

Numerical approach for performance study of hybrid PV/Thermal collector

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Abstract - According to their thermophysical properties, the solar collectors using the working fluid (air, water) show considerably poor efficiency. In this paper, we study the combination of the collector with a photovoltaic module as an efficient method for improving the system performance, particularly the electrical and thermal performance. The mathematical model presented here is based on the energy transfer phenomenon within the various components of the collector. Thus, the transfer equations discretization is carried out using the finite difference method. Our results clearly show the direct impact of various parameters, in particular the inclination angle of the collector and the flow mass rate, on the overall efficiency of the collector. The proposed approach achieves a significant efficiency.

Résumé - Les propriétés thermophysiques du fluide caloporteur (air, eau) utilisé habituellement dans les capteurs solaires ne donnent que des rendements médiocres. Dans le but d'obtenir des performances énergétiques élevées, la combinaison du capteur avec un module de cellules solaires paraît être une solution intéressante. L'objectif de ce travail est d'étudier les performances électriques et thermiques d'un capteur hybride PV/T à eau. Le modèle mathématique développé est basé sur le phénomène de transfert d'énergie dans les différents composants du capteur. La discrétisation des équations de transfert est effectuée par la méthode des différences finies. Les résultats obtenus mettent clairement en évidence l'impact direct de différents paramètres, notamment l'angle d'inclinaison du capteur et le débit massique d'eau sur l'efficacité totale du capteur.

Key words: Hybrid PV/T collector - Photovoltaic cells - Solar energy.

1. INTRODUCTION

The energy policy of many countries per concern of reducing their dependence of fossil energies whose exhaustion is inescapable and in order to safeguard the environment, is based on the promotion and the development of renewable energies. Today the direct use of solar energy gives largely promising results. Nevertheless, the usual systems of direct conversion of solar energy, collector and photovoltaic cells, give poor outputs. With an aim of increasing the conversion rate, the combination of the two systems was considered.

The hybrid photovoltaic/thermal collector was the subject of several researches in 25 last years [1]. L.W. Florschuetz [2] modified the Hottel *et al.* [3] analytical model concerning the thermal performances of a flat plate thermal collector to apply it to

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hybrid PV/T collector. T. Bergene *et al.* [4] proposed one model for the performance of hybrid PV/T, based on the energy transfer analysis.

B Sandnes *et al.* [5] developed an analytical model for PV/T collector by the modification of the Hottel and Willier model for the flat plate collectors. They found a good agreement between the experimental results and simulation. K.S. Sopian *et al.* [6] study analytically the performances of simple and double pass hybrid PV/T air collector.

H.A. Zondag *et al.* [7] Developed four numerical models to determine the performances of the hybrid collector. T.T. Chow [8] proposed an explicit dynamic model for a simple cover hybrid collector. The model can give results for the horary analysis of performance and include the instantaneous efficiency thermal and electrical.

S.A. Kalogirou [9] modelled and simulated PV/T collector by using the TRNSYS simulation program and the typical meteorological data for Nicosia and Cyprus. Y. Tripanagnostopoulos *et al.* [10] built and examined covered and uncovered PV/T collector with water and air as working fluid.

J.S. Coventry [11] studies the CHAPS (Combined Heat and Power Solar) hybrid collector PV/T with a panel of the single crystalline silicon cells. It finds an efficiency thermal of about 58 % and one electric efficiency approximate equal to 11 %. G Notton *et al.* [12] developed a simulation model for a PV/T double glass with a multi crystalline silicon photovoltaic module.

A. Joshi *et al.* [13] study the hybrid collector performances for four climatic states concerning the Srinagar site in India. J. Ji *et al.* [14] examine the flow mass rate and the packing factor effects on the thermal and electrical collector performances.

The aim of this work is the numerical study of the thermal and electrical performances of a collecting hybrid PV/T water collector. Our interest will relate particularly by examining the inclination angle of the collector and flow mass rate effects on the overall efficiency of the collector.

2. MATHEMATICAL MODEL

2.1 System description

The hybrid PV/T collector consists of a glass plate, a photovoltaic panel, which is composed of polycrystalline silicon cells, and a metal plate in backside is welded, using tin, parallel tubes intended for the circulation of the working fluid.

The solar cells are inserted in encapsulated materials, which included on the top the transparent TPT (tedlar-polyestertedlar) and EVA (ethylene–vinyl acetate), and by layers EVA and the opaque tedlar on the bottom part (Fig. 1).

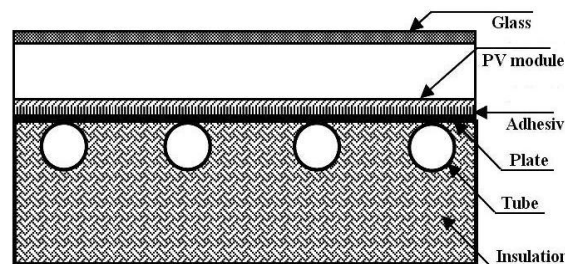


Fig. 1: Hybrid PV/T collector

2.2 The energy balance

The developed thermal model is based on the energy transfer phenomenon in the various components of the collector.

- The outside glass

$$\delta_v \rho_v \frac{C_{p_v}}{2} \frac{\partial T_{ve}}{\partial t} = \frac{G \alpha_v}{2} + h_{r,v-c} (T_c - T_{ve}) + h_{vent} (T_a - T_{ve}) + h_{c,v} (T_{vi} - T_{ve}) \quad (1)$$

where, G is the intensity of the incident solar radiation; δ_v , ρ_v , C_{p_v} and α_v are respectively thickness, mass density, specific heat and absorption of glass cover. T_c , T_a are respectively temperature of sky and temperature of ambient air. $h_{r,v-c}$ is radiation heat transfer coefficient between glass cover and sky; h_{vent} is the convective heat transfer coefficients due to the wind. $h_{c,v}$ is the conduction heat transfer coefficient in the glass cover.

α_v , depends on the incident angle of the solar radiation θ_1 , and can be obtained by [15]:

$$\alpha_v = 1 - \tau_v \quad (2)$$

where τ_v is the transmittance of glass cover determined in the reference [16].

$$\tau_v = e^{-\lambda \delta_v / \cos \theta_2} \quad (3)$$

$$\theta_2 = \arcsin \left(\frac{\sin \theta_1}{n_v} \right) \quad (4)$$

with: θ_2 is the refraction angle. n_v is the refraction index of the glass and λ is the extinction coefficient of the glass.

- The inside glass

$$\delta_v \rho_v \frac{C_{p_v}}{2} \frac{\partial T_{vi}}{\partial t} = \frac{G \alpha_v}{2} + (h_{v,pv-v} + h_{r,pv-v}) (T_{pv} - T_{vi}) + h_{c,v} (T_{ve} - T_{vi}) \quad (5)$$

$h_{v,pv-v}$, $h_{r,pv-v}$ are respectively the convective coefficient and radiation heat transfer coefficient between glass cover and PV panel.

- PV module

$$\delta_{pv} \rho_{pv} C_{p_{pv}} \frac{\partial T_{pv}}{\partial t} = G(\alpha \tau)_p - E + h_{c,p-pv} (T_p - T_{pv}) + (h_{v,pv-p} + h_{r,pv-p}) (T_{vi} - T_{pv}) \quad (6)$$

where, E is the electrical power output. δ_{pv} is the thickness of PV. ρ_{pv} , C_{pv} is the mass density and the specific heat of PV panel respectively.

$(\alpha\tau)_p$ is the effective absorptance of the PV panel given by the relation [16].

$$(\alpha\tau)_p = \frac{\tau_v \tau_r \alpha_p}{1 - (1 - \alpha_p)r} \quad (7)$$

τ_r and r are the transmission (due to the reflexion partial of the incidental radiation) and the reflexion coefficients respectively [16].

$$\tau_r = \frac{1 - r}{1 + r} \quad (8)$$

and

$$r = \frac{1}{2} \left[\frac{\sin^2(\theta_2 - \theta_1)}{\sin^2(\theta_2 + \theta_1)} + \frac{\tan^2(\theta_2 - \theta_1)}{\tan^2(\theta_2 + \theta_1)} \right] \quad (9)$$

The electrical power output E depends on the temperature of the cells T_{pv} ; it is calculated by as [17].

$$E = GP\tau_v\eta_0 \times [1 - \varphi_c (T_{pv} - 25)] \quad (10)$$

where, η_0 is the electrical conversion efficiency at the reference temperature $T_r = 25$ °C. φ_c is the temperature coefficient of the solar cell with $\varphi_c = 0.0045$ °C⁻¹. P is the packing factor of the cell.

$h_{c,pv-p}$ is the conduction heat transfer coefficients in the adhesive layer.

- The plate absorber

$$\delta_p \rho_p C_{pp} \frac{\partial T_p}{\partial t} = h_{c,p-pv} (T_{pv} - T_p) + \frac{A_{pt}}{A_p} h_{c,p-t} (T_t - T_p) \quad (11)$$

Where δ_p , ρ_p , A_p are respectively thickness, mass density and specific heat of the plate. $h_{c,p-t}$: conduction heat transfer coefficients between plate absorber and tubes. A_{pt} : surface between the plate absorber and the tubes, given by the following relation:

$$A_{pt} = \frac{\pi}{4} N D_e l \quad (12)$$

With, N is the total number of tubes and D_e is the external diameter of tubes.

- Tubes

$$\delta_t \rho_t C_{pt} \frac{\partial T_t}{\partial t} = \frac{A_{pt}}{A_t} h_{c,p-t} (T_p - T_t) + \frac{A_f}{A_t} h_{v,t-f} (T_f - T_t) + \frac{A_i}{A_t} h_{c,i-t} (T_{ii} - T_t) \quad (13)$$

where δ_t , ρ_t and C_{p_t} are respectively thickness, mass density and specific heat of the tubes. $h_{v,t-f}$ are the convective heat transfer coefficients between the tubes and the fluid.

A_f is the contact surface of tube- fluid given by the relation

$$A_f = N \pi D_i l \quad (14)$$

A_i is the insulation surface.

- The fluid

$$\dot{m} C_{p_f} (T_f - T_f^*) = A_t h_{v,t-f} (T_t - T_f) + A_{if} h_{v,i-f} (T_{ii} - T_f) \quad (15)$$

with, \dot{m} : Flow mass rate of fluid. C_{p_f} : Specific heat of the fluid. $h_{v,i-f}$: Convective heat transfer coefficients between the insulation and fluid, T_f^* the fluid temperature of the preceding section.

A_{if} : the fictitious surface of the fluid flow on insulation with

$$A_{if} = N D_e l \quad (16)$$

- The inside insulation

$$\delta_i \rho_i \frac{C_{p_i}}{2} \frac{\partial T_{ii}}{\partial t} = \frac{A_{if}}{A_i} h_{v,i-f} (T_f - T_{ii}) + h_{c,i} (T_{ie} - T_{ii}) + \frac{A_t}{A_i} h_{c,i-t} (T_t - T_{ii}) \quad (17)$$

where, δ_i , ρ_i and C_{p_i} are respectively thickness and mass density and specific heat of insulation; $h_{c,i}$ conduction heat transfer coefficients in insulation.,

- The outside insulation

$$\delta_i \rho_i \frac{C_{p_i}}{2} \frac{\partial T_{ie}}{\partial t} = h_{r,i-s} (T_s - T_{ie}) + h_{c,i} (T_{ii} - T_{ie}) + h_{vent} (T_a - T_{ie}) \quad (18)$$

$h_{r,i-s}$ is radiation heat transfer coefficient between insulation and the ground.

2.3 Heat transfer coefficients

- Convective heat transfer coefficients due to the wind is given according to McAdams by the relation [15]

$$h_{vent} = 5.7 + 3.8 V_{vent} \quad (19)$$

V_{vent} is the wind velocity ($m.s^{-1}$)

- Radiation heat transfer coefficients between the outside glass and the sky on the one hand and between the outside insulation and the ground in addition are respectively given as [15]

$$h_{r,v-c} = \varepsilon_v \sigma (T_c^2 + T_{ve}^2) (T_c + T_{ve}) \quad (20)$$

$$h_{r,i-s} = \varepsilon_i \sigma (T_s^2 + T_{ie}^2) (T_s + T_{ie}) \quad (21)$$

- Natural convective heat transfer coefficient between the glass and the PV is given by the expression

$$h_{v,v-pv} = \frac{Nu k_{air}}{b} \quad (22)$$

k_{air} is the thermal conductivity of the air. b is the distance between the glass and the PV. Nu is the Nusselt number calculated by using the following correlations [16].

$$Gr < 1700 + 47.8\phi \quad Nu = 1.013 \quad (23)$$

$$Gr > 80000 \quad Nu = 2.5 + 0.0133(90 - \phi) \quad (24)$$

$$\text{Otherwise} \quad Nu = \left[0.06 + 3.10^{-4} (90 - \phi) \right] Gr^{0.33} \quad (25)$$

Gr is the Grashof number defined by

$$Gr = \frac{g \beta \Delta T b^3}{\nu^2} \quad (26)$$

ϕ the inclination angle of the collector($^\circ$). β is being the thermal dilation coefficient; for the air $\beta \approx T^{-1}$

The heat transfer between the tubes and the working fluid is done by forced convection. For the circular piping, one can use the following correlations to determine the heat transfer coefficients by convection between the fluid and the tubes $h_{v,t-f}$, on the one hand and between the fluid and insulation $h_{v,i-f}$, on the other hand.

a - In the case of a laminar flow: $Re < 2100$

- for $Gz < 100$

$$Nu = 3.66 + \frac{0.085 Gz}{1 + 0.047 Gz^{2/3}} \left[\frac{\mu_f}{\mu_n} \right]^{0.14} \quad (27)$$

- for $Gz > 100$

$$Nu = 1.86 Gz^{1/3} \left[\frac{\mu_f}{\mu_b} \right]^{0.14} + 0.87 \left(1 + 0.015 Gz^{1/3} \right) \quad (28)$$

b - In the case of a transitory flow: $2100 < Re < 10000$

$$Nu = 0.116 \left(Re^{2/3} - 125 \right) Pr^{1/3} \left[1 + \left(\frac{D_i}{l} \right)^{2/3} \right] \left(\frac{\mu_f}{\mu_n} \right)^{0.14} \quad (29)$$

c - In the case of a turbulent flow

$$Nu = 0.023 Re^{0.8} Pr^{1/3} \left[\frac{\mu_f}{\mu_n} \right]^{0.14} \quad (30)$$

μ is the dynamic viscosity of the fluid at the temperature considered. Re , Pr , Gz , respectively Reynolds, Prandtl and Graetz numbers.

$$h_{v,t-f} = \frac{Nu k_t}{\delta_t} \quad (31)$$

$$h_{v,i-f} = \frac{Nu k_i}{\delta_i} \quad (32)$$

- Conduction heat transfer coefficients in the various layers of the collector are obtained by the relation

$$h_{c,n} = \frac{k_n}{\delta_n} \quad (33)$$

k_n is the thermal conductivity of the layer. δ_n is the thickness of the layer
For $\delta_i \gg D_e$, taking the approximation of the reference [8]: $h_{c,i-t} \approx h_{c,i}$

2.4 The thermal and electrical efficiencies of the collector

The thermal efficiency of the collector is given by

$$\eta_t = \frac{Q_u}{A_c G} \quad (34)$$

The useful power Q_u is

$$Q_u = \dot{m} C_p (T_{fs} - T_{fe}) \quad (35)$$

T_{fe} , T_{fs} are the outlet and inlet temperatures of the fluid. One can evaluate Q_u according to the average temperature of the plate $T_{p,m}$ by the relation [1].

$$Q_u = A_c [S - U_L (T_{p,m} - T_a)] \quad (36)$$

with

$$S = G \times (\alpha \tau)_p \quad (37)$$

U_L is the overall loss coefficient.

Hottel *et al.* [3] modified the above relation and proposed the expression

$$Q_u = A_c F_t [S - U_L (T_{fe} - T_a)] \quad (38)$$

with, F_t is the thermal transfer factor.

The calculation of the factors U_L and F_t is given in appendix A.

- The electrical efficiency is written [18]:

$$\eta_e = \eta_0 [1 - \varphi_c (T_{pv} - 25)] \quad (39)$$

- The distribution of the temperature in a photovoltaic panel, which consists of polycrystalline silicon cells, is given by [19]:

$$T_{pv} = 30 + 0.0175(G - 150) + 1.14(T_a - 25) \quad (40)$$

3. RESULTS AND DISCUSSION

The simulation program elaborated and developed in Fortran language is used to study the variation of many parameters of operation. The input parameters of the hybrid collector are displayed in **Table 1**. All the results are carried out for the typical day of 21-June with the use of the meteorological conditions of Constantine town, Algeria [20].

Table 1: Input parameters of the simulation model

Component	Size	Value
Glass	Thickness	0.003 m
	Mass density	2530 kg.m ⁻³
	Specific heat	836 J.K ⁻¹ .m ⁻¹
	Thermal conductivity	0.93 W.K ⁻¹ .m ⁻¹
	Extinction coefficient	0.2 cm ⁻¹
	Absorptance	0.066
	Transmission	0.88
	Emissivity	0.83
	Refraction index	1.5
	Distance between the glass and the PV	0.045 m
PV module	Thickness	0.002 m
	Mass density	2702 kg.m ⁻³
	Specific heat	903 J. K ⁻¹ .m ⁻¹
	Thermal conductivity	237 W.K ⁻¹ .m ⁻¹
	Absorptance	0.85
	Emissivity	0.95
	Packing factor	0.97
Plates	Thickness	0.003 m
	Mass density	8940 kg.m ⁻³
	Specific heat	38 J. K ⁻¹ .m ⁻¹
	Thermal conductivity	300 W.K ⁻¹ .m ⁻¹
Tubes	Mass density	8940 kg.m ⁻³
	Specific heat	38 J. K ⁻¹ .m ⁻¹
	Thermal conductivity	300 W.K ⁻¹ .m ⁻¹

	External diameter	0.028 m
	Internal diameter	0.018 m
	Tubes numbers	10
	Length	1 m
Insulation	Thickness	0.03 m
	Mass density	24 kg.m ⁻³
	Specific heat	919 J . K ⁻¹ .m ⁻¹
	Thermal conductivity	0.039 W.K ⁻¹ .m ⁻¹
	Emissivity	0.11

The curves of figure 2 illustrate the temporal variation of the photovoltaic polycrystalline silicon cells temperature for system (PV/T)_{sv} (without glass), system PV/T and PV panel with tedlar layer insulation.

We can see clearly that the temperature of the photovoltaic cells of PV module is higher than that of system (PV/T)_{sv} with a flow mass rate equal to 0.1 kg.s⁻¹ which makes it possible to cool the fluid in the tubes thus preventing the excess of the temperature in PV module.

For system PV/T, it is obvious that the presence of the glass cover caused an increase in the temperature (greenhouse effect) in the solar cells contrary to system (PV/T)_{sv} which is ventilated naturally.

In figure 3, we can remark that the variation of the electrical efficiency is very significant in the three systems. It is higher in system (PV/T)_{sv}. For maximum solar radiation intensity about 1057 W.m⁻², it varies between 14.2 % and 15.3 % due to the fact that the electrical efficiency of a PV module depends on the operating temperature. The efficiency decreases linearly with the increase in the temperature of the cell [15].

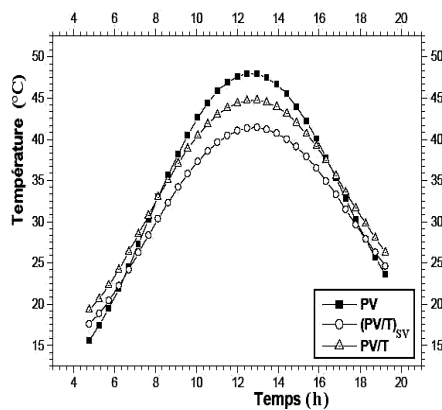


Fig. 2: Daily temperature variation for the three systems

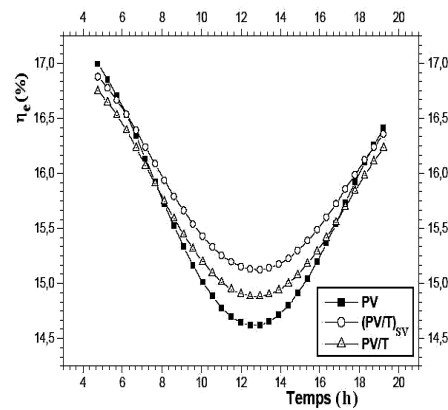


Fig. 3: Daily variation of the electrical efficiency for the three systems

The evolution of the thermal efficiency according to the flow mass rate for various conduction heat transfer coefficients $h_{c,pv-p}$ in the adhesive layer (between the PV and the plate) and for a maximum solar radiation intensity (1057 W.m⁻²) is illustrated in figure 4.

We can notice that for all cases studied the thermal efficiency undergoes significant variations for a flow mass rate of water included between 0.001 kg.s^{-1} and 0.04 kg.s^{-1} and reached its maximum for the value of 0.2 kg.s^{-1} [21].

The analysis of the curves of figure 4 makes it possible moreover to note that the maximum thermal efficiency is about 50-70 for values of $h_{c,pv-p}$ respectively equal to 25 and $10000 \text{ W.m}^{-2}\text{K}^{-1}$.

Concerning the maximum electrical efficiency, figure 5 shows that for the flow mass rate value of 0.2 kg.s^{-1} the efficiency varies between 14.2% and 15.5 % for the four values of $h_{c,pv-p}$ considered here. Thus, it seems clearly that the thermal properties of the adhesive layer exploit a significant role on the thermal and electrical efficiency of the collector and consequently the use of materials of good thermal conductivity leading to an increase in solar conversion.

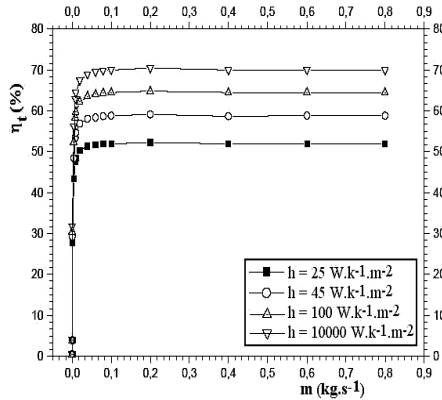


Fig. 4: Thermal efficiency variation according to the flow mass rate

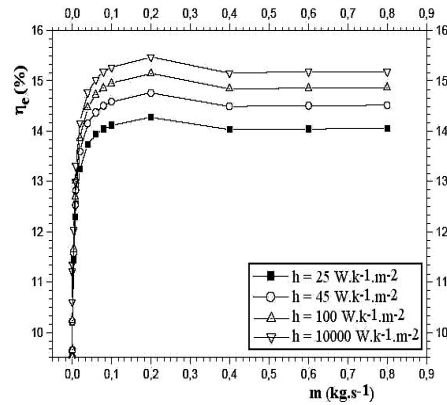


Fig. 5: Electrical efficiency variation according to the flow mass rate

The effect of the inclination angle and the flow mass rate on the characteristics current-voltage (I-V) and power-voltage (P-V) were also examined.

The curves of figures 6 and 7 show that when the inclination angle of the collector ϕ varies, the characteristics I-V and P-V of the collector change considerably. Thus, and as one can note it, in these figures the increase in the inclination angle causes a reduction in the incidental direct solar radiation intensity on the collector [22] which leads to a reduction in the electrical power (Fig. 7) and the short-circuit current I_{cc} (Fig. 6), whereas the tension in open circuit V_{co} is almost constant.

In addition, the increase in the flow mass rate of water decreases the temperature of the cells it results from it an increase in the maximum power (Fig. 9) with an increase in the tension in open circuit V_{co} between 25 and 28 volts. The short-circuit current I_{cc} (Fig. 8) is almost constant.

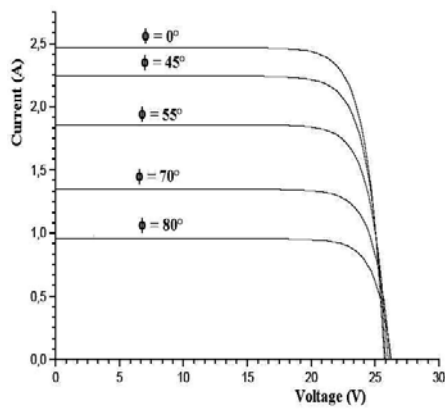


Fig. 6: I-V characteristic according to the inclination angle

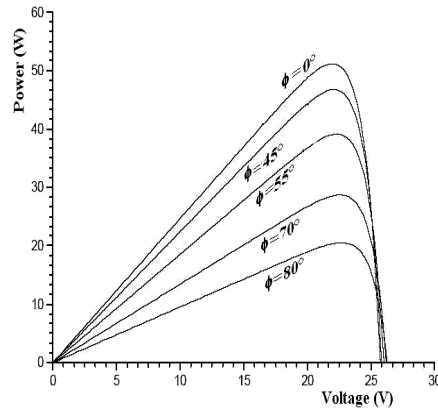


Fig. 7: P-V characteristic according to the inclination angle

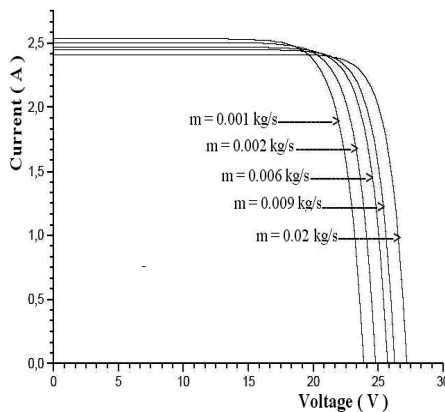


Fig. 8: I-V characteristic according to the mass flow rate

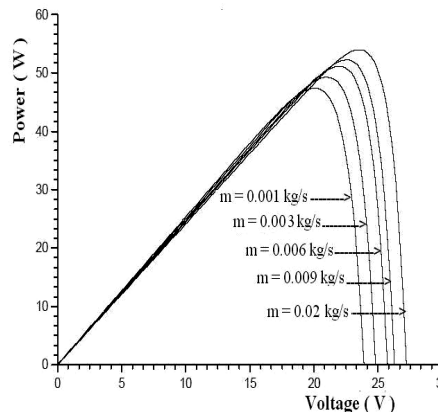


Fig. 9: P-V characteristic according to the mass flow rate

4. CONCLUSION

This paper has presented the thermal and electrical performances of a photovoltaic/thermal hybrid collector. The results of the study demonstrated the effect of the inclination angle, the flow mass rate of water and the conduction heat transfer coefficient in the adhesive layer, on the overall efficiency of the collector.

We have deduced that the increase in the inclination angle leads to a reduction in the electrical power. Concerning the effect of the flow mass rate of water, it is concluded that by increasing this last, the temperature of the cells decreases and it results an increase in the maximum power from it.

The results obtained here also made it possible to note that the use of materials of good thermal conductivity for the adhesive layer makes it possible to clearly improve the overall efficiency (thermal and electrical) of system PV/T.

Appendix A

The overall loss coefficient U_L is calculated using the following relation

$$U_L = U_b + U_t + U_e \quad (41)$$

The bottom loss coefficient is given by

$$U_b = \frac{k_i}{\delta_i} \quad (42)$$

The edge loss coefficient is calculated by the relation:

$$U_e = \frac{(UA)_{\text{edge}}}{A_c} \quad (43)$$

with

$$(UA)_{\text{edge}} = \frac{k_e}{\delta_e} P_e \quad (44)$$

k_e , δ_e , P_e are respectively thermal conductivity, thickness and perimeter of the case. The top loss coefficient is calculated by the expression

$$U_t = \frac{1}{\frac{c}{T_{pv,m} \left(\frac{T_{pv,m} - T_a}{N_v + f} \right)^e + \frac{1}{h_{\text{vent}}}} + \frac{\sigma (T_{pv,m} + T_a) (T_{pv,m}^2 + T_a^2)}{\frac{1}{\varepsilon_{pv} + 0.00591 N_v h_{\text{vent}}} + \frac{2 N_v + f - 1 + 0.133 \varepsilon_{pv} - N_v}{\varepsilon_v}} \quad (45)$$

with

$$c = 520 \left(1 - 0.000051 \phi^2 \right) \quad (46)$$

$$f = \left(1 + 0.089 h_{\text{vent}} - 0.1166 h_{\text{vent}} \varepsilon_{pv} \right) \times \left(1 + 0.07866 N \right) \quad (47)$$

$$e = 0.430 \left(1 - \frac{100}{T_{pm}} \right) \quad (48)$$

The thermal transfer factor is given by the equation

$$F_t = \frac{m' C_p}{A_c U_L} \left[1 - e^{-\frac{A_c U_L F'}{m' C_p}} \right] \quad (49)$$

$$F' = \frac{1/U_L}{w \left[\frac{1}{U_L (D_i + (w - D_i) F)} + \frac{1}{w h_{c,pv-p}} + \frac{1}{\pi D_i h_{v,f-i}} \right]} \quad (50)$$

with

$$F = \frac{\tanh \left[m' \frac{w - D_e}{2} \right]}{m' \frac{w - D_e}{2}} \quad (51)$$

$$m' = \sqrt{\frac{U_L}{k_{pv} \delta_{pv} + k_p \delta_p}} \quad (52)$$

w is the distance between n the tubes

N_v is the glass numbers

$T_{pv,m}$ is the average temperature of the PV panel

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