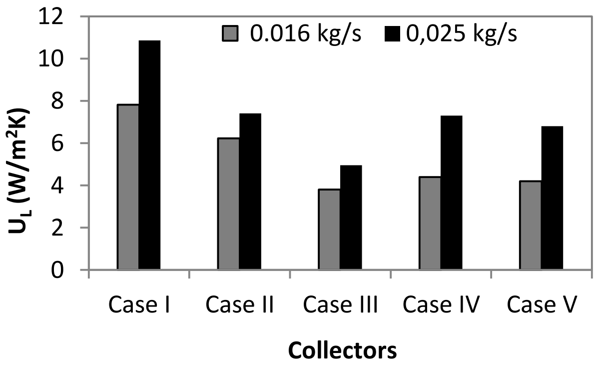


Fig. 7. Heat loss coefficient of the five collectors

Table 3 shows a comparison of efficiency values obtained from present work with available literature values with the same conditions such as single pass flow, single glass cover, collector dimensions and almost for same mass flow rate. As given in the Table, the efficiency value for this work as high as Akpinar and Kocyigit (2010) and the system has more efficient from other work. It means that the technique of porous partition location can be useful for solar air heating systems.

Table 3. Comparison of obtained results with literature for same single glass cover, collector dimensions and single pass flow conditions at almost same mass flowrate

AUTHORS	FLOW	GLASS	η (%)
Akpinar and Kocyigit (2010)	Single pass flow	Single glass cover	82
Gill et al. (2012)	Single pass flow	Single glass cover	37.45
Omojaro and Aldabbagh (2010)	Single pass flow	Single glass cover	59.62
Present Study	Single pass flow	Single glass cover	78

Thus, the chosen method of location of porous plate into solar air collector can be used as both scientifically and industrially to improve and control of heat and fluid flow for solar air heating processes.

CONCLUSIONS

In the present study, performances of four solar air collectors were tested. According to the experimental results, the type of solar air collectors has been introduced for increasing the heat-transfer area, leading to improved thermal efficiencies. The following conclusions can be derived:

- The efficiency of solar air collectors increases depending on the surface geometry, absorber plate and air mass flowrate of the collector.
- The changing of heat transfer area of the surface geometry, affects the local velocities inside the collector and efficiency of collector is affected from this variation.
- Based on the efficiencies of the obstacle located collectors are higher values than that of the collector without obstacle (Case I). The highest heat loss coefficient value is 9.98 W/m^2K for Case I at the 0.025 kg/s air mass flow rate, while the lowest value is 8.48 W/m^2K for Case III at the 0.016 kg/s.
- The highest and lowest collector efficiencies are obtained for Case III with 77% at the 0.025 kg/s and Case I with 0.37% at the 0.016 kg/s, respectively.

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