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Experimental Investigation of Various Type Absorber Plates for Solar Air Heaters

Nuri ÇAĞLAYAN[a], Zeliha Deniz ALTA[b], Can ERTEKİN[b]

[a] *Akdeniz University, Faculty of Engineering, Department of Mechatronics Engineering, Campus, 07058, Antalya, TURKEY*

[b] *Akdeniz University, Faculty of Agriculture, Department of Agricultural Machinery, Campus, 07058, Antalya, TURKEY*

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Corresponding Author: Nuri ÇAĞLAYAN, E-mail: nuricaglayan@akdeniz.edu.tr, Tel: +90 (242) 310 60 16
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ABSTRACT

In this study, four different types' absorber plates were designed and compared of their energetic performances. These absorber plates were formed as a flat plate (Type I), V-shaped (Type II), wedge-shaped (Type III) and wavy-shaped (Type IV). Each type absorber plate was manufactured in both aluminum (Al) and copper (Cu) materials. Energy efficiencies of the heaters were investigated with airflow velocities of 2, 3 and 4 m s^{-1} experimentally and compared with each other. The results showed that efficiency of the heater with the copper absorber plate better than aluminum plate however, the resulting air temperature from heater with aluminum absorber plate higher than cooper plate. The experimental results have shown that Type IV and Type II achieved the highest energy efficiency, respectively.

Keywords: Solar air heater; Absorber plate; Airflow velocity; Energy efficiency

Hava Isıtıcılı Güneş Kollektörleri İçin Farklı Tip Yutucu Plakaların Deneysel İncelenmesi

ESER BİLGİSİ
Araştırma Makalesi
Sorumlu Yazar: Nuri ÇAĞLAYAN, E-posta: nuricaglayan@akdeniz.edu.tr, Tel: +90 (242) 310 60 16
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ÖZET

Bu çalışmada hava ısıtıcı kollektörler için dört farklı tip yutucu plaka tasarlanmış ve bunların enerjik performansları karşılaştırılmıştır. Bu yutucular, düz (Tip I), V (Tip II), trapez (Tip III) ve dalga (Tip IV) şeklindeki plakalardan oluşmaktadır. Her tip yutucu hem alüminyum (Al) hem de bakırdan (Cu) imal edilmiştir. Isıtıcıların enerji verimleri 2, 3 ve 4 m s^{-1} hava hızında deneysel olarak incelenmiş ve karşılaştırılmıştır. Elde edilen sonuçlara göre, bakır yutucu plakalı ısıtıcının, alüminyum yutucu plakalıdan daha verimli olduğu, ancak alüminyum yutucu plakalı ısıtıcının çıkış hava sıcaklığının bakır plakalıdan daha sıcak olduğu görülmüştür. Deneysel sonuçlara göre en yüksek enerji verimleri sırasıyla IV.ve II. Tip yutucu plakalı ısıtıcılarda elde edilmiştir.

Anahtar Kelimeler: Hava ısıtmalı kollektör; Yutucu plaka; Hava akış hızı; Enerji verimi

1. Introduction

The use of solar air heater has been increasing in recent years, because of their simplicity, cheapness, ease of their maintenance and operation, friendly for environment and non-fuel operation. Such heaters are implemented in many applications that require low to moderate temperature below 60 °C (Gupta & Kaushik 2008).

In agricultural area, the main application of solar air heater is drying by means of solar drying. Using a solar dryer, the drying time can be shortened by about 65% compared to natural sun drying because, inside the dryer, it is warmer than outside; the quality of the dried products can be improved in terms of hygiene, cleanliness, safe moisture content, color and taste; the product is also completely protected from rain, dust, insects, rodents; and its payback period ranges from 2 to 4 years depending on the rate of utilization (Eliçin & Saçılık 2005). The quality of dried product is mostly dependent on drying air temperature, velocity and drying time (Aktaş et al 2012; Tülek & Demiray 2014). On the other hand, the thermal efficiency of solar air heater has been found to be poor due to the low heat transfer capacity and low heat conductivity of air. Therefore, several researchers have studied to design several types of solar air heaters to improve their performance (Youcef-Ali 2005; Kurtbaş & Turgut 2006; Gao et al 2007; Esen 2008; Varol & Öztop 2008; Luna et al 2010).

Ayadi et al (2014) investigated the performance of two components of a solar drying unit (collector and storage system) without drying energy supplement. They used a V-corrugated absorber and single glazing in the air collector and metal parallelepiped system for storage unit. According to their experimental results, average collector efficiency and outlet temperature were found as 30.52% and 54.06 °C, respectively.

Karim & Hawlader (2006) presented a performance study on V-groove solar air collector for drying application and V-corrugated collector was found better thermal efficiency (about 12% more efficiency) compared to flat plate collector. In their study the height of the 'V' and the dimensions of absorber plate were selected as 10 cm and 1.8 m x 0.7 m, respectively. Absorber material was black-painted mild steel and the number of glazing was one. Karim et al (2012) also developed a mathematical model for this type collector and compared the simulation results carried out using MATLAB with experimental study.

Ho et al (2011) investigated the collector efficiency of upward-type double-pass flat plate solar air heaters with fins attached and external recycle theoretically. Collector efficiency increases as airflow rate, number of fins attached and incident solar radiation increase. Considerable improvement in collector efficiency is also obtainable if the operation is carried out with external recycling.

Ben-Amara et al (2005) presented experimental results of a new-design plate collector used to heat air in a new desalination humidification–dehumidification process. In addition, the effects of different parameters on the collector efficiency, such as solar radiation, wind velocity, ambient temperature, air mass flow rate, air temperature and humidity through the collector was investigated.

Kurtbaş & Durmuş (2004) investigated the effect of airflow line on the performance of solar collectors with absorber slices having four different surface geometries. The efficiency of collectors increases depending on the collector surface geometry and extension of the airflow line. As a result, it appears that if the surface roughness is increased, the heat transfer and pressure loss increases.

Karslı (2007) determined the first and second law efficiencies of four types of air heating flat plate solar collectors; finned with an angle of 75°, finned with an angle of 70°, with tubes and a base collector. As a result, the highest energy efficiency (80%) and air temperature rise were found for the collector finned with angle of 75°, whereas the lowest values were obtained for the base collector.

Mittal et al (2007) compared effective efficiency of a solar collector with different geometry type having roughness elements on the absorber plate. In this study, the relative roughness height is considered as strong parameter of roughness element for effective efficiency of solar collector. It is observed that among all the roughness elements investigated, the inclined ribs having low values of roughness height resulted in better effective efficiency in higher range of Reynolds number (more than 12000). However, in lower range of Reynolds number (less than 12000), the better effective efficiency is observed for the solar air heaters having expanded metal mesh as artificial roughness element. Further, it is observed also that the effective efficiency of solar air heater is better than the roughened solar air heaters in the range of very high Reynolds number.

Karwa & Chauhan (2010) presents results of the performance of solar air heater with 60° v-down discrete rectangular cross-section repeated rib roughness on the airflow side of the absorber plate. The effects of various ambient, operation and design parameters on the thermal and effective efficiencies of air heaters have been investigated. The study shows that, at air mass flow rates less than about 0.04 kg s^{-1} m^{-2} of the absorber plate, roughened duct solar air heaters provide significant performance advantage over the smooth ducted solar air heater. At the mass flow rate of about 0.045 kg s^{-1} m^{-2}, the effective efficiencies of the roughened and smooth duct solar air heaters are practically the same.

Alta et al (2010) investigated the effects of the mass flow rate (25, 50 and 100 m^3 m^{-2} h^{-1}) and title angle (0°, 15° and 30°) on the efficiencies of different types of designed flat-plate solar air heaters. It was found that attaching fins on absorber surface increases the efficiency of solar air heater. The energy efficiency of the heater also improved with increasing airflow rates due to an enhanced heat transfer to the airflow while temperature difference of fluid decreases at constant tilt angle.

This paper presents a comparison of energy efficiencies of solar air heaters having plates with four different geometry types, two different materials (Al and Cu) and for 2, 3 and 4 m s^{-1} airflow velocities. Comparisons of the energetic and economic advantages of the collectors with absorber plate in different dimensions, geometries and materials are distinctive properties of this study.

2. Material and Methods

2.1. Experimental setup and measurement procedure

In the study, two experimental solar collectors were used and mounted as shown in Figure 1. A single glazing was chosen in order to maximize the radiation impact on the absorber. Dimension of the collectors are 1.92 x 0.82 x 0.10 m and they have insulation thickness of 0.05 m in the bottom and sides. The gap between the absorber plate and bottom is 0.043 m. Al and Cu absorber plates thickness of 2 mm and their surfaces are painted matt black. All plates are designed as a portable.

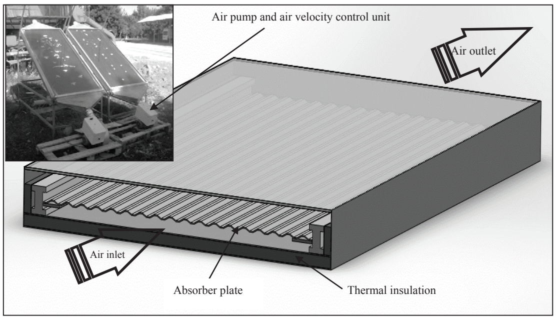

Figure 1- The experimental solar air heater

Şekil 1- Deneysel hava ısıtmalı güneş kollektörü

The schematic diagrams and cross-sections of the absorber types are presented in Figure 2. The surface areas of collectors are 1.5744, 1.7602, 1.6412 and 1.6284 m^2 and the airflow areas are 0.033616, 0.033426, 0.033550, 0.033558 m^2, respectively.

In this study, collector inlet and outlet air temperature, ambient temperature, airflow rate, solar radiation, pressure drop and wind velocity was measured and all of data recorded by a data logger. A radial fan with a capacity of 0.41 m^3 s^{-1} was used for each collector to provide the airflow. The fan speed and airflow rate can be adjusted by an electrical controller unit.

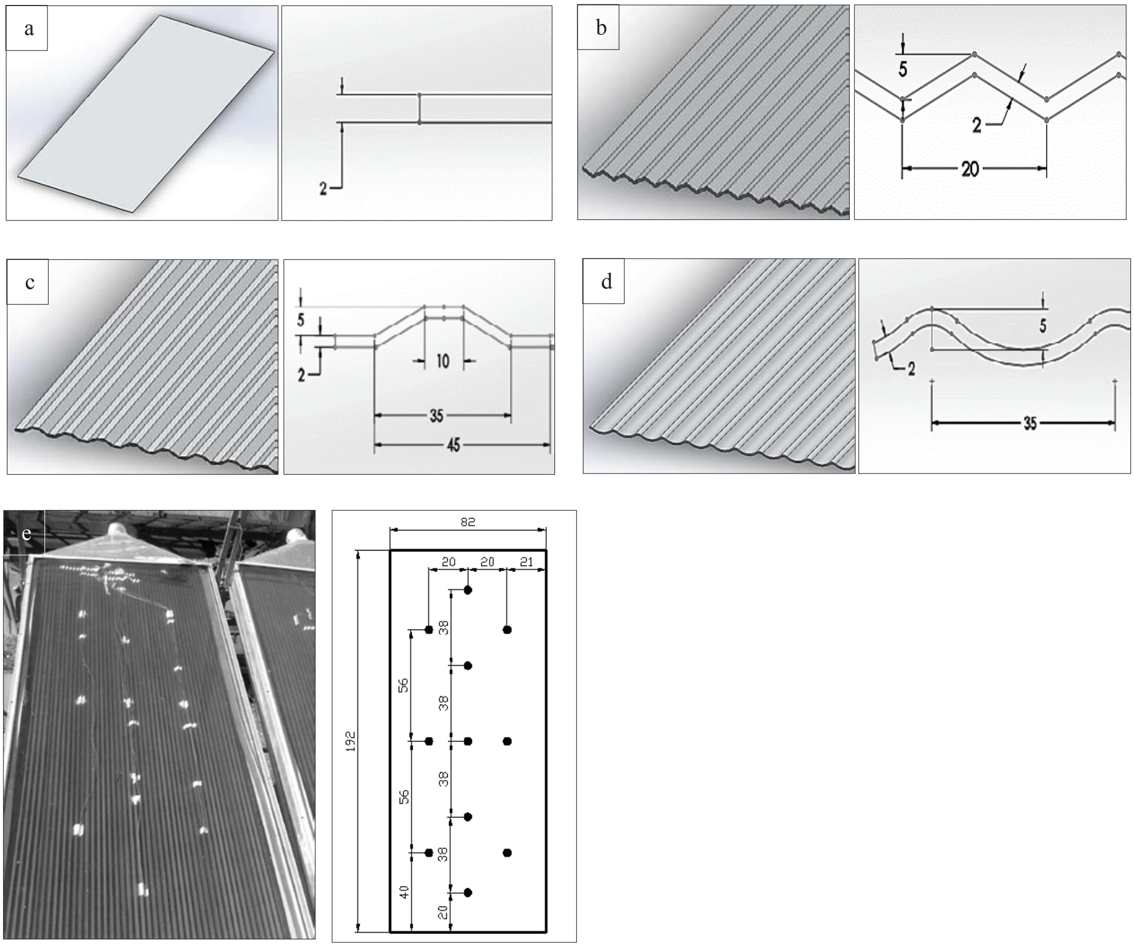

Figure 2- The absorber plates which are used in experiments: a, flat plate (Type-I); b, V-shaped (Type-II); c, wedge-shaped (Type-III); d, wavy-shaped (Type-IV) and e, K-type thermocouples which are fixed on a plate

Şekil 2- Denemelerde kullanılan yutucu plakalar: a, Düz plaka (Tip-I); b, V şekilli (Tip-II); c, trapez (Tip-III); d, dalga şekilli (Tip-IV) ve e, hava ısıtmalı güneş kollektörü üzerine yerleştirilmiş K-tipi ısıl çiftler

Inlet and outlet air temperature, absorber surface and ambient temperature were measured using K-type thermocouples. Wind velocity was measured using a cup anemometer (Delta-T A100 R model, accuracy: 1% ± 0.1 m s^{-1}). Anemometer was placed about 1 m above the collector. A flow meter (Testo 405, accuracies: ± 0.1 m s^{-1} ± 5% of m.v. at 0-2 m s^{-1}) was used to measure the air inlet velocity for the solar collector. Incident radiation on the collector was measured using a global radiation sensor (Delta-T ES2 accuracy: ±3% at 20 °C). The radiation sensor was placed on the glass cover of the collector. All of sensors were connected to a data logger Delta-T Model DL2e and measurements were stored in 5 minutes intervals.

In order to obtain best thermal efficiency of the collector the sunshine should be fully used through the whole year. For a collector operating through the whole year the best effect would be obtained when the panel was set with tilt angle of 35˚. Collector parameters are summarized in Table 1.

2.2. Energy analysis and uncertainty

The energy balance for solar air heaters are given in Equation 1 (Hottel & Woertz 1942; Duffie & Beckman 2006; Karwa & Chauhan 2010).

$$\eta_I = Q_u / (A_c\ G_T) \tag{1}$$

Where; Q_u is the useful energy gain; A_c is the heater aperture area and G_T is the solar radiation intensity on the heater surface. The useful energy gain can be calculated by the Equation 2.

$$Q_u = A_c\ F_R\ \left[S - U_L\ (T_i - T_a)\right] \tag{2}$$

Where; F_R is the heat removal factor; S is the solar energy absorbed by heater; U_L is the overall heat loss coefficient; T_i is the inlet air temperature and T_a is the ambient air temperature. The heat removal factor of the collector is defined as shown in Equation 3.

$$F_R = \left[(m\ c_p)/(A_c\ U_L)\right]\left\{1 - \exp\left[-A_c\ U_L\ F'/(m\ c_p)\right]\right\} \tag{3}$$

Where; m is airflow rate; c_p is the specific heat of air at constant pressure and $F¢$ is the heater efficiency factor as shown in Equation 4.

$$S = G_T \left\{\frac{\tau\ \alpha_p}{\left[1-\left(1-\alpha_p\right)\rho_c\right]}\right\} \tag{4}$$

Where; t is transmittance of transparent cover; a_p is absorptance of absorber plate and r_c is reflectance of transparent cover. The overall heat loss coefficient is the sum of top, bottom and edge heat loss coefficients as shown in Equation 5.

$$U_L = U_t + U_b + U_e \tag{5}$$

The top heat loss coefficient is presented in Equation 6 (Duffie & Beckman 2006).

Table 1- The properties of experimental solar air heater

Çizelge 1- Deneysel hava ısıtmalı kollektörün özellikleri

Collector parameters	*Value*
Absorber material	Copper or aluminum
Plate thickness	2 mm
Absorber coating	Dull black paint
Glazing	Single glass (thickness of 4 mm)
Agent fluid in flow ducts	Air
Width of the duct, W	0.9 m
Collector side wall height, h_e	0.1 m
Air flow duct height, D	43 mm
Length of the collector, L	1.9 m
Emissivity of the glass cover, ε_g	0.85
Emissivity of the absorber plate, ε_p	0.95
Emissivity of the bottom plate, ε_b	0.95
Tilt angle, β	35°
Insulation thicknesses, t_b, t_e	50 mm
Thermal conductivity of insulation, λ	0.043 W m^{-1} K^{-1}
Heat transfer coefficient of copper, λ_{Cu}	385 W m^{-1} K^{-1}
Heat transfer coefficient of aluminum, λ_{Al}	210 W m^{-1} K^{-1}
Heat capacity of copper, $c_{p, Cu}$	0.385 J g^{-1} $°C^{-1}$
Heat capacity of aluminum, $c_{p, Al}$	0.90 J g^{-1} $°C^{-1}$

$$U_t = \left\{ \frac{N}{\left(\frac{C}{T_{pm}}\right)\left[\frac{\left(T_{pm}-T_a\right)}{(N+f)^e}\right]} + \frac{1}{h_w} \right\}^{-1} + \frac{\sigma\left(T_{pm}+T_a\right)\left(T_{pm}{}^2+T_a{}^2\right)}{\frac{1}{\varepsilon_p+0.00591\,N\,h_w}+\frac{2\,N+f-1+0.133\,\varepsilon_p}{\varepsilon_g}-N} \tag{6}$$

Where; N is number of transparent cover; $C = 520\,(1-0.000051\,\beta^2)$; b is the heater tilt angle; T_{pm} is temperature of absorbing plate, $f = (1+0.089\,h_w - 0.1166\,h_w\,\varepsilon_p)\,(1+0.07866\,N)$; e_p is emissivity of absorbing plate, $h_w = 5.7+3.8\,V_r$; V_r is wind velocity; $e = 0.430\left[1-\left(100/T_{pm}\right)\right]$; e_g is emissivity of transparent cover. The bottom and edge heat loss coefficients are presented in Equation 7 and 8, respectively:

$$U_b = \lambda / L \tag{7}$$

$$U_e = (\lambda\; c\; h / L)\,(1/A_c) \tag{8}$$

Where; l is thermal conductivity of insulation material; L is the thickness of insulation material; c is the perimeter of heater; h is the heater height. The collector efficiency factor $F¢$ is calculated by the Equation 9 and 10.

$$F' = \alpha / (\alpha + U_L) \tag{9}$$

$$\alpha = Nu\;\lambda / D_h \tag{10}$$

Errors and uncertainties in the experiments can arise from instrument selection, condition, calibration, environment, observation, reading and test planning. In these experiments, mass flow rate, ambient, inlet and outlet air temperatures, wind speed and solar radiation were measured with appropriate instruments. The result R is a given function in terms of the independent variables. Let w_R be the uncertainty in the result and $w_1, w_2, \ldots, w_n$ be the uncertainties in the independent variables. If the uncertainties in the independent variables are all given with same odds, then uncertainty in the result having these odds is given in Equation 11 (Akpınar 2006).

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \ldots.. + \left(\frac{\partial R}{\partial x_n} w_n\right)^2\right]^{1/2} \tag{11}$$

For example, the total uncertainty in the measurement of the ambient air temperature (w_{Ta}) may be calculated as from the Equation 12 and 13 (Ayadi et al 2014).

$$w_{Ta} = \left[(w_{th})^2 + (w_{cp})^2 + (w_{tm})^2\right]^{1/2} \tag{12}$$

$$w_{Ta} = \left[(0.25)^2 + (0.05)^2 + (0.25)^2\right]^{1/2} = 0.36 \tag{13}$$

Where; w_{th}, arisen from thermocouple; w_{cp}, arisen from connection points; w_{tm}, arisen from temperature measurement.

During the experiments, total uncertainties of the measured parameters were presented in Table 2.

Table 2- Uncertainties of the parameters during experiments

Çizelge 2- Denemelerdeki parametrelerin belirsizlikleri

Parameter (unit)	*Comment*
Uncertainty in the measurement of temperature	
- Ambient air temperature (°C)	±0.368
- Inlet air temperature (°C)	±0.652
- Outlet air temperature (°C)	±0.368
Uncertainty in the measurement of mass flow rate (m s^{-1})	±0.165
Uncertainty in the measurement of wind speed (m s^{-1})	±0.152
Uncertainty in the measurement of solar radiation (W s^{-2})	±0.531

Total uncertainty for collector thermal efficiency can be written as in Equation 14 and 15.

$$w_\eta = \left[\left(\frac{\partial \eta}{\partial \dot{m}} w_{\dot{m}} \right)^2 + \left(\frac{\partial \eta}{\partial T_a} w_{T_a} \right)^2 + \left(\frac{\partial \eta}{\partial T_{in}} w_{T_{in}} \right)^2 + \left(\frac{\partial \eta}{\partial T_{out}} w_{T_{out}} \right)^2 + \left(\frac{\partial \eta}{\partial V_r} w_{V_r} \right)^2 + \left(\frac{\partial \eta}{\partial G_T} w_{G_T} \right)^2 \right]^{1/2} \quad (14)$$

$$w_\eta = \left[(0.165)^2 + (0.368)^2 + (0.652)^2 + (0.368)^2 + (0.152)^2 + (0.531)^2 \right]^{1/2} = 1.014 \quad (15)$$

3. Results and Discussions

Experiments were performed between 19th July and 1st September 2012 at the Akdeniz University, Antalya, Turkey (36° N latitude; 30° E longitude). All the heaters were placed facing south and with a tilt angle of 35°. The experiments were carried out at the same time periods between 08:30 and 17:00 of the days for a fixed air flow rate and the data collected each 5 min during the experiments, but the results were discussed and evaluated where the solar radiation are more than 630 W m^{-2} (Ion & Martins 2006).

The energy efficiencies of Al and Cu absorber types (Type I, II, III and IV) were compared with each other for airflow velocity of 2 m s^{-1} (Figure 3-6).

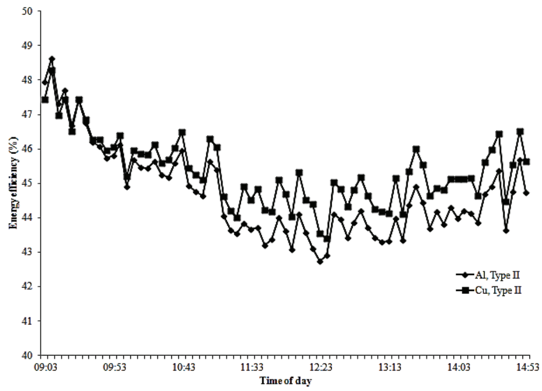

Figure 4- Energy efficiencies of the Type-II collectors with the Al and Cu absorber plate (2 m s^{-1} airflow velocity)

Şekil 4- Al ve Cu yutucu plakalı Tip-II kollektörlerinin enerji verimleri (2 m s^{-1} hava hızında)

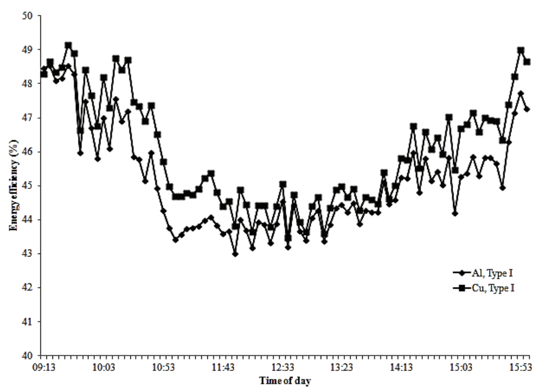

Figure 3- Energy efficiencies of the Type-I collectors with the Al and Cu absorber plate (2 m s^{-1} airflow velocity)

Şekil 3- Al ve Cu yutucu plakalı Tip-I kollektörlerinin enerji verimleri (2 m s^{-1} hava hızında)

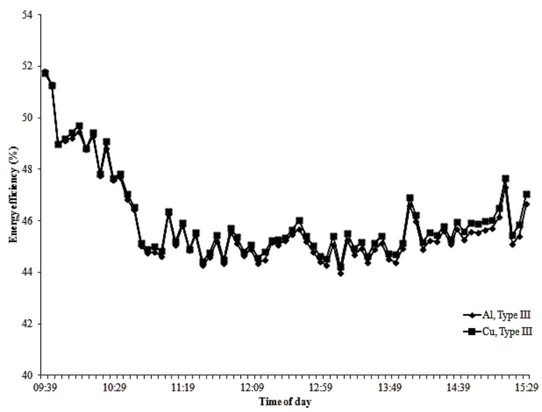

Figure 5- Energy efficiencies of the Type-III collectors with the Al and Cu absorber plate (2 m s^{-1} airflow velocity)

Şekil 5- Al ve Cu yutucu plakalı Tip-III kollektörlerinin enerji verimleri (2 m s^{-1} hava hızında)

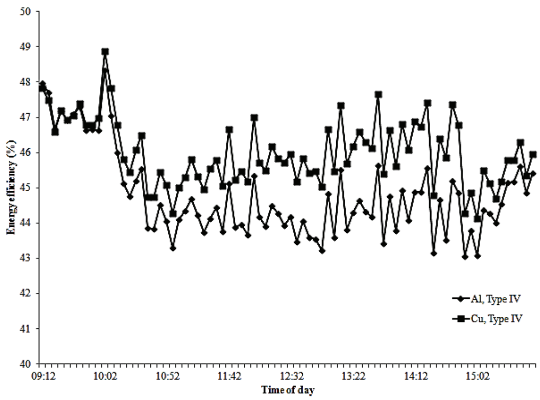

Figure 6- Energy efficiencies of the Type-IV collectors with the Al and Cu absorber plate (2 m s^{-1} airflow velocity)

Şekil 6- Al ve Cu yutucu plakalı Tip-IV kollektörlerinin enerji verimleri (2 m s^{-1} hava hızında)

Maximum and minimum energy efficiencies of the collectors are 48.54 and 43.02 for the Type I with Al absorber plate, 49.14 and 43.47 for the Type I with Cu absorber plate, 48.63 and 42.74 for the Type II with Al absorber plate, 48.70 and 43.41 for the Type II with Cu absorber plate, 51.82 and 43.97 for the Type III with Al absorber plate, 51.84 and 44.22 for the Type III with Cu absorber plate, 48.33 and 43.05 for the Type IV with Al absorber plate, 48.88 and 44.13 for the Type IV with Cu absorber plate, respectively. Minimum efficiency values are shown in the midday in consequence of the reduction of the rates between the inlet and outlet temperature difference and solar radiation.

The highest energy efficiencies were obtained in the air collectors with Cu absorber plate. The similar results were found of 3 and 4 m s^{-1} airflow velocities (Table 3).The differences of energy efficiency results between the collectors with Al and Cu absorbers are about 1% and quite close especially in Figure 5, as a result of the proximity of the surface and airflow areas of the plates.

The average efficiencies of the collectors with Al absorber were ordered as $h_{Type\ III}>h_{Type\ I}>h_{Type\ II}>h_{Type\ IV}$ for 2 m s^{-1} airflow rate,$h_{Type\ III}>h_{Type\ II}>h_{Type\ IV}>h_{Type\ I}$ for 3 m s^{-1} airflow rate and $h_{Type\ III}>h_{Type\ IV}>h_{Type\ II}>h_{Type\ I}$ for 4 m s^{-1} airflow rate. Type III was found more effective than the others for all airflow rate. The orders of the energy efficiencies of the collectors with Cu absorber were found as $h_{Type\ IV}>h_{Type\ III}>h_{Type\ I}>h_{Type\ II}$ for 2 m s^{-1} airflow rate,$h_{Type\ IV}>h_{Type\ II}>h_{Type\ III}>h_{Type\ I}$ for 3 m s^{-1} airflow rate and $h_{Type\ IV}>h_{Type\ I}>h_{Type\ II}>h_{Type\ III}$ for 4 m s^{-1} airflow rate. Type IV was found more effective than the others for all airflow rates.

Table 3- The achieved values of efficiencies of Type-I, II, III and IV collectors in 2, 3 and 4 m s^{-1} airflow velocities and under solar radiation

Çizelge 3- Tip- I, II, III ve IV kollektörlerinin 2, 3 ve 4 m s^{-1} hava hızında ve güneş ışınımı altında elde edilen verim değerleri

Shape of absorber plate		*Airflow velocity*	*Wind velocity*	$T_{p,1}$	$T_{p,2}$	T_a	G_T	$T_{i,1}$	$T_{o,1}$	$T_{i,2}$	$T_{o,2}$	$h_{I,1}$	$h_{I,2}$	dT/G_T	dT_2/G_T
1st collector	*2nd collector*	*m s^{-1}*	*m s^{-1}*	*°C*	*°C*	*°C*	*W m^{-2}*	*°C*	*°C*	*°C*	*°C*	*%*	*%*		
Type I (Al)	Type I (Cu)	2	1.69	98.66	96.61	37.46	878	41.98	79.08	41.28	75.30	44.76	45.64	0.0363	0.0357
		3	1.65	99.63	96.35	40.10	856	44.62	78.29	42.77	74.05	45.03	46.61	0.0393	0.0380
		4	1.60	100.07	97.54	41.00	925	45.10	73.34	43.39	73.30	45.59	46.87	0.0367	0.0331
Type II (Al)	Type II (Cu)	2	1.97	111.69	108.07	36.87	939	41.62	79.46	40.86	77.50	44.70	45.36	0.0403	0.0390
		3	2.35	100.98	90.74	34.93	862	38.51	75.49	38.03	75.13	45.65	46.68	0.0430	0.0425
		4	2.16	99.41	94.15	35.52	868	39.23	75.44	39.76	73.77	45.79	46.85	0.0418	0.0392
Type III (Al)	Type III (Cu)	2	2.02	101.02	101.30	35.46	826	39.47	76.63	39.11	74.24	45.56	45.78	0.0450	0.0428
		3	2.25	94.18	91.60	34.59	812	37.44	73.62	37.44	71.46	46.43	46.63	0.0436	0.0402
		4	2.37	96.37	96.77	35.41	879	37.74	72.86	38.09	70.10	46.45	46.67	0.0408	0.0380
Type IV (Al)	Type IV (Cu)	2	1.40	114.45	103.46	43.37	899	47.92	81.54	47.08	77.25	44.49	45.86	0.0360	0.0336
		3	1.41	113.21	98.63	43.31	911	46.90	80.50	46.13	76.73	45.23	46.81	0.0370	0.0334
		4	1.03	108.98	93.39	43.31	865	46.33	80.27	46.15	76.61	45.80	47.18	0.0407	0.0354

The energy efficiencies of the collectors with corrugated absorber plate were expected to find higher than collector with flat plate in all circumstances. Increasing the airflow areas of the absorber plates will help to understand the reasons of these differences.

In Figure 7-10, the results are also plotted for Al absorber plate at 2, 3 and 4 m s^{-1} airflow velocities. As is expected, with increasing the mass flow, the thermal efficiency increased.

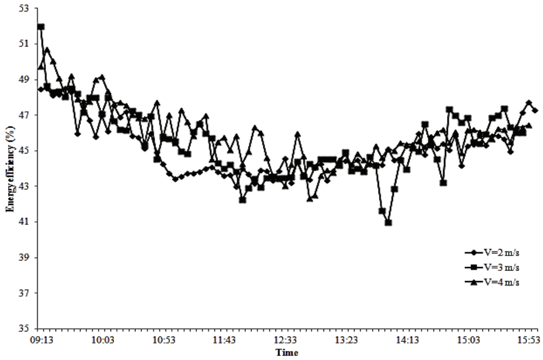

Figure 7- Energy efficiencies of the Type-I collectors with the Al absorber plate in in 2, 3 and 4 m s^{-1} airflow velocity

Şekil 7- 2, 3 ve 4 m s^{-1} hava hızlarında Al yutucu plakalı Tip-I kollektörlerinin enerji verimleri

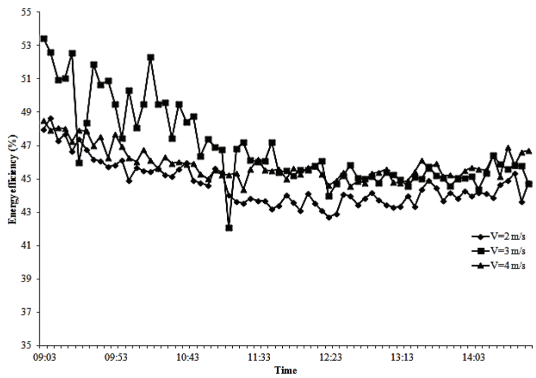

Figure 8- Energy efficiencies of the Type-II collectors with the Al absorber plate in in 2, 3 and 4 m s^{-1} airflow velocity

Şekil 8- 2, 3 ve 4 m s^{-1} hava hızlarında Al yutucu plakalı Tip-II kollektörlerinin enerji verimleri

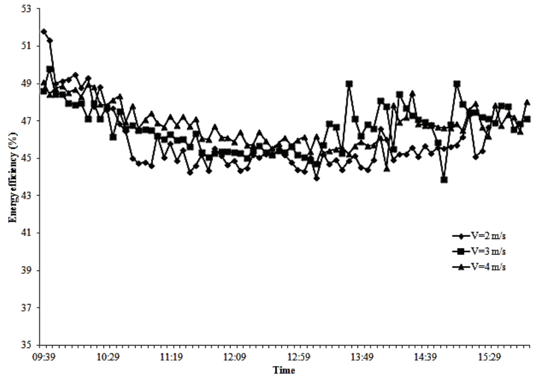

Figure 9- Energy efficiencies of the Type-III collectors with the Al absorber plate in 2, 3 and 4 m s^{-1} airflow velocity

Şekil 9- 2, 3 ve 4 m s^{-1} hava hızlarında Al yutucu plakalı Tip-III kollektörlerinin enerji verimleri

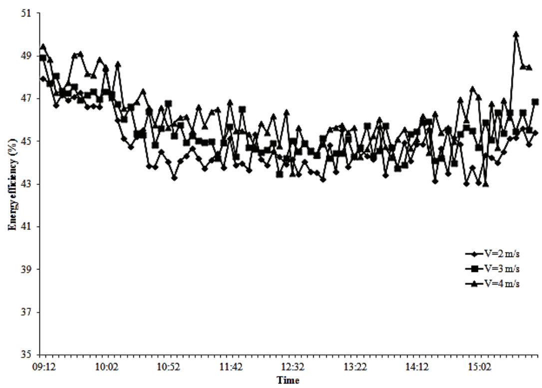

Figure 10- Energy efficiencies of the Type-IV collectors with the Al absorber plate in 2, 3 and 4 m s^{-1} airflow velocity

Şekil 10- 2, 3 ve 4 m s^{-1} hava hızlarında Al yutucu plakalı Tip-IV kollektörlerinin enerji verimleri

The temperature differences between outlet and inlet airflows for each type of collectors with Al absorber are plotted in Figure 11-13. The average temperature differences of the collectors were found as 31.35, 37.84, 37.15, 32.35 °C for 2 m s^{-1} airflow velocity, 33.66, 36.93, 35.42, 33.74 °C for 3 m s^{-1} airflow velocity, 33.98, 36.26, 35.88 and 35.21 °C for 4 m s^{-1} airflow velocity, respectively.

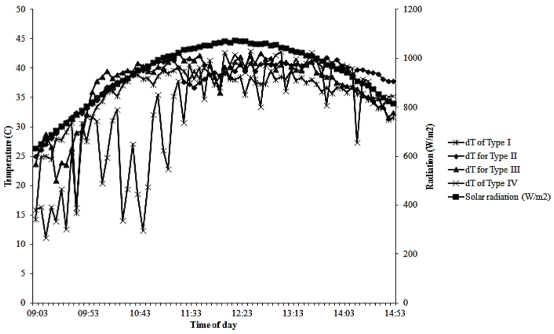

Figure 11- The variations of temperature of Type- I, II, III and IV collectors with the Al absorber plate in 2 m s^{-1} airflow velocity and under solar radiation

Şekil 11- Al yutucu plakalı Tip-I, II, III ve IV kollektörlerinin güneş ışınımı altındave 2 m s^{-1} hava hızındaki sıcaklık değişimleri

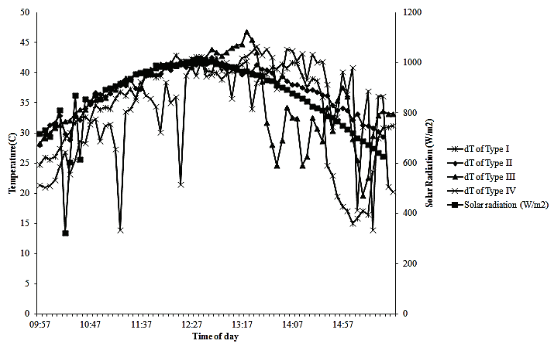

Figure 12- The variations of temperature of Type- I, II, III and IV collectors with the Al absorber plate in 3 m s^{1} airflow velocity and under solar radiation

Şekil 12- Al yutucu plakalı Tip-I, II, III ve IV kollektörlerinin güneş ışınımı altında ve 3 m s^{-1} hava hızındaki sıcaklık değişimleri

The average variation of temperature difference with solar radiation for the collector with Al and Cu absorbers were found for 2 m s^{-1} air velocity as 0.0363 and 0.0357, for 3 m s^{-1} air velocity as 0.0393 and 0.0380, for 4 m s^{-1} air velocity as 0.0367 and 0.0331 for Type I; for 2 m s^{-1} air velocity as 0.0403 and 0.0390, for 3 m s^{-1} air velocity as 0.0430 and 0.0428, for 4 m s^{-1} air velocity as 0.0418 and 0.0392 for Type II; for 2 m s^{-1} air velocity as 0.0450 and

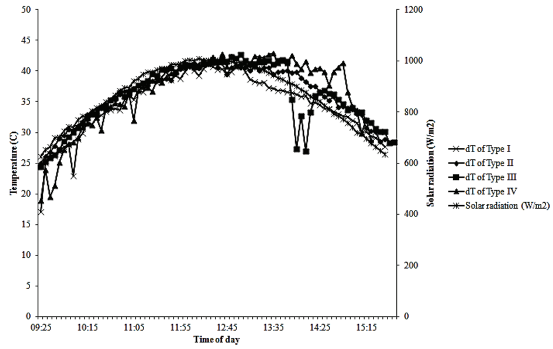

Figure 13- The variations of temperature of Type- I, II, III and IV collectors with the Al absorber plate in 4 m s^{1} airflow velocity and under solar radiation

Şekil 13- Al yutucu plakalı Tip-I, II, III ve IV kollektörlerinin güneş ışınımı altında ve 4 m s^{-1} hava hızındaki sıcaklık değişimleri

0.0425, for 3 m s^{-1} air velocity as 0.0436 and 0.0402, for 4 m s^{-1} air velocity as 0.0408 and 0.0380 for Type III; for 2 m s^{-1} air velocity as 0.0360 and 0.0336, for 3 m s^{-1} air velocity as 0.0370 and 0.0334, for 4 m s^{-1} air velocity as 0.0407 and 0.0354 for Type IV, respectively.

The flow Reynolds number in solar air heaters ranges as about 6646-7274, 9969-10912 and 13293-13953 for 2, 3 and 4 m s^{-1}, respectively.

4. Conclusions

In this experimental study, four-collector type's combination with Al and Cu absorber plate was tested for 2, 3 and 4 m s^{-1} airflow velocity and compared among themselves. From this study, the following conclusions could be drawn.

- Efficiency of the collector is very much dependent on airflow rate and as efficiency increased with flow rate, outlet temperature decreased correspondingly.
- The maximum collector efficiencies were found for all types of collector with Cu absorber plate, because the heat transfer coefficient of Cu (385 W m^{-1} K^{-1}) is higher than Al (210 W m^{-1} K^{-1}).

- The maximum values of the variation of temperature difference with solar radiation were found for all types of collectors with Al absorber.
- The heat capacity of Al (0.90 J g^{-1} $°C^{-1}$) is higher than Cu (0.385 J g^{-1} $°C^{-1}$), thus it seems that the aluminum absorber plate accumulates more heat and its heat transfer takes longer time than cooper absorber plate. In addition, temperature of aluminum absorber plate is higher than cooper absorber plate.
- According to comparison of experimental results for collector types, the highest collector efficiencies were found as $h_{Type\ III}$= 45.56, $h_{Type\ III}$= 46.63 and $h_{Type\ III}$= 46.45 for Type III with Al absorber plate in 2, 3 and 4 m s^{-1} airflow velocity, respectively. Al absorber plate in 2, 3 and 4 m s^{-1} airflow velocity, respectively and $h_{Type\ IV}$= 45.86, $h_{Type\ IV}$= 46.81 and $h_{Type\ IV}$= 47.18 for Type IV with Cu absorber plate in 2, 3 and 4 m s^{-1} air flow velocity, respectively.
- The highest variation of temperature difference with solar radiation was achieved by the collector of Type III for 2 m s^{-1} (dT/GT= 0.0450 for Al absorber and dT/GT= 0.0428 for Cu absorber.
- According to the findings, the differences of outlet temperatures and energy efficiencies between the aluminum and cooper absorber plates are insignificant to recommended temperature for solar air heaters. Especially, cost of collector with aluminum absorber plate cheaper than with cooper absorber plate and aluminum material weight less than cooper material.
- For the future investigations, it is suggested that, increasing the differences of the surface and airflow areas of the absorber plates will help to compare the absorber plates in terms of efficiency and economy, more clearly.

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