Performance analysis of photovoltaic-thermal collector by explicit dynamic model

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Abstract

Although the performance of hybrid photovoltaic-thermal (PV/T) collector had been studied both experimentally and numerically for some years, the thermal models developed in previous studies were mostly steady-state models for predicting the annual yields. The operation of a PV/T collector is inherently dynamic. A steady-state model is not suitable for predicting working temperatures of the PV module and the heat-removal fluid during periods of fluctuating irradiance or intermittent fluid flow. Based on the control-volume finite-difference approach, an explicit dynamic model was developed for a single-glazed flat-plate water-heating PV/T collector. A transport delay fluid flow model was incorporated. The proposed model is suitable for dynamic system simulation applications. It allows detailed analysis of the transient energy flow across various collector components and captures the instantaneous energy outputs.

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1. Introduction

Different kinds of hybrid photovoltaic-thermal (PV/T) collectors had been proposed in the past. One with a practical design is the flat-plate water-heating collector inserted with a PV module. Comparing with the conventional collectors, the advantages of this hybrid design are the saving in space, lower operating temperature, and the increase in overall efficiency with simultaneous electricity generation and service-water heating. Comparing with other hybrid designs, its advantage lies in the simple construction, lower production costs and very likely, lower life-cycle costs. Although the performance of this kind of collector had been studied both experimentally (e.g. Lalovic et al., 1986; Huang et al., 2001; Tripanagnostopoulos et al., 2002) and numerically (e.g. Florschuetz, 1979; Bergene and Lovvik, 1995; Garg and Agarwall, 1995), the thermal models accompanying these studies were basically for steady-state analysis. The reason was that the collector system performance could be adequately evaluated through the quasi-steady-state analysis using hourly weather data.

The operation of a PV/T collector is inherently dynamic. The excitations like solar irradiance and wind are transient in nature. The circulation pump operation can be intermittent. The numerical process of a dynamic system simulation is no longer a real burden in these days in terms of computer resources. Jones and Underwood (2001) pointed out that a steady state model of the PV module temperature cannot be justified during periods of rapidly fluctuating irradiance, when the response time caused by the thermal mass of the PV material becomes significant. The time constant of the panel was found typically in the order of minutes. A dynamic model is particularly useful if the interest lies in studying system control or the extensive thermal analysis of various collector components. Zondag et al. (2002) reported the use of four numerical models (one 3D dynamic model and three steady-state models) for evaluating the thermal yield of a combi-PV/T prototype, which was based on single-tube serpentine design. The 3D dynamic model was an extensive one, typically using 2.5-h simulation time for 1-h real-life equipment operation. The model was therefore not developed for system simulation purpose. In this paper, an explicit dynamic thermal
A model suitable for system simulation is introduced. With the control-volume finite-difference approach, the model on one hand can generate results for hourly performance analysis, and on the other hand, can provide information on the transient performance, including the instantaneous thermal/electrical gains, efficiencies, and thermal state of various collector components. With an extension of the nodal scheme, the model is able to give a complete thermal analysis of the equipment.

2. Energy flow analysis

Fig. 1 shows the front view of one water-heating PV/T collector, and a cross-section at one water tube between two mid-planes of the absorber plate. The front glass cover is separated from the PV plate by an air gap. The PV plate is fixed on to the absorber plate through a thin adhesive layer. This adhesive layer is a compound one that includes also the EVA (ethylene-vinyl acetate) layer and the Tedlar layer. The flat absorber plate forms the fins of the heat-exchange water tubes, which are arranged at equal spacing throughout the panel width. The water tubes are boned to the absorber plate. The arrangement of the two common headers at the top and bottom ends allows balanced water flow in all tubes, and makes the mid-distance between any two adjacent water tubes thermally the plane of symmetry. There will be no heat flow across this plane at any time under proper operation. The edges and bottom surface of the panel are inserted with thermal insulation. For a compact and thin panel design, the losses of the absorbed solar energy

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>( \Theta ) absolute temperature, K</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \theta_1 )</td>
<td>angle of incident, degree</td>
</tr>
<tr>
<td>( \theta_2 )</td>
<td>angle of refraction, degree</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>normalized distance, –</td>
</tr>
<tr>
<td>( \mu )</td>
<td>weighting factor, –</td>
</tr>
<tr>
<td>( \rho )</td>
<td>reflectance, –</td>
</tr>
<tr>
<td>( \sigma )</td>
<td>Stefan–Boltzman constant, ( 5.67 \times 10^{-8} ) W/m² K⁴</td>
</tr>
<tr>
<td>( \tau )</td>
<td>transmittance, –</td>
</tr>
<tr>
<td>( \tau_a )</td>
<td>transmittance of glass cover (considering only absorption losses), –</td>
</tr>
<tr>
<td>( \tau_r )</td>
<td>transmittance of glass cover (considering only reflection losses), –</td>
</tr>
<tr>
<td>( \tau_{\alpha} )</td>
<td>effective absorptance, –</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Subscripts</th>
<th>a air</th>
</tr>
</thead>
<tbody>
<tr>
<td>ad adhesive layer</td>
<td></td>
</tr>
<tr>
<td>b absorber plate</td>
<td></td>
</tr>
<tr>
<td>bo bond</td>
<td></td>
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<tr>
<td>c convective</td>
<td></td>
</tr>
<tr>
<td>cell solar cell</td>
<td></td>
</tr>
<tr>
<td>e environmental; electrical</td>
<td></td>
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<tr>
<td>g glass cover</td>
<td></td>
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<tr>
<td>gr ground</td>
<td></td>
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<tr>
<td>i inner; insulation material</td>
<td></td>
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<tr>
<td>m mean</td>
<td></td>
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<tr>
<td>o outer</td>
<td></td>
</tr>
<tr>
<td>p PV plate; power</td>
<td></td>
</tr>
<tr>
<td>r radiation; reference</td>
<td></td>
</tr>
<tr>
<td>s sky</td>
<td></td>
</tr>
<tr>
<td>t tube; tube bonding</td>
<td></td>
</tr>
<tr>
<td>u surrounding</td>
<td></td>
</tr>
<tr>
<td>w water</td>
<td></td>
</tr>
</tbody>
</table>

Greek

| \( \alpha \) | absorptance, – |
| \( \beta \) | temperature coefficient, /K |
| \( \delta \) | thickness or depth, m |
| \( \epsilon \) | emissivity, – |
| \( \eta \) | efficiency, – |
to the surrounding are mainly through the front and back panel surfaces; the edge loss is negligible especially for a large panel.

An explicit dynamic analysis can be worked on through solving the transient energy balance equations for the various collector components. In the control-volume finite-difference technique (Clarke, 2001), a control volume is created by defining a fictitious boundary enclosing a physical space within which the conservation laws of physics (mass and energy balance in particular) are applied. This control volume appears as a node in the simulation network. For the given collector, adequate uniformity in material properties and physical dimensions in each panel component can be assumed. Further, the water flow rate and temperature conditions in all parallel tubes can be taken as the same. The thermal behavior of the entire panel is then well defined by analyzing the heat transfer in the vicinity of a single water tube, as in section Z–Z of Fig. 1(b). In the case of a small panel, the edge loss can be absorbed into the back loss with an adjusted surface area. Accumulating heat transfer along the water flow direction (X-direction) lead to a positive temperature gradient at all components. This temperature gradient is treated separately with that in the transverse direction (Y-direction). In this way, the energy exchange across various components can then be handled by considering their mean temperatures.

A PV/T collector described above can be represented by seven nodes, and mathematically, by a matrix equation set derived from the instantaneous energy and mass flow balance at these nodes. As denoted in Fig. 1(b), the first node ‘g’ represents the glass cover. The second node ‘p’ is for the PV plate. The third one ‘b’ is for the thin-plate absorber. Then ‘t’ is for the metallic bonding between the plate and the tube, ‘i’ is for insulation layer, and ‘w1’ is for the water in tube. The last node ‘w2’ not shown in this figure is for the leaving water node.
This node passes on the outlet water condition to the downstream component, and facilitates the transport delay analysis. Fig. 2 gives an energy flow diagram of the PV/T collector based on the R–C circuit representation.

At the glass cover, there are two interfaces to cause reflection losses. The solutions of its transmittance $\tau_g$, reflectance $\rho_g$, and absorptance $\alpha_g$, allowing for both reflection and absorption can be obtained via ray-tracing techniques (Duffie and Beckman, 1991). If at any time instant the solar radiation flux falling on the glass surface is at a rate of $G$ in Watts, the radiation energy absorbed by the glass cover is given by

$$Q_g = G\alpha_g = G(1 - \tau_g)$$

where $\tau_g$ is the transmittance of the glass considering only absorption loss. With the angles of incidence and refraction of the direct beam represented by $\theta_1$ and $\theta_2$, applying the Bouguer’s law gives

$$\tau_g = e^{-A\delta_g/\cos\theta_2} = \exp \left[ -A\delta_g \left( 1 - \sin^2 \theta_1 / R_g^2 \right)^{-0.5} \right]$$

where $A$ is the extinction coefficient and $R_g$ is the refractive index of the glass cover.

The energy accumulated in glass is a result of solar radiation absorbed ($Q_g$) and the net heat exchange with: (i) the ambient air at temperature $T_a$ for convective heat exchange, (ii) the background equivalent (sky, ground and surroundings) environment at temperature $T_e$ for longwave radiation heat exchange, and (iii) the PV plate at temperature $T_p$ through the confined air gap for combined mode heat exchange, i.e. convection + radiation.

The heat transfer coefficients at the outer surface are given by

$$h_{ag} = (h_c)_{ag} = 3u_a + 2.8$$

$$h_{eg} = (h_e)_{eg} = \varepsilon_g \sigma (\Theta_g^2 + \Theta_e^2)(\Theta_g + \Theta_e)$$

where $u_a$ is the wind velocity in m/s (Watmuff et al., 1977); $\sigma$ is the Stefan–Boltzmann’s constant, $\Theta$ and $\varepsilon$ are respectively the absolute temperature and the emissivity of the corresponding layer. If $f_s$, $f_g$ and $f_u$ are respectively the view factors of the glass surface to the sky, ground and surroundings, then

$$\Theta_e^4 = f_s \Theta_s^4 + f_g \Theta_gr^4 + f_u \Theta_u^4$$
The background equivalent temperature has been found not important for solar collector performance. Our results show that at 15 °C air temperature, the change of \( T_e \) from 0 to 15 °C cause the change in electricity and thermal gains by less than 1%. Hence for its practical applications in buildings, \( T_e \) can be taken the same as \( T_e \).

For the inner surface, the heat transfer coefficient is given by

\[
h_{gp} = (h_1)_{gp} + (h_c)_{gp} = \frac{\sigma \left( \Theta_x^2 + \Theta_p^2 \right) (\Theta_x + \Theta_p)}{T_c + \frac{1}{T_c} - \frac{1}{T_p}} + \frac{N_u k_s}{\delta_a}
\]

(6)

where \( k \) and \( \delta \) are respectively the thermal conductivity and the thickness of the corresponding layer. Hollands et al. (1976) derived the relationship between the Nusselt number and Raleigh number for natural convection between parallel flat plates.

Heat absorbed by the PV plate is

\[
Q_p = G(\tau z)_p - E_p
\]

(7)

\((\tau z)_p\) is its effective absorptance which is given by

\[
(\tau z)_p = \frac{\tau_p \tau_x \tau_p}{1 - (1 - \tau_x)r}
\]

(8)

where \( \tau_p \) is the transmittance of the glass considering only reflection loss, and \( r \) the reflectance of glass for diffuse radiation. Some radiation will reach the absorber plate if the PV plate is partially transparent. The generated DC power \( E_p \) varies with the temperature-dependent solar cell operating efficiency \( \eta_{cell} \). If \( r_c \) is the ratio of cell area to aperture area, then

\[
E_p = G r_c \eta_{cell}
\]

(9)

and

\[
\eta_{cell} = \eta_i \left[ 1 - \beta_i (T_p - T_i) \right]
\]

(10)

\( \eta_i \) is the reference cell efficiency at the reference operating temperature \( T_i \), and \( \beta_i \) is the temperature coefficient.

The heat flow from the PV plate to the tube metallic bond position can be considered as through two separate paths: (i) through the adhesive layer then along the absorber plate and (ii) along the PV plate then through the adhesive layer at the tube bonding position.

Heat flow from the metallic bond to the water in tube is by means of conduction and convection. The thermal conduction is determined by the bond conductance given by

\[
h_{bw} = \frac{k_{bo} W_{bo}}{\delta_{bo}}
\]

(11)

where \( k_{bo} \) is the bond thermal conductivity, \( \delta_{bo} \) the bond average thickness, and \( W_{bo} \) the bond width. The convective heat transfer \( h_w \) can be obtained from the Dittus–Boelter equation for fully developed turbulent flow (Holman, 1989), i.e.

\[
Nu_D = 0.023 \Re D^{0.8} \Pr^{0.4}.
\]

(12a)

For fully developed laminar flow,

\[
h_w = 4.364 k_w D_i
\]

(12b)

where \( D_i \) is the inner tube diameter. A more accurate representation of the average convective heat transfer coefficient requires to consider the entrance effects and the combined forced and natural convection modes.

It can be shown that

\[
\frac{1}{h_{bw} A_{bw}} = \frac{1}{h_w \pi D_i L} + \frac{1}{c_{bo} L}.
\]

(13)

In case there is a sudden change of incoming water condition such as its temperature \( T_{w0} \) or the mass flow rate \( m_w \), \( T_{w1} \) and \( T_{w2} \) will be determined by transport delay analysis. This is often required when the flow velocity is low or the simulation time step is small (Chow, 1997). Within one finite time increment \( \delta t \), a fluid element in the tube advances by a distance of \( u_w \cdot \delta t \). If the entire water tube of length \( L \) is divided evenly into \( N \) number of segments such that \( N \) is the smallest whole number given by

\[
N = \frac{1}{\delta \lambda} \geq \frac{L}{u_w \delta t}
\]

(14)

\( \delta \lambda \) is the normalized length of a fluid element. It can be seen that \( N \) is water velocity dependent and can be a variable during simulation. The fluid element which moves from segment \( "j-1" \) to segment \( "j" \) will have traveled a distance of \( (\lambda_j - \lambda_{j-1}) \) where \( \lambda_j \) is the normalized distance of the point \( "j" \) from the tube entrance and has a value in the range of 0–1. This fluid element will undergo the following temperature change in the future time step:

\[
T_{w} (\lambda_j) = \frac{m_w C_w T_{w0} (\lambda_{j-1}) + h_j A_j \delta \lambda T_i^*}{m_w C_w + h_j A_j \delta \lambda}
\]

(15)

where \( T_{w0} (\lambda_j) \) and \( T_i^* \) denote the present time step values. All other variables are expressed for the future time step. The average water temperature \( T_{w1} \) is given by the weighted mean of the array of the axial temperature values along the tube such that

\[
T_{w1} = \frac{1}{N} \left[ \frac{T_{w0} + T_{w2}}{2} + \sum_{j=1}^{N-1} T_w \left( \frac{j}{N} \right) \right]
\]

(16)
The outlet temperature of the last segment gives $T_{w2}$.

The thermal efficiency of the collector is given by the ratio of the heat power $H_p$ received by the leaving water to the corresponding radiation $G$ falling over the aperture area $A$, i.e.

$$\eta_t = \frac{H_p}{G} = \frac{\dot{m}_wC_w(T_{w2} - T_{w1})}{G}$$  \hspace{1cm} (17)

and the electrical efficiency is

$$\eta_e = \frac{E_p}{G}.$$  \hspace{1cm} (18)

### 3. Matrix template

Consider the energy balance at the portion of the collector defined by Section Z–Z in Fig. 1(b) with surface area $A$, length $L$ and spacing $W$. For the glass node ‘g’,

$$M_gC_g \frac{dT_g}{dt} = h_{eg}A_{eg}(T_a - T_g) + h_{eg}A_{eg}(T_e - T_g)$$

$$+ h_{eg}A_{eg}(T_p - T_g) + Q_g$$  \hspace{1cm} (19)

where $M_g$ and $C_g$ are respectively the mass and the specific heat of glass; $A_{eg} = A_{eg} = A_{eg} = A$.

For the PV plate (node ‘p’),

$$M_pC_p \frac{dT_p}{dt} = h_{gp}A_{gp}(T_g - T_p) + h_{bp}A_{bp}(T_b - T_p)$$

$$+ h_{pl}A_{pl}(T_i - T_p) + Q_p$$  \hspace{1cm} (20)

where

$$A_{bp} = A \left(1 - \frac{D_o}{W}\right)$$  \hspace{1cm} (21)

$$h_{bp} = \frac{k_{ad}}{\delta_{ad}}$$  \hspace{1cm} (22)

$$h_{pl}A_{pl} = \frac{\delta_p L}{x_p + \frac{\delta_{ad}}{\delta_p} + \frac{\delta_p}{k_{ad} D_o}}$$  \hspace{1cm} (23)

$$x_p = W/4.$$  \hspace{1cm} (24)

$k_{ad}$ and $\delta_{ad}$ are respectively the thermal conductivity and the thickness of the adhesive layer.

For the absorber plate (node ‘b’),

$$M_bC_b \frac{dT_b}{dt} = h_{bp}A_{bp}(T_p - T_b) + h_{bp}A_{bp}(T_i - T_b)$$

$$+ h_{bp}A_{bp}(T_i - T_b)$$  \hspace{1cm} (25)

where

$$A_{bi} = A \left(\frac{W - D_o}{W}\right)$$  \hspace{1cm} (26)

$$A_{bi} = \delta_p L$$  \hspace{1cm} (27)

$$h_{bi} = \frac{2k_b}{x_b}$$  \hspace{1cm} (28)

$$h_{bi} = \frac{2k_b}{\delta_t}$$  \hspace{1cm} (29)

$$x_b = (W - D_o)/4.$$  \hspace{1cm} (30)

For the tube bonding node ‘t’,

$$M_tC_t \frac{dT_t}{dt} = h_{bt}A_{bt}(T_b - T_t) + h_{bt}A_{bt}(T_i - T_t)$$

$$+ h_{tw}A_{tw}(T_w - T_t) + h_{pt}A_{pt}(T_p - T_t)$$  \hspace{1cm} (31)

where

$$A_{tw} = \pi D_t L$$  \hspace{1cm} (32)

$$A_{ti} = \left(\frac{\pi}{2} + 1\right)D_o L = 2.571D_o L.$$  \hspace{1cm} (33)

Lumped in $M_t$ are the masses of metallic tube, metallic bond and the small portion of absorber plate above the bond (with width = $D_o$).

For an insulation layer with $\delta_t \gg D_o$, an approximation gives $h_{ti} = h_{ti}$.

For the insulation node ‘i’,

$$M_iC_i \frac{dT_i}{dt} = h_{ii}A_{ii}(T_b - T_i) + h_{ii}A_{ii}(T_i - T_i) + h_{ii}A(T_s - T_i)$$  \hspace{1cm} (34)

where

$$\frac{1}{h_{ii}} = \frac{\delta_i}{2k_i} + \frac{1}{h_{ag}}.$$  \hspace{1cm} (35)

The radiation exchange can be neglected at the bottom side, since the temperature difference between the shaded surface and the surrounding is negligibly small.

For the water in tube (node ‘w1’),

$$M_wC_w \frac{dT_{w1}}{dt} = h_{tw}A_{tw}(T_1 - T_{w1}) + \dot{m}_wC_w(T_{w0} - T_{w2})$$  \hspace{1cm} (36)

In most simulation time steps the transport delay will not be invoked, then $T_{w1}$ is the arithmetic mean of the inlet and outlet conditions, hence

$$T_{w1} - 0.5T_{w2} - 0.5T_{w0} = 0.$$  \hspace{1cm} (37)

Accordingly, the energy balance matrix equation based on the finite-difference formulation is as follows:
In the model, the mass flow balance is straightforward since only the water nodes are involved in the fluid flow network. If incompressible liquid flow is assumed,
\[
m_{w0} = m_{w1} = m_{w2}.
\]
In the mass-balance matrix equation using the same matrix template as in Eq. (38), all the diagonal coefficients carry a value of 1; C(24) is −1, C(31) is 1, and the remaining coefficients are zeros.

### 4. Simulation studies

This explicit model can perform energy analysis for arriving at optimum designs. Fig. 3 shows the variations in thermal and electrical efficiencies for a range of water flow rate in tube. The collector represents a panel with an aperture area 2 m long by 1 m wide, and is inclined at 45°. The glass cover is of non-iron type with \( \epsilon_g = 0.88 \). The PV plate is of \( \epsilon_p = 0.88 \) and \( \alpha_p = 0.9 \). Both the absorber plate and tubes are in copper. \( T_v, T_c \) and \( T_w0 \) are at 30 °C. \( I_t \) is 800 W/m², \( u_w \) is 1.5 m/s. Other technical data includes: \( D_o = 0.01 \) m, \( W = 0.2 \) m, \( r_e = 0.8 \), \( \eta_t = 12\% \) at 25 °C, \( \beta_t = 0.0045 \) °C⁻¹. Two key manufacturing defects found in PVT collectors are: (i) the imperfect adhesion between the PV plate and the absorber plate, and (ii) the imperfect bonding between the absorber plate and the metallic tubes. Obtained from steady-state simulation with the above data are the four efficiency curves versus water flow rate shown in Fig. 3(a) and (b). The group of curves represents a range of collector quality from perfect to defective, i.e.

1. \( h_{bp} = 10000 \) W/m² K and \( c_{bo} = 10000 \) W/m K,
2. \( h_{bp} = 100 \) W/m² K and \( c_{bo} = 100 \) W/m K,
3. \( h_{bp} = 45 \) W/m² K and \( c_{bo} = 45 \) W/m K,
4. \( h_{bp} = 25 \) W/m² K and \( c_{bo} = 25 \) W/m K.

Under the given condition the maximum combined efficiency (\( \eta_t + \eta_e \)) of a perfect collector can be over 70%. This may decrease to less than 60% for a low-quality collector. These are in line with the previous studies. The maximum efficiency state however is only achieved outside the normal design range of water flow rate.

In this explicit model, we have assumed that the temperature gradient along the \( X \)-direction will not affect the temperature distribution in the \( Y \)-direction, and one nodal temperature is used for each component. The temperature gradient in the \( X \)-direction can be obtained by dividing the collector into several panel segments along its length, and the whole collector is then represented by an interconnection of these panel segments. Fig. 4 shows the temperature variation of the nodes when the collector is divided into four equal segments of length 0.5 m each. The two power output lines of \( H_p \) and \( E_p \) are also shown. The simulation was based on aluminum absorber and copper tubes. The water flow was 0.005 kg/s per tube. All nodal temperatures were found increase linearly, though the gradients may not be the same. The power output lines drop linearly. The change of \( E_p \) is not apparent on the graph owing to its small change in magnitude. The mean nodal temperatures of the four segments are listed in Table 1, with \( T_{w2} \) given by the leaving water node of the last segment. The results
obtained from using one single segment and two segments are also listed in the table for ready comparison. It can be seen that the three sets of results are almost identical, in particular for the three most important performance parameters: $T_w$, $E_p$ and $H_p$. It is expected that when the thermal conduction for all components along the X-direction is also taken into consideration, the differences in the listed results will be even smaller.

The variation of plate temperatures in the Y-direction is non-linear. However, in our results the position of the mean temperature point was found not sensitive to the outputs. Using the same simulation condition as above, the change of $\alpha_b$ by $\pm 30\%$ caused the change of $H_p$ and $\eta_i$ by $\pm 2\%$, and $E_p$ and $\eta_e$ by less than $\pm 1\%$. The above tests confirmed the justification of using a single node for each component.

A step increase of water flow rate from 0.002 to 0.004 kg/s has been used to compare the dynamic response of various components making use of the time constant (TC). The simulation results are in Fig. 5(a–c). The values of TC were found in the range of 47–530 s, with the water node at the low end and the glass node the high end. These TC values vary with the actual collector operating conditions and the type of disturbances. The TC of the water node in tube is in the same order of magnitude as the tube flushing time. This justifies the use of transport delay analysis to model the transient behavior under finite simulation time steps. The delay analysis is particularly important in those systems with very lengthy panels and the liquid flow is led by natural circulation. During the sudden increase of water flow in this case, the improved convective heat transfer at the tube interior surface led to a momentary sharp increase of heat gain by 73%. This is captured in Fig. 5(b). Notwithstanding this, the electrical gain curve in Fig. 5(c) showed a gradual improvement of cell performance.

Table 1
Comparison of simulation results for mean temperatures (in °C) and energy gains (in W) with panel divided into different number of segments

<table>
<thead>
<tr>
<th></th>
<th>Single segment</th>
<th>Two segments</th>
<th>Four segments</th>
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<tbody>
<tr>
<td>$T_g$</td>
<td>33.14</td>
<td>33.16</td>
<td>33.17</td>
</tr>
<tr>
<td>$T_p$</td>
<td>51.72</td>
<td>51.78</td>
<td>51.79</td>
</tr>
<tr>
<td>$T_b$</td>
<td>51.46</td>
<td>51.51</td>
<td>51.53</td>
</tr>
<tr>
<td>$T_i$</td>
<td>46.05</td>
<td>46.11</td>
<td>46.12</td>
</tr>
<tr>
<td>$T_{w2}$</td>
<td>37.50</td>
<td>37.53</td>
<td>37.54</td>
</tr>
<tr>
<td>$E_p$</td>
<td>46.40</td>
<td>46.40</td>
<td>46.39</td>
</tr>
<tr>
<td>$H_p$</td>
<td>27.03</td>
<td>27.02</td>
<td>27.02</td>
</tr>
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</table>

Fig. 4. Temperature distribution in °C and energy gain in Watts at the four panel segments.

Fig. 5. Dynamic response of PV/T collector subject to a step increase in water flow rate.
The water flow rate was 0.004 kg/s. The air temperature was 17 °C. When the irradiation reduced by 25%, \( E_p \) dropped immediately to a level only slightly higher than 25% of the original (since \( T_p \) has dropped). \( H_p \) dropped gradually, like \( T_p \). The instantaneous \( E_p \) momentarily jetted up to more than 90% within the first minute. This amplification was owing to the remaining high level of \( T_p \) at a time of low irradiation level. \( T_{w2} \) showed a mild change, decreasing at the same pace as \( T_t \). At the 660 s, \( T_{w2} \) dropped to a level only 2 °C from \( T_{w0} \). The temperature differential sensor passed a signal to call for 'pump-off'. \( H_p \) soon approached zero when the pump stopped in 30 s and only very minor flow remained. Meanwhile \( E_p \) remained stable. Without the cooling effect, \( T_t \) climbed up to reach \( T_b \) which is close to \( T_p \). From this point onward, the entire plates were virtually at their uniform temperatures.

The computation time of the last two dynamic analyses were in the order of seconds on the Pentium PC platform. This PV/T model is therefore suitable for both dynamic and year round integrated building simulation. With an extension of the nodal scheme, say increasing the number of nodes at the PV plate and the absorber plate for handling multi-dimensional heat conduction, the model is able to give a comprehensive thermal analysis of the equipment. The latter will be useful for product design applications.

5. Conclusions

The operation of a hybrid PV/T collector is inherently dynamic. In the study of transient solar system performance that involves fluctuating irradiance and/or an imposed system control scheme, dynamic analysis has to be performed. An explicit dynamic model of a water-heating PV/T collector suitable for dynamic system simulation has been presented. The seven-node model, derived from the control-volume finite-difference formulation and incorporated with a transport delay subprogram, can provide information on the transient performance, including the instantaneous thermal/electrical gains, their efficiencies, and thermal conditions of various collector components. The appropriateness of the nodal scheme had been tested by using both steady-state and dynamic simulations. The results supported the use of a single node for each component. Because of the relatively short simulation time required, the model is also suitable for hourly analysis of equipment energy performance. With an extension of the nodal scheme to include multi-dimensional thermal conduction on PV and absorber plates, the model is able to perform complete energy analysis on the hybrid collector.

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References

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