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Energy Procedia 74 (2015) 1090 – 1099

Energy Procedia

International Conference on Technologies and Materials for Renewable Energy, Environment and Sustainability, TMREES15

Model Validation of an Empirical Photovoltaic Thermal (PV/T) Collector

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Abstract

Within solar energy technologies, the hybrid photovoltaic–thermal (PV/T) systems offer an attractive option because the absorbed solar radiation is converted into thermal and electrical energy (the conversion can be done separately or simultaneously). In this paper, an attempt is made to investigate the thermal and electrical performance of a solar photovoltaic thermal collector. A detailed thermal model is developed to calculate the thermal parameters of a typical PV/T collector. The thermal parameters of this collector include solar cell temperature, outlet water temperature, thermal efficiency and useful thermal energy. Some corrections are done on heat loss coefficients in order to improve the thermal model of a PV/T collector. A computer simulation program is developed in order to calculate the thermal and electrical parameters of a PV/T collector. The absorber is realized with the galvanized iron of high quality, allowing a good transfer of heat with lower cost compared to copper. This PVT has the advantage of its simple implementation and its low cost compared to other configurations. The results of numerical simulation are in good agreement with the experimental measurements noted in the previous literature. It is observed that the thermal simulation results obtained in this paper is more precise than the one given by the previous literature. It is also found that the thermal efficiency of PV/T collector is about 54.51% in the mode of water heat exchanger, and 16.24% in air heat extraction, the electrical efficiency is 11.12%, for a sample climatic, operating and design parameters.

Peer-review under responsibility of the Euro-Mediterranean Institute for Sustainable Development (EUMISD)

Keywords: Hybrid solar system; solar photovoltaic thermal (PV/T) collector; thermal performance, Performance analysis, simulation.

Peer-review under responsibility of the Euro-Mediterranean Institute for Sustainable Development (EUMISD)
doi:10.1016/j.egypro.2015.07.749

1. Introduction

A photovoltaic thermal (PVT) system is a combination of photovoltaic cells and a thermal collector. Such a collector, when exposed to sunlight, will produce electricity and thermal energy. In such a system, the thermal system component acts as a heat sink to the photovoltaic cells, thus improving its efficiency. Energy, that is not absorbed by the PV cells, is utilized in the thermal system, and hence, both these systems interdependent. The heat extracted from the photovoltaic thermal system can be used for domestic water heating system or for space heating [1].

The hybrid PVT systems can be classified according to the used type of working fluid, commonly PVT air heating systems (PVT/a) and PVT water heating systems (PVT/w). Some theoretical and experimental researches on these two types of hybrid PVT systems have been conducted in recent years. Hegazy [2] evaluated the performances of four popular designs of PVT/a systems including the air flowing over the absorber or under it, and on both sides of the absorber in single pass or double pass mode, this study provided water significant information about the design and operation of such types of PVT systems. Tonui et al. [3, 4] improved the performance of PVT/a system by reasonable structural modifications to enhance heat abstraction process. By model analysis and experimental verification, Tiwari et al. [5] evaluated the overall efficiency (electrical and thermal) performance of a PVT/a system on climatic conditions in India. Afterwards, Raman and Tiwari [6, 7] conducted further analyses based on energy and exergy for a double-pass hybrid PVT collector on different climatic conditions in India. Sarhaddi et al. [8, 9] evaluated the thermal, electrical and exergetic performance of a typical PVT/a system by a series of detailed energy and exergy analytical approaches. Kumar et al. [10] performed a comprehensive steady state analysis to evaluate and compare the thermal and electrical performance of a double-pass PVT/a system with and without specific configuration of vertical fins. Recently, Al-Alili et al. [11] proposed a hybrid desiccant assisted air conditioner system powered by PVT collector. The simulation and computation results showed such a proposed system could meet the corresponding requirements of buildings in humid and hot climates. Compared with other solar air conditioners, the overall performance of this air conditioner system was improved. Besides, Huang et al. [12] analyzed the performance of an integrated PVT solar water heater with a commercial polycrystalline PV module. Using the concept of primary-energy saving efficiency, the performance of this system was evaluated and measurements for performance improvements were mentioned. By developing a series of steady and dynamic simulation models, Zondag et al. [13, 14] conducted energy performance analyses for a PVT/w sys-tem. Kalogirou et al. Kalogirou et al. [15] presented TRNSYS simulation results for a hybrid PVT solar systems for domestic hot water applications working with either passive (thermo-syphon) or active (pump circulating) mode. Ji et al. [16] constructed a flat-box aluminum-alloy PVT/w system designed for natural circulation, conducted out-door tests and performed dynamic simulations on an improved prototype in a moderate climate region. Chow et al. [17] experimentally analyzed a facade-integrated PV/water-heating system; with which tremendous energy savings can be achieved. Suggestions on further performance improvements of this system were given consequently. Robles-Ocampo et al. [18] constructed an experimental model of a hybrid PVT/w system with the application of a specific bifacial PV module for enhancing electric energy production and estimated the overall solar energy utilization efficiency of this system. Erdil et al. [19] constructed a hybrid PVT domestic water pre-heating system and made experimental measurements for this system in Cyprus. The pay-back period was estimated considering the required modification costs on PV module. It was concluded this low-cost hybrid system with a relatively short pay-back period was economically attractive. With a theoretical model, Dubey et al. [20] evaluate the thermal, exergetic and electrical performance of PVT flat plate water collectors by changing the number of collectors connected in series, alternative series/parallel combination modes and different weather conditions of India. Results showed the flat plate water collectors partially covered by PV module were beneficial for those users whose priority requirements were hot water generation, and those fully covered were beneficial for users whose priority requirements were electricity generation. With numerical simulations, Santbergen et al. [21] constructed a one-coversheet-and-tube PVT system for domestic hot water heating. The mechanisms determining the electrical and thermal yields we reanalyzed and improvement measures were proposed in this work. Pei et al. [22] designed a detailed simulation model of a heat-pipe PVT system, analyzed and compared the annual electrical and thermal performance of such HP-PVT systems with and with-out auxiliary heating equipment in different climate conditions of China.

A new approach of a hybrid photovoltaic thermal collector (PVT) through an experimental study has been proposed by K. Touafek [23, 24, 27 and 28]. This novel collector constitutes a new technical approach to maximize the total output of conversion with lower cost compared to the traditional hybrid collectors, it is characterized by a galvanized steel absorber placed at the lower part of the module, and this absorber is an enclosure containing the coolant, a prototype is made at the Unit of Applied Research in Renewable Energy in Ghardaïa.

The sheet and tube configuration which is adapted in the design of plan thermal collectors is used. The galvanized steel was used for its low cost. The advantages of this hybrid collector are better heat absorption and lower production cost compared to other configurations of hybrids collectors [25, 26].

In this paper, an attempt is made to investigate the thermal performance of the collector studied by Touafek [23, 28]. A detailed thermal model will be developed to calculate the thermal parameters of this prototype. Some corrections are done on heat loss coefficients in order to improve the thermal model of a PV/T water and air collector.

2. Numerical Approach

Fig. 1. Shows the simplified diagram of the new proposed PVT hybrid collector. It consists of a monocristallin's photovoltaic module (with its three layers: tempered glass, layer of the cells with the ethylene vinyl acetate (EVA), and lay down Tedlar) type (UDTS50) of 1.29 m length and 0.33 m width, and galvanized steel absorber placed at the lower part of the module, and this absorber is an enclosure containing the coolant, which can be air, water, or glycol. An insulation of the hybrid collector is necessary, it allows better thermal performances, and this insulation is ensured by glass wool.

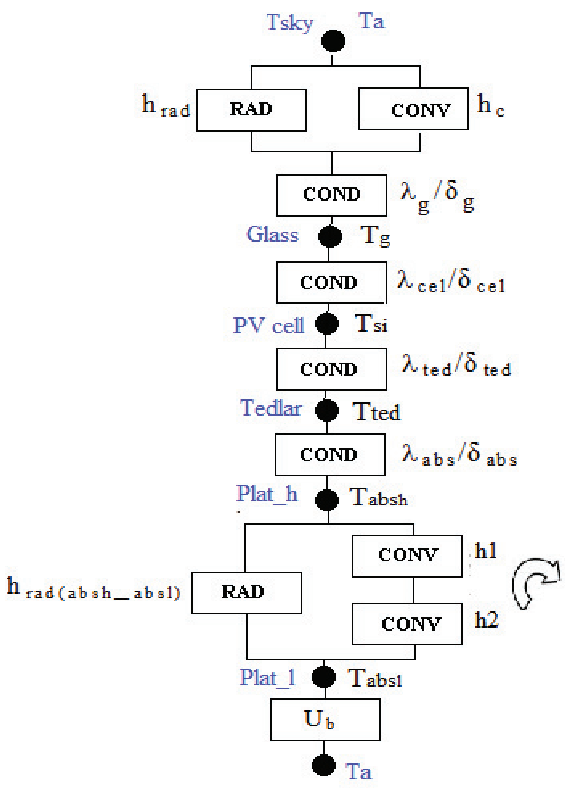

Fig 1. Simplified diagram of new PV/T collector

In order to write the energy balance equation for each component of a PV/T water collector, the following assumptions have been made:

- One dimensional heat conduction is good approximation for the present study.
- The glass cover is at uniform temperature.
- The mass flow rate of the working fluid is the same in all sections of the tubes.
- The material properties of the glass, cover, absorber, and the insulation are constant.

For an element of length *dy* in the direction of flow, the heat balance equations for each component of (PV/T) solar water heating system when is subjected to the illumination *G*, selecting dq_u the quantity extract heat from water:

- *For solar cells of PV module:*

$$\delta_c \rho_c Cp_c \frac{\partial T_c}{\partial t} = G\alpha_c \tau_g + AU_1\left(T_a - T_c\right) - AU_2\left(T_c - T_{abs_h}\right) - \eta_e \tau_g G \tag{1}$$

An expression for temperature dependent electrical efficiency of a PV module Schott [29] is given by

$$\eta_e = \eta_0\left(1 - 0.0045\left(T_c - T_{ref}\right)\right) \tag{2}$$

- *For top part of absorber surface*

$$\delta_p \rho_p Cp_p \frac{\partial T_{absh}}{\partial t} = AU_2\left(T_c - T_{absh}\right) - Ah_{rad\left(abs_h - abs_b\right)}\left(T_{absh} - T_{absb}\right) - Ah_f\left(T_{absh} - T_f\right) \tag{3}$$

- *For the bottom part of absorber surface*

$$\delta_p \rho_p Cp_p \frac{\partial T_{absb}}{\partial t} = Ah_{rad\left(abs_h - abs_b\right)}\left(T_{absh} - T_{absb}\right) - AU_b\left(T_{absb} - T_a\right) - Ah_f\left(T_{absb} - T_f\right) \tag{4}$$

- *For water flowing through the duct*

The energy balance of flowing water inside the duct

$$M_f Cp_f \frac{\partial T_f}{\partial t} = Ah_1\left(T_{absb} - T_f\right) + Ah_2\left(T_f - T_{absb}\right) \tag{5}$$

For the heat transfer inside the water channel, in solar thermal water collectors, the water channel bottom (h_2) and top (h_1) heat transfer coefficients are typically assumed to be identical. The fluid heat transfer coefficient in the channel is a function of the Nusselt number, *Nu*, fluid thermal conductivity λ and hydraulic diameter *D*.

At some location long the flow direction the absorbed solar energy heats up the plate to a temperature T_p. Energy is transferred from the plate to the ambient air at Ta through the back loss coefficient U_b, to the fluid at T_f trough the convection heat transfer coefficient h_2, and to the low part of the absorber through the linearized radiation heat transfer coefficient $h_{rad(absh\text{-}absb)}$. Energy is transferred to the low part of absorber from the fluid through the heat transfer coefficient h_1. The part of energy is lost to the ambient air through the combined radiation and convection coefficient.

The radiative heat transfer coefficient term (h_r) has been added to overall heat transfer coefficient from glass external to ambient (U_l). Further, the corresponding coefficient of convective loss caused by the wind can be calculated from equation (h_c).

$$U_1 = \left(\frac{1}{h_r + h_c} + \frac{\delta_g}{\lambda_g}\right)^{-1} \tag{6}$$

The convective heat flux depends on the wind speed and can be calculated by [31]:

$$h_c = 2.8 + 3V_w \tag{7}$$

$$h_{rad} = \varepsilon_g \sigma\left(T_c^2 + T_{sky}^2\right)\left(T_c + T_{sky}\right) \tag{8}$$

The effective temperature of the sky (T_{sky}) is calculated from the following empirical relation [32]:

$$T_{sky} = 0.0552(T_a)^{1.5} \tag{9}$$

In the estimation of overall loss coefficient from cell to absorber (U_2), has been calculated from Eq. (10)

$$U_2 = \left(\frac{\delta_c}{\lambda_c} + \frac{\delta_{ted}}{\lambda_{ted}} + \frac{\delta_p}{\lambda_p} \right)^{-1} \tag{10}$$

A radiative loss exists from the top to the bottom part of absorber surface $h_{rad(absh\_absb)}$. [33]

$$h_{rad(abs_h - abs_b)} = \frac{4\sigma \overline{T}^3}{(1/\varepsilon_p)(1/\varepsilon_p) - 1} \tag{11}$$

In the estimation of overall back loss coefficient (U_b), the convective heat transfer coefficient on the back surface of PV/T, ($h_{conv,b}$) has been calculated from Eq (7)

$$U_b = \left(\frac{\delta_p}{\lambda_p} + \frac{\delta_i}{\lambda_i} + \frac{\delta_{tol}}{\lambda_{tol}} + \frac{1}{h_{conv,b}} \right)^{-1} \tag{12}$$

Solving equations expressing the conservation of energy for an element of volume between the glass, cell, the absorber and the fluid expressed in terms of different coefficient function of U_1, U_2, h_1, h_2, $h_{rad(absh\text{-}absb)}$ and T_a , leads us to determine the U_L and F_R for the second collector coefficient of overall loss.

$$dq_u = F'\left[(\tau\alpha)G - U_L\left(T_f - T_a\right)\right]bdy \tag{13}$$

$$\dot{m} C_p \frac{dT_f}{dy} = bF'\left[(\tau\alpha)G - U_L\left(T_f - T_a\right)\right] \tag{14}$$

The Hottel-Whiller model [30] is adopted here. The actual useful energy gain by the water is equal to the collector heat removal factor time's maximum possible useful energy gain. Therefore the heat collected by the water can be rewritten as

$$q_u = AF_R\left[(\tau\alpha)G - U_L\left(T_{f_i} - T_a\right)\right] \tag{15}$$

Where

$$U_L = \frac{2h_f U_b U_3 + \left(2h_f h_{rad(abs_h - abs_b)} + h_f^2\right)\left(U_3 + U_b\right)}{2h_f h_{rad(abs_h - abs_b)} + h_f^2 + h_f U_b} \tag{16}$$

U_L for this collector is not just the top loss coefficient in the absence of back losses but also accounts for heat transfer between the top and bottom of absorber surface.

$$F' = \frac{2h_f h_{rad(abs_h - abs_b)} + h_f^2 + h_f U_b}{h_{rad(abs_h - abs_b)}\left(2h_f + U_3 + U_b\right) + h_f^2 + h_f U_b + h_f U_3 + U_3 U_b} \tag{17}$$

The amount of solar energy available for the thermal system is reduced since electrical energy is extracted by the solar cells:

$$(\tau\alpha)G = \left(\left(\tau_g \alpha_c\right) - \tau_g \eta_{pv} \frac{A_c}{A_T} \right) G \tag{18}$$

The heat removal efficiency factor (F_R) can be calculated as:

$$F_R = \frac{\dot{m} C_p}{AU_L}\left[1 - \exp\left(-\frac{F' AU_L}{\dot{m} C_p}\right)\right] \tag{19}$$

Where F_R is the heat removal factor, defined as the ratio of heat extracted by the collector to heat extracted when the whole collector is at fluid inlet temperature. For a fixed mass flow rate, F_R and U_L are almost constant quantities for a thermal collector. Therefore, if thermal efficiency (η_{th}) is plotted against $(T_{fi}-T_a)/G$, a straight line is obtained.

The cell efficiency represented as a function of the module temperature. As is shown in the eqt (2). Where the η_0 is the reference efficiency of the solar cell at T_{ref} =25°C which is in our study *15%*.
The temperature of the photovoltaic panel may be written as :

$$T_c = \left(X_1 - X_2\eta_e\right)G + X_3T_a + X_4T_{fi} \tag{20}$$

The expressions for X_1, X_2, X_3, and X_4 suggest that these should be constant. It can be noted that a linear relationship exists between the temperature of the PV panel (T_c) and electrical efficiency (η_e). For constant value of solar radiation (G), the electrical efficiency of the system can be expressed as:

$$\eta_e = Y_1 - Y_2\left(\frac{T_{fi} - T_a}{G}\right) \tag{21}$$

If Y_1, Y_2 are constant values, then the variation of electrical efficiency (η_e) of the system with T_a and T_{fi} should give a plane.

The climatic, operating and design parameters of the PV/T collector during validation process are described in Table 1.

Table1. Ambient conditions and design used in simulations

Solar PV/T collector parameters	**Value**
The solar radiation intensity at the reference conditions, G_{ref}	1000 W/m^2
The ambient temperature at reference conditions, $T_{a,ref}$	298 K
The length of PV/T collector, L	1.29 m
The width of PV/T collector, H	0.33 m
The solar cell temperature at reference conditions, $T_{cell,ref}$	298 K
The electrical efficiency at the reference conditions, $\eta_{el,ref}$	0.15
The thickness of glass cover, δ_g	0.003 m
The conductivity of glass cover, λ_g	1 W/m K
The transmittivity of glass cover, τ_g	0.92
The emissivity of glass, ε_g	0.88
The emissivity of cell, module, ε_{cell}	0.8
The absorptivity of cell, module, α_c	0.75
The thickness of silicon solar cell, δ_{si}	35 x10^{-5} m
The conductivity of silicon solar cell, λ_{si}	131 W/m K
The absorptivity of tedlar, α_T	0.26
The thickness of tedlar, δ_T	0.0002 m
The conductivity of tedlar, λ_T	163W/m K
The emissivity of absorber, ε_p	0.4
The thickness of absorber (galvanized iron) , δ_p	0.003 m
The conductivity of absorber, λ_p	65 W/m K
The thickness of back insulation, δ_i	0.05 m
The conductivity of back insulation, λ_i	0.035 W/m K
The wind speed, V_w	1 m/s
Collecteur angle, ϕ	35°

Table 2. The typical results of computer simulation program under the sample conditions of Table 1, $T_{fi} = T_a = 300K$, $V_w = 1m/s$, $G = 1000\ W/m^2$.

Solar PV/T collector parameters	Value
Case of air	
The thermal efficiency, η_{th}	16.24 %
The electrical efficiency, η_{el}	11.12 %
The outlet air temperature, T_{fo}	303.13 K
The solar cell temperature, T_c	355.42 K
The rate of useful thermal energy, Q_u	69.36 W
The electrical energy , E_e	44.30 W
The solar absorbed flux, S	597.68 W
The heat capacity of flowing air, C_p	1.006 kJ/kg K
Case of water	
The thermal efficiency, η_{th}	54.51 %
The electrical efficiency, η_{el}	11.13 %
The outlet air temperature, T_{fo}	302.53 K
The solar cell temperature, T_c	355.29 K
The rate of useful thermal energy, Q_u	232.83W
The electrical energy , E_e	44.30 W
The solar absorbed flux, S	597.60 W
The heat capacity of flowing water, C_p	4.180/kg K

3. Results and discussion

From the presented results of Fig. 2, we can see that the maximum electrical energy is reached at 1 p.m. and is 44.34 W powers.

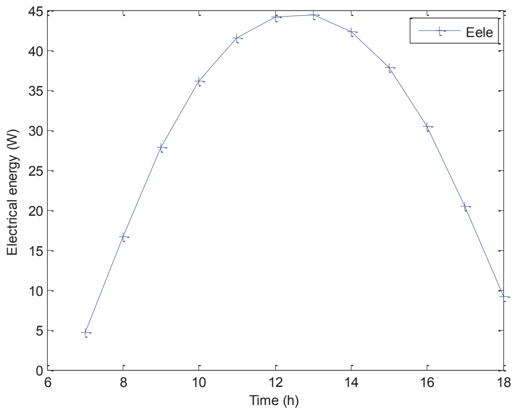

Fig.2 The hourly variation of electrical energy

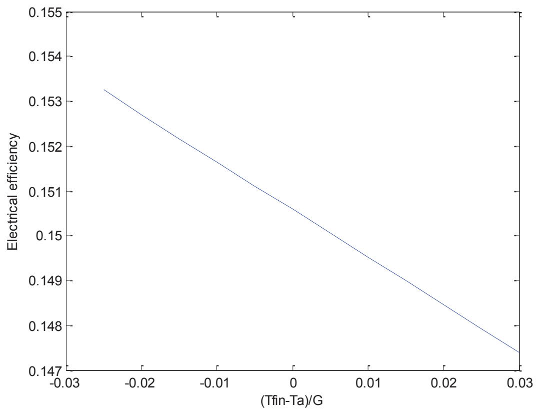

Fig. 3. Variations of the electrical efficiency according to the reduced temperature

A simulation was performed to predict the electrical efficiency of PVT-panel. Fig. 3 show electrical efficiency variation, the ration $\Delta T/G$ (C°W^{-1}m²) is the reduced temperature, with $\Delta T = T_i - T_a$ (K). At zero reduced temperature it was found to be 15.06%.

Increasing the solar radiation intensity, thermal power of the PV/T-collector increases initially and then decreases after attaining solar radiation intensity of about a maximum point. The Hourly variation of thermal power with time in air and water heating are displayed in Fig. 4. It was found to be 69.36W in air heating and 232.83W in the mode of water heat exchanger.

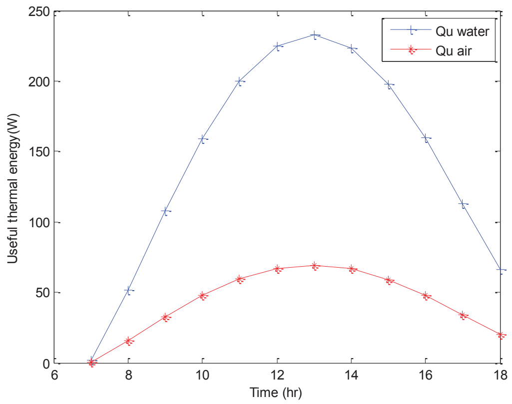

Fig.4 The hourly variation of useful thermal energy

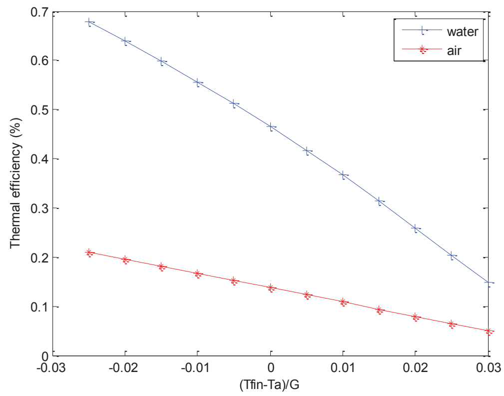

Fig. 5 Variations of the thermal efficiency according to the reduced temperature in the mode of water and air heat exchanger

Fig. 5 shows thermal efficiency variation at PVT-panel, these results are referred to thermal efficiency of water $\eta_{th/w}$ in the mode of water heat exchanger, and air $\eta_{th/a}$ heat extraction, according to the reduced temperature. The PV/T air collector is obviously performing poorest at zero reduced temperature and PVT water collector represents a higher value.

4. Experimental study

To study experimentally the PVT hybrid solar collector, a prototype is made at the Unit of Applied Research in Renewable Energy. It is constituted by the monocrystalline photovoltaic module (UDTS 50 type) below which circulates the air.


Fig. 6 Photograph of the prototype of hybrid PV/T air collector at URAER Unity, Ghardaïa.

The absorber is realized with the galvanized iron of high quality, allowing a good transfer of heat with lower cost compared to copper. Aims to increase its electric and thermal conversion energy effectiveness with a low cost compared to the already existing hybrid collector.

The measured data include the solar radiation intensity, ambient temperature, inlet and outlet air temperature; the simulated values of outlet air temperature, in present work have been validated by their corresponding experimental values in Ref. [23]. The test measurements were made on September 2008. The output temperature reached 37 ◦C for an input of 34 ◦C and an ambient temperature (T_a) of 33◦C. the increase in the temperature of the coolant of an average of 3 ◦C between the entry and the exit. The experimental results for a PV/T air collector make it possible to verify the results obtained by our computer simulation.

5. Conclusion

In this paper, the performance evaluation of a PV/T collector was carried out. A detailed thermal model was developed to calculate the thermal and electrical parameters of a typical PV/T collector. Some corrections were done on heat loss coefficients in order to improve the thermal model of a PV/T collector. A natural circulation of air is the easiest and cheapest way to remove the heat at the module and consequently increase the efficiency. The used absorber satisfying simplicity and cost constraints, it is made by galvanized iron which is cheaper and the manufacturing time is reduced compared with the classical hybrid collectors.

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