Simulation of a Hybrid Photovoltaic-Thermal (PV-Th) Air Heating System for Regenerating Desiccant Gel in an Air Conditioning Room

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Abstract: A simulation model is developed for predicting the performance of a hybrid PV-Th air heating system incorporating with the wasted heat recovered from the condenser for regenerating desiccant in an air conditioned room under prevailing meteorological conditions at the site location. The model consists of submodels for five main system components, namely, PV-Th collector, desiccant dehumidification and regeneration unit, air conditioning system, air
mixing unit and air-conditioning room. Mathematical equations for modeling the performance of these individual components are developed based on the balances of mass and energy. The investigation of the developed simulation model is carried out on a typical living room equipped with a 1.5-ton air conditioner and a three-42-W hybrid PV-Th air heating collector. Under varied meteorological data inputs, the model can illustrate dynamic performances of various components of the system. The simulation result shows that the heat gains from PV-Th collector and condenser can produce dry air as high as 48°C and 21% relative humidity, and the desiccant dehumidification unit can save the electrical energy consumption of the air-conditioner up to 12%.

**Keywords:** Hybrid Photovoltaic-Thermal System; Dehumidification; Desiccant Cooling.

### 1. INTRODUCTION

A hybrid photovoltaic-thermal collector and an air heating system can be combined into a photovoltaic-thermal (PV-Th) system which can produce low temperature heat and electricity simultaneously. The solar energy from the sun is partly converted to electricity by photovoltaic cells in thermal contact with a solar heat absorber so that the excess heat generated in the photovoltaic cells can be served as input for the thermal system. During operation, a heat carrier fluid removes heat from the absorber and solar cells. Those cooled cells increase the solar cell electrical power output. In tropical countries
like Thailand, an air conditioner consumes over 70% of the total 
electrical energy used in a small house [1]. It is generally used to 
remove the sensible and latent heat in order to maintain the comfort 
zone in the living space. To reduce the latent heat, the room moist 
air is dehumidified by the dry desiccant so that the room air becomes 
dry and warm. The warm, dry air then passes through the cooling 
coils at the evaporator of the A/C system to decrease its temperature. 
The cooling of air results in the condensation of part of the moisture 
in the air and the cool-dry air is routed directly to the room. This 
would result in decreasing the total energy consumption of the air 
conditioner. When the desiccant is saturated, it needs a dry-up 
process to remove the absorbed moisture so that it can be reused. 
The process is so called “regeneration”. In the A/C system, when the 
room air is cooled down, the refrigerant in the evaporator absorbs the 
heat from the air and rejects it at the condenser. The outdoor ambient 
air passing though the condenser coil carries the rejected heat from 
the condenser and becomes the hot air. Meanwhile, the hot air from 
the PV-Th collector flows to the mixing unit and mixes with the hot 
air from the condenser of the A/C system. The mixed hot air then 
flows to the outer portion of the desiccant unit to regenerate the 
moist desiccant. After the moist desiccant is dried, it is reused in the 
dehumidifying process in the room.

A simple diagram of a system employing the heat from a 
hybrid PV-Th air heating collector incorporating with the recovered 
heat from the condenser of an A/C system for regenerating desiccant 
to be used in an air-conditioned room can be illustrated in figure 1.
There are five main components, those are: (1) PV-Th air heating collector, (2) desiccant dehumidification and regeneration unit, (3) air-conditioning system, (4) air mixing unit, and (5) living space (room).

**Figure 1.** Schematic diagram of a hybrid photovoltaic-thermal (PV-Th) air heating system for regenerating desiccant for an air-conditioned room.

The objective of the study is to develop a simulation model which can predict the thermal performance of the above mentioned system when operating under various climatic conditions. Key performance indicators, those are: the PV-Th collector conversion
efficiencies, the amount of adsorbed moisture, the saving amount of energy consumption of the A/C system with desiccant dehumidification unit, and the proportions of heat from the PV-Th system and the condenser in the desiccant regeneration process, can be analyzed from the developed model.

2. THE SIMULATION MODEL

The simulation model consists of mathematical equations of the five main components of the system shown in figure 1. It can simulate the transient performance of each system component at a time interval of five minutes. The specifications, dimensions and keyed performance parameters of those five components can be specified as desired. The required meteorological inputs include the total solar irradiance, the temperature and relative humidity ratio of the ambient air and the wind speed. The simulated outputs contain the electrical power generated by the PV cells, the temperatures and moisture contents of processed air at the inlet and outlet of each component. These output data can be analyzed and used to estimate the keyed performance indicators of each individual components and the system as a whole.

2.1 PV-Th Air Heating Collector Model

Figure 2 presents the physical configuration of the double-pass PV-Th air heating collector used in this study. It is a double-pass collector consisting of five significant components, those are: a glass
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cover (g), flow channel 1 (f1), a PV-Th absorber plate (p), flow channel 2 (f2), and a back metal plate (b). The ambient air enters the PV-Th collector at flow channel 1, which is formed between the glass cover and the front side of the PV-Th absorber plate. Once reaching the end of the channel 1, it is forced to flow down to pass through flow channel 2, which is formed between the rear side of the PV-Th absorber plate and the back metal plate, before leaving the collector. This results in the heat removal from the PV-Th absorber and the back plate, and the higher air temperature at the collector outlet [2,3]. It is assumed that the heat distribution in each component is uniform, therefore the heat capacitance of the component is lumped and its temperature represents as the effective temperature. The air mass flow rate is also assumed to be uniform throughout the PV-Th collector.

The energy balances for various components of the PV-Th collector can be expressed by the following equations [4,5]:

For Glass Cover (g),
\[ \dot{Q}_{\text{store, } g} = \dot{Q}_{\text{in, } g} - \dot{Q}_{\text{rad, } g \rightarrow a} - \dot{Q}_{\text{conv, } g \rightarrow a} - \dot{Q}_{\text{conv, } g \rightarrow f1} + \dot{Q}_{\text{rad, } p \rightarrow g} \]

or
\[ m_g C_g (dT_g / dt) = \alpha_g G A_g - A_g F_{gsk} \sigma (\varepsilon_g T_g^4 - \varepsilon_{sky} T_{sky}^4) - A_g h_{cga} (T_g - T_a) - A_g h_{cfl} (T_g - T_{f1}) + A_g h_{rpg} (T_p - T_g) \]  

(1)

For Air Flowing in Channel 1 (f1),
\[ \dot{Q}_{\text{store, } f1} = \dot{Q}_{\text{conv, } g \rightarrow f1} + \dot{Q}_{\text{conv, } p \rightarrow f1} - \dot{Q}_{u1} \]

or
\[ m_a C_a (dT_{f1} / dt) = A_g h_{cfl} (T_g - T_{f1}) + A_g h_{cpf} (T_p - T_{f1}) - m_{a,PV-Th} C_a (T_{f1out} - T_{f1in}) \]  

(2)

For PV-Th Absorber (p),

\[
\hat{Q}_{\text{store},p} = \hat{Q}_{\text{in},p} - \hat{Q}_{\text{rad},p \rightarrow g} - \hat{Q}_{\text{conv},p \rightarrow f_1} - \hat{Q}_{\text{conv},p \rightarrow f_2} - \hat{Q}_{\text{rad},p \rightarrow b} - P_{\text{elec}}
\]

or

\[
(m_p C_p) \frac{dT_p}{dt} = \alpha_s \tau_g G A_p - A_p h_{rg}(T_p - T_g) - A_p h_{cpg}(T_p - T_{fl}) - A_p h_{cpg2}(T_p - T_{f2}) - A_p h_{rp}(T_p - T_b) - \eta_s \tau_g G A_p
\]

(3)

For Air Flowing in Channel 2 (f2),

\[
\hat{Q}_{\text{store},f_2} = \hat{Q}_{\text{conv},p \rightarrow f_2} + \hat{Q}_{\text{conv},b \rightarrow f_2} - \hat{Q}_{\text{u_2}}
\]

or

\[
m_a C_a \frac{dT_{f2}}{dt} = A_p h_{cpg2}(T_p - T_{f2}) - A_b h_{cb}(T_b - T_{f2}) - m_{a, PV-Th} C_a (T_{f2out} - T_{f2in})
\]

(4)

For Back Metal Plate (b),

\[
\hat{Q}_{\text{store},b} = \hat{Q}_{\text{rad},p \rightarrow b} - \hat{Q}_{\text{conv},b \rightarrow f_2} - \hat{Q}_{\text{cond},b \rightarrow a}
\]

or

\[
m_b C_b \frac{dT_b}{dt} = A_p h_{rp}(T_p - T_b) - A_b h_{cb}(T_b - T_{f2}) - A_b U_b(T_b - T_a)
\]

(5)

Figure 2. Geometric configuration of PV-Th collector: (1) assembly of the main components, and (2) cross sectional view with temperatures and heat transfer coefficients of various components in the PV-Th collector.
The sky temperature is approximately assumed to be close to the ambient temperature as [6]:

$$T_{sky} \approx 0.914T_a.$$  \hspace{1cm} (6)

As the back metal plate and collector casing are thin, the overall heat loss coefficient from back plate to ambient air $U_{ba}$ can be approximately determined from:

$$U_{ba} = \left[1/(L_{ins}/k_{ins})\right].$$ \hspace{1cm} (7)

The convective heat transfer coefficient from the glass cover to the ambient air, $h_{cga}$, for panel tilted at an angle $\beta$ below 25° and for low moderate wind speed, $V_{wind}$, is expressed as [6]:

$$h_{cga} = 1.2475 \left[(T_g - T_a) \cos \beta \right]^{0.33} + 2.685V_{wind}.$$ \hspace{1cm} (8)

The forced convective heat transfer coefficient, $h_c$, of air flowing between any parallel flat plates with spacing, $D$, can be found by using the dimensionless quantities, which are Nusselt number ($N_u$), and Reynolds number ($R_e$), as followed [6]:

$$R_e = (\rho V D)/\mu, \hspace{0.5cm} N_u = 0.0158 R_e^{0.8}, \hspace{0.5cm} \text{and} \hspace{0.5cm} h_c = (N_u k)/D.$$ \hspace{1cm} (9)

The radiative heat transfer coefficient, $h_r$, between any two surfaces 1 and 2 can be determined by

$$h_r = 4\sigma T^3/(1/\varepsilon_1 + 1/\varepsilon_2 - 1).$$ \hspace{1cm} (10)

The output from the PV-Th model is the outlet air temperature, $T_{f2out}$. The amount of water vapor in the atmospheric
outlet air at any temperature, $T$, can be specified by the absolute humidity or humidity ratio, $\omega$. It is a function of relative humidity, $\phi$, and saturated air pressure, $P_{sat}$; i.e. expressed as [7]:

$$\omega = 0.622\phi P_{sat} / (P - \phi P_{sat})$$  \hspace{1cm} (11)

where $P_{sat} = 0.6125 + 0.0436T + 1.484 \times 10^{-3}T^2 + 2.531 \times 10^{-5}T^3$  \hspace{1cm} (12)

In addition, the enthalpy of air, $h$, at any temperature and absolute humidity can be determined by

$$h = 1.01T + \omega(1.84T + 2,502)$$  \hspace{1cm} (13)

The average daily PV-Th collector efficiency can be written as the summation of thermal efficiency, $\eta_{th}$ and solar cell efficiency, $\eta_S$ or determined by the integration of thermal and electrical energy obtained over the daytime period in comparison with total solar energy over the same period, which can be expressed as [8]:

$$\eta_{PV-Th} = \eta_{th} + \eta_S = \left(\int Q_{th} dt + \int P_{elec} dt\right) / \left(A_g \int G dt\right)$$  \hspace{1cm} (14)

2.2. Desiccant Dehumidification and Regeneration Model

As shown in figure 1, the desiccant in this study undergoes two processes which are the dehumidification and regeneration. The mathematical models for these two processes are aimed at the predictions of the air conditions (i.e. temperature and humidity) at the outlet of each process, which can be further used to determine the amount of water adsorbed or desorbed by the desiccant.
2.2.1 Dehumidification Process

The room air is circulated through the dehumidification unit where its moisture is adsorbed by the desiccant. Considering the energy balance of desiccant, the rate of change of energy in desiccant is transferred into the heat lost by convection to processed air and the sensible heat of adsorption which is expressed as [9]:

\[ m_D (C_D + m_{WD} C_W) \frac{dT_D}{dt} = h_{c,D \rightarrow as} A_D (T_D - \bar{T}_r) + R_d h_s \]  

(15)

For the processed air temperature, the energy gain in the air stream is from the adsorbent and the raise of the energy of adsorbed water. It can be written as [9]:

\[ m_{a,r1} \left[ C_a + (C_W + dh_v/dT_{r1}) \omega_{r1} \right] (T_{r2} - T_{r1}) = h_{c,D \rightarrow as} A_D (T_D - \bar{T}_r) + R_d C_w (T_D - \bar{T}_r) \]  

(16)

The adsorption rate, \( R_d \), of water vapor between desiccant and air during the dehumidification processor can be further applied to determine the moisture content in the desiccant. It can be expressed as

\[ R_d = \left( h_{m,D \rightarrow as} A_D / R_w \right) \left( P_{w,as} / T_{r1} - P_{w,D} / T_D \right) \]  

(17)

Knowing \( R_d \) and \( \omega_{r1} \), which is calculated from \( \phi_{r1} \), the absolute humidity at the outlet can be calculated by

\[ \omega_{r2} = \omega_{r1} + R_d / m_{a,r1} \]  

(18)

After \( T_{r2} \) and \( \omega_{r2} \) are determined from eqs. (16) and (18), the relative humidity at the outlet \( \phi_{r2} \) can be calculated by eq. (11). These parameters are the input of the air conditioning system model.
The mass transfer coefficient of the water vapor between desiccant and air, $h_{m,D→a,s}$, when the air stream passing through the dehumidification unit can be calculated by the dimensionless Sherwood number, $S_h$, Reynolds Number, $R_e$, and Schmidt Number, $S_c$, as [10]:

$$S_h = h_{m,D→a,s} \frac{D_h}{D_{W→a,s}} = 0.023 R_e^{4/5} S_c^{1/3}, \quad R_e = 2 \dot{m}_{a,r1}/\left[\mu(X+Y)\right],$$

and $S_c = \nu/D_{W→a,s}$ \hspace{1cm} (19)

The convective heat transfer coefficient, $h_{c,D→a,s}$, when the air stream passing through the baffles, can be determined by

$$N_u = h_{c,D→a,s} \frac{D_h}{k_a} = 0.023 R_e^{4/5} P_r^{1/3} \hspace{1cm} (20)$$

### 2.2.2 Regeneration Process

The main driving equations used for desiccant regeneration process are similar to those of dehumidification except the conditions of air at the inlet and outlet of the unit. As the regenerated hot air stream is flown from the air mixing unit to the desiccant regeneration unit, the inlet parameters of the regeneration process becomes $\dot{m}_{a,mix}, T_{a,mix}, \phi_{a,mix}$ and $\omega_{a,mix}$ and are used in all driving equations in place of those of dehumidification process, $\dot{m}_{a,r1}, T_{r1}, \phi_{r1}$ and $\omega_{r1}$, respectively (see figure 1). Similarly, the outlet parameters $\dot{m}_{a,r2}, T_{r2}, \phi_{r2}$ and $\omega_{r2}$ for dehumidification are replaced by $\dot{m}_{a,mix}, T_{a, out}, \phi_{a, out}$ and $\omega_{a, out}$, respectively, for regeneration. It is noted that $\dot{m}_{a,r2}$ is equal to $\dot{m}_{a,r1}$. 
2.3 Air Conditioning System Model

The air conditioning process can be formulated as a steady-flow process and analyzed by applying the principles of conservation of mass and energy. The mass flow rate at the outlet of cooling coil is assumed to be equal to that at the inlet. The concerned governing equations at the evaporator of the A/C system, which is placed inside the room, are as follows [11,12]:

**Dry air mass:**
\[
\dot{m}_{a,r2} = \dot{m}_{a,r3} \quad (21)
\]

**Water mass:**
\[
\dot{m}_{a,r2} \omega_{a,r2} = \dot{m}_{a,r3} \omega_{a,r3} + \dot{m}_W \quad (22)
\]

**Energy:**
\[
\dot{Q}_{evap} = \dot{m}_{a,r3} h_{r3} + \dot{m}_W h_W - \dot{m}_{a,r2} h_{r2} \quad (23)
\]

For the sake of simplicity in developing the A/C system model, it is assumed that the friction losses in the piping and expansion valve, and the heat loss in the refrigeration cycle are small and negligible. In such a case, the rate of rejected heat at the condensing unit can be expressed as:

\[
\dot{Q}_{con} = \dot{Q}_{evap} + \dot{W}_{com} \quad (24)
\]

where \( \dot{W}_{com} \) is equal to the compressor efficiency, \( \eta_{com} \), multiplied by the electrical power required by compressor, \( \dot{P}_{com} \),

\[
\left( \dot{W}_{com} = \eta_{com} \dot{P}_{com} \right).
\]

The thermal performance of the A/C system or the coefficient of performance (COP), is determined by
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\[ COP = \frac{\dot{Q}_{\text{evap}}}{\dot{W}_{\text{com}}} \]  \hspace{1cm} (25)

It is noted that generally the \( COP \) of A/C system varies with the operating temperatures of the evaporator and condenser. However, as the operating temperatures of these two components in this study do not vary greatly, the \( COP \) is assumed to be constant.

2.4 Air Mixing Chamber Model

At the condenser of the A/C system, the heat removed from the room is transferred to the outdoor ambient air which is forced to flow through the condenser by a fan. The resulting warm air then flows through a well-insulated duct routing to a mixing chamber where it is mixed with the warm air flowing from the PV-Th collector. The mass and energy balance equations for the mixing process can be written as a function of absolute humidity, mass flow rate and enthalpy of the two air streams as follows [7]:

Mass of dry air: \[ \dot{m}_{a,\text{PV-Th}} + \dot{m}_{a,\text{con}} = \dot{m}_{a,\text{mix}} \]  \hspace{1cm} (26)

Mass of water vapor: \[ \omega_{a,\text{PV-Th}} \dot{m}_{a,\text{PV-Th}} + \omega_{a,\text{con}} \dot{m}_{a,\text{con}} = \omega_{a,\text{mix}} \dot{m}_{a,\text{mix}} \]  \hspace{1cm} (27)

Energy: \[ \dot{m}_{a,\text{PV-Th}} h_{a,\text{PV-Th}} + \dot{m}_{a,\text{con}} h_{a,\text{con}} = \dot{m}_{a,\text{mix}} h_{a,\text{mix}} \]  \hspace{1cm} (28)

Eliminating \( \dot{m}_{a,\text{mix}} \) from the relations above, thus the \( h_{a,\text{mix}} \) and \( \omega_{a,\text{mix}} \) can be found by
Knowing \( h_{a,\text{mix}} \) and \( \omega_{a,\text{mix}} \), the air temperature and the air relative humidity at the outlet of the mixing unit can be calculated by eqs. (13) and (11), respectively.

### 2.5. Living Space Model

The comfortable condition of a living space can be maintained by removing the heat load in the space to outside by the use of an air conditioner which is known as the cooling load. The cooling load consists of sensible heat and latent heat which are the heat gained by the increases in temperature and moisture, respectively. It can be separated into infiltration load and space cooling load which includes heat gain across the building envelope and heat gain from internal sources such as occupants and electrical appliances. Thus, the total cooling load, \( Q_{\text{cooling}} \), of a living space can be expressed as

\[
Q_{\text{cooling}} = Q_{\text{wall}} + Q_{\text{window}} + Q_{\text{roof}} + Q_{\text{internal}} + Q_{\text{infil,si}} + Q_{\text{infil,la}} \tag{30}
\]

A method involving the use of sol-air temperatures, \( T_{so} \), is used to determine the cooling load for a room. The instantaneous flow of heat through wall, glazed windows, and roof into a room is given by [13]

\[
\begin{align*}
\dot{Q}_{\text{wall}} &= A_{\text{wall}} U_{\text{wall}} (T_{so} - T_{r1}) , \\
\dot{Q}_{\text{window}} &= A_{\text{win}} U_{g} (T_{so} - T_{r1}) + SC \cdot G , \\
\dot{Q}_{\text{roof}} &= A_{\text{roof}} U_{\text{roof}} (T_{so} - T_{r1}) \tag{31}
\end{align*}
\]
The sol-air temperature can be expressed as

\[ T_{so} = T_a + \left( \alpha_{wall} G / h_{so} \right) \]  

(32)

The heat gained by natural infiltration is calculated by

\[ \dot{Q}_{infil,si} = 0.33 n V_{room} (T_a - T_{r1}) \]

and

\[ \dot{Q}_{infil,la} = 0.8 n V_{room} (\omega_a - \omega_{r1}) \]  

(33)

According to the diagram of figure 3, the governing equation for determining the room air conditions, i.e., \( T_{r1} \) and \( \phi_{r1} \), can be expressed in terms of energy balance as:

\[ m_{a,r1} (h_{r3} - h_{r1}) = \dot{Q}_{cooling} \]  

(34)

Figure 3. Direction of air circulation in the room.

3. CONFIGURATION OF SIMULATION SYSTEM

A simulation has been carried out for a hybrid PV-Th solar air heating system consisting of a 2-m\(^2\) single-glass collector with double-pass air flow channels and three 42-W\(_p\) amorphous-silicon (a-Si) thin-film solar cell modules pasting over an aluminum absorber
plate. The living space is 3.5-m wide, 4.8-m long and 2.5-m high and there is a 0.7-m² window on the east and west walls. According to ASHRAE standard [14], for two occupants, an infiltration rate at 60% of one air change per hour is used. Two occupants, four 36W fluorescent lamps, and a 50W television are simulated as its internal cooling load. Five kg silica gel desiccant and 1.5-ton air conditioner are assumed to maintain the living space within comfort zone, which is 25°C and 50% relative humidity. The system is assumed to be operated under the typical climate conditions of Bangkok, which is located at the latitude 14°N. Hourly total solar irradiance for Bangkok simulating from the Exell’s solar radiation model [15] are used in the simulation while the daily ambient temperature variation is assumed to be sinusoidal and a constant wind speed of 1 m/s is used for the sake of simplicity. The simulation is carried out for two consecutive days during winter.

The constant parameters used in the simulation model are shown in Table 1. Some parameters are obtained from the manufacturer’s specifications while others are obtained from the standard property of each material.
### Table 1. Constant parameters used in the simulation model.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Dimension:</strong></td>
<td>(W x L x Thickness)</td>
<td></td>
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<tr>
<td>Glass Cover</td>
<td>0.95 x 2.35 x 0.004</td>
<td>m x m x m</td>
</tr>
<tr>
<td>PV-Th Absorber Plate</td>
<td>0.95 x 2.15 x 0.003</td>
<td>m x m x m</td>
</tr>
<tr>
<td>Back Metal Plate</td>
<td>0.95 x 2.35 x 0.001</td>
<td>m x m x m</td>
</tr>
<tr>
<td>Insulation</td>
<td>0.95 x 2.35 x 0.04</td>
<td>m x m x m</td>
</tr>
<tr>
<td>Room</td>
<td>2.40 x 3.60 x 3.60</td>
<td>m x m x m</td>
</tr>
<tr>
<td><strong>Density:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( \rho_g )</td>
<td>2,515</td>
<td>kg/ m³</td>
</tr>
<tr>
<td>( \rho_f )</td>
<td>1.184</td>
<td>kg/ m³</td>
</tr>
<tr>
<td>( \rho_b )</td>
<td>7,870</td>
<td>kg/ m³</td>
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<tr>
<td><strong>Mass:</strong></td>
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<td></td>
</tr>
<tr>
<td>( m_g )</td>
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</tr>
<tr>
<td>( m_a )</td>
<td>0.261</td>
<td>kg</td>
</tr>
<tr>
<td>( m_b )</td>
<td>24.382</td>
<td>kg</td>
</tr>
<tr>
<td>( m_D )</td>
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<td>kg</td>
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<td><strong>Specific heat:</strong></td>
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<tr>
<td>( C_g )</td>
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<td>( C_a )</td>
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<tr>
<td>( C_b )</td>
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<td>( C_D )</td>
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<td>( C_W )</td>
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<td><strong>Emissivity:</strong></td>
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<td>( \varepsilon_g )</td>
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<td>( \varepsilon_p )</td>
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<td>( \varepsilon_{sky} )</td>
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<td>( \varepsilon_g )</td>
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<td><strong>Overall heat transfer coefficient</strong></td>
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<tr>
<td>( U_{ba} )</td>
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<td>W/m²·K</td>
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<td>( U_{wall} )</td>
<td>4.0</td>
<td>W/m²·K</td>
</tr>
<tr>
<td>( U_g )</td>
<td>5.9</td>
<td>W/m²·K</td>
</tr>
<tr>
<td>( U_{roof} )</td>
<td>3.6</td>
<td>W/m²·K</td>
</tr>
<tr>
<td><strong>Viscosity, ( \mu_a )</strong></td>
<td>1.562x10⁻⁵</td>
<td>m²/s</td>
</tr>
</tbody>
</table>
Table 1. Constant parameters used in the simulation model (continued).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate of the air:</td>
<td>0.01</td>
<td>kg/s</td>
</tr>
<tr>
<td>$\dot{m}_{a, PV-Th}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\dot{m}_{a, con}$</td>
<td>0.25</td>
<td>kg/s</td>
</tr>
<tr>
<td>$\dot{m}_{a, r1}$</td>
<td>0.10</td>
<td>kg/s</td>
</tr>
<tr>
<td>Efficiency:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_0$</td>
<td>9</td>
<td>%</td>
</tr>
<tr>
<td>$\eta_{con}$</td>
<td>95</td>
<td>%</td>
</tr>
<tr>
<td>Absorptivity:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\alpha_{wall}$</td>
<td>0.300</td>
<td></td>
</tr>
<tr>
<td>$\alpha_S$</td>
<td>0.040</td>
<td></td>
</tr>
<tr>
<td>$\alpha_S[16]$</td>
<td>0.816</td>
<td></td>
</tr>
<tr>
<td>Shading coefficient $SC$</td>
<td>0.5</td>
<td></td>
</tr>
<tr>
<td>Gas Constant $R_W$</td>
<td>461.52</td>
<td>J/kg·K</td>
</tr>
<tr>
<td>Area:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$A_{wall}$</td>
<td>2.4x3.6</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$A_{window}$</td>
<td>0.6x1.2</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$A_{roof}$</td>
<td>3.6x3.6</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$A_D$</td>
<td>0.005</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Transmissivity, $\tau_g$</td>
<td>0.88</td>
<td></td>
</tr>
<tr>
<td>PV-Th collector tiled angle, $\beta$</td>
<td>15°</td>
<td></td>
</tr>
<tr>
<td>Coefficient of Performance $COP$</td>
<td>2.14</td>
<td></td>
</tr>
<tr>
<td>Effective thermal capacity of PV-Th absorber plate($m_pC_p$) [16]</td>
<td>9,950</td>
<td>J/K</td>
</tr>
</tbody>
</table>

4. RESULTS AND DISCUSSION

The prevailing variations of the solar irradiance, wind speed, ambient temperature and ambient relative humidity during those two days are shown in figure 4. The simulated irradiance from the Exell’s
The simulated PV generated electrical power shown in figure 5 varies in the same pattern as the solar irradiance input. The highest generated power of 118 W is predicted at noon when the solar irradiance is highest. During the night, there is no solar energy input to the collector, therefore the generated electrical power is zero. The simulated power consumption of air conditioning system is also shown in the
At the beginning, as the initial room temperature is set to be equal to the outdoor ambient temperature, continuous power is required by the A/C system to cool down the room air to be in the setting range of 24-26°C. After that, the A/C system operates at a stabilized state in which the compressor is automatically switched on and off to maintain the room temperature within the setting range. The operation is represented by the rectangular-wave curve shown in the figure. The power taken by the compressor is about 1.8 kW. When the compressor is off, small power is still required by the A/C system for its accessories, such as blower of evaporator and control circuit, in the order of 20 W. The frequency of on-off operation during the day is found to be higher than that during the night due to the solar heat gain and higher daytime ambient temperature.

**Figure 5.** Simulated results of PV generated power and A/C system consumed power.
Figure 6 presents the variations of temperature and relative humidity (RH) of the outdoor ambient air passing through the fin of the condenser unit of the A/C system and of the air at the outlet of PV-Th collector. The temperature $T_{f2out}$ and $\phi_{f2out}$ represents the temperature and relative humidity of air flowing at the end of Flow Channel 2 of PV-Th collector, respectively. They are initially set to be equal to the ambient condition. During the day, the PV-Th collector is heated up by solar irradiance. The solar energy is partly converted to electricity and transferred to the absorber plate as thermal energy. This useful energy then heats up the air which flows in channel 1 (between the glass cover and PV-Th absorber plate) and channel 2 (between the PV-Th plate and the back plate). When the air is heated up, the relative humidity is decreased. The simulated results show that the highest temperature and the lowest RH of the hot air flowing out from PV-Th collector are about 49°C and 17%, respectively. During the night, as no heat can be obtained from the PV-Th collector, the air at the collector outlet has the same conditions as those of the outdoor air.

For the condensing unit, the outdoor ambient air flows through the condenser in order to carry away the rejected heat from the living space. Both temperature ($T_{a,con}$) and relative humidity ($\phi_{a,con}$) appear to have the similar up-and-down fluctuation patterns which are actually resulted from the on/off operating duty cycle of the compressor of the A/C system as shown in figure 6. When the compressor is on, the temperature of air passing through the condenser increases while its relative humidity decreases. When the compressor
is off, the temperature and relative humidity of the air at the condenser outlet remains the same as those of the ambient air before the condenser (indicated in the figure by $T_a$ and $\phi_a$, respectively). The outlet air temperatures at the condenser during the compressor running period range from $38^\circ$C during the night to $48^\circ$C during the day, while their corresponding relative humidities are about 35% during the night and 19% during the day. This hot outlet air from the condenser is mixed with the hot air from the PV-Th collector at the air mixing unit before delivering to the desiccant regeneration unit.

Figure 6. Simulated results of temperature and relative humidity of outlet air from PV-Th collector and condensing unit.

The predicted results of the temperature and relative humidity of the mixed air from the air mixing unit are presented in figure 7. Both simulated curves appear in the rectangular-wave patterns.
similar to those for the on-off operation of the compressor of the A/C system during the stabilizing period. Moreover, the temperature and relative humidity of outdoor ambient air also produce effects to those of the mixed air that the curves swing up and down during day and night like a sine-wave pattern similar to those of the outdoor ambient variation. The results show that the predicted daytime temperature and relative humidity of hot mixed air stream during the compressor running period reach the best air condition of 48°C and 21%RH. However, during the night, as there is no hot air from the PV-Th collector, the mixed hot air has the same conditions as those of the outdoor air passing through the condenser. This mixed air stream is fed into the desiccant regeneration unit.

Figure 7. Simulated results of temperature and relative humidity of the mixed air stream.
Figure 8 shows the moisture content in the desiccant during the sorption and desorption processes. In the desiccant sorption process, starting with dry desiccant, the moisture in the room is adsorbed at a high adsorption rate until the adsorbent is saturated at the moisture content of 0.3 kg/kg. Once the desiccant in the dehumidification unit is saturated, it will be replaced with the dry one and the saturated pack is then taken to the regeneration unit. The simulated period for each sorption process during the day is about 6 hours while that during the night is 8 hours. For regeneration process, the saturated desiccant is heated by the hot air from the mixing unit, thus the moisture is evaporated. The length of regeneration depends on the temperature and relative humidity of hot mixed air stream from the air mixing unit. In the morning, the first saturated pack takes about 5 hours to regenerate while the second pack takes only about 3 hours because in the afternoon the air from the mixing unit has higher temperature and lower relative humidity than that in the morning. It should be note that there is more heat gain to the room due to higher ambient temperature resulting in the more heat rejected at the condenser. Moreover, the higher outdoor ambient air temperature in the afternoon would result in the higher temperature of the outlet air from the PV-Th collector. Hence, there is more energy contained in the mixed air to regenerate the desiccant consequently leading to the shorter time for regeneration. However, during the night, there is no heat collected by the PV-Th collector and the OFF period of compressor is longer than the ON period, hence the average air temperature from the air mixing unit is lower. This results in the longer regeneration cycle, which is about 12 hours.
Figure 8. Simulated results of the desiccant moisture content.

Table 2 displays various values of energy obtained from the simulation model during those two consecutive days. The PV-Th collector model predicts the total PV generated electrical energy of about 6.52 MJ from the 98.15 MJ total solar energy falls on the PV-Th collector. Therefore, the average daily PV-Th electrical efficiency is 6.64%. As the blower used to circulate the process air in the PV-Th collector requires 150 Wh or 0.5 MJ per day, the total energy gain for PV-Th is about 5.52 MJ. The total 2-day thermal energy generated by the PV-Th collector is found to be 10 times larger than the generated electrical energy. On the average, the PV-Th thermal efficiency is approximately 59% and the total PV-Th conversion efficiency is therefore about 66%. This indicates the significance of the recovery of thermal energy from the PV panel.
### Table 2. Various Values of Energy and Efficiency Predicted by the PV-Th Model.

<table>
<thead>
<tr>
<th>Day</th>
<th>Time</th>
<th>Solar Irradiance on the PV/T collector</th>
<th>PV Generated Electrical Energy</th>
<th>PV/T Electrical Efficiency</th>
<th>PV/T Generated Thermal Energy</th>
<th>PV/T Thermal Efficiency</th>
<th>Total PV/T Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>MJ/m²</td>
<td>MJ</td>
<td>Q elec (MJ)</td>
<td>P elec (kWh)</td>
<td>η elec (%)</td>
<td>Q th (MJ)</td>
</tr>
<tr>
<td>1</td>
<td>6:00-18:00</td>
<td>21.81</td>
<td>49.07</td>
<td>3.26</td>
<td>0.91</td>
<td>6.64</td>
<td>28.93</td>
</tr>
<tr>
<td></td>
<td>18:00-6:00</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>6:00-18:00</td>
<td>21.81</td>
<td>49.07</td>
<td>3.26</td>
<td>0.91</td>
<td>6.64</td>
<td>28.93</td>
</tr>
<tr>
<td></td>
<td>18:00-6:00</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TOTAL</td>
<td></td>
<td>43.62</td>
<td>98.15</td>
<td>6.52</td>
<td>1.81</td>
<td>6.64</td>
<td>57.87</td>
</tr>
</tbody>
</table>

### Table 3. Various Values of Energy Predicted by the A/C System Model and Desiccant Process Model.

<table>
<thead>
<tr>
<th>Day</th>
<th>Time</th>
<th>A/C Energy Consumption (Without Desiccant Dehumidification Unit)</th>
<th>Wasted Energy from Condenser</th>
<th>Ratio Q/A</th>
<th>Adsorped Water by Desiccant in Dehumidification Unit</th>
<th>Approximated A/C Energy Consumption (Without Desiccant Dehumidification Unit)</th>
<th>Energy Saving</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>W com (MJ)</td>
<td>P com (kWh)</td>
<td>Q com (MJ)</td>
<td>%</td>
<td>kg</td>
<td>MJ</td>
</tr>
<tr>
<td>1</td>
<td>6:00-18:00</td>
<td>91.42</td>
<td>25.39</td>
<td>268.77</td>
<td>10.77</td>
<td>6.68</td>
<td>16.38</td>
</tr>
<tr>
<td></td>
<td>18:00-6:00</td>
<td>41.52</td>
<td>11.53</td>
<td>122.07</td>
<td>0.00</td>
<td>4.98</td>
<td>12.19</td>
</tr>
<tr>
<td>2</td>
<td>6:00-18:00</td>
<td>90.42</td>
<td>25.12</td>
<td>265.83</td>
<td>10.88</td>
<td>6.68</td>
<td>16.38</td>
</tr>
<tr>
<td></td>
<td>18:00-6:00</td>
<td>41.52</td>
<td>11.53</td>
<td>122.07</td>
<td>0.00</td>
<td>4.98</td>
<td>12.19</td>
</tr>
<tr>
<td>TOTAL</td>
<td></td>
<td>264.88</td>
<td>73.58</td>
<td>778.75</td>
<td>7.43</td>
<td>23.32</td>
<td>57.13</td>
</tr>
</tbody>
</table>
In Table 3, the simulated A/C electrical power consumption during the two days is found to be about 264.88 MJ while the total rejected heat at the condenser is about 778.74 MJ. The percentage ratio of the thermal energy from the PV-Th collector ($Q_{th}$) to that from the condenser ($Q_{con}$) is also presented. The average $Q_{th}/Q_{con}$ ratio during the day is about 10% while it is zero as there is no solar irradiance during the night. When taking the night into account, the overall average thermal energy generated from PV-Th collector is only about 7% of the total energy that is used to regenerate the desiccant.

The amount of water in kilogram adsorbed by the dehumidification unit can be converted into the amount of energy that it is required to evaporate or condense by multiplying it with 2,450 kJ/kg, which is the enthalpy of vaporization of water at 25°C. It is obviously seen that the total energy that the dehumidification unit absorbed is 57.13 MJ.

If the air-conditioned room does not have the desiccant dehumidification unit, the A/C system must consume additional energy in order to handle the condensation of water vapor at the cooling coil in place of the desiccant dehumidification unit. In such a case, the total energy consumption required by the A/C system can be approximated equal to the summation of the amount of energy of the water adsorbed by the desiccant in the dehumidification unit, which is divided by COP and efficiency of compressor, and the existing energy consumption of the A/C system. The value obtained from simulation is 294.95 MJ. The last column of Table 3 presents the total electrical energy saving for the hybrid PV-Th air heating
collector and the desiccant dehumidification unit combined with the A/C system. The energy saving is approximately 12%.

5. CONCLUSION

It can be concluded that the developed simulation model is able to predict the performance of a hybrid photovoltaic-thermal air heating system for regenerating desiccant in an air conditioning room. The dynamic variations of several system performance parameters such as the electrical power output, the mount of heat that can be drawn from the system, temperature and humidity of each considered parts can be simulated. The PV-Th air heating can be used to supply the hot air, which is about 50°C, for mixing with those of condensing unit of air conditioning system to regenerate the saturated desiccant. In addition, the PV-Th efficiency is high up to 66% while the generated electrical energy of 6.52 MJ can be used to supply the electrical device in the system.

The developed model also provides the total amount of energy saving of the A/C system with and without the desiccant dehumidification unit. It showed that the system was able to save about 12% of the total energy consumption. These results initiate a conclusion that the devised system is feasible to install and can be used in the house in the tropical climate such as that in Thailand. However, the simulated results are needed to directly compare with those of relevant experiments with a longer time span of simulation and experimental validation. Finally, it is expected that once the developed simulation model is successfully verified by experimental...
results, it will be useful not only for predicting the whole system transient performance, but also for designing the system to meet the load requirements at any operating site location provided that the local meteorological data is available.

6. NOMENCLATURE

\( A \) = Area \((m^2)\), \( C \) = Specific heat \((J/kg \, K)\), \( COP \) = Coefficient of performance, \( D \) = space between the parallel flat plate \((m)\), \( D_h \) = Hydraulic diameter of air flow channel \((m)\), \( D_{W-as} \) = Binary mass diffusion, \( F \) = View Factor, \( G \) = Global Radiation \((W/m^2)\), \( h \) = Heat Transfer Coefficient \((W/m^2K)\), \( k \) = Thermal conductivity \((W/m^2)\), \( L \) = Thickness \((m)\), \( m \) = Mass \((kg)\), \( \dot{m} \) = Mass flow rate \((kg/s)\), \( n \) = infiltration rate \(\%\), \( N_u \) = Nusselt number, \( P \) = Pressure \((Pa)\), \( P_{elec} \) = Electrical power \((W)\), \( P_r \) = Prandl number, \( Q \) = amount of thermal energy \((MJ)\), \( \dot{Q} \) = Rate of thermal energy \((W)\), \( R_d \) = adsorption rate \((kg/s)\), \( R_e \) = Reynolds number, \( R_W \) = gas constant, \( S_c \) = Schmidt Number, \( S_h \) = Sherwood number, \( SC \) = Shading coefficient, \( T \) = Temperature \((K)\), \( \bar{T} \) = Average temperature \((K)\), \( t \) = Time \((s)\), \( U \) = Overall heat loss coefficient \((W/m^2K)\), \( V \) = air velocity \((m/s)\), \( W \) = amount of work done \((MJ)\), and \( \dot{W} \) = rate of total work done \((W)\).

**Greek letters:** \( \alpha \) = absorptance, \( \sigma \) = Stephan-Boltzmann constant, \( \tau \) = transmittance, \( \eta \) = efficiency, \( \varepsilon \) = emissivity, \( \beta \) = tilted angle, \( \rho \) = air density \((kg/m^3)\), \( \omega \) = absolute humidity, \( \phi \) = relative humidity, \( \mu \) = dynamic viscosity of air, \( \nu \) = kinematic viscosity of air, and \( h \) = enthalpy.
Subscripts: \(a = \text{ambient},\) \(as = \text{air stream},\) \(b = \text{back plate},\) \(c\) or \(\text{conv} = \text{convective},\) \(com = \text{compressor},\) \(con = \text{condenser},\) \(D = \text{desiccant},\) \(elec = \text{electrical},\) \(evap = \text{evaporator},\) \(f1 = \text{working fluid (air) at channel 1},\) \(f2 = \text{working fluid (air) at channel 2},\) \(g = \text{glass cover},\) \(in = \text{input},\) \(infil = \text{infiltration},\) \(ins = \text{insulation},\) \(la = \text{latent heat},\) \(mix = \text{mixing unit},\) \(out = \text{output},\) \(p = \text{absorber plate},\) \(PV-Th = \text{PV-Th collector},\) \(r\) or \(\text{room} = \text{room},\) \(rad = \text{radiative},\) \(roof = \text{roof},\) \(S = \text{solar cell},\) \(sat = \text{saturated},\) \(si = \text{sensible heat},\) \(sky = \text{sky},\) \(so = \text{sol-air},\) \(store = \text{stored by},\) \(th = \text{thermal},\) \(u = \text{useful energy},\) \(v = \text{water vapor},\) \(W = \text{water},\) \(wall = \text{wall},\) \(win = \text{window},\) \(\text{wind} = \text{wind},\) \(1 = \text{first state},\) \(2 = \text{second state},\) \(3 = \text{third state},\) and \(\rightarrow = \text{to}.

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References


