EXPERIMENTAL RESEARCH ON THE IMPACT OF THE APPLICATION OF FINS ON ABSORBER SURFACE ON THERMAL EFFICIENCY OF PASSIVE SOLAR AIR COLLECTOR

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ABSTRACT

In this paper an experimental results on passive solar air collectors are presented. An impact of application of waveform fins on the absorber surface on thermal efficiency was investigated. The results were compared with experiment on passive solar air collector with flat absorber surface. The experiment was conducted under steady state artificial radiation conditions in a laboratory environment. This enabled the acquisition of differences in flow and thermal parameters of the two designs. At the experimental stand following parameters were measured: irradiance, air temperature at inlet and outlet to the collector, mean air velocity at the inlet duct of the collector. The impact of irradiance on temperature increase, volumetric flow rate and thermal efficiency is shown. It was established that thermal efficiency of the collector with expanded absorber surface in form of fins is 10-15% higher than the efficiency of flat absorber collector.

INTRODUCTION

Solar collectors may be divided into two categories in regards to the working fluid: liquid or air (Miyazaki T., 2006, Yusoff W. F. M., 2010). Major amount of solar collectors that find application to home or industrial areas are liquid based. The air operated collectors are seeing increasing popularity. Air solar collectors can be used for a variety of drying applications, e.g., agricultural products, timber, biomass cultivation, waste biomass, building materials, etc. (Afriyie J. K., 2009, Karim M. A., 2004). It is also possible to use them for regulating microclimate in agricultural products storage facilities (Bouadila S., 2013), as well as for ventilating and heating enclosed industrial areas and storage. They can be also applied to passive solar chimneys (Karim M.A., 2006, Ozgen F., 2009, PN EN ISO 9488), additional greenhouse heating (Benli H., 2009, Benli H., 2013) and many other processes that require hot air (Nematollahi O., 2014).

However, the air solar collectors have been of little interest to the investors when compared to liquid solar collectors due to lack of universal, reliable data on energy efficiency under operating conditions. Tests of various prototype air solar collectors are being conducted in many countries.

Arce et al. (Arce J., 2009) studied a solar collector (solar chimney) with dimensions L x H = 4.5 x 1.0 m and an air gap thickness of 0.3 m. Thermal and flow investigations were conducted under natural conditions. At the instance of maximum solar irradiance (604 W/m^2), the increase in temperature of the air flowing through the collector was 70°C and the convective air flow rate was 50-374 m^3/h (corresponding to an average daily level of flow rate of 177 m^3/h). Experimental studies have confirmed that the flow of air through the collector mainly depends on temperature increase and pressure changes caused by variations in the density of the air outside and inside the chimney.

Alvarez et al. (Alvarez G., 2004) presented the results of experimental investigations of a solar collector proto-type that featured an absorber made from aluminum cans obtained from recycling. Sixteen parallel channels made from an arrangement of 128 black painted cans were placed under a glass cover. The study investigated the experimental efficiency of the proposed collector. The results were compared with the results of other authors. In addition to the undoubted advantages associated with using waste materials for design, a more than 60% increase in collector efficiency, relative to other expanded area solar absorber surfaces, was found.

The concept of using waste aluminum cans to build an air collector is also described in the work of Ozgen et al. (Ozgen F., 2009). The authors experimentally determined the efficiency of three solar prototypes featuring dual-flow (above and below the absorber). For the experimental investigations, different arrangements were tested, namely, cans placed in lines on both sides of the absorber and cans arranged in a zigzag pattern along both sides of the absorber. All designs worked concurrently in natural sunlight and with forced air flow with mass flow rates of 0.03 kg/s or 0.05 kg/s. Greater efficiency was obtained for mass flow rate 0.05 kg/s. The highest efficiency was obtained with the air collector with cans arranged in a zigzag. During the maximum solar radiation, the efficiency of the collector exceeds 70%.

This article aims to present the results of the experimental investigations of two types of prototype passive solar air collector. First design is made as a collector with a flat absorber and second one with finned absorber. Some laboratory experiments were made to obtain important parameters of collector operating conditions. The aim of research is to choose better collector design in terms of efficiency and in order to further its improvement.

EXPERIMENT

The solar air collector

Figure 1 shows an external view of the examined passive air solar collector prototypes. First o them (Fig. 1a) consists of an aluminum casing with dimensions 1.04 m (width) x 2.08 m (height) x 0.18 m (depth). The air inlet to the collector is a circular duct with diameter, d_{in} = 110 mm and a length of 0.5 m. The air outlet duct has an internal diameter, d_{out} = 130 mm and a length of 0.5 m. The axes of inlet and outlet channels are located in the middle of collector widthwise, 90 mm from the lower edge of the collector and 110 mm from the upper edge. The absorber is placed on the surface of the back insulation and is made from 0.5 mm thick aluminum plate coated with a thin layer of matte black paint.

Second collector (Fig. 1b) consists of an aluminum casing with dimensions 0.85 m (width) x 2.00 m (height) x 0.18 m (depth). This air collector has additional absorber surface made of aluminium sheet of 0.5 mm thick. The shape of this surface is on Fig. 1b presented. The air outlet and inlet to the collector is a circular duct with diameter, d_{in} = d_{out} = 125 mm and a length of 0.5 m. The axes of inlet and outlet channels are located in the middle of collector widthwise, 110 mm from the lower and upper edges of the collector.

The back wall of the collectors are insulated with mineral wool, 50 mm thick. The side walls, as well as the top and bottom, are insulated with mineral wool 20 mm thick. The transparent covers of the collectors are made of solar glass sheet 3,2 mm thick.

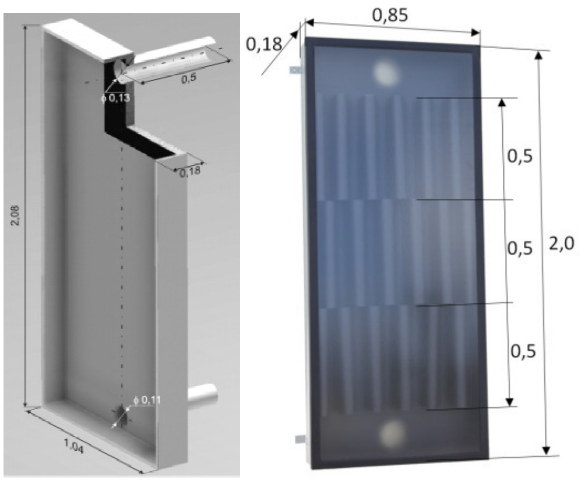


Fig. 1. View of tested solar collectors; a) with flat absorber, b) with finned absorber

Experimental set-up and procedure

A schematic of the experimental set-up is presented in Fig. 2. The main part is the prototype air solar collector, which was mounted on the supporting structure. The collector was operated in a vertical position. The distance between the lower edge of the collector and the floor was 0.5 m.

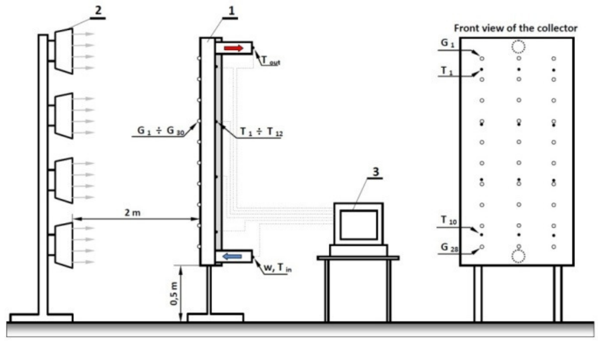


Fig. 2. Experimental set-up: 1 - air collector, 2 - adjustable radiation source, 3 - data acquisition system, T_{in} - inlet air temperature, T_{out} - the outlet air temperature, w – mean air velocity, G_1-G_{30} - irradiance measurement locations

The source of radiation were infrared Philips HeLeN lamps. They were arranged equidistant from the collector front. The maximum value of the irradiance achievable under laboratory conditions was G = 1 000 W/m^2.

Detailed description of experimental set-up, measuring devices accuracy and measuring procedure in articles (Esen H., 2009, Fudholi A., 2010) is presented.

The temperature increase, ΔT, of air flowing through the collector was determined as the difference between the temperature of air leaving the outlet channel, T_{out}, and the air temperature at the inlet to the collector, T_{in}. It should be noted that the air temperature at the entrance of the collector is constant and corresponds to the ambient temperature, T_{amb}, hence $T_{in} = T_{amb} = 20\ ^0C \pm 2\ ^0C$.

$$\Delta T = T_{out} - T_{in} = T_{out} - T_{amb}. \quad (1)$$

The volume flow rate of air through the collector was calculated from the velocity values measured on the inlet channel using the equation

$$\dot{V} = w \cdot \left(0.25 \cdot \pi \cdot d_{in}^2\right) \quad (2)$$

where w is mean air velocity in the inlet of the channel, and d_{in} is the diameter of the inlet channel. Heat output of the solar collector was determined according to the formula

$$Q = \dot{m} \cdot c_p \cdot \Delta T = \rho \cdot \dot{V} \cdot c_p \cdot \Delta T \quad (3)$$

where ρ is the density of air (at the inlet), and c_p is the specific heat of air (at ambient temperature and constant pressure).

The thermal efficiency of the passive air collector is defined as

$$\eta = \frac{Q}{G \cdot A} = \frac{\rho \cdot \dot{V} \cdot c_p \cdot \Delta T}{G \cdot A} \quad (4)$$

where G is the average value of the irradiance reaching the front surface of the collector, and A is the collector aperture area.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Experimental studies were carried out using the following procedure. Following the turning on the measurement equipment the value of electric current supplied to the radiation source was set using an autotransformer. Once steady-state conditions were achieved, the local air velocity at the entrance to the inlet channel and the distribution of the irradiance on the front surface of the solar air collector were measured. Measurements of air velocity, irradiance, and the various temperatures were used to calculate the desired characteristic parameters.

Air temperature increase

Figure 3 shows the increase of the air temperature ΔT through the collector as a function of irradiance G. It is important to highlight that presented charts show the air temperature increase. The comparison of two designs is not yet possible as the main geometrical dimensions differ. In order for the comparison to be possible, a non-dimensional parameters must be employed.

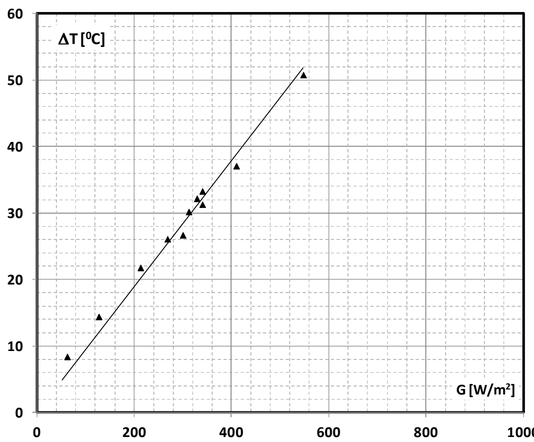


Fig. 3a. Air temperature difference ΔT vs. irradiation G; (collector with flat absorber)

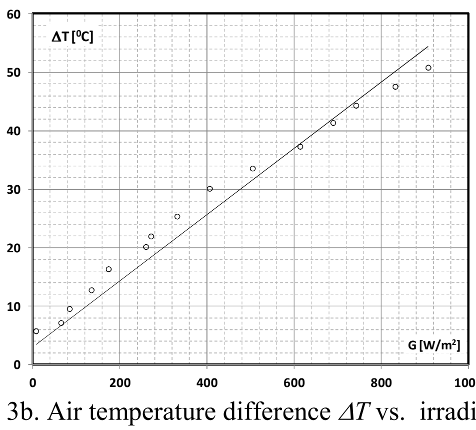
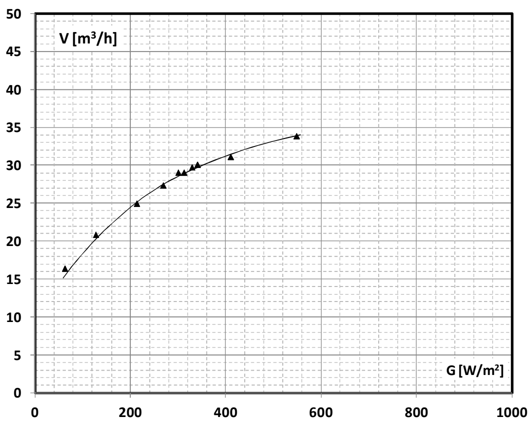


Fig. 3b. Air temperature difference ΔT vs. irradiation G; (collector with finned absorber)

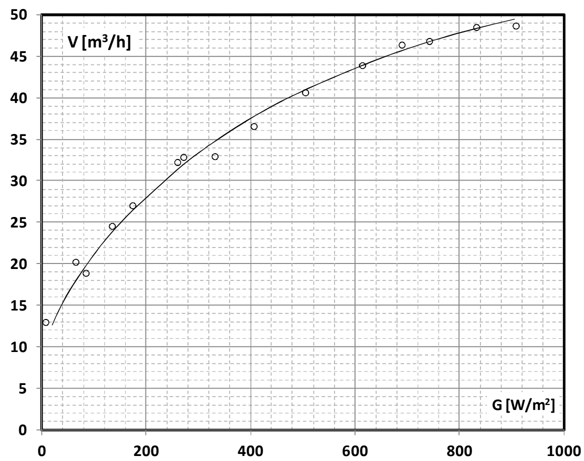
It was found that the air temperature increased proportionally to the intensity of incident radiation. The linear nature of the ΔT change related to irradiance G is observed for both forced air flow through collector (Lewandowski W. M., 2009, Maneewan S., 2003) and during natural convection (Gan G., 2006, Toure S., 2001). It is noticed that for the same amount of irradiance G (i.e. G = 500 W/m^2) the temperature increase is higher for the flat absorber collector than in the design with finned absorber (namely ΔT = 45 K and ΔT = 30 K).

Volume flow rate of air

The volumetric flow rate of air was calculated using equation 2 and is dependent on average flow velocity. As shown in Fig. 4, the maximum flow rate of air through the passive collector with flat absorber was $\dot{V}$ = 35 m^3/h, for the collector with finned absorber (for the same irradiance G = 500-600 W/m^2) equals about $\dot{V}$ = 44 m^3/h. The shown comparison indicates that the amount of flowing air and consequently its mean velocity is higher in the design with fins. Higher values of volumetric flow rate can explain the lower values temperature increase (Fig. 3).



a)



b)

Fig. 4. Volume air flow rate $\dot{V}$ vs. irradiance G; a) collector with flat absorber, b) collector with finned absorber

Heat output of the collector

Figure 5 shows the effect of irradiance *G* on the collector heat output *Q*, defined by the equation (2.3). The increase of solar irradiance causes an increase in heat output of the collector. The maximum heat output for flat absorber collector was approximately Q = 570 W when the irradiance was 550 W/m^2 and for the finned absorber - 830 W for G = 910 W/m^2. For the same level of irradiance absorbed, the heat transferred to the fluid is slightly lower in the finned collector.

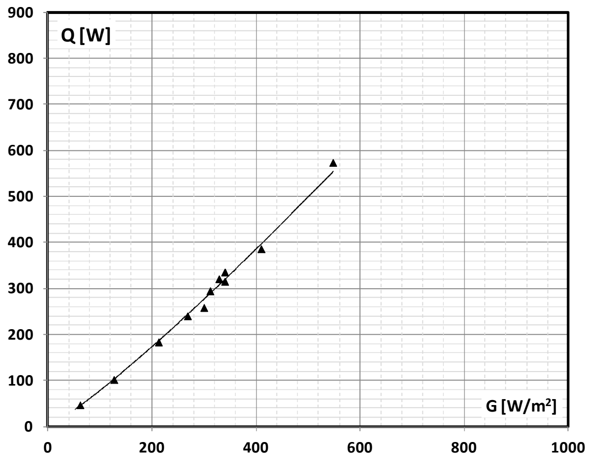


Fig. 5a. Collector heat output, *Q*, vs. irradiance *G*; (collector with flat absorber)

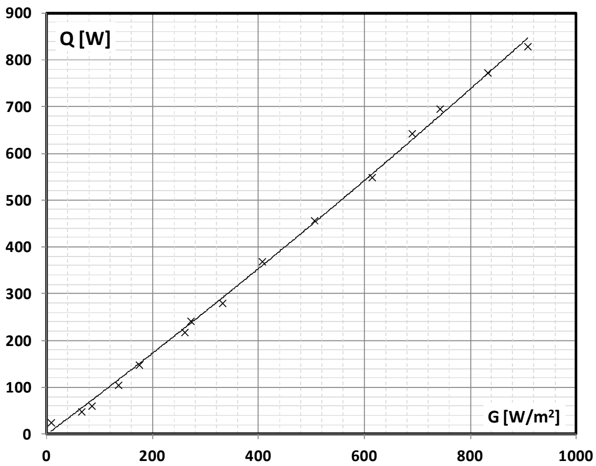


Fig. 5b. Collector heat output, *Q*, vs. irradiance *G*; (collector with finned absorber)

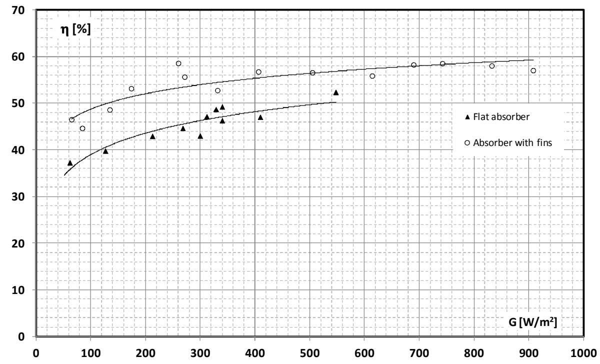


Fig. 6. Influence of irradiance *G* on the thermal efficiency η of the collector

Thermal efficiency

The thermal efficiency of the collector, expressed as the ratio of useful heat supplied to the fluid and thermal power that reaches the collector through radiation, is defined by equation 4. Fig. 6 shows the collector efficiency values for different values of the irradiance *G*.

The results show that an increase in the irradiance increases the thermal efficiency of the collector. The maximum efficiency, obtained in this paper was approximately 40% and 50% for collector with flat absorber and for finned one. For the same values of irradiance, thermal efficiency for the collector with finned absorber is over 10-15% higher than the efficiency of collector with flat absorber. This is caused by the fact that while heat output for the two designs is similar, the aperture area of the collector with finned absorber is lower. Therefore, from lower surface, the same amount of heat is obtained, thus efficiency is higher.

SUMMARY AND CONCLUSIONS

In this paper, results of an experimental investigation on collectors with flat and finned absorber have been presented. The collectors worked under natural convection regime. The study was performed on a laboratory set-up and led to obtaining basic performance characteristics of this collector type. The measured parameters consisted of: collector irradiance, air temperature increase between inlet and outlet, air velocity at the inlet duct. From these measured values it was possible to determine heat output of the collector and its thermal efficiency. The tests were performed for a range of irradiance G = 0 – 540 W/m^2 (collector with flat absorber) and G = 0 – 1000 W/m^2 (collector with finned absorber). Due to the fact that dimensions of each designs are different, the final comparison is made through non-dimensional parameters, namely the thermal efficiency. It is shown that collector with finned absorber has thermal efficiency about 10-15% higher than analogous design with flat absorber surface.

NOMENCLATURE

A	collector aperture surface	m^2
B	collector gap width	m
c_p	specific heat at constant pressure	J/(kgK)
d	diameter	m
$\dot{m}$	mass flow rate	kg/s
H	height	m
G	irradiance	W/m^2
L	length	m
Q	heat output	W
T	temperature	°C
$\dot{V}$	volume flow rate	m^3/s
w	mean velocity	m/s

Greek symbols

ρ	mass density	kg/m^3
ΔT	difference of air temperature	K
η	thermal efficiency	-

Subscripts

amb	ambient
in	inlet
out	outlet

REFERENCES

Afriyie J. K., Nazha M. A. A., Rajakaruna H., Forson F. K., 2009, *Experimental investigations of a chimney-dependent solar crop dryer*, Renew. Energy, 34, 217–222

Alvarez G., Arce J., Lira L., Heras L.M.R., 2004, *Thermal performance of an air solar col-lector with an absorber plate made of recyclable aluminum cans*, Sol. Energy, 77, 107–113

Arce J., Jiménez M. J., Guzmán J. D., Heras M. R., Alvarez G., Xamán J., 2009, *Experimental study for natural ventilation on a solar chimney*, Renew. Energy, 34, 2928–2934

Benli H. and Durmus A., 2009, *Performance analysis of a latent heat storage system with phase change material for new designed solar collectors in greenhouse heating*, Sol. Energy, 83, 2109–2119

Benli H., 2013, *Experimentally derived efficiency and exergy analysis of a new solar air heater having different surface shapes*, Renew. Energy, 50, 58–67

Bouadila S., Kooli S., Lazaar M., Skouri S., Farhat A., 2013, *Performance of a new solar air heater with packed-bed latent storage energy for nocturnal use*, Appl. Energy, 110, 267–275

Esen H., Ozgen F., Esen M., Sengur A., 2009, *Modelling of a new solar air heater through least-squares support vector machines*, Expert Syst. Appl., 36, 10673–10682

Fudholi A., Sopian K., Ruslan M. H., Alghoul M. A., Sulaiman M. Y., 2010, *Review of solar dryers for agricultural and marine products*, Renew. Sustain. Energy Rev., 14, 1–30

Gan G., 2006, *Simulation of buoyancy-induced flow in open cavities for natural ventilation*, Energy Build., 38, 410–420

Karim M. A., Hawlader M. N. A., 2004, *Development of solar air collectors for drying applications*, Energy Convers. Manag., 45, 329–344

Karim M. A., Hawlader M. N. A., 2006, *Performance evaluation of a v-groove solar air collector for drying applications*, Appl. Therm. Eng., 26, 121–130

Lewandowski W. M., 2007: *Proecological, renewable energy sources [in Polish]*, *WNT*

Maneewan S., Khedari J., Zeghmati B., Hirunlabh J., Eakburanawat J., 2003, *Experimental investigation on generated power of thermoelectric roof solar collector,* Int. Conf. Thermoelectr. ICT, Proc., 574–577

Miyazaki T., Akisawa A., i Kashiwagi T., 2006, *The effects of solar chimneys on thermal load mitigation of office buildings under the Japanese climate*, Renew. Energy, 31, 987–1010

Nematollahi O., Alamdari P., Assari M. R., 2014, *Experimental investigation of a dual pur-pose solar heating system*, Energy Convers. Manag., 78, 359–366

Ozgen F., Esen M., Esen H., 2009, *Experimental investigation of thermal performance of a double-flow solar air heater having aluminium cans*, Renew. Energy, 34, 2391–2398

PN EN ISO 9488. Solar Energy - Vocabulary

Toure S., 2001, *Characteristic temperatures in a natural convection solar air heater*, Energy Convers. Manag., 42, 1157–1168

Yusoff W. F. M., Salleh E., Adam N. M., Sapian A. R., Yusof S. M., 2010, *Enhancement of stack ventilation in hot and humid climate using a combination of roof solar collector and vertical stack*, Build. Environ., 45, 2296–2308