Photovoltaic Thermal (PV/T) Collector Performance Evaluation For Partially Covered with PV Modules

Jalal Mohammed Jalil, Ahmed Abdulqader Hussein, Anwar Jabbar Faisal
Electromechanical Engineering Department, University of Technology
E-mail: jalalmjalil@gmail.com, ahmedabdulqaderhussein@gmail.com, anwarjabarfaisal@gmail.com

Abstract. This study experienced the performance of air biased PV/T collector partially covered with glass at various values of irradiance (400, 600, 800, 1000) W/m² and constant ambient temperature. A three various configurations have been demonstrated in this work, case A (glass at inlet portion), case B (glass at mid portion) and case C (glass at outlet portion). The investigation has been performed in term of outlet air temperature and electrical power. A numerical model was developed via using the computational fluid dynamic (CFD) program. 3D steady state, turbulent forced convection model is utilized to solve Navier Stokes and energy equations of air flowing inside the duct. The results compared with the experimental measurements that carried out from indoor conditions using solar simulator, a strong agreement has been achieved between experimental and numerical outcomes. It has been observed that case C (glass at outlet portion) is more favorable in electrical and thermal gain, which generates 44.3°C hot air and 26.6 W electric power, while the maximum outlet air temperature and electric power reach 39°C, 25.4 W and 43°C, 21.05 W for configurations A and B respectively. Furthermore, the analysis of PV/T system concluded that the PV/T system should be operate at moderate air flowrate (0.013 kg/s) which is the specific or optimum flowrate.

Keywords: PVT air collector; PV/T partially covered with PV; Thermal efficiency; Thermal Analysis

1. Introduction
PV/T systems is one of the most popular renewable energy resource that facing the energy crisis all around the world [1]. Its allow us to utilize the solar radiation efficiently and reduce the usage of expensive fossil fuel consumed [2]. The electrical power generated from photovoltaic system depends on the cells temperature and the solar radiation incident directly on it. PV system can converts 6-16% of solar incidence into electricity at 25°C. The remaining 80% of this incident is converted to heat and acted as losses dissipated to the environment. The heat dissipation is not desirable and must be converted into useful energy such as forced air or water circulation with backside of the PV surface. This sophisticated system is recognized as hybrid photovoltaic thermal systems, these systems able to generate electric energy as well as heat energy simultaneously. The generated heat is used for various application, such as the hot water/air can be used for heating space, solar dryers and solar stills. Solar cells power
and efficiency is a leaner decreasing function of temperature this mean when the operating cells temperature is high, the electrical power becomes low so that this circulation of fluid below the panel will act as coolant to the PV cells, enhance its performance and helps to maintain the PV system in a good condition during its operation [3]. In this system, the working fluid used to extracting the heat from PV cells is the air. The system is consist of a basic air channel placed at the back of the PV cells. PV modules is enclosed in a duct and air is circulated in channel by fans. A part of energy generated is required to operate the fans, therefore it must be considered when designing the duct. Optimum channel high and mass flow rate of air are the most effecting parameters in this type [4]. PV/T system is partially covered when designing the duct. A part of energy generated is required to operate the fans, therefore it must be considered when designing the duct. Optimum channel high and mass flow rate of air are the most effecting parameters in this type [4]. PV/T system is partially covered when designing the duct. A part of energy generated is required to operate the fans, therefore it must be considered when designing the duct. Optimum channel high and mass flow rate of air are the most effecting parameters in this type [4]. PV/T system is partially covered when designing the duct.

| Nomenclature | Description |
|--------------|-------------|
| \( A_e \) | Area in the direction of east and west (m²) |
| \( A_y \) | Area in the direction of top and bottom (m²) |
| \( A_z \) | Area in the direction of north and south (m²) |
| \( k_{ap} \) | Thermal conductivity of the absorber plate (W/m.K) |
| \( T_{air} \) | Bulk temperature of air flowing (°C) |
| \( \alpha_{ap} \) | Absorber plate absorptivity |
| \( \tau_g \) | Transmissivity of glass surface |
| \( U, V, W \) | Velocities in X, Y, Z directions (m/s) |
| \( x, y, z \) | Cartesian coordinates |
| \( V_f \) | Air following velocity (m/s) |
| \( i, j, k \) | Indices which indicate positions in the (x, y, z) direction |
| \( \rho \) | Density of air (kg/m³) |
| \( G \) | Solar radiation |
| \( h_{in} \) | Internal heat transfer coefficient (W/m.K) |
| \( \nu_e \) | Effective kinematics viscosity (m²/s) |
| \( \Gamma \) | Diffusion coefficient (N.s/m²) |
air-based photovoltaic thermal collectors with four different packing factor namely: double-path parallel flow design, double-path, counter flow, double-pass, returning flow and single-path design. The experiment done indoor using the solar simulator, the result showed that single-path design is better than the other in electricity generation and for heat generation, double-path parallel flow design is the better one, its showed the maximum overall efficiency of 51% - 67%, while the single-path design showed minimum overall efficiency of 28% - 49% for packing factor equal (0.7). Mishra and Tiwari (2013) [5] studied the influence of coverage rate with PV module on the effectiveness of water biased PV/T system at constant collection temperature, with two configuration are: case A (PV/T system partially covered with PV model) and case B (PV/T system fully covered with PV model). It’s seen that the thermal gain is better for case A due to lesser area of the collector covered with PV in case A than that for case B. Electrical gain in case B is higher than case A due to larger area covered with PV module in case B than that for case A. This work will investigate the electrical and thermal optimization for air based PV/T collector partially covered with glass at three different configurations and investigate the specific flow rate and optimum glass configuration that combination with PV modules.

Figure 1: Sketch of the experimental rig.
2. System Description
A sketch of the experimental rig is illustrated in Fig.(1), and Fig.(2) shows the photograph of three cases for PV/T system that carried out in this work, which is mainly consists of tow pieces of solar cells and one piece of glass attached to the aluminum absorber plate, a three DC fans for forced cooling system, DC voltage regulator for control the flowrate. Solar simulator with microcontroller for tungsten control, sensors and measuring Units. Other important components are solar charge controller and battery for power storage. Air is the working medium and supplied to the thermal channel by three DC fans each of 3 W capacity. The experiment was situated indoor to preserve ambient temperature and the intensity of solar radiation and to avoid rabidly change in outdoor condition. The detail description and dimensions of PV solar cells specification is illustrates in Table (1).

Table 1: Specification of PV modules.

| Two (20 watt) solar cells |  |
|----------------------------|---|
| Dimensions                 | 47cm * 35cm |
| Current at P max           | 1.1 Amp |
| Voltage at P max           | 17.5 V |
| Short circuit current (Ish)| 1.2 Amp |
| Open circuit voltage       | 20 V   |
3. Numerical Modeling

This study includes mathematical formulation, which govern partial differential equations (PDEs) to depict turbulent fluid flow field and heat transfer in a PV/T system. The hybrid PV/T system, which studied numerically, has full scale of experimental duct. The duct has 0.35 m width and 1.4 m length. The cross-section of the duct is (0.35 * 0.03) m². A mesh size of (62 * 22 * 22) elements involved in numerical solution of PV/T system. The mathematical model of the airflow problem is governed by basic conservation of momentum, mass and energy balance equations. Turbulent flow is considered in the present work (the Reynolds number values of the air inlet is 6133). The following assumptions have been considered:

(i) Assumption has been considered steady state with negligible side and bottom losses, body forces and air leakage.
(ii) Conversation equation of three-dimensional model.
(iii) Forced convection of operation and the streamline of air through the duct is uniform.
(iv) Three-dimensional heat transfer conduction through PV modules, glass and absorber plate.
(v) Turbulent airflow has been considered in the channel.

3.1. Governing Equations

The flow of air inside the PV/T system is turbulent with, 3-D, incompressible and steady state. Navier Stokes equations (NSEs) (continuity, momentum and energy) are resolved by control volume (CV) in Cartesian Coordinates and after simplification the equations are as follows [11]:

(i) The Continuity Equation

\[
\frac{\partial}{\partial x} (U) + \frac{\partial}{\partial y} (V) + \frac{\partial}{\partial z} (W) = 0 \tag{1}
\]

(ii) The Momentum Equation

(a) x-Direction

\[
\frac{\partial}{\partial x} (UU) + \frac{\partial}{\partial y} (UV) + \frac{\partial}{\partial z} (UW) = - \frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left( \nu_e \frac{\partial U}{\partial x} \right) + \frac{\partial}{\partial y} \left( \nu_e \frac{\partial U}{\partial y} \right) + \frac{\partial}{\partial z} \left( \nu_e \frac{\partial U}{\partial z} \right) + \\
\frac{\partial}{\partial x} \left( \frac{\partial U}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{\partial U}{\partial y} \right) + \frac{\partial}{\partial z} \left( \frac{\partial U}{\partial z} \right) \tag{2}
\]

(b) y-Direction

\[
\frac{\partial}{\partial x} (UV) + \frac{\partial}{\partial y} (WW) + \frac{\partial}{\partial z} (VW) = - \frac{1}{\rho} \frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left( \nu_e \frac{\partial V}{\partial x} \right) + \frac{\partial}{\partial y} \left( \nu_e \frac{\partial V}{\partial y} \right) + \frac{\partial}{\partial z} \left( \nu_e \frac{\partial V}{\partial z} \right) + \\
\frac{\partial}{\partial x} \left( \frac{\partial V}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{\partial V}{\partial y} \right) + \frac{\partial}{\partial z} \left( \frac{\partial V}{\partial z} \right) \tag{3}
\]

(c) z-Direction

\[
\frac{\partial}{\partial x} (UW) + \frac{\partial}{\partial y} (VW) + \frac{\partial}{\partial z} (WW) = - \frac{1}{\rho} \frac{\partial p}{\partial z} + \frac{\partial}{\partial x} \left( \nu_e \frac{\partial W}{\partial x} \right) + \frac{\partial}{\partial y} \left( \nu_e \frac{\partial W}{\partial y} \right) + \frac{\partial}{\partial z} \left( \nu_e \frac{\partial W}{\partial z} \right) + \\
\frac{\partial}{\partial x} \left( \frac{\partial W}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{\partial W}{\partial y} \right) + \frac{\partial}{\partial z} \left( \frac{\partial W}{\partial z} \right) \tag{4}
\]

(iii) The Energy Equation
\[ \frac{\partial}{\partial x} (UT) + \frac{\partial}{\partial y} (VT) + \frac{\partial}{\partial z} (WT) = \frac{\partial}{\partial x} \left( T_e \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( T_e \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( T_e \frac{\partial T}{\partial z} \right) \quad (5) \]

The \textit{k-\epsilon model} is used to simulate the turbulent flow numerically [12]. This model has couple of equations, which join between turbulent kinetic energy (k) equation and turbulence dissipation rate (\epsilon). All partial differential equations can be prescribed in the following general equation [11]:

\[ \frac{\partial}{\partial x} (\rho u \phi) + \frac{\partial}{\partial y} (\rho v \phi) + \frac{\partial}{\partial z} (\rho w \phi) = \frac{\partial}{\partial x} \left( \Gamma_{\phi} \frac{\partial \phi}{\partial x} \right) + \frac{\partial}{\partial y} \left( \Gamma_{\phi} \frac{\partial \phi}{\partial y} \right) + \frac{\partial}{\partial z} \left( \Gamma_{\phi} \frac{\partial \phi}{\partial z} \right) + S_{\phi} \quad (6) \]

In this equation, the left side is represent the convection terms expressions, while the right side represent diffusion and source expressions.

3.2. Thermal Analysis of Solid Walls

3.2.1. Absorber Plate:

Thermal energy is barge to the blackened Absorber Plate from the direct radiation passed through glass cover in the non-packing area of PV module and fall on it. In addition by the connection between back surface of PV module and the absorber plate due to temperature difference between them, the air flowing in the channel get heated by the thermal energy brought from absorbing plate to the following air [13-14]. At node 3 of absorber plate, the energy balance is shown in Fig.(3) and represented as following:

\[ q_e + q_w + q_t + q_b + q_n + q_s + q_{flux} = 0 \quad (7) \]

\( q_e, q_w, q_b, q_s, q_n \) are the heat transfer by conduction to the east, west, top, bottom, north and south of the node respectively which can be given by [15]

\[ q_e = k_{ap} A_x \frac{\partial T}{\partial x} \frac{T_{ap}(i, j, k) - T_{ap}(i + 1, j, k)}{\Delta x} \quad (8) \]

\[ q_w = k_{ap} A_x \frac{\partial T}{\partial x} \frac{T_{ap}(i, j, k) - T_{ap}(i - 1, j, k)}{\Delta x} \quad (9) \]

\[ q_t = k_{ap} A_z \frac{\partial T}{\partial z} \frac{T_{ap}(i, j, k) - T_{ap}(i, j, k + 1)}{\Delta z} \quad (10) \]

\[ q_b = k_{ap} A_z \frac{\partial T}{\partial z} \frac{T_{ap}(i, j, k) - T_{ap}(i, j, k - 1)}{\Delta z} \quad (11) \]

\[ q_n = q_{\text{conv}}(ap, af) = h_{in} A_y (T_{ap}(i, j, k) - T_{af}) \quad (12) \]

\[ q_s = k_{ap} A_y \frac{\partial T}{\partial y} \frac{T_{ap}(i, j, k) - T_{ap}(i, j - 1, k)}{\Delta y} \quad (13) \]

\( q_{\text{cons}(ab-af)} \) is the heat transferred by convection from absorber plate to the flowing air

\[ h_{in} = 2.8 + 3 v_f \quad (14) \]

Where \( h_{in} \) is the coefficient of heat transferred by convection between the absorber plate and air flowing inside the collector [16]. Where \( v_f \) is the air following velocity.

\[ q_{\text{flux}} = (t_g a_{ab}) A_y G \quad (15) \]
for the portion covered by glass where $\alpha_{ab}$ is the absorber plate absorptivity and equal 0.8 [17], $\tau_g$ is the transmissivity of glass surface and equal 0.9 [18], and

$$q_{flux} = 0 \quad (16)$$

for the portion covered by PV modules. Eq.(7) can be represented as:

$$K_{ab}A_x \left( T_{ab}(i,j,k) - T_{ab}(i+1,j,k) \right) + \frac{K_{ab}A_x}{\Delta x} \left( T_{ab}(i,j,k) - T_{ab}(i-1,j,k) \right) + \frac{K_{ab}A_z}{\Delta z} \left( T_{ab}(i,j,k) - T_{ab}(i,j,k+1) \right) + \frac{K_{ab}A_z}{\Delta z} \left( T_{ab}(i,j,k) - T_{ab}(i,j,k-1) \right) + h_{in}A_y \left( T_{ab}(i,j,k) - T_{amb} \right) + \frac{K_{ab}A_y}{\Delta y} \left( T_{ab}(i,j,k) - T_{ab}(i,j,k-1) \right) + A_y q_{flux} = 0 \quad (17)$$

Suppose:

$$a_e = a_w - \frac{K_{ap}A_x}{\Delta x}$$
$$a_t = a_b = \frac{K_{ap}A_x}{\Delta z}$$
$$a_e = \frac{K_{ap}A_y}{\Delta y}$$
$$a_n = h_{in}A_y$$
$$a_p = a_e + a_w + a_t + a_b + a_n + a_s$$

these terms are substituted in Eq. (17) to give:

$$T_{ab}(i,j,k) = \left[ a_e * T_{ab}(i+1,j,k) + a_w * T_{ab}(i-1,j,k) + a_t * T_{ab}(i,j,k+1) + a_b * T_{ab}(i,j,k-1) + a_n * T_{amb} + a_p * T_{ab}(i,j-1,k) + a_s * q_{flux} \right] / a_p \quad (18)$$

The absorber plate temperature of any point can be calculated by Eq.(18)

4. Results and Discussion

For the three cases, 0.33% of PV/T area is covered with glass at three different position (A- inlet portion, B- mid portion, C- outlet portion), in order to increase heat gain. Figure (4) reveals the variation of air temperature along the distance of PV/T system at different values of irradiance for the three configurations. Figure (5) illustrates the variation of absorber plate temperature along the distance of PV/T system at different values of irradiance for the three configurations.
Figure (6) shows the variation of (PV cell and glass) temperature along the distance of PV/T system at different values of irradiance for the three positions. From Fig.(4) one can notice that the maximum temperature of outlet air was obtained at configuration C reached 44.3 °C at 1000 W/m² and this sharp increase in outlet air temperature is due to the direct radiation fall into glass at outlet portion and restricts the heat gain in this position without make this radiation to passing parallel with PV modules. It was also observed that there is gradual increase in air temperature in case B due to the existence of glass at mid portion make the air get heated by direct radiation in the mid side moreover, to the thermal energy acquired by convection with PV modules back surface at outlet portion. Minimum outlet air temperature was obtained at case A because of that the direct radiation fall in the inlet portion when the cooling air was supplied. From Fig.(5) one can observe that the configuration A have convergent temperatures along the distance of absorber plate, while configuration B have the higher temperature along the distance of absorber plate at mid portion rather than the last end of absorber plate because of the solar radiation fall directly in mid portion and obscured by PV modules in the front and last end of plate, configuration C have the optimum rising in absorber plate temperature reached 60 °C. The maximum PV modules temperature achieved at configuration A reached 70 °C at 1000 W/m² due to the existence of PV module at the last side of PV/T air system and get heated by heat transferred from the hot air below it as well as the thermal energy gained from the direct radiation as illustrates in Fig.(6a). The PV cell and glass temperature have the same raising in temperature in configuration B due to the existence of glass in mid portion it will withdraw thermal energy from PV modules that connected with it from two direction, as shown in Fig. (6b). For configuration C Fig. (6c) shows that the maximum PV modules temperature was 63 °C at 1000 W/m² which is the less among the other configuration as a result of existence the PV modules at cooling side of PV/T system. Additionally, the reduction in area of PV cell is lead to reduction on electric power generated also, the effect of the position of the glass on the electric power is depicted in Fig.(7) it’s obvious that the configuration B (glass at mid portion) have the lowest generation due to high cell temperature because of receive the radiation from two direction. Furthermore, the electrical power is augmented in configuration C (glass at outlet portion) due to cooling effect and restricted the direct radiation at last end of PV/T system. To express the effect of cooling air more in partially covered PV/T with glass at outlet portion Fig(8) shows the variation of outlet air temperature with electrical power generated depending on the air flowrate variation and at constant irradiance (600 W/m²). It is clear that the optimum air flowrate was (0.013 kg/s) when the electric power and outlet air temperature were 18 W and 37°C. Thermal efficiency of PV/T system partially covered with glass at three different configurations analyzed in Fig.(5,9), one can observe from this figure that case C have the maximum thermal efficiency reached 48.7% due to the effect of direct radiation fallen in the glass and hot air flowing in the outlet portion lead to high thermal energy, in contrast to the case A which the direct radiation from the glass fallen in to inlet portion when the cooling air was supplied. In contours Figures, the three dimensional plot for absorber plate and air temperature is plotted at airflow of 0.013 kg/s for configuration C (partially covered with glass at outlet portion), Figs.(10) and (11) shows isotherm contours of absorber plate and air respectively. The results clearly show that the initial increase in the temperature of absorber plate edges is due to the connection between PV modules and absorber plate in this point and for air, Fig (11) reveals that the upper layers of air that attached to the back surface of PV modules have got higher temperature and at last portion of PV/T system this increase will be at the upper and lower layers when the glass is existence and the radiation fallen in the absorber plate directly. The PV modules, plate and air temperatures obtained by experimental work has been compared with the numerical results and plot in Fig.(12) for the case C glass at outlet portion, a strong agreement has been achieved between the experimental and numerical outcomes. Fig. (13) shows the flow field of three-dimensional PV/T system at air flowrate equal 0.013 kg/s, this figure shows the creation of boundary layers, its observed that the thickness of boundary layers is small near the wall.
5. Conclusion
On the basis of study the hybrid PV/T air collector with different configurations the following points have been concluded:

(i) PV/T system partially covered with different glass configurations concluded that placing the glass at outlet portion give the higher electrical and thermal gain reached 26W and 44°C so that its prefer over other two configurations (inlet portion and mid portion).

(ii) Optimizing the electrical power and outlet air temperature with mass flow rate, it is advised to operate the PV/T collector at moderate flow rate (0.013 kg/s) to obtain the required hot air and electrical power.

(iii) Glass cover located at mid portion (case B) will maximize PV modules temperature hence, the electrical power generated will be worst in this configuration amongst other two configurations.
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Figure 4: Experimental variation of air temperature along the distance of PV/T collector for the three cases at different values of solar radiation.
Figure 5: Experimental variation of absorber plate temperature along the distance of PV/T collector for the three cases at different values of solar radiation.
Figure 6: Experimental variation of the PV cell and glass temperature along the distance of PV/T collector for the three cases at different values of solar radiation.
Figure 7: Experimental variation of electrical power generated with respect to the irradiance variation at different glass configuration.

Figure 8: Variation of electrical power and outlet air temperature with respect to air flowrate variation for configuration C (glass at outlet portion) at constant solar radiation (600 W/m²).

Figure 9: Experimental variation of thermal efficiency with respect to the irradiance variation at different glass configurations.
Figure 10: Isotherm contours of absorber plate in PV/T system for configuration C (glass at outlet portion).

Figure 11: Isotherm contours of air in PV/T system for configuration C (glass at outlet portion)

Figure 12: Comparison between the experimental and numerical results for the configuration C (glass at outlet portion)
Figure 13: Velocity vector (u, v and w) at flowrate of 0.013 kg/s.