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# A mathematical model to investigate on the thermal performance of a flat plate solar air collector and its experimental verification

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ABSTRACT

A mathematical model based on numerical finite-difference approach under forced convection mode was presented for the SAC. Airflow channel, absorber plate, glass cover, thermal insulation board and fan power were taken into consideration in this model and analyzed in detail. In order to verify the accuracy of this model, an indoor experimental system was built to study the performance of a double pass flow SAC. The effect of the inlet mass flow rate of the collector on the thermal performance was investigated under various environmental conditions. The outlet air temperature obtained from the theoretical and experimental studies are in reasonable agreement, which supports the validity of the theoretical model. By considering the energy gained and the fan power consumed under real conditions, the optimum mass flow rates were discussed and simulated with different ambient temperature and solar irradiance, which showed that for this flat plate SAC, $\dot{m}_{opt}$ equals 0.03 kg/s at $I = 400$ W/m$^2$, equals 0.04 kg/s at $I = 700$ W/m$^2$, and equals 0.045 kg/s at $I = 1000$ W/m$^2$. The results are useful for analyzing and designing new SACs.


## 1. Introduction

Solar collectors are devices that absorb the incoming solar radiation, convert it into heat and transfer the heat to the fluid flowing through them. According to the heat transfer medium flowing through the collector, solar collectors are divided into two types: Solar Liquid Collector (SLC) and Solar Air Collector (SAC). SACs are extensively used in air conditioning, agricultural product drying and industrial process heat [1,2]. Compared with SLCs, mass production of SACs appears to be inherently cheaper and easier to realize, since conductivity and corrosion of the absorber plate are secondary considerations [3]. However, the disadvantages of SACs are the low energy density, low thermal capacity and small heat conductivity of air. To improve the performance of SACs, qualitative or quantitative optimization suggestions to design efficient SACs are presented in many previous studies. These studies include different types of covers, selective surfaces, porous media, airflow channel geometries, fin structures, mass flow rate, inlet temperature, etc.

According to the types of flow channels, the conventional SACs can be constructed into three basic designs: SACs with air flow either over the absorber (type I) or under it (type II) and even on both sides of the absorber (type III). Close [3] compared the collection efficiencies of these three designs and found that the type with a stagnant air gap above the absorber plate might have the best performance. Duffie and Beckman [1] provided an example of an equation for flow over the absorber. In these two studies, the insulation of the back board was neglected which enhanced the impact of the heat losses through the cover. Parker et al. [4] improved Close's work by presenting procedures to compute thermal performance for the three types of collectors and compared the results with the experimental data. Thermal performance of SACs was also been predicted both theoretically and experimentally by Ong [5,6] and Njomo et al. [7].

Most studies focused on the enhancement of the thermal efficiency of SACs. For instance, Fudholi et al. [8] developed a theoretical model of a finned double-pass solar collector and carried out its experimental validation. Wei et al. [9] came up with a combined system of solar Kang and SAC to make the most of heat in cold winter and reduce air pollution. Stanciu [10] presented a theoretical study on the optimum tilt angle for flat plate collectors at different geographical locations and different time moments over a year. Also, tracking methods were analyzed and verified by Maia et al. [11], with which higher useful gain and higher efficiency could be reached. Baritto and Bracamonte [12,13] presented a dimensionless model for the outlet temperature of a non-isothermal

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**Nomenclature**

| | |
|---|---|
| $k$ | thermal conductivity ($W\,m^{-1}\,K^{-1}$) |
| $c_p$ | heat capacity ($J\,kg^{-2}\,K^{-1}$) |
| $h$ | heat transfer coefficient ($W\,m^{-2}\,K^{-1}$) |
| $hr$ | radiative heat transfer coefficient ($W\,m^{-2}\,K^{-1}$) |
| $hc$ | convective heat transfer coefficient ($W\,m^{-2}\,K^{-1}$) |
| $T$ | temperature (K) |
| $p$ | pressure (Pa) |
| $u$ | velocity in x direction ($m\,s^{-1}$) |
| $v$ | velocity in y direction ($m\,s^{-1}$) |
| $\dot{m}$ | mass flow rate ($kg\,s^{-1}$) |
| $Pr$ | Prandtl number |
| $I$ | solar radiation intensity ($W\,m^{-2}$) |
| $P$ | power (W) |
| $K$ | K-factor |
| $\dot{V}$ | volume flow rate |
| $l$ | total length of the SAC |
| $a$ | area ($m^2$) |
| $w$ | thickness (m) |
| $W$ | flow channel height |
| $U$ | non-dimensionalized parameter of $u$ |
| $V$ | non-dimensionalized parameter of $v$ |
| $P$ | non-dimensionalized parameter of $p$ |
| $X$ | non-dimensionalized parameter of $x$ |
| $Y$ | non-dimensionalized parameter of $y$ |

*Greek symbols*

| | |
|---|---|
| $\nu$ | kinetic viscosity (Pa s) |
| $\rho$ | density ($kg\,m^{-3}$) |
| $\alpha$ | absorptivity |
| $\tau$ | transmissivity |
| $\varepsilon$ | emissivity |
| $\eta$ | efficiency |
| $\sigma$ | uncertainties/Stefan–Boltzmann constant |
| $\Theta$ | non-dimensionalized parameter of $T$ |

*Subscripts*

| | |
|---|---|
| $f1$ | airflow in channel I |
| $f2$ | airflow in channel II |
| $opt$ | optimum value |
| $g$ | glass cover |
| $g-$ | the lower surface of glass cover |
| $g+$ | the upper surface of glass cover |
| $b$ | insulation board |
| $b-$ | the lower surface of insulation board |
| $b+$ | the upper surface of insulation board |
| $p$ | absorber plate |
| $p-$ | the lower surface of absorber plate |
| $p+$ | the upper surface of absorber plate |
| $pg$ | value between absorber plate and glass cover |
| $pb$ | value between absorber plate and insulation board |
| $ga$ | value between glass cover and ambient |
| $gp$ | value e between glass cover and absorber plate |
| $ba$ | value between insulation board and ambient |
| $bp$ | value between insulation board and absorber plate |
| $in$ | inlet parameter |
| $out$ | outlet parameter |
| $a$ | ambient parameter |
| $theo$ | theoretical value |
| $exp$ | experimental value |
| $fan$ | fan |
| $flow$ | flow pump |
| $f$ | flow friction |
| $l$ | local resistance |
| $u$ | useful thermal output value |

SAC and it showed good agreement with experimental data. Another model was solved for a wide range of aspect ratios and mass flow numbers. Optimization studies on size, structure, fin geometry and flow control for flat plate SAC were also conducted by Badescu [14–16].

To balance the outlet temperature and thermal efficiency, exergy analysis could be considered in the evaluation of the airflow rate. A comparison of three types of SACs was conducted by Alta et al. [17] based on the energy and exergy analyses by which the optimized SAC designs were suggested. However, it is obvious that without any air leakage of the SAC, exergy efficiency always increases with the increasing of flow rate. Badescu proposed an optimal operation strategy for exergy gain maximization by controlling mass flow rate [18]. Farahat et al. [19] developed an exergetic optimization of flat plate solar collectors to determine the optimal performance and design parameters of these solar-to-thermal energy conversion systems. Jafarkazemi and Ahmadifard [20] carried out energetic and exergetic evaluation of SACs and found that increasing inlet water temperature and decreasing water mass flow rate can be effective on decreasing most of exergy destructions. Bahrehmand et al. [21] built a mathematical model for simulating the thermal behavior of single and two glass cover SAC systems with forced convection flow.

Another way to assess the performance of SAC is taking the fan power into consideration. El-Sebaii et al. [22] carried out a theoretical study on thermohydraulic efficiency of two types of SACs. One conventional way to enhance the thermal performance of SACs is to employ higher airflow rates but costs additional fan power and reduced the outlet temperature. An improved method is to lengthen the absorber and/or increase the depth of the channel. Hegazy [23,24] developed an analytical criterion for determining the optimal channel geometry in considering the fan power consumption.

Since most of the previous studies focused on the enhancement of the thermal efficiency of solar collectors with experimental study or numerical simulation, the theoretical analysis is usually conducted with only part of the components of the SACs. In their theoretical models, empirical equations or complex CFD (Computational Fluid Dynamics) simulation models are adopted. The work presented in this paper aims at providing a new, complete and fast mathematical model constructed in finite difference method to predict the thermal performance of a flat plate SAC and guide the design of flat plate SACs. The model is validated by comparing the simulation results with those obtained in an experimental model. It also aims at using the developed model to investigate the effects of the airflow rate on the comprehensive performance of a flat plate SAC and coming out with the optimum mass flow rates under different environmental conditions.

## 2. Theoretical study

### 2.1. Description of the presented SAC

The schematic of the SAC geometry is shown in Fig. 1. The SAC is covered by a single, high transmittance tempered glass with 3.2 mm thick, dimension size of 1944 × 932 mm. The heights of