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Modeling, Identification and Control of Photovoltaic/Thermal Solar Panel

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Abstract: This paper considers a bond graph approach to model a solar photovoltaic-thermal panel (PV/T) system as an alternative to the method used in [3]. Based on the obtained 1d model representing the dynamics of the PV/T, we identify transfer functions connecting the air output temperature to the air input flow for different operating points. Further, a PI-type controller is proposed and proven to guarantee good performance. All the techniques presented are illustrated through simulation results.

Index Terms: Renewable energy, photovoltaic/thermal systems, modeling, identification, PI control.

NOMENCLATURE

PCM \(\text{phase change material} \)
PV/T \(\text{photovoltaic/thermal} \)
PV \(\text{photovoltaic} \)
ST \(\text{solar thermal} \)

Symbols
\(A_{\text{mod}}\) \(\text{module area of a PV panel} \)
\(A_{\text{hr}}\) \(\text{horizontal area of air duct and collector area} \)
\(C\) \(\text{heat capacity} \)
\(C_{\text{p}}\) \(\text{thermal power conversion factor for absorber plate} \)
\(d_{\text{hy}}\) \(\text{hydraulic diameter} \)
\(I_{\text{sun}}\) \(\text{solar radiation} \)
\(l\) \(\text{thickness} \)
\(M\) \(\text{mass} \)
\(T\) \(\text{temperature} \)
\(V_{w}\) \(\text{wind velocity} \)
\(x_{\text{duc}}\) \(\text{width of air duct and hybrid collector} \)
\(x_{\text{mod}}\) \(\text{width of PV module} \)

Greek letters
\(\alpha\) \(\text{absorptivity} \)
\(\beta\) \(\text{packing factor} \)
\(\beta_{\text{p}}\) \(\text{temperature coefficient} \)
\(\delta\) \(\text{solar radiation coefficient} \)
\(\varepsilon\) \(\text{emissivity} \)
\(\lambda\) \(\text{thermal conductivity} \)
\(\eta_{\text{ref}}\) \(\text{reference efficiency} \)
\(\rho\) \(\text{density} \)
\(\tau\) \(\text{transmissivity} \)
\(\mu\) \(\text{dynamic viscosity} \)
\(\zeta\) \(\text{duct area/PV module area} \)

Subscripts
\(\text{am}\) \(\text{ambient} \)
\(c\) \(\text{PV cell} \)
\(f\) \(\text{fluid} \)
\(fi\) \(\text{inlet} \)
\(g\) \(\text{glass cover} \)
\(i\) \(\text{thermal insulator} \)
\(t\) \(\text{tedlar} \)

I. INTRODUCTION

Photovoltaic/thermal (PV/T) hybrid panels are those which provide heat and low-carbon electricity simultaneously by combining PV and ST modules, more efficient than a PV or ST alone. A lot of research work was done in the past few decades but today most of the research work is related to optimization and improvement of previous models. PV/T systems are preferred because of more benefits as compared to drawbacks. The design and the material of the PV/T hybrid system play a vital role in achieving high energy conversion efficiency. The thermal part not only provides heat energy but also comes up with cooling; reduces the cell temperature which results in high efficiency for the electrical part. There are two common types of PV/T collectors; one comes up with water and the other is air-based. The air-based PV/T collector has few advantages over water-based collector; there is no freezing, no boiling and leakage will not harm the panel. The one major drawback with air-based collector is the low heat transfer due to its low heat capacity and low thermal conductivity. Apart from solar radiation and ambient temperature air-based PV/T collector highly depends on mass flow rate and wind speed. These Hybrid PV/T's have an efficiency of more than 50%; between 15 to 20% of electrical and rest is
work on hybrid PV/T system was first presented by Florschuetz [1] in the late 70’s where a principal idea was presented and indeed a lot of research is going on today. In the past few decades, various models with improved performance and efficiency have been presented with a large number of theoretical and experimental results. Based on 1-D energy balance model the performance of a PV/T hybrid system adding a PCM has been studied theoretically and experimentally by Malvi et al. [2]. Various layers of PCM were considered which are attached in series and each of the phase is determined separately. The performance evaluation of air as a working material has been done by Slimani et al. [3] using four different configurations (e.g. Simple PV, air duct PV, glazed PV with air duct and air duct above and below PV). The numerical models were built using heat balance equations and the experimental results are validated through previous studies. A comparative study has been done with and without glazing to test the performance of the PV/T systems [3]-[5]. Each arrangement is compared with the non-glazed PV/T collector, it is observed that there is an increase in the efficiency and better output results with glazed PV/T collectors. Naima et al. [6] developed a finite element method to calculate the thermal parameters of PV/T and evaluate the performance of the PV/T with air as a coolant. It is also concluded that an increase in the collector’s length increases the temperature of the component. Bhargava et al. [7] discussed some aspects of solar air heater hybrid system where an air heater module is integrated with a photovoltaic system. The fluctuation in photovoltaic efficiency due to temperature dependence is calculated using a linear relation. Hejazy [8] presented a detailed experimental investigation of four different air-based PV/T solar collector models. Under the same working conditions, performance is evaluated with 4 dissimilar cooling techniques and with the variation in parameters. It is concluded that an increase in specific mass flow rate enhances thermal efficiency and degrades electrical efficiency. The experimental performance of the PV/T air collector with a mono-crystalline PV module was analyzed with an air flow rate of 240 m³/h by Kim et al. [9] which come to an end with a thermal efficiency of 22% and 15% electrical efficiency. Theoretical and experimental studies for an unglazed PV/T collector have been carried out by Tiwari et al. [10] for the composite climate of India. The thermal model is developed using energy balance equation for each of the component. They observed that overall efficiency is increased by 18% of the PV/T air collector. Sopian et al. [11] examined the performance of single and double pass PV/T air heaters; double pass-type PV/T collector displayed better performance, can bring out more heat, better cooling effect to reduce PV cell temperature than a single pass-type PV/T air heater.

Solar cogeneration system providing heat and electricity simultaneously is studied in this work and a PV/T thermal model is presented in section 2 using a bond graph technique. In section 3, a PI controller is proposed to regulate the output-air temperature. The aim is to control the air outlet temperature to the desired value using air input flow as a control variable under different operating conditions. Section 4 provides the results and the paper ends with a conclusion.

II. MODELLING

The governing equations of the model were determined using a bond graph which is a graphical representation of a dynamic system. Few assumptions have been considered for this model such as pressure losses are over sighted and heat transfer between the ground and the panel is neglected.

![Fig. 1. Air-based PV/T thermal model](image)

The Fig. 1(a) shows that cold air is supplied from one side and heat (hot air) is collected along with the electrical gain on the other end side of the panel. The Fig. 1(b) shows the division of panel in various layers; a glass layer PV, solar cells layer, tedlar layer (thermoplastic floor), air channel and thermally insulated frame.

Bond graph modeling can include multiple domains. The bond graph model presented in Fig. 2 contains two domains; one is the thermal and the other is hydraulic (in a dotted rectangle). Effort and flow are the power variables; in thermal sub-model heat flow, solar flow, and enthalpy are flow variables whereas temperature is an effort variable. In hydraulic sub-model, mass flow is flow variable and pressure is the effort variable. Sources represent the input of the system and supply power to the system; it can be a flow source (S_f) or effort source (S_e) and have a specific flow and effort level. Fig. 2 shows various flow sources; in the thermal sub-model, it represents solar flow absorbed by different layers whereas in hydraulic it represents air mass flow entering the collector. Another source which is effort source represents the atmospheric temperature. Air enthalpy flow is represented by (S_{d}) which is a controlled flow source entering the collector. Passive elements which are single port elements (C & R) receive energy from the system. Capacitive element (C) has a static relation between effort.
and displacement whereas resistive element (R) has a static relation between effort and flow.

\[ Q = \frac{1}{R} (T_1 - T_2) A \]  \hspace{1cm} (6)

Where \( R \) field models the heat-transfer phenomena between two layers with temperatures \( T_1 \) and \( T_2 \). In (5) and (6), thermal heat flow is flow variable whereas temperature is an effort variable. Using flow equation (1), the energy balance equations for glass, cell, Tedlar, air channel and thermal insulator are written as:

\[
\begin{align*}
\dot{Q}_1 &= g_g - \dot{Q}_{r,gs} - \dot{Q}_{v,ga} - \dot{Q}_{c,gc} \\
\dot{Q}_2 &= g_c + \dot{Q}_{c,gc} - \dot{Q}_{c,ct} - E_p \\
\dot{Q}_3 &= g_t + \dot{Q}_{c,ct} - \dot{Q}_{v,tf} - \dot{Q}_{r,ti} \\
\dot{Q}_4 &= \dot{Q}_{v,tf} - \dot{Q}_{v,fi} \\
\dot{Q}_5 &= \dot{Q}_{r,ti} + \dot{Q}_{v,fi} - \dot{Q}_{v,ia}
\end{align*}
\]

At a rate of \( I \) in Watts, solar radiation energy absorbed by the glass cover \( G_g \) is \( A_{mod} g_{sun} \) and the energy \( G_c \) absorbed by the PV cell layer is \( A_{mod} g_{cell} \). \( \alpha_g \) and \( \alpha_c \) are absorptance of glass cover and PV cell whereas \( \tau_g \) is the transmittance of the glass cover. The values of the different parameters used are given in Table 1. Temperature of sky is given by Swinbank in 1963 associate’s sky temperature with local air temperature:

\[ T_s = 0.0552 T_{am}^{1.5} \]  \hspace{1cm} (7)

Convective heat coefficient due to wind by dimensional equation [13].

\[ h_{v,am} = 5.7 + 3.8 W_v \]  \hspace{1cm} (8)

Each of the layers can be interpreted in the following five equations (9-13);

\[
\begin{align*}
M_g c_g \dot{g}_g &= A_{mod} \alpha_g I_{sun} - h_{r,gs} (T_g - T_s) - \\
h_{v,am} (T_g - T_{am}) - h_{c,gc} (T_g - T_c)
\end{align*}
\]

\[ R_{r,gs} \] is the radiation heat flow between sky and glass cover. Where \( \sigma \) is Stefan Boltzmann constant and radiation heat transfer coefficient [11] between two surfaces (glass – sky) is;

\[ h_{r,gs} = \sigma \epsilon_g \left( T_g^2 - T_s^2 \right) \left( T_g^2 + T_s^2 \right) \frac{1}{(T_g - T_{am})} \]

\[ R_{c,gc} \] is the conduction heat flow and conductive heat transfer coefficient between two parallel elements (glass - cell) [3] is given as:

\[ h_{c,gc} = \left( \frac{1}{R_g} + \frac{1}{R_c} \right) \]

Junctions are multi-port elements, differ on the basis of how effort and flow are carried across. There exists two types of junctions, the 1-junction, and the 0-junction. 0 – Junction:

- Effort junction: all connected bonds constrained to have the same efforts.
- Flow equation: represents energy flow balances.

\[ \sum_{k=1}^{n} f_{ik} = \sum_{k=1}^{n} e_{ok} \]  \hspace{1cm} (1)

- Effort equation:

\[ e_{i1} = \ldots = e_{im} = e_{o1} = \ldots = e_{on} \]  \hspace{1cm} (2)

1–Junction:

- Flow junction: all connected bonds constrained to have same flows
- Flow equation:

\[ f_{i1} = \ldots = f_{im} = f_{o1} = \ldots = f_{on} \]  \hspace{1cm} (3)

- Effort equation:

\[ \sum_{k=1}^{n} e_{ik} = \sum_{k=1}^{n} e_{ok} \]  \hspace{1cm} (4)

Using relationship between generalized effort and generalized flow corresponding to compliance element, temperature can be written as:

\[ T = \frac{1}{c} \int Q \, dt \]  \hspace{1cm} (5)
TABLE 1

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_{\text{mod}}$</td>
<td>0.4096 m$^2$</td>
</tr>
<tr>
<td>$A_{\text{et}}$</td>
<td>0.384 m$^2$</td>
</tr>
<tr>
<td>$a_f$</td>
<td>0.06</td>
</tr>
<tr>
<td>$a_c$</td>
<td>0.85</td>
</tr>
<tr>
<td>$a_l$</td>
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<tr>
<td>$\beta$</td>
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<td>$b_{\text{ft}}$</td>
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<tr>
<td>$\delta$</td>
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</tr>
<tr>
<td>$\delta_f$</td>
<td>0.93</td>
</tr>
<tr>
<td>$\delta_l$</td>
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<tr>
<td>$\nu_l$</td>
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</tr>
<tr>
<td>$r_l$</td>
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</tr>
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<td>$k_l$</td>
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</tr>
<tr>
<td>$k_i$</td>
<td>0.3 mm</td>
</tr>
<tr>
<td>$h_i$</td>
<td>50 mm</td>
</tr>
<tr>
<td>$\lambda_f$</td>
<td>0.98 W/m-K</td>
</tr>
<tr>
<td>$\lambda_t$</td>
<td>0.02551 W/m-K</td>
</tr>
<tr>
<td>$\lambda_0$</td>
<td>0.035 W/m-K</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>5.67x10^{-8} W/m$^2$K$^4$</td>
</tr>
<tr>
<td>$\eta_{\text{el}}$</td>
<td>12 %</td>
</tr>
</tbody>
</table>

$M_c C_e \frac{dT_e}{dt} = A_{\text{mod}} \left[ T_0 \alpha_c I_{\text{sun}} \beta + h_{c,\text{el}}(T_0 - T_c) - h_{c,\text{ct}}(T_c - T_e) \right] - E_p$ (10)

Electric power [1] produced by panel can be written as;

$E_p = I_{\text{sun}} A_{\text{mod}} \eta_c = I_{\text{sun}} A_{\text{mod}} \eta_{\text{ref}} \left[ 1 - \beta_p(T_c - T_{c,\text{ref}}) \right]$  

$R_{c,\text{ct}}$ is the conduction heat flow and conductive heat transfer coefficient between two parallel elements (cell - tellar) [3] is given as;

$h_{c,\text{ct}} = \frac{1}{\frac{1}{k_c} + \frac{1}{\lambda_c}}$  

$M_t C_t \frac{dT_t}{dt} = A_{\text{mod}} \left[ T_0 \alpha_t I_{\text{sun}} (1 - \beta) + h_{c,\text{ct}}(T_c - T_t) + h_{v,\text{ft}} (T_f - T_t) - h_{r,\text{fi}} \Delta_l(T_t - T_i) \right]$ (11)

$R_{v,ft}$ and $R_{v,fi}$ are the convection heat flows and convective heat transfer coefficients between two elements (air - tellar and air - insulator) are;

$h_{v,ft} = h_{v,fi} = N_u \frac{\lambda_f}{d_{by}}$  

Radiative heat transfer between two parallel elements (tellar - insulator) is;

$h_{r,ft} = \frac{1}{\varepsilon_l + \frac{1}{\varepsilon_l} \Delta_l}$

$R_{ct}$ is the heat exchange between tellar, fluid and thermal insulator. The useful energy gain $Q_u$ is;

$Q_u = \Delta H_f - \Delta H_{fi} = m_f C_f(T_f - T_f)$

$m_f C_f \frac{dT_f}{dx} = x_{dwc} \left[-h_{v,ft}(T_f - T_t) - h_{v,fi}(T_f - T_i)\right]$ (12)

Solution of (9) can be written as;

$T_f(x) = \left[(T_{fi} - \omega)e^{-\psi} + \omega\right]$  

where

$\omega = \frac{\tau_e h_{v,ft} + \tau_{h,fi}}{h_{v,ft} + h_{v,fi}}$ and $\psi = x_{mod} \left(\frac{h_{v,ft} + h_{v,fi}}{m_f C_p}\right)x$

whereas Nusselt number, Reynold number and Prandtl number [13] are given as;

$N_u = 0.023 R_e^{0.8} P_r^{0.4}$  

$P_r = \frac{\mu f}{\lambda_f}$ and $P_r = \frac{\mu f}{\lambda_f}$

$M_i C_i \frac{dT_i}{dt} = A_{\text{Ref}}[h_{v,fi}(T_i - T_i) + h_{v,ft}(T_f - T_i) - h_{v,am}(T_i - T_{am})]$ (13)

The five equations (9-13) obtained using bond graph technique are compatible with the paper published by Slimani et al [3]. However, in this paper, we use bond graph technique to model the PV/T instead of the energy conservation principle used in [3]. Hopefully, both approaches are consistent.

III. IDENTIFICATION AND CONTROLLER DESIGN

The first part concerns system identification that refers to the problem of estimating an unknown system and the main purpose is to construct a model that approximates the behavior of a system using input and output data. In automated control, the dynamic system model is a mathematical description of the relationship between inputs and outputs of the system. In this section identification of the thermal part of the PV/T system, with a mass flow rate (kg/sec) as input and outlet temperature of the duct (°C) as output is taken. To this end, an input-output data is required from real measurements or from simulation. Herein, we consider the latter to generate data for two different values of mass flow rate, each with different irradiance: 250, 500, and 750 (W/m$^2$).

Using a system identification toolbox, different order of transfer functions were tested in order to find the one which fits the best. It was found that a simple order 1 gives the closest response to our system. Using six sets of input-output data corresponding to 6 operating points, we obtained six different first-order transfer functions with time constants (resp. gain) ranging [31.25, 41.67, 50, 34.48, 47.66, 76] seconds (resp. [-15, -19.17, -22.5, -14.14, -17.14, -24.61]) with an average of $T_{avg} = 46.83$ s (resp. $k_{avg} = -18.76$). Therefore, the average transfer function denoted $G_{avg}$ can be written as;

$G_{avg}(s) = \frac{T_f}{m_f} = \frac{k_{avg}}{1 + Ts_{avg}} = -\frac{0.42}{s + 0.024}$ (14)

This simple model gives a satisfactory approximation of the behavior of the non-linear system considering the numerical values of the parameters. It is good enough for the purpose of control.
In fact, for the sake of simplicity, we design a PI controller based on the transfer function (14). The advantage of using a PI controller is that the steady-state error is eliminated which is important when using an average transfer function.

\[
G_{avg}(s) = \frac{b}{s+a} \quad \text{and} \quad C(s) = K_p + \frac{K_i}{s}
\]

The transfer function of the closed loop system from reference to output is \( F(s) \).

\[
F(s) = \frac{b(K_p s + K_i)}{s^2 + (a + bK_p)s + bK_i}
\]

This transfer function \( F(s) \) denominator which is closed loop characteristic polynomial is compared with the second order general equation (15), assuming that desired characteristic polynomial is:

\[
s^2 + 2\zeta \omega_n s + \omega_n^2
\]  

(15)

The controller parameters are then given by

\[
K_p = \frac{2\zeta \omega_n - a}{b} \quad \text{and} \quad K_i = \frac{\omega_n^2}{b}
\]

The parameter \( \omega_n \) determines the response speed while \( \zeta \) relates to the damping. The controller parameters are \( K_p = -0.0126 \) and \( K_i = -0.0014 \) with \( \zeta = 0.61 \). To avoid the overshoot a small value of 0.025 is chosen for \( \omega_n \). These parameters are used for the controller part but for the plant the whole non-linear system is considered using MATLAB function in Simulink.

IV. RESULTS AND DISCUSSION

The typical distribution of solar radiation and wind speed of daytime is given in Fig. 4.

![Fig. 4. Hourly variation of solar radiation and wind speed](image)

During the summer season the intensity of the sun is high and so does values of solar irradiance. The maximum value for the solar irradiance is 870 W/m². The solar irradiance data considered is for the daytime, it is zero or with very low values during the rest of the day. The wind profile also seems normal with a maximum value of 5.5 m/sec. The temperature profiles of glass, PV cell, tedlar, working fluid (air), and insulation along with ambient temperature are given below in the Fig. 5.

![Fig. 5. Temperature evolution during a typical summer daytime](image)

The temperature for each arrangement is calculated using (9-13). The inlet air temperature is almost the same as ambient temperature. The outcomes convey that the maximum ambient temperature is around 35°C and the outlet air temperature from the duct goes to 40°C. The heat transfer is small because of low heat capacity and low thermal conductivity. The electrical efficiency of the panel highly depends on the PV cell temperature and with this cell temperature there will be good electrical efficiency. The temperature profiles are shown in Fig. 5 are with the mass flow rate of 0.005 kg/sec. The temperature of the insulator is almost the same as the ambient temperature.

Outlet air temperature from the duct highly depends on the mass flow rate. It can be seen in Fig. 6 that when the mass flow rate starts increasing, the temperature starts decreasing. The mass flow rate rise degrades the outlet temperature from the duct.

![Fig. 6. Variation of outlet air temperature with the increase in mass flow rate at T_{in} = T_{amb} = 30 (°C) and I_{in} = 500 (W/m^2)](image)

Based on the need to control the mass flow rate, the evolution of the temperature rise using PI-controller is shown in Fig. 8. To make the solution simple a linear controller approach is used instead of a non-linear approach. PI controller is widely used in industries and it works. For PI-controller the parameters \( K_p \) and \( K_i \) are obtained using the average transfer function (14). For the plant, the whole non-linear model is taken using MATLAB function in Simulink and different reference points are considered.
We have a complex model, and it wasn’t obvious to control it but in our approach, we obtain good results. The controller tracks the reference point with a high settling time and a closed-loop time constant of 68 sec. The desired temperature is in light blue color whereas the measured temperature is in the dark blue color. Initially, constant solar radiation is applied to see the response of the controller and it can be seen in Fig. 7 that controller gets the temperature to the desired setting with the decrease and increase in reference point.

The effect of solar radiation with a step as shown in Fig. 8 can be seen in Fig. 9, even with the change in solar radiation the controller rejects the change and tracks the desired temperature.

V. CONCLUSION

This paper presents the modeling, identification, and control of a thermal model of a PV/T. The study of thermal PV/T is presented under a sample of southern France climatic conditions. The obtained results confirm that the heat transfer using air as a working material is small because of its low heat capacity and low thermal conductivity. It confirms that the increase in the mass flow rate reduces the outlet temperature of the duct. It is also found that the system requires a long time for the outlet air temperature to reach a reference value but settles out to the exact target with PI control even with the change in solar irradiance. The proposed technique in this work will be implemented in real time on a laboratory platform to check the concrete applicability of the proposed solution.

REFERENCES