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Contents

Evaluation of Various Hybrid Solar Collector Configurations for Water and Air Heating

Qahtan Adnan Abed, Viorel Badescu and Iuliana Soriga

Abstract In this paper a mathematical model has been developed to study the thermal efficiency of a water-air hybrid solar collector (HSC). The model, implemented in MATLAB, was used to evaluate the collector in the real operation conditions of Timisoara, Romania. The effect of solar irradiation, ambient temperature and air inlet temperature on the useful energy collected by air and water has been investigated. Absorber plate temperature variation, water and air outlet temperatures, collector efficiency and air flow rate effects are presented. Moreover, the efficiency of the collector for three different configurations of the air channel is computed and compared: triangular fins, rectangular fins and without fins. The simulation results show that the configuration with triangular fins has better performance compared with the others.

Keywords Hybrid solar collector · Water heater · Air heater · Modeling · Effectiveness

1 Introduction

The use of solar heating systems had increased based on the reasonable initial costs and relatively simple structure, especially the solar water heating and air heating are widely used in most of the countries. There are many studies about the hot water and air heated by solar radiations. Among these, one analysis focused on the performance of solar thermosiphon water heaters with heat exchangers in storage tanks [1]. Another study [2] made a review on element geometries used as artificial roughness in solar air heaters in order to improve the heat transfer capability of solar air heater

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ducts. In the agricultural domain, a simple solar air heater from cheap plastic wrapping film with air bubbles, for use in drying operations on a farm, has been proposed [3]. These studies had been performed on a single-purpose collector, where the operating fluid is only air or liquid. For the collectors whose working fluid is water, thermal transfer rate is acceptable because water is a good conductor of heat. However, for solar air collectors, heat transfer is low, therefore, the use of fins or corrugated surfaces is required, for the improvement of the heat transfer [4]. In order to increase the efficiency, some researchers studied the thermal performance of solar collectors working with two different types of fluids simultaneously. In this regard, studies were made on the performance of an air and water collector combined in a single solar collector, also called a Dual Purpose Solar Collector (DPSC) [5]. Energy and exergy study of air-water combined solar collector was also investigated [6] observing the water inlet temperature and air flow rate effects. A dynamic numerical model has been developed and validated by experimental data [7], for a new proposed building- integrated dual-function solar collector that is able to provide passive space heating in cold winter, and water heating in warm seasons.

In this work the thermal performance of the Hybrid Solar Collector (HSC) for water and air heating was studied. The mathematical model used for the HSC assessment in the real operating conditions of Timisoara, Romania, was implemented in MATLAB.

2 Description of the HSC

The studied collector is a flat-plate collector, in which both working fluids receive heat from the absorbing plate, as shown in Fig. 1. The length of collector is 1.94 m and width 0.94 m [8]. The transparent cover is made from glass. The gap between the glass cover and the absorber plate is 3 cm. The water is flowing through pipes located in the top of the absorber plate, and the air flows through a channel in the

Fig. 1 Schematic layout of a typical solar water and air heater system

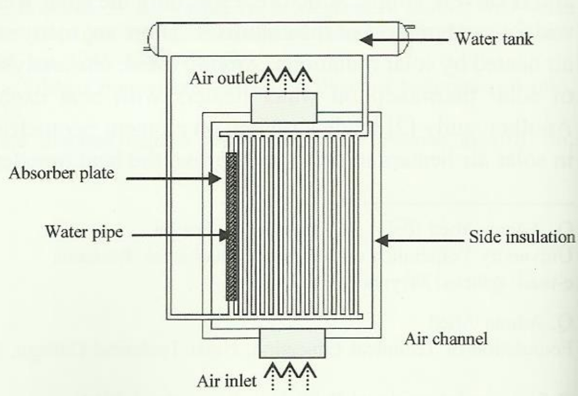

Table 1 Detail specification and input parameter of the hybrid solar collector

HSC detail specification		
Transparent cover and absorber plate	Material	Glass and aluminium
	Emissivity	0.88 and 0.17
	Thickness	4 and 2 mm
Insulation	Thermal conductivity	0.04 W/(m.K)
	Rear and the side thickness	5 and 2 cm
Water pipes	Number	10
	Diameter	7 mm
Air channel (heat transfer area)	Without fins	4.24 m^2
	Rectangular fins	7.6 m^2
	Triangular fins	8.16 m^2
HSC input parameter		
Solar global irradiance		11.5–883.8 W/m^2
Inlet air temperature		294.25–308.05 K
Water inlet temperature		313.15 K
Mass flow rate of water and air		0.02 and 0.1 kg/s
Wind speed		1.5 m/s

bottom side of the absorber plate with height 6 cm. The rear and the side insulation are provided by a polystyrene sheet. The detail configuration and input parameter of HSC for water and air are tabulated in Table 1.

3 Theoretical Analysis: Assumptions and Collector Model

The suitable sizing of the components of the solar system is a complex problem which includes both predictable (collector and other performance characteristics) and unpredictable (weather data) components, so the following assumptions were made to simplify the analysis of the HSC:

(a) The absorber plate, water tube surface and air channel surface, have the same temperature.
(b) The conduction and radiation heat transfer in water tubes and air channel was neglected.
(c) The heat losses by convection and radiation from the rear and side surfaces was neglected.
(d) Flow inside water pipes and air channels was assumed to be laminar.

The first step in the HSC thermal model is the computation of the heat loss from the collector. Top, rear and side heat loss was calculated according to Ref. [8]. The heat transfer coefficient between water and the tubes is calculated from the Nusselt number:

$$h_w = \frac{k_w}{d} Nu_w \tag{1}$$

where: k_w—thermal conductivity (W/(m.K)), d—diameter of the tube (m) and Nu_w—Nusselt number.

For the circular tube the average Nusselt number for the thermal entrance region can be determined from [9]:

$$Nu_w = 3.66 + \frac{0.0667(\frac{d}{L_1})\mathrm{Re}_w.\,\mathrm{Pr}}{1 + 0.04\left[(\frac{d}{L_1})\mathrm{Re}_w.\,\mathrm{Pr}\right]^{\frac{2}{3}}} \tag{2}$$

where: Re_w—Reynold number, L_1—length of absorber plate (m) and Pr—Prandtl number.

Also, the convection heat transfer coefficient in the air channel of the collector for the three different configurations (without fins, rectangular fins and triangular fins), can be estimated by Eqs. (3)–(5), respectively [10–12].

$$Nu_D = 0.0158\mathrm{Re}_D^{0.8} \tag{3}$$

$$Nu_D = 0.023\mathrm{Re}_D^{0.8}.\,\mathrm{Pr}^{0.4} \tag{4}$$

$$Nu_D = Nu_o + \beta\frac{b}{L_1}n, \left\{\begin{array}{lll} Nu_o = 2.821 & \beta = 0.126\mathrm{Re}_D & \mathrm{Re}_D < 2800 \\ Nu_o = 1.9 * 10^{-6}\mathrm{Re}_D^{1.79} & \beta = 225 & 2800 < \mathrm{Re}_D < 10^4 \\ Nu_o = 0.0302\mathrm{Re}_D^{0.74} & \beta = 0.242\mathrm{Re}_D^{0.74} & 10^4 < \mathrm{Re}_D < 10^5 \end{array}\right\} \tag{5}$$

where: β—tilt angle (degrees), b—half height of the triangular fins (m) and n—number of collectors connected.

The heat exchange effectiveness is defined as heat delivery to maximum heat delivery that can transfer to fluids. Heat exchange effectiveness for water can be obtained from the following equation [13]:

$$\varepsilon_w = 1 - \exp\left[-\frac{h_w A_w}{\dot{m}_w C_{p,w}}\right] \tag{6}$$

The performance of a hybrid solar collector for water and air can be described by the useful heat gain from the collector, which is defined as the difference between the absorbed solar energy and the thermal loss of a collector. Finally the amount of useful heat gain (q_u), is calculated from summation of the absorbed heat for water and air by using the two following equations respectively [8]:

$$q_{u,w} = \left(\frac{\varepsilon_w \dot{m}_w C_{p,w}}{U_{L,w} A_p + \varepsilon_w \dot{m}_w C_{p,w}} \right) \left[A_p S - U_{L,w} A_p \left(T_{w,in} - T_{amb} \right) \right] \tag{7}$$

$$q_{u,a} = \left(\frac{\varepsilon_a \dot{m}_a C_{p,a}}{U_{L,a} A_p + \varepsilon_w \dot{m}_a C_{p,a}} \right) \left[A_p S - U_{L,a} A_p \left(T_{a,in} - T_{amb} \right) \right] \tag{8}$$

where: ε_w—heat exchange effectiveness of water and air, $\dot{m}_w$—water mass flow rate (kg/s), $C_{p,w}$—specific heat (J/kg. K), A_p—collector area (m^2), S–absorbed solar radiation (W/m^2), $T_{w,in}$—water inlet temperature (K), $T_{a,in}$—air inlet temperature (K) and T_{amb}—ambient air temperature (K).

The thermal efficiency of the HSC is defined as the ratio of useful absorbed energy by the fluid to the solar energy received by the collector.

$$\eta = \frac{q_u}{A_p I_T} \tag{9}$$

The individual component models are coded into "MATLAB" software and the simulation program is used to predict system parameters such as hot water and air temperature, fluid absorbed useful energy, heat exchange effectiveness and the performance of HSC for various solar global irradiance values various air inlet temperatures and constant water inlet. The computation algorithm is represented in the flow chart shown in Fig. 2.

4 Results and Discussions

Useful heat gain, efficiency and working fluids outlet temperatures, were obtained from the numerical simulation, using as input data values characteristic to Timisoara, Romania (45° 44′N / 21° 13′E). Fig. 3 shows the variation of solar global irradiance and ambient air temperature on a sunny day (01/July/2010). The ambient air temperature was high throughout the day, with maximum value 308.05 K and a minimum value 294.25 K, and the maximum solar global irradiance was 883.8 W/m^2 at the midday.

The variation of water outlet and air outlet temperatures of HSC for three types of air channels: without fins, with rectangular fins and triangular fins are shown in Fig. 4. Here, it can be observed that the collector without fins in the air channel (first case), obtained highest water outlet temperature and lowest air outlet temperature, while in the other two cases temperatures were identical.

Figure 5 shows the useful heat gain of water and air during the day that are obtained from HSC model. The average value of the useful heat gain for water is 534 W, ranging between 24 and 924 W, and the average value of the useful heat gain for air is 124 W, ranging between 37.8 and 173.4 W.

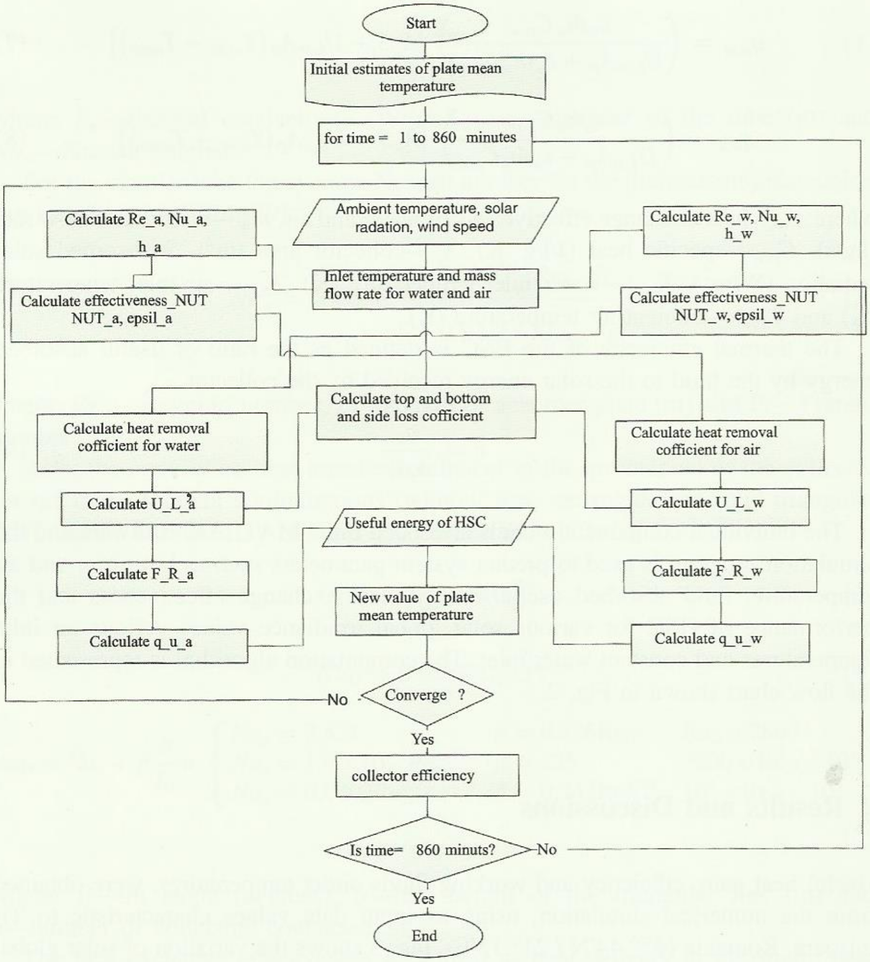

Fig. 2 Simple flow chart of simulation process

The useful heat gain of air increases when adding the fins for constant water and air mass flow rate equal to 0.02 and 0.1 kg/s, respectively. Figure 6 shows the maximum heat gain for air in the collector that uses triangular fins. The average value of the useful heat gain for the three types of air channel: without fins, rectangular fins and triangular fins was 72.3 W varying from 3.4 to 113.7 W, 110.4 W varying from 4.6 to 173.5 W and 111.8 W varying from 4.6 to 176.2 W, respectively.

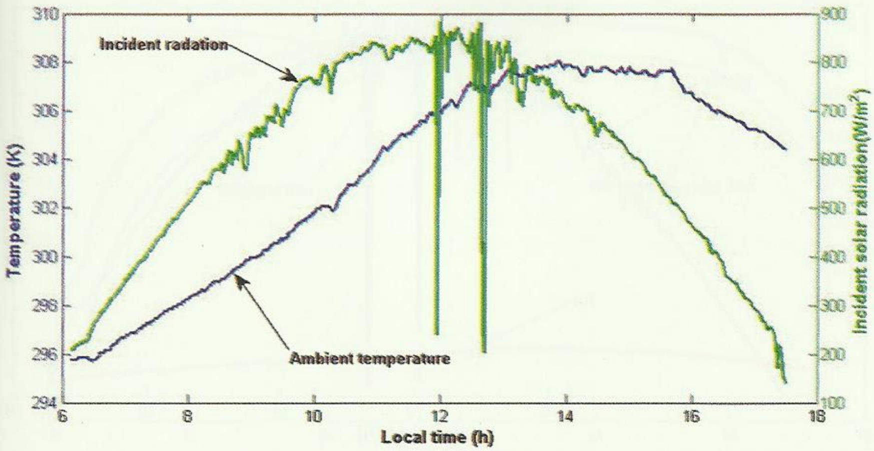

Fig. 3 Variation of ambient air temperature and solar radiation intensity on 01/July/2010

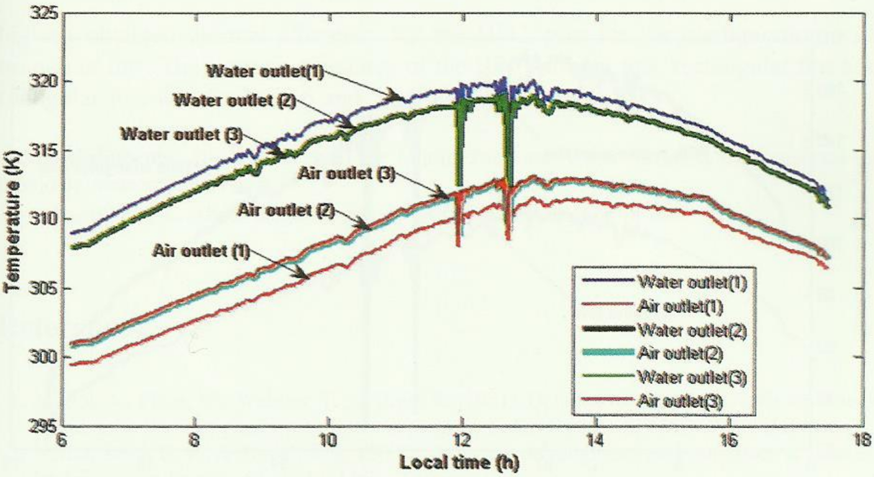

Fig. 4 Variation of water outlet and air outlet temperature for three types of air channels: (*1*) without fins, (*2*) with rectangular fins and (*3*) with triangular fins

The thermal efficiency of HSC is shown in Fig. 7, also for the three different types of air channels. It can be observed that it is higher for the configurations with fins. The maximum HSC efficiency was 67.9 % and it was obtained at noon for the air channel with triangular fins and constant air mass flow rate of 0.1 kg/s. This efficiency increased from morning until noon and then decreased. The maximum efficiency of the air channel without fins was 61 % and with rectangular fins, 65.8 % (Fig. 7).

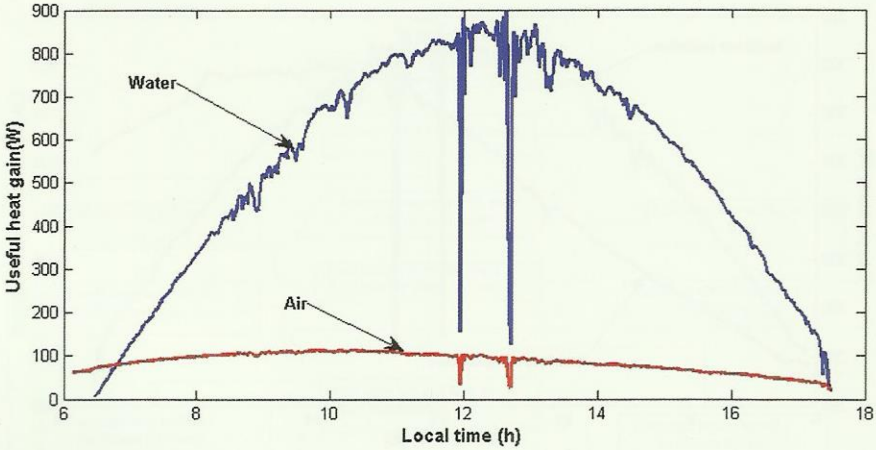

Fig. 5 Useful heat gain from HSC for air cannel without fins

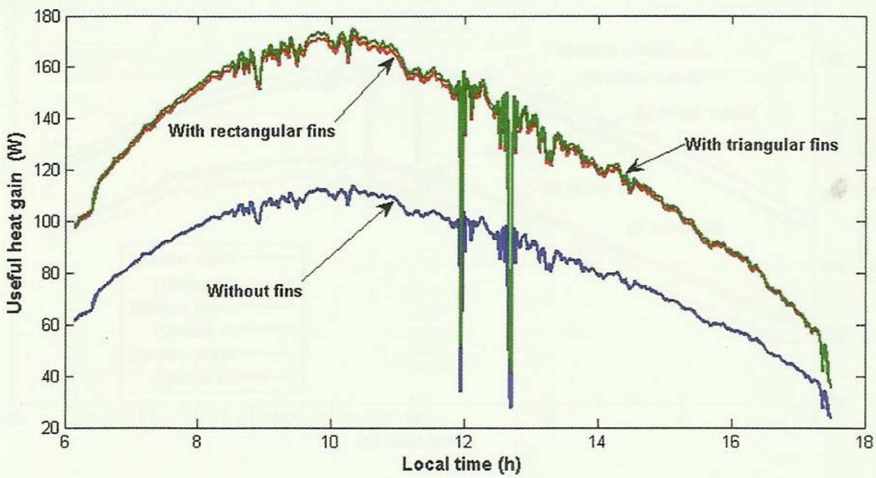

Fig. 6 Comparison of air useful heat gain for three types of air channel

5 Conclusion

A mathematical model for the simulation of HSC for water and air heating was developed. The performance of the solar collector was simulated with variable solar radiation, variable air inlet temperature, and constant water inlet temperature. The air channel type has an effect on the thermal efficiency. Because the heat transfer area of the air channel with triangular fins is higher compared to the other types, the

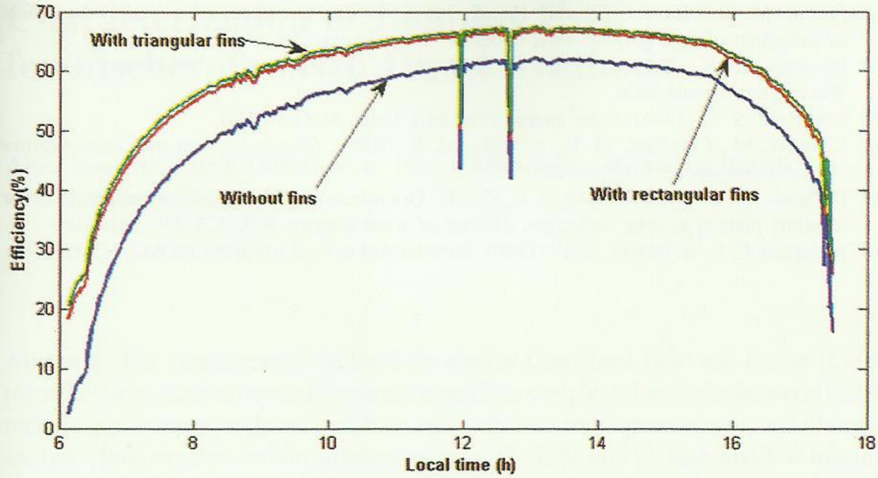

Fig. 7 Thermal efficiency for HSC during the day for different types of air channels

highest obtained thermal efficiency for the HSC, was for the configuration with triangular fins. The average efficiency of the HSC without fins, rectangular fins and triangular fins was 50.2, 56.6 and 57.4 % respectively.

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