Performance Evaluation of Solar Photovoltaic Thermal Air Collector Based on Energy and Exergy Analysis

A.M. Shaneb
Misurata University /Mechanical
Engineering Department
Misurata, Libya

A. Abouda
Misurata University /Mechanical
Engineering Department
Misurata, Libya

M. Taha
Misurata University /Mechanical
Engineering Department
Misurata, Libya

M. Abdullrahem
Misurata University /Mechanical
Engineering Department
Misurata, Libya

m.shaneb@eng.misuratau.edu.ly

Abstract— **This work presents a comprehensive parametric study of thermal and electrical performance of four different designs of photovoltaic/ thermal air heaters (PV/T) based on energy and exergy analysis. The results are compared to that from the conventional design of flat duct PV/T. The main goal of these designs is to enhance the heat transfer inside the air duct and consequently to increase the system's electrical power output. A computer simulation program has been developed in order to calculate the electrical and thermal PV/T collector parameters. The results indicate that an increase of 52, 52.4, and 68.25 % in the total system efficiency has been achieved from the corrugated duct, double pass design and finned duct respectively. Furthermore, at a low mass flow rates the electrical efficiency has also been improved for the three types by 9.8, 14.6 and 12.6 % respectively compared to that of flat duct design. The temperature of the PV arrays has a major effect on the exergy efficiency and removing heat from the PV module keeps the net power output at satisfactory level.**

Index Terms: **Solar energy, exergy, photovoltaic/thermal.**

I. INTRODUCTION

Photovoltaic solar systems are one of the most promising sources of electrical energy. The low efficiency of PV modules is caused by the lower conversion efficiency of their cells which is estimated to be from 6-15% for silicon solar cells [1]. Different techniques have been used to improve the PV modules performance. Some of these techniques are based on increasing the income radiation on the PV cells by using concentrators, lenses, or using solar tracking. However, the problem associated with these techniques is the increase of the PV cells temperature above the operating limit. In fact, the increase in cells temperature decreases the solar cells efficiency and hence decrease the maximum power output from about 0.3- 0.5% per degree Celsius [2, 3].

The reduction of electrical efficiency can be partially avoided by heat extraction using cooling media. The setup is established by attaching a rectangular duct to the back surface of the PV module with controlled circulation of fluid at low inlet temperature. Through the duct, heat is extracted from the PV module to maintain the electrical efficiency at satisfactory values. In addition, the extracted heat can be used as a thermal energy for low temperature applications which, in turn, increase the total system efficiency [4]. The new configuration calls photovoltaic thermal system (PV/T) that generates both heat and electricity simultaneously.

During the past four decades, the performance of PV/T systems has been studied widely using experimental and numerical methods. Among those studies is the steady-state model presented by Sopian *et al*. [5]. The authors investigated the performance of a single and double pass photovoltaic thermal solar collector suitable for solar drying applications. Garg, had developed a computer simulation model for predicting the transient performance of conventional PV/T air heating collector with single and double pass configurations [6]. Kalogirou, Tripanagnostopoulos, presented TRNSYS simulation resulted for hybrid PV/T solar system for domestic hot water application [7]. Alfegi *et al*. investigated a double duct PV/T air system experimentally with fin, to increase heat transfer, and with compound parabolic concentrating, to concentrate solar radiation on solar cells [8]. Elsfi and Gandhidasan, presented a simulation model for predicting the thermal and electrical performance of PV/T air heater. The authors evaluated a double-pass flat plate hybrid PV/T with attached vertical fins of different configurations and different material [9]. Ben cheikh el hocine *et al*., presented a detailed thermal model to calculate the thermal parameters of typical PV/T collector. Some corrections were conducted on heat loss coefficient in order to improve the thermal model of PV/T collector. The authors indicated that their results are in a good agreement with the experimental measurements observed in their previously published research [10]. The studies mentioned above examine the PV/T systems performance from the point of view of energy. However, exergy analysis usually provides more realistic and practical aspects of the process than the energy analysis.

Joshi and Tiwari evaluated exergy and energy analysis of PV/T air system for cold climate of India. They observed that an instantaneous energy and exergy efficiency of PV/T air heater varies between 55–65% and 12–15%, respectively. Chow *et al.* developed energy and exergy analysis of PV/T - water with and without glass cover. Their numerical model results indicated that the higher energy efficiency for a model with a glass cover but the exergy efficiency was higher for the model without cover [12]. The exergy analysis of integrated photovoltaic thermal solar water heater under constant flow rate and constant collection temperature modes was investigated by Tiwari *et al.* [13]. Agrawal and Tiwari evaluated energy and exergy of a hybrid micro-channel PV/T module under constant mass flow rate of air [14]. Sarhaddi *et al.* evaluated the exergy performance of solar photovoltaic thermal (PV/T) air collector. Their results showed that the overall exergy efficiency and energy efficiency are about 45 and 10.75%, respectively for an unglazed PV/T model [15]. Joshi *et al.* also studied the performance characteristics of a photovoltaic (PV) and photovoltaic-thermal (PV/T) system in terms of energy efficiency and exergy efficiency, respectively. They proposed equations for the energy, electrical, and exergy efficiency of a PV system [16]. Sarhaddi, Farahat and Ajam investigated exergetic optimization of typical flat plate solar collector under given operating conditions. They considered the geometric and the other operating parameters as variables [17].

In this study, the thermal and electrical performance of four types of PV/T photovoltaic thermal air heaters based on energy and exergy analysis is studied. The type-1 module is the flat rectangular duct PV/T air heater as shown in Fig.1- a. Type-2 has a corrugated duct for both the rear plate and the back surface of solar cells as shown in Fig.1-b. However the wavelike shape of the back surface of the cells and that of the bottom plate is along the flow direction. Type 3 module is the double-pass PV/T solar air heater with vertical fins attached to the lower surface of the solar cells as shown in Fig.1-c. A number of ten rectangular fins are attached in the air passage between the back surface of cells and the rear plate in Type 4 (Figure.1-d.). All of the PV modules are polycrystalline PV solar modules (SOL-060-01). The electrical specifications are shown in table1.

II. ENERGY ANALYSIS

A. Energy balance Equations:
A.1. Energy balance for the glass cover:

On a glass cover, the energy gains are the solar energy absorbed by the glass cover $G\alpha_g$ where α_g is the absorptivity of the glass cover and the convection and radiation heat transfer from the cell surface to the glass cover. The energy losses through the glass cover are the convection heat transfer from the glass surface to the ambient due to the wind and the thermal radiation from the glass surface to the sky.

Table.1 Thermo-Physical Parameters and Operating Parameters.

Solar(PV/T) air collector parameters	Value
(PV) module type	Polycrystalline silicon (60 W)
The length of (PV) module, L_1	1.5 m
The width of PV module, L_2	0.5 m
The short-circuit current at the reference conditions, $I_{sc,ref}$	3.79 A
The open-circuit voltage at the reference conditions, $V_{oc,ref}$	21.6 V
The maximum power point current (actual) at the reference conditions, $I_{mp,ref}$	3.45 A
The maximum power point voltage (actual) at the reference conditions, $V_{mp,ref}$	17.4 V
The solar radiation intensity at the reference conditions, G_{ref}	1000 $W/_{m^2}$
The ambient temperature at reference conditions, $T_{amb,ref}$	298.15K
The solar cell temperature at reference conditions, $T_{cell,ref}$	298.15K
The electrical efficiency at the reference conditions, $\eta_{el,ref}$	15 %

$$G\alpha_g + \left(h_{c,cell\rightarrow g} + h_{r,cell\rightarrow g}\right) * \left(T_{cell} - T_g\right)$$
$$= \left(h_{r,g\rightarrow sky} + h_w\right) * \left(T_g - T_{sky}\right) \qquad (1)$$

A. 2. Energy balance for the cell:

On the cell surface, the incoming absorbed insolation $S = \tau_g \alpha_c I_c$ where τ_g is the transitivity of the glass, is distributed to thermal losses to the glass cover by natural convection and radiation, convection heat transfer from the back surface of cells to the working fluid, radiation heat transfer from the cells back surface to the rear plate and electrical energy produced by the cells.

$$S = \left(h_{c,cell\rightarrow g} + h_{r,cell\rightarrow g}\right) * \left(T_{cell} - T_g\right) + \left(h_{c,cell\rightarrow f}\right)$$
$$* \left(T_{cell} - T_f\right) + \left(h_{r,cell\rightarrow p}\right)\left(T_{cell} - T_p\right)$$
$$+ P_{el} \qquad (2)$$

Where $P_{el} = \left(1 - \alpha_g\right) * \alpha_c \eta_c GA_s$ (The electrical energy produced by solar cell).

A. 3. Energy balance for the fluid:

For the working fluid, the heat gained from the cells back surface by convection is distributed to the heat carried away by the fluid and the convection heat transferred from the fluid to the rear plate.

$$\left(h_{c,cell\rightarrow f}\right) * \left(T_{cell} - T_f\right)$$
$$= \dot{m}c_p(T_{out} - T_{in}) + \left(h_{c,f\rightarrow p}\right)$$
$$* \left(T_f - T_p\right) \qquad (3)$$

A. 4. Energy balance for rear plate:

On the rear plate, the heat gain from the fluid via convection and the heat gain from the cells via radiation are balanced by the conduction loss to ambient.

$$\left(h_{c,f\rightarrow p}\right) * \left(T_f - T_p\right) + \left(h_{r,cell\rightarrow p}\right) * \left(T_{cell} - T_p\right) = h_p(T_p - T_{amb}) \quad (4)$$

III. DETERMINATION OF HEAT TRANSFER COEFFICIENTS

The convection heat transfer coefficient from the glass cover due the wind is given by the dimensional equation recommended by McAdams (1954) [2].

$$h_w = 5.7 + 3.8V_w \quad (5)$$

Where V_w is the wind speed in m/s and usually taken equal to $1.5\ m/s$ [18].

The radiation heat transfer coefficient from the glass to the sky can be obtained by the following correlation [19].

$$h_{r,g\rightarrow sky} =$$

$$\sigma\varepsilon_g\left(T_g + T^*_{sky}\right)\left[\left(T^2_g + T^{*2}_{sky}\right) * \left(\frac{(T_g - T^*_{sky})}{(T_g - T_{amb})}\right)\right] \quad (6)$$

Where $\sigma = 5.67 \times 10^{-8}\ W/m^2k^4$ is the Stefan-Boltzmann constant, ε_g is the emissivity of the glass cover and T^*_{sky} is the sky temperature which can be obtained from the correlation recommended by Swinbank [20].

$$T^*_{sky} = 0.0552T^{1.5}_{amb} \quad (7)$$

The radiation heat transfer coefficient between the cell and the glass cover and between the cell and the rear plate can be obtained respectively, by [2].

$$h_{r,cell\rightarrow g} = \frac{\sigma\left(T^2_{cell} + T^2_g\right) * \left(T_{cell} + T_g\right)}{1/\varepsilon_c + 1/\varepsilon_g - 1} \quad (8)$$

And

$$h_{r,cell\rightarrow p} = \frac{\sigma\left(T^2_{cell} + T^2_p\right) * \left(T_{cell} + T_p\right)}{1/\varepsilon_c + 1/\varepsilon_p - 1} \quad (9)$$

The conduction heat transfer coefficient cross the insulation to the ambient can be estimated by

$$h_p = \frac{K_i}{\Delta_i} \quad (10)$$

where K_i is the thermal conductivity of the insulation and Δ_i is the mean thickness of the insulation.

The convection heat transfer coefficient between the glass cover and the cell surface for the flat, finned, and corrugated duct is calculated by

$$h_{c,cell\rightarrow g} = N_{u,cell\rightarrow g} * \frac{K}{H_c} \quad (11)$$

Figure.1. a. Flat Duct PV/T Air Heater.

Figure.1. b. Corrugated Duct PV/T Air Heater.

Figure.1. c. Double-Pass PV/T Air Heater.

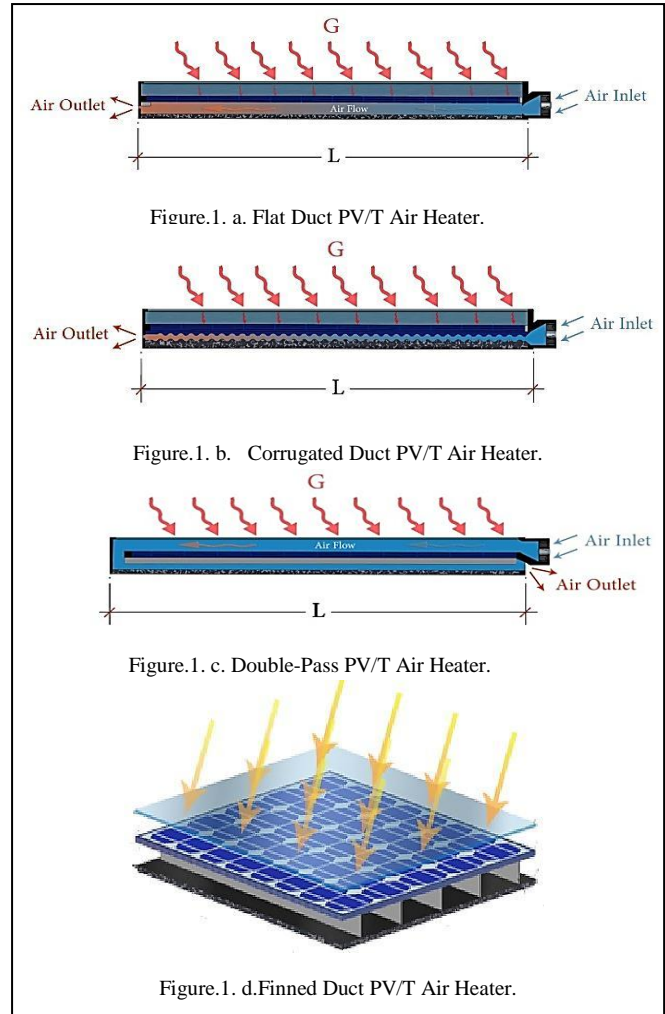Figure.1. d.Finned Duct PV/T Air Heater.

Figure.1. Proposed Channel Designs of PV/T Air Collector.

where K is the thermal conductivity of the air, H_c is the mean gap thickness between the cells surface and the glass cover, and $N_{u,cell\rightarrow g}$ is the Nusselt number for the natural convection in the channel.

For all types except the double-pass design, $N_{u,cell\rightarrow g}$ can be estimated by the following correlation [21].

$$N_{u,cell\rightarrow g} = 1 + 1.441\left[1 - \frac{1708(sin1.8\theta)^{1.6}}{R_a\ cos\theta}\right]\left[1 - \frac{1708}{R_a\ cos\theta}\right]^+$$

$$+ \left[\left(\frac{R_a\ cos\theta}{5830}\right)^{1/3} - 1\right]^+ \quad (12)$$

where $\theta°$ is the inclination angle of the solar panel and R_a is Reyleigh number which can be obtained using the following correlation [2].

$$R_a = \frac{\rho^2 Cp_{air}\ g\ \beta\ \left(T_{cell} - T_g\right)H_c^3}{K\ \mu} \quad (13)$$

g = gravitational constant.
β = volumetric coefficient of expansion.
Cp_{air} = heat capacity of the air.

μ = dynamic viscosity.

where the meaning of (+) exponent is that only positive values of the terms in the square brackets are to be used (i.e., use zero if the term is negative).

The Nusslet number between the solar cell and the glass cover for the double pass design can be estimated from the correlation:

$$N_{u\,,cell \to g} = 0.052R_e^{0.78}P_r^{0.4} \tag{14}$$

The forced convection heat transfer coefficient between the cells back surface and the fluid and between the fluid and the rear plate are calculated by

$$h_{c,f \to p} = h_{c,cell \to f} = N_{u\,,cell \to f} * \frac{K}{D_h} \tag{15}$$

D_h (m) is the hydraulic diameter of the air flow channel formed by the cells surface and the bottom plate which for flat and corrugated design can be calculated from:

$$D_h = {2\text{W} * H_g}\Big/{W + H_g} \tag{16}$$

While for the finned and double pass design is

$$D_h = {2\text{W} * H_g}\Big/{W + H_g(1 + x)} \tag{17}$$

where W (m) is the PV module width, H_g (m) is the mean gap thickness between the cell back surface and the bottom plate, x is number of fins attached to the fluid duct and $N_{u\,,cell \to f}$ is the Nusselt number for the convection of fluid moving in the air flow channel.

For the flat duct, $N_{u\,,cell \to f}$ can be estimated by the following correlation [22,23].

$$N_{u\,,cell \to f} = 0.01583R_e^{0.8} \tag{18}$$

Where R_e Reynolds number is defined as follows

$$R_e = \frac{\rho \overline{U_f} D_h}{\mu} \tag{19}$$

$\overline{U_f}$ is the mean velocity of the fluid in the channel. This correlation is valid for $3,000 \le R_e \le 50,000$,

For the corrugated duct, Nusselt number for forced convection in the air flow channel can be estimated using the following correlation. [24]

$$N_{u\,,cell \to f} = 0.0743R_e^{0.76} \tag{20}$$

For the finned duct, Nusselt number for the convection of fluid moving in the air flow channel can be estimated using the following correlation [2].

$$N_{u\,,cell \to f} = 0.0293R_e^{0.8} \tag{21}$$

While for the double-pass design, Nusselt number can be calculated from the following correlation [25].

$$N_{u\,,cell \to f} = 1.017R_e^{0.471}P_r^{0.4} \tag{22}$$

Reynolds number can be calculated as follows.

$$R_e = \frac{2\rho_c(\overline{U_f} * H_g)}{\mu_c} \tag{23}$$

To increase the results accuracy, density, thermal conductivity and dynamic viscosity of the air can be estimated from the following empirical correlations [3].

$$\rho_c = 3.9147 - 0.016082T + 2.9013 * 10^{-5}T^2 - 1.9407$$
$$* 10^{-8}T^3 \tag{24}$$

$$K_c = (0.0015215 + 0.097459T - 3.3322 * 10^{-5}T^2)$$
$$* 10^{-3} \tag{25}$$

$$\mu_c = (1.6157 + 0.06523T - 3.0297 * 10^{-5}T^2)$$
$$* 10^{-6} \tag{26}$$

The correlations above valid for $(280k - 470k)$.

The pressure drop experienced by the air flow through the channel due to the flow friction and by neglecting the effect of surface roughness and the effect of the air compressibility can be computed from the relation [21] [22] [23].

$$\Delta \text{P} = \frac{2f\rho \overline{U_f}^2 L}{D_h} \tag{27}$$

Where f the friction factor for turbulent flow and for type1 is given by [2].

$$f = 0.0791R_e^{-0.25} \tag{28}$$

And for the corrugated and finned duct can be calculated by [24].

$$f = 0.059R_e^{-0.2} \tag{29}$$

While for the double-pass design, it is calculated from the following correlation:

$$f = 0.0791R_e^{-0.25} + 0.059R_e^{-0.2} \tag{30}$$

The pumping power required to pass the air through the channel can be obtained by using the relation [23].

$$P_{fan} = \frac{\dot{m}\Delta P}{\rho} \tag{31}$$

IV. EXERGY ANALYSIS

The exergy efficiency of PV/T air collector system can be expressed in two different ways; it can be defined base on net output exergy (thermal and electrical exergy output) and /or in terms of exergy loses (exergy destruction). In this study, we will use only the first technique for calculating the exergy efficiency as follows:

$$\eta_{ex} = \frac{Net\ (desired)\ output\ exergy\ rate}{Net\ (supplied)\ input\ exergy\ rate}$$
$$= \frac{\sum Ex_{out}}{\sum Ex_{in}} \tag{32}$$

The exergy balance equation for a control volume can be written as follows [26].

$$\dot{E}_{x_{fin}} - \dot{E}_{x_{ini}} = \dot{E}_{x_Q} - \dot{E}_{x_w} + \dot{E}_{x_{in}} - \dot{E}_{x_{out}}$$
$$- \dot{I}_{c,v} \tag{33}$$

Where:

$\dot{E}_{x_{fin}}$ is exergy rate of final mass in control volume.

$\dot{E}_{x_{ini}}$ is exergy rate of initial mass in control volume.

The exergy change of final and initial mass in control volume presents the thermal exergy gain for the fluid occupied in the system from the PV and it can be defined as [15] [26].

$$\dot{E}_{x_{fin}} - \dot{E}_{x_{ini}} = \left(\frac{m_{cell}C_{P,cell}}{\Delta t}\right)\left[T_{cell} - T_{amb} - T_{amb}ln\left(\frac{T_{cell}}{T_{amb}}\right)\right]$$
$$- \left(V_{oc}I_{sc} - V_{mp}I_{mp}\right)\frac{T_{cell}}{T_{sun}} \qquad (34)$$

The specific heat capacity of silicon solar cell can be calculated from [27].

$$C_{p,cell} = 0.844 + 1.8 \times 10^{-4}T_{cell} - 1.55 \times 10^4 T_{cell}^{-2} \qquad (35)$$

$\dot{E}_{x_Q}$ is heat transfer exergy rate or the inlet exergy rate for PV/T system that includes only solar radiation intensity exergy and it can be expressed according to Petela theorem as follows [26].

$$\dot{E}_{x_{Q.sun}} = GA_s\left[1 - \frac{4}{3}\left(\frac{T_{amb}}{T_{sun}}\right) + \frac{1}{3}\left(\frac{T_{amb}}{T_{sun}}\right)^4\right] \qquad (36)$$

$\dot{E}_{x_w}$ is work exergy rate which presents the net electrical power output and it can be calculated as follows:

$$\dot{E}_{x_w} = P_{el} - P_{fan} = V_{mp}I_{mp} - \frac{\dot{m}\Delta P}{\rho\eta_{fan}} \qquad (37)$$

P_{fan} is the electrical power consumed by the fan to pass the air inside the duct. two DC 12V fans attached to the PV/T module to blow air into the air duct and dimensions of $(12'' * 12'')$.

$\dot{E}_{x_{in}}$ is exergy rate of inlet mass and it can be calculated as follows [26].

$$\dot{E}_{x_{in}} = \dot{m}C_p\left[T_{in} - T_{amb} - T_{amb}ln\left(\frac{T_{in}}{T_{amb}}\right)\right]$$
$$+ \dot{m}RT_{amb}ln\left(\frac{P_{in}}{P_{amb}}\right) \qquad (38)$$

$\dot{E}_{x_{out}}$ is exergy rate of outlet mass and it can be calculated as follows [26].

$$\dot{E}_{x_{out}} = \dot{m}C_p\left[T_{out} - T_{amb} - T_{amb}ln\left(\frac{T_{out}}{T_{amb}}\right)\right]$$
$$+ \dot{m}RT_{amb}ln\left(\frac{P_{out}}{P_{amb}}\right) \qquad (39)$$

$\dot{I}_{c,v}$ is the irreversibility rate in control volume [26].

$$\dot{I}_{c,v} = \sum \dot{E}_{x_{loss,ext}} + \sum \dot{E}_{x_{loss,int}} = \sum \dot{E}_{x_{loss,ext}} + \sum \dot{E}_{x_{loss,des}} \qquad (40)$$

The exergy efficiency for the PV/T system in term of net output exergy can be written by substituting equations 33-40 into equation 32.

η_{ex}
$$= \frac{\left(\frac{m_{cell}C_{P,cell}}{\Delta t}\right)\left[T_{cell} - T_{amb} - T_{amb}ln\left(\frac{T_{cell}}{T_{amb}}\right)\right] - \left(V_{oc}I_{oc} - V_{mp}I_{mp}\right)\frac{T_{cell}}{T_{sun}}}{GA_s\left[1 - \frac{4}{3}\left(\frac{T_{amb}}{T_{sun}}\right) + \frac{1}{3}\left(\frac{T_{amb}}{T_{sun}}\right)^4\right]}$$
$$+ \frac{\dot{m}C_p\left[T_{out} - T_{in} - T_{amb}ln\left(\frac{T_{out}}{T_{in}}\right)\right] + \dot{m}RT_{amb}ln\left(\frac{P_{out}}{P_{in}}\right) + V_{mp}I_{mp} - P_{fan}}{GA_s\left[1 - \frac{4}{3}\left(\frac{T_{amb}}{T_{sun}}\right) + \frac{1}{3}\left(\frac{T_{amb}}{T_{sun}}\right)^4\right]} \qquad (41)$$

V. PERFORMANCE PARAMETERS:

A. Thermal efficiency:

$$\eta_{th} = \frac{\dot{m}c_p(T_0-T_i)}{GA_s} \qquad (42)$$

B. Solar cell efficiency: [28]

$$\eta_c = \eta_r[1 - \beta_r(T_c - T_r)] \qquad (43)$$

Where η_r is the reference efficiency at the reference temperature, T_r is the reference temperature which always takes equal to 25^0C, and β_r is the cell efficiency temperature coefficient (K^{-1}) which can be found from the following correlation:

$$\beta_r = 1/(\text{T}^*_c - T_r) \qquad (44)$$

Where $\text{T}^*_c = 270^0C$ is the solar cell temperature at which its efficiency drops to zero.

C. Electrical efficiency:

The electrical efficiency is given by:

$$\eta_e = \eta_c(1 - R_g)(1 - \alpha_g) \qquad (45)$$

V. SOLUTION FOR THE TEMPERATURE AND EFFICIENCIES

It is obvious that it is difficult to find an analytical solution for the temperatures T_c, T_f, T_g and T_b for the equations 1-4, and the efficiencies for the equations 41-45 as most of the heat transfer coefficients are functions of these temperature. Hence, the values of these parameters can be obtained numerically with an iteration method. The matrix inverse method $[T] = [h]^{-1}[C]$ was used to find the values of the temperatures of PV/T system components, where $[T]$ is the variables matrix, $[h]$ is the coefficients matrix (heat transfer coefficients) and $[C]$ is the constants matrix. A MATLAB script has been developed to solve this system of equations numerically for the four types of configurations.

VI. RESULTS AND DISCUSSION

A. Solar PV/T system components temperatures:

The results show that by increasing the mass flow rate $\dot{m}_f$ from $0.016 - 0.301 kg/s$, the glass cover; solar cell; rear plate and the air outlet temperature decrease significantly for the four types of PV/T configurations. That is due to the increase in the turbulence inside the air
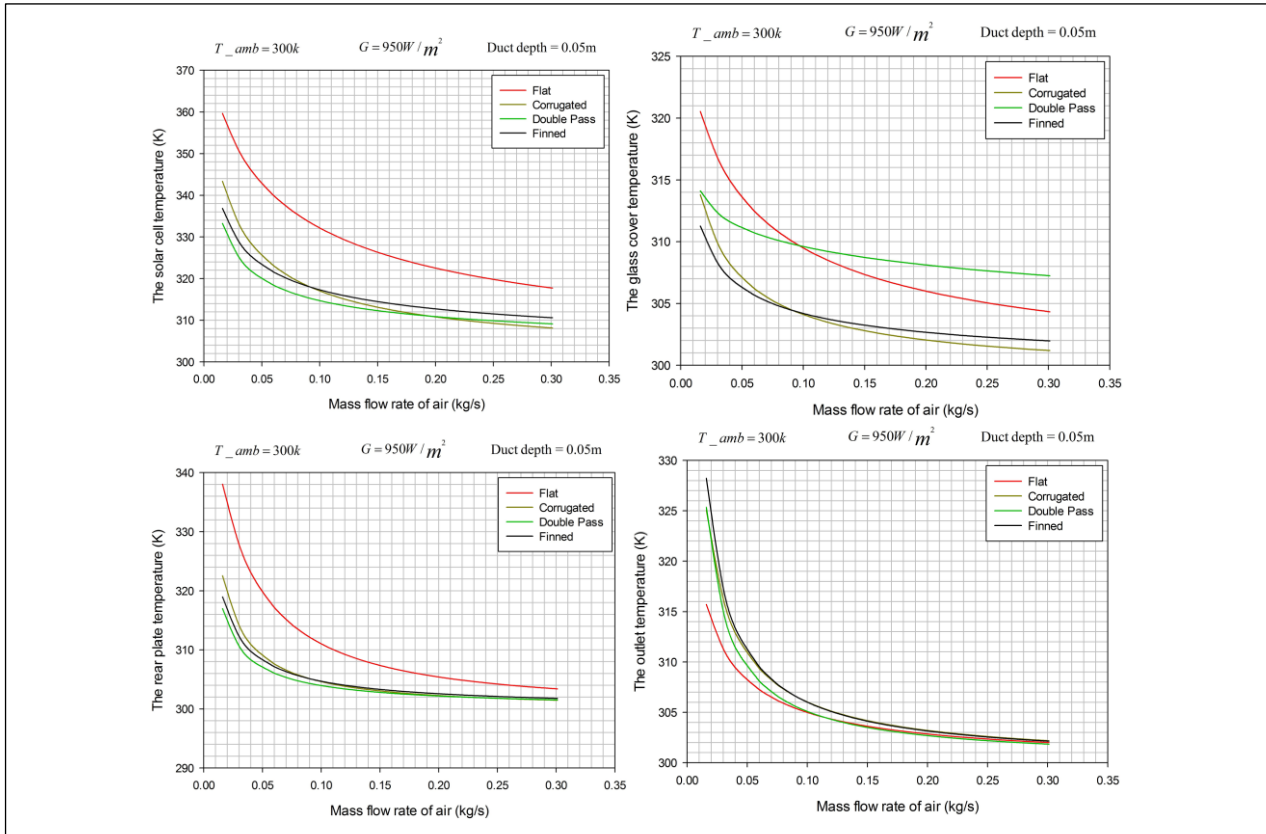
Figure.3. Effect of Mass Flow Rate on the PV/T Component Temperature

duct and the increase in the forced convection of heat transfer coefficient as shown in Figure 3. The temperature increases in type 1 more than that of types 2; 3and 4 for the solar cell, the glass cover and the rear plate while the condition is different with regard to the outlet air temperature. In addition, the corrugated duct design gives slightly higher temperatures than the finned and double-pass types for both the solar cell and the rear plate at low air flow velocities. Nevertheless, it gives approximately the same air outlet temperatures for all mass flow rate variations.

B. Solar PV/T system efficiencies:

The simulation also shows for all types of PV/T configuration that, by increasing air mass flow rate inside the channel, the electrical and solar cell efficiencies increase. That is due to cooling caused by adding more air mass flow rate at ambient temperature. The increase in the heat transfer coefficient between the back surface of cells and the working fluid, results in lower solar cell temperatures. Furthermore, the amount of heat extracted from the PV which is increased by the mass flow rate increase, results in rapid thermal efficiency improvement.

However, the corrugated; finned; and double-pass PV/T configuration give minor differences in electrical efficiency for all mass flow rate variations above 0.1kg/s. In the meantime, all of them result in a higher electrical, solar cell, and thermal efficiencies than the flat PV/T module as shown in Figure 4.

For the exergy efficiency of type1, a significant increase is seen as a result of increasing the mass flow rate until it
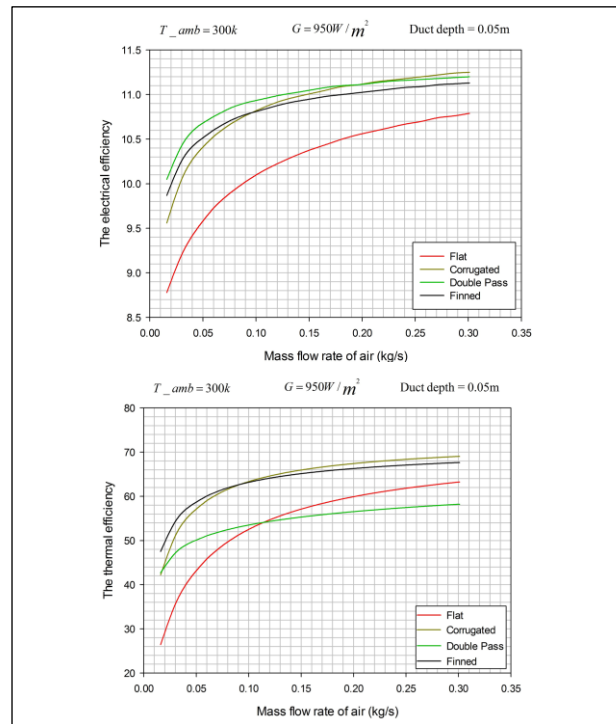

Figure.4. Effect of Air Mass Flow Rate on the PV/T Efficiencies.

reaches 0.2 kg/s then it starts to decrease. Type 2, type 3, and type 4 show decreasing in the exergy efficiency even at low air mass flow rates as shown in Figure5. That is probably due to increasing the pressure drop inside the
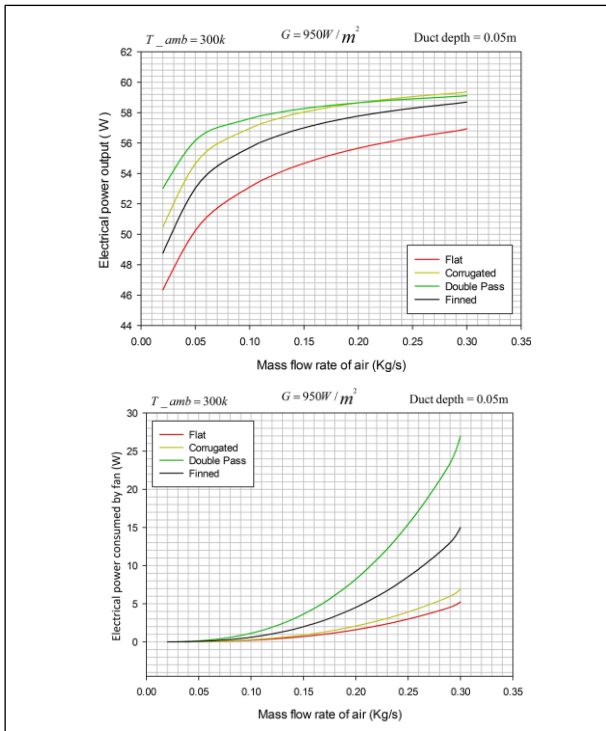
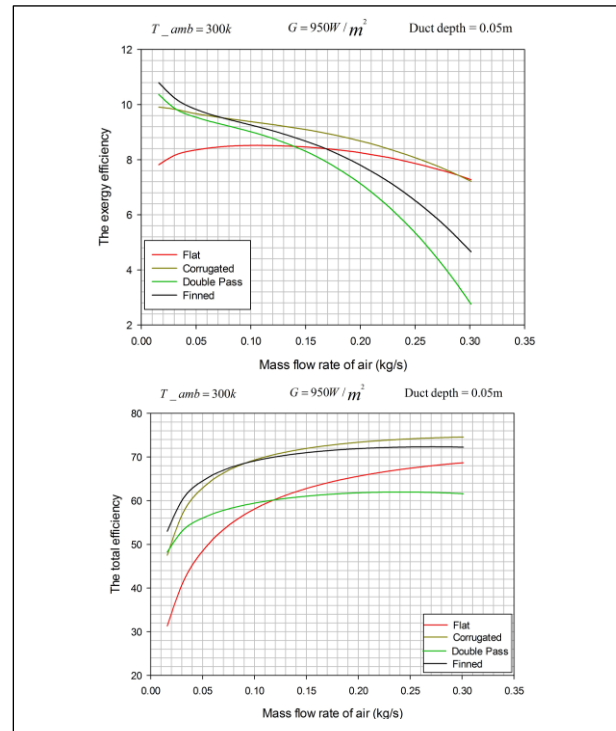Figure.5. Effect of Air Mass Flow Rate on Electrical Power.

Figure.6. Effect of Air Mass Flow Rate on the Total & Exergy efficiencies.

channel which results in more power consumption by the fan to deliver more air inside the channel.

In general, type 1 gives lower exergy than type 2, type 3, and type 4 for wide variation of mass flow rates. However, when the mass flow rate riches 0.15 kg/s, the finned and double-pass duct PV/T start to give lower exergy efficiency than the flat one because of the reason mentioned in the previous paragraph. The finned, corrugated, and the double-pass PV/T show almost similar exergy efficiency until the mass flow rate reaches 0.12 Kg/s. After that, the finned and the double-pass PV/T exergy efficiency drops rapidly with any increase in the mass flow rate.

The amount of electrical energy produced by the solar cell is increasing with any increase in the mass flow rate as shown in Figure 6. However, type 1 produces slightly higher electrical power than type 2 at low mass flow rate and it gives a similar electrical energy output at any mass flow rate higher than 0.061kg/s. Both type 1 and type 2 produce much higher electrical energy than type 3 for all mass flow rate variations.

VII.   CONCLUSION

This study has dealt with performance evaluation of a hybrid photovoltaic thermal (PV/T) air collector system based on energy and exergy analysis. The four types of PV/T modules namely flat rectangular duct PV/T, corrugated duct PV/T, double-pass PV/T with vertical

fins, and finned duct PV/T are considered for comparison purpose. From the study, the following conclusions can be drawn:

- The corrugated duct PV/T gives superior hydraulic performance than the other PV/T modules, which means less pumping power requirements.
- The finned duct PV/T shows a maximum enhancement in heat transfer, especially at low mass flow rates.
- Unlike the corrugated design, the double-pass and finned design show significant drop in exergy efficiency with increase the fan velocity, which is due to the high-pressure drop associated to their design configuration.
- The double-pass PV/T shows higher electrical efficiency for mass flow rates less than 0.18 Kg/s.
- The finned duct PV/T and corrugated duct PV/T show almost similar total efficiency for a long mass flow rate variation. Both of these configuration results higher total efficiency than the double-pass and the flat duct PV/T.

REFERENCES

[1] S.E, A.M, Z.A, M.A, Increasing Efficiency in the Conversion of solar Energy to Electric Power, Energy   Source,21 (1999) 367-377
[2] Duffy and W. A. Beckman, "*Solar engineering of thermal processes*", 2006, Hoboken, N.J. Wiley.

[3] L.W, G.W, L.T, A parametric study in the thermal performance of cross-corrugated solar air collector, Applied Thermal Engineering 26 (2006) 1043-1053.

[4] T.Y, S.M, B.R, C.A, Application Aspects of Hybrid PV/T Solar systems, unknown publisher.

[5] ADNAN IBRAHIM, M.Y. OTHMAN, M.H. RUSLAN, M.A. ALGHOUL, M. YAHYA, AND A. z AHARIM AND K. SOPIAN, " *Performance of photovoltaic Thermal collector (PV/T) With Different Absorbers design* ", Solar Energy Research Institute, University Kebangsaan Malaysia.

[6] H.P. GARG and R.S. ADHIKARI, "*Conventional hybrid photovoltaic/thermal (PV/T) air heating collectors: steady – state simulation* ", Centre for Energy studies, Indian Institute of Technology, Hauz Khas, New Delhi, India, 1997.

[7] S. A. Kalogirou and Y. Tripanagnostopoulos, " *Hybrid PV/T solar systems for domestic hot water and electricity production* ", 2006.

[8] K. Sopian, M.A. Alghoul, Ebrahim M. Alfegi, M. Y. Sulaiman, E. A. Musa, " *Evaluation of thermal efficiency of double-pass solar collector with porous nonporous media* ", Malasia, 2008.

[9] A. M. Elsafi and P. Gandhidasan, "*Comparative study of double-pass flat and compound parabolic concentrated photovoltaic-thermal systems with and without fins''*, Mechanical engineering Department, King Fahd University of petroleum and Minerals, Dhahran, Saudi Arabia, 2015.

[10] Khalifa A. Touafek K. Ben Mousa H. Tabet I. Ben cheikh El hocine H. Haloui H, " *Analysis of a Hybrid Solar Collector Photovoltaic Thermal (PV/T)*", International Conference on Technologies and Materials for Renewable Energy, Environment Sustainability, TMREES15, 2015.

[11] Anand S. Joshi and Arvind Tiwari, " *Energy and Exergy efficiencies of a hybrid photovoltaic-thermal (PV/T) air collector* ", Centre for Energy studies, Indian Institute of Technology, Hauz Khas, New Delhi, India, 2007.

[12] T. T Chow, G. Pei, K. F Fong, Z. Lin, A.LS Chan, J.J *" Energy and exergy analysis of photovoltaic-thermal collector with and without glass cover* ", China, 2008.

[13] Arvind Tiwari, Swapnil Dubey, G.S. Sandhu, M.S. Sodha, S.I. Anwar, " *Exergy analysis of integrated photovoltaic thermal solar water heater under constant flow rate and constant collection temperature modes''*, India, 2009.

[14] Sanjay Agrawal, G.N. Tiwari, " *Energy and exergy and analysis of hybrid micro-channed photovoltaic thermal module* ", Centre for Energy studies, Indian Institute of Technology, Hauz Khas, New Delhi, India, 2010.

[15] F. sarhadi, S. farhat, H. Ajam, A. Behzadmehr, *"Exergetic performance assessment of a solar photovoltaic thermal (PV/T) air collector* ", University of sistan, Iran, 2010.

[16] Anand S. Joshi, Ibrahim Dincer, Bale V. Reddy, " *Effects of various parameters on energy and exergy efficiencies of a solar thermal hydrogen production system* ", University of Ontario Institute of Technology, 2016.

[17] F. sarhadi, S. farhat, H. Ajam, "*Exergetic optimization of flat plate solar collectors* ", university of sistan, Iran, 2008.

[18] W.H. McAdams, Heat Transmission, third ed., McGraw-Hill, New York, 1954.

[19] X.Q. Zhai, Y.J. Dai, R.Z. Wang, Comparison of heating and natural ventilation in a solar house induced by two roof solar collectors, Applied Thermal Engineering 25 (2005) 741–757.

[20] W.C. Swinbank, Long-wave radiation from clear skies, Quarterly Journal of the Royal Meteorological Society 89 (1963) 339.

[21] K.G.T. Hollands, T.E. Unny, G.D. Raithby, L.J. Konicek, free convection heat transfer across inclined air layers, Transcations of the ASME, Journal of Heat Transfer 98 (1976) 189–193.

[22] Y. Piao, E.G. HauptmannG, M. Iqbal, forced convective heat transfer in cross-corrugated solar air heaters, Transcations of the ASME, Journal of Solar Energy Engineering 116 (1994) 212–214.

[23] Y. Piao, Natural, forced and mixed convection in a vertical cross corrugated channel, M. Sc. Thesis, The University of British Columbia, Canada, 1992.

[24] Lin. W, W. Gao, and T. Liu," *A parametric study on the thermal performance of cross corrugated solar air collector*", Applied Thermal engineering, (2006).

[25] Hedayatizadeh. Mahdi, Sarhaddi. Faramarz, Safavinejad. Ali, Ranjbar. Faramarz, Chaji. Hossein, "Exergy loss-based efficiency optimization of a double pass/glazed v-corrugated plate solar air heater",   Energy,   Iran,(2016).

[26] F. Sobhnamayan, A. Hamidi, H. R. Monavari, F. Sarhaddi, S. Farahat, and M. A. Alavi, *"performance evaluation of a solar photovoltaic thermal air collector using energy and exergy analysis*", article in jornal of renewable and sustainable energy, Iran, august 2011.

[27] O.S, B.M, Improving Photovoltaic Module Efficiency Using Water cooling, Heat Transfer Engineering 30(2009) 499-505.

[28] Zondag, H. A. "Flat-plate PV–thermal collectors and systems: a review. Renewable and Sustainable Energy Reviews,12: 891–959, 2008.