Annual analysis of heat pipe PV/T systems for domestic hot water and electricity production

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ABSTRACT

Heat-pipe photovoltaic/thermal (HP-PV/T) systems can simultaneously provide electrical and thermal energy. Compared with traditional water-type photovoltaic/thermal systems, HP-PV/T systems can be used in cold regions without being frozen with the aid of a carefully selected heat-pipe working fluid. The current research presents a detailed simulation model of the HP-PV/T system. Using this model, the annual electrical and thermal behavior of the HP-PV/T system used in three typical climate areas of China, namely, Hong Kong, Lhasa, and Beijing, are predicted and analyzed. Two HP-PV/T systems, with and without auxiliary heating equipment, are studied annually under four different kinds of hot-water load per unit collecting area (64.5, 77.4, 90.3, and 103.2 kg/m²).

1. Introduction

The photovoltaic/thermal (PV/T) collector is an integration of PV cells and a solar thermal collector into a single unit that can generate electrical and thermal energy simultaneously. The PV/T collector can reduce required space and initial cost because a common frame and bracket are used in the setup, unlike a separate PV module and a solar thermal collector that are separately installed [1]. The effective rate of solar energy utilization per unit collecting area can also be increased in this PV/T collector.

The first study on a PV/T system was presented by Martin Wolf in 1976 [2]. A large number of investigations on PV/T collectors/systems have been subsequently conducted. For example, Raghuraman [3] presented two separate one-dimensional mathematical models for predicting the performance of water and air PV/T flat plate collectors and compared the results with the experimental data. Design recommendations were also made for the maximization of the total energy extracted from the collectors. Similarly, Bergene and Lovvik [4] presented a detailed model for predicting the performance of a PV/T system. Sandnes and Rekstad [5] constructed a PV/T collector by laminating single-crystalline silicon cells onto a black plastic absorber plate. The photovoltaic and thermal performances of the PV/T collector were experimentally and numerically studied in their work. Kalogirou and Tripanagnostopoulos [6] presented a TRNSYS simulation for PV/T systems for domestic hot water production with both thermosyphon and forced water flow. In addition, an economic analysis of the PV/T systems was also given. Ji et al. [7] constructed a flat-box aluminum-alloy, photovoltaic/water-heating system. In their work, the system exhibited a daily thermal efficiency of approximately 40% and primary-energy savings of up to 65% when natural water circulation was adopted. Charalambous et al. [8], Zondag [9], and Ramos et al. [10] presented comprehensive reviews of the PV/T collectors/systems. Chow et al. [11] used energy and exergy to analyze the performance of PV/T collectors with and without a glass cover. Other studies on PV/T collector/systems, such as the performance study of PV/T collector combined with a heat pump [12], system model improvement [13–16], PV/T solar still [17], and performance analysis of a PV/T module for concentrator [18,19], have also been conducted recently. However, a traditional water-type PV/T collector/system is unsuitable for cold regions because the freezing of water will damage the collectors. A heat pipe is a structure with very high thermal conduction that enables the transport of heat without any drop in temperature. Freezing can be eliminated by carefully selecting the working fluid, which can also reduce corrosion. Hence, a heat pipe and a solar collector can be incorporated into a practical design for a PV/T collector.

Several studies on heat pipes integrated with solar thermal collectors have been conducted. Hammad [20] studied flat-plate heat pipe collectors, and the results of his study indicated that the thermal efficiency of a heat pipe collector is comparable with that of a water-cooled solar collector. Furthermore, the transient thermal performances of wickless heat pipe, flat-plate solar collectors are affected by various parameters, such as solar radiation intensity, temperature of cooling water, absorber plate material and thickness, and the ratio of condenser-section length to total wickless heat pipe length [21]. Hiroshi and Yasuhiro [22] designed a vertical...
multiple-effect diffusion-type solar still coupled with a heat pipe solar collector. Theoretical research showed that the still could produce 21.8 kg/(m² d) of distilled water on a sunny autumn equinox day with a solar radiation of 22.4 MJ/(m² d). Esen [23], on the other hand, designed a solar cooker by using a vacuum-tube collector with heat pipes. Experiments were conducted to examine the performance of the solar cooker with three different heat-pipe working fluids. The results of the experiments revealed that a cooking time between 27 min and 70 min could be obtained using this type of cooker. Riffat et al. [24] developed a theoretical model investigating the thermal performance of a thin-membrane heat pipe solar collector. Bourdoukan et al. [25] experimentally studied the potential of solar heat pipe vacuum collectors in the desiccant-cooling process. Azad [26] presented a theoretical model based on the effectiveness of the number of transfer units method in evaluating the thermal performances of heat pipe solar collectors. The results were validated using experimental data. An optimum evaporator length to condenser length ratio also was suggested in his work. Although both the heat pipe thermal and the water-type PV/T collectors have been studied, heat pipes integrated with a PV/T panel have rarely been reported previously.

Based on the concept of integrating heat pipes and a PV/T flat-plate collector into a single unit, a heat-pipe PV/T (HP-PV/T) collector was designed and constructed in a previous work [27]. The HP-PV/T collector can be used in cold regions without freezing, and corrosion can be reduced as well. The transient performance of the HP-PV/T system in Dongguan, China, was also studied both experimentally and numerically in a previous work. The current study presents a detailed simulation model for the system. Unlike PV/T system models presented in the previous studies [13–16,27], the proposed simulation model can be used to predict the annual performance of the HP-PV/T system at different geometrical locations, and the additional thermal energy requirement for hot water is also considered. Using the simulation model, the annual electrical and thermal behavior of two HP-PV/T systems, with and without auxiliary heating equipment, are predicted and analyzed when used in three typical climate areas of China, namely, Hong Kong, Lhasa, and Beijing. Performances of the HP-PV/T systems with different kinds of hot-water load per unit collecting area are also studied in the current work. The present work provides useful reference data for the prediction of the energy-saving performance of the HP-PVT system used in different climate areas of China.

2. Description of the heat pipe PV/T system

Fig. 1 shows the structure of the HP-PV/T solar collector. A piece of aluminum plate is chosen as the base panel. The evaporator section of the heat pipes is connected to the back of the aluminum plate, and the condenser section is inserted into a water box. Fig. 2 depicts a section of the HP-PV/T solar collector. The PV cells (single-crystalline silicon) are laminated onto the surface of the aluminum plate. Between the two is a layer of black tedlar–polyester–tellar (TPT), playing a role in the electrical insulation of the PV cells and enhancing the absorption of solar irradiation. Anterior to the PV cells, a low-iron tempered glass plate is used as the upper glaze for the collector, permitting sunlight passage but preventing thermal loss and the entry of dust particles and rain. A thermal insulation layer is placed behind the aluminum plate to prevent thermal loss. The characteristics of the HP-PV/T solar collector considered in the current work are shown in Table 1.

When solar radiation passes through the glass cover and penetrates the PV layer, most of the radiation is absorbed by the PV cells and the black TPT layer. Part of this radiation is converted into electricity, and the rest is eventually converted to heat energy. The electricity is transported by a solar controller to the electricity consumer or to an accumulator battery. Heat energy is conducted along the aluminum plate to the evaporator section of the heat pipes. The heat pipes then transfer this energy to the water flowing through the water box through the condenser sections.

Fig. 3 shows a schematic diagram of the HP-PV/T system. The system comprises a storage tank, a gas heater (for the HP-PV/T...
system with auxiliary heating equipment), a circulation pump, and a number of HP-PV/T solar collectors. The collectors are arranged parallel to one another at the water-circulation loop. The water pump circulates the water between the water-storage tank and the collectors, such that the heat energy of the collectors is removed from the storage tank by the circulating water.

3. Thermal analysis

In the present study, a detailed simulation model for an HP-PV/T system was developed. The mathematical model comprises six primary equation sets: (i) heat-balance equation of the glass cover; (ii) heat-balance equation of the PV module; (iii) uni-dimensional heat conduction of the base panel (aluminum plate); (iv) heat-balance equation of the heat pipe; (v) heat-balance equation for water in the water box; and (vi) heat-balance equation for water in the storage tank. To simplify the calculation, the following assumptions were made in the proposed model:

- Heat conduction in the longitudinal direction of the aluminum plate was neglected.
- The temperatures of the adhesive layer [ethylene–vinyl acetate (EVA) and TPT) and PV cells in the same direction were considered equal.
- The heat capacity of the adhesive layer (EVA and TPT) was neglected.

For the glass cover, the heat-balance equation is as follows:

\[ \delta \rho C_x \frac{\partial T_g}{\partial t} = h_a(T_a - T_g) + h_{e,g}(T_e - T_g) + h_{e,pv}(T_{pv} - T_g) + G_x T_g \]  

and \( T_a \) is given as [28]:

\[ T_a^4 = f_{sky} \cdot T_{sky}^4 + f_g \cdot T_g^4 + f_{sur} \cdot T_{sur}^4 \]

where \( h_a \) and \( h_{e,g} \) are convective and radiant heat-transfer coefficients, respectively, between the glass cover and environments; \( h_{e,pv} \) is the total heat-transfer coefficient between the glass cover and the PV layer; \( T_{sky} \), \( T_g \), and \( T_{sur} \) are the temperatures of the...
sky, ground, and surroundings, respectively; and \(f_{\text{sky}}, f_{\text{gr}},\) and \(f_{\text{sur}}\) are the view factors of the glass surface to the sky, ground, and surroundings, respectively.

The convective and radiant heat-transfer coefficients between the glass cover and environments are respectively given as [29]:
\[
h_a = 2.8 + 3.0u_a
\]
and
\[
h_{cg} = \varepsilon_g\sigma(T_g^4 + T_e^4)(T_e + T_g)
\]
By combining the radiation and convection heat transfer mechanisms, the heat transfer coefficient between the PV and the glass cover, \(h_{g,pv}\), could be derived from the following equation:
\[
h_{g,pv} = \sigma(T_g^4 + T_e^4)(T_e + T_g)
\times \left( \frac{\gamma}{1/\varepsilon_p + \gamma(1/\varepsilon_g - 1)} + \frac{1 - \gamma}{1/\varepsilon_{pv} + (1 - \gamma)(1/\varepsilon_g - 1)} \right)
\div \frac{Nu}{k_a} + \frac{k_a}{k_a}
\]
(5)
where \(\gamma\) is PV cell covering factor, and \(\gamma = A_{pv}/A_c\).

For tilt angles ranging from 0° to 75°, Hollands et al. [30] presented the relationship between the Nusselt and Raleigh numbers as:
\[
Nu = 1 + 1.14 \left( 1 - \frac{1708 \cdot (\sin 1.8\theta)^{1.6}}{Ra \cdot \cos \theta} \right) + \left( \frac{Ra \cdot \cos \theta}{5830} \right)^{1/3} - 1
\]
(6)
where + exponent indicates that only positive values for terms within the square brackets are to be used. In case of negative values, zero is used.

Based on these assumptions, the heat-balance equation for the PV layer, including PV cells, EVA, and TPT, is as follows:
\[
\gamma \rho_p c_p \varepsilon_p \frac{\partial T_{pv}}{\partial t} = h_{g,pv}(T_g - T_{pv}) + \frac{(T_b - T_{pv})}{R_{pv}} + G(\tau \varepsilon)_{pv} - E_{pv}
\]
where \(R_{pv}\) is the thermal resistance between the PV layer and the base panel (aluminum plate), expressed as \(R_{pv} = \delta_{al}/k_{ad}\).

\(E_{pv}\) is given by the instantaneous PV efficiency (\(\eta_{pv}\)) expressed as:
\[
E_{pv} = \gamma G\eta_i(1 - B_i(T_{pv} - T_i))
\]
(8)
where \(\eta_i\) is the reference cell efficiency at the reference operating temperature, \(T_i = 298.15\) K; \(B_i\) is the temperature coefficient, and \(B_i = 0.0045\) K\(^{-1}\); and \(\tau\) is the total transmittance of the covers, given by Eq. (9) [29]:
\[
\tau = \frac{1}{2} \left( \frac{\tau_p \tau_i}{1 - \rho_p \rho_i} + \frac{\tau_o \tau_i}{1 - \rho_o \rho_i} \right)
\]
(9)
where \(\tau_p\) and \(\tau_i\) are the transmittances of the outer (glass) and inner covers (adhesive layer), respectively; \(\rho_p\) and \(\rho_i\) are the reflectances of the outer (glass) and inner covers (adhesive layer), respectively; and \(\perp\) and \(\parallel\) indicate perpendicular and parallel components of unpolarized radiation passing through the covers, respectively.

\(\tau \varepsilon \) is the transmittance-absorptance product and is determined by the following equation:
\[
\tau \varepsilon = \frac{\tau (1 - \tau)}{1 - \tau \varepsilon}
\]
(10)
where \(\tau\) is the effective absorptance of PV/T plate, given by the weighted average of the absorptances of PV cell (\(\varepsilon_{pv}\)) and black TPT (\(\varepsilon_{tpt}\)), \(\varepsilon = \gamma \varepsilon_{pv} + (1 - \gamma)\varepsilon_{tpt}\); and \(\rho_p\) is the reflectance of the inner cover for diffuse radiation, which can be estimated by the formula \(\rho_p = 1 - \tau_{ad} - \tau_{sd}\) at a radiation incidence angle of 60° [29].

For the base panel, differential grids were divided, as shown in Fig. 4. Two types of grids are labeled, with one grid being connected to a heat pipe (heat pipe node), whereas the other is not (middle node). The heat-conduction equations in these two types of grids are different and are given by Eqs. (11a) and (11b), respectively.

The heat pipe node is expressed as:
\[
\rho_b c_b \frac{\partial T_b}{\partial t} = k_b \frac{\partial^2 T_b}{\partial x^2} + \frac{1}{\rho_b} \left[ (T_a - T_b)/R_{ba} + (T_{pv} - T_b)/R_{pv} \right] + G(\tau \varepsilon)_{pv} - E_{pv}
\]
(11a)
and the middle node is expressed as:
\[
\rho_b c_b \frac{\partial T_b}{\partial t} = k_b \frac{\partial^2 T_b}{\partial x^2} + \frac{1}{\rho_b} \left[ (T_a - T_b)/R_{ba} + (T_{pv} - T_b)/R_{pv} \right] + C(\tau \varepsilon)_{pv} - E_{pv}
\]
(11b)
$R_{ba}$ is the thermal resistance between the base panel and the ambient air, determined by the following formula:

$$R_{ba} = \delta_b/k_i + 1/h_a \quad (12)$$

$R_{pb}$ is the thermal resistance between the base panel and the heat pipe, expressed as:

$$R_{pb} = \delta_{pb}/(k_p \cdot A_{pb}) \quad (13)$$

where $A_{pb}$ and $\delta_{pb}$ are the contact area and thickness between the base panel and evaporator section of the heat pipe, respectively.

For the heat pipe, heat-balance equations were provided for the evaporator and the condenser sections. The transfer from the evaporator to the condenser section was calculated using a total thermal resistance, $R_{vcan}$. Given that the pressure drop resulting from the vapor flow along the axial length of the heat pipe was very small, the vapor space was assumed to operate at a constant saturation pressure. Therefore, the temperature gradient of the working fluid along the axial length of the heat pipe can be neglected. The value for $R_{vcan}$ can be derived based on the resistance of the following components: conduction resistance across the pipe wall and the wick of the heat pipe evaporator, thermal resistance associated with the condensing process, and conduction resistance across the pipe wall of the heat pipe condenser, expressed as:

$$R_{vcan} = R_{vap} + R_{vwick} + R_{con,i} + R_{con,p} \quad (14)$$

where

$$R_{vap} = \frac{\ln(D_{eva}/D_{eva,i})}{2\pi L_{eva} k_p} \quad (15)$$

$$R_{vwick} = \frac{\ln(D_{wick}/D_{wick,i})}{2\pi L_{wick} k_w} \quad (16)$$

An axial groove structure was designed as the wick of the heat-pipe evaporator, as shown in Fig. 5. The equivalent thermal conductivity of the wick can be calculated using the following formula [31]:

$$k_{wick} = \frac{w_g k_{w_g} d + w_k (0.185 w_g k_{w_g} + k_d)}{(w + d) (0.185 w_g k_{w_g} + k_d)} \quad (17)$$

where $w_g$ is the rib width of the groove, $w$ is the width of the groove, and $d$ is the depth of the groove.

The thermal resistance associated with the condensing process is defined as:

$$R_{con,i} = \frac{1}{\pi D_{con,i} h_{con,i}} \quad (18)$$

where $h_{con,i}$ is the condensing film coefficient and can be obtained by the following formula [21]:

$$h_{con,i} = \left[0.997 - 0.334 \cos \theta \right]^{0.108} \left[\frac{g \rho^2 k_w h_p}{h_{con,i} \Delta T_{con}}\right]^{0.25} \left[\frac{L_{con,i}}{D_{con,i}}\right]^{0.25} \left[\cos \theta \right]^{0.108}$$

and

$$R_{con,p} = \frac{\ln(D_{con,p}/D_{con,i})}{2\pi L_{con} k_p} \quad (19)$$

The heat-balance equation of the evaporator section is expressed as:

$$M_{p, eva} C_p \frac{\partial T_{p, eva}}{\partial t} = \frac{(T_{p, eva} - T_{eva,i})}{R_{eva} + (T_{eva} - T_{eva,i})}/R_{eva} + (T_{eva} - T_{eva,i})/R_{eva} \quad (20.a)$$

whereas that of the condenser section is expressed as:

$$M_{p, con} C_p \frac{\partial T_{p, con}}{\partial t} = \frac{(T_{p, con} - T_{eva})}{R_{eva} + (T_{eva} - T_{eva,i})}/R_{eva} + (T_{eva} - T_{eva,i})/R_{eva} \quad (20.b)$$

where $R_{vcan}$ is the thermal resistance between the heat-pipe condenser and water, which can be calculated using the following formula:

$$R_{vcan} = \frac{1}{A_{wa} h_w} \quad (22)$$

In the formula, $h_w$ is the convection heat transfer coefficient between the heat-pipe condenser and water, which can be obtained using the following formulas [32]:

$$h_w = \frac{Nu k_w}{D} \quad (23)$$

and

$$Nu = \frac{CRe^\theta Pr^{n}}{Pr_s^{1/4}} \quad (24)$$

when $Pr \leq 10$, $n = 3.7$, and when $Pr > 10$, $n = 3.6$. The values of $C$ and $m$ depend on the Reynolds number.

The differential grid partition for the water in the water box is shown in Fig. 6. The upwind scheme is used in the water differential equation. For grid $(j)$, the equation can be expressed as:

$$m_{wa} C_p \frac{\partial T_{wa,j}}{\partial t} + m_w C_w (T_{wa,j} - T_{wa,j-1}) = \frac{(T_a - T_{wa})}{R_{wa} + (T_{p, con} - T_{wa,j})}/R_{wa} \quad (25)$$

where $m_w$ is the mass of water in a single control volume; and $m_w$ is the mass flow rate of the water, that is, $m_w = \rho_w m_a A_i$; and $R_{wa}$ is the equivalent thermal resistance between water and ambient air.

For the water in the storage tank, the heat-balance equation is given by:

$$m_{w, tank} C_w \frac{\partial T_{w, i}}{\partial t} + n m_w C_w (T_{w, out} - T_{w, in}) = \frac{(T_{w, i} - T_{w, i-1})}{R_{w, tank} + n m_w C_w (T_{w, out} - T_{w, in})} \quad (26)$$

where $m_{w, tank}$ is the mass of water in the storage tank; $R_{w, tank}$ is the equivalent thermal resistance between the water and the ambient air; $T_{w, in}$ and $T_{w, out}$ are the inlet and outlet water temperatures of solar collectors, respectively; and $n$ is the number of solar collectors.

The useful heat gain of the system is expressed as:

$$Q_w = m_{w, tank} C_w (T_{w, i} - T_{w, i-1}) \quad (27)$$

where $T_{w, i}$ and $T_{w, i-1}$ are respectively the initial and final water temperatures of the storage tank.

The model of the HP-PV/T system given above has been validated by the experimental results in a previous work [27]. The deviations between the predicted and measured values, such as that for the
daily average electrical and heat gains, are all within ±2.4%. Based on the validated system model, a simulation will be performed to predict the annual electrical and thermal behavior of the HP-PV/T system in the present paper.

4. Simulation studies

Simulation is performed to predict the annual electrical and thermal behavior of two HP-PV/T systems (systems with and without auxiliary heating equipment) used in three typical climate areas of China, namely Hong Kong (114.17°E, 22.32°N), Lhasa (91.13°E, 29.67°N), and Beijing (116.28°E, 39.93°N). Hong Kong stands south of China where the available solar radiation is low and represents the area with hot summers and mild winters. Lhasa stands west of China, where the available solar radiation is highest, and represents the area with mild summers and cold winters. Beijing stands north of China, where the available solar radiation is highest, representing the area with hot summers and cold winters. The average ambient temperatures in daytime and the average solar irradiations of the three areas are shown in Fig. 7. The weather data of the three areas are provided by EnergyPlus [33]. The weather types are City-UHK for Hong Kong and CSWD for Lhasa and Beijing. The weather of Lhasa and Beijing is cold in winter, so ammonia–aluminum heat pipes were used in the HP-PV/T system to avoid freezing. However, in Hong Kong, the weather is mild in winter and water is not freezing, allowing for the use of water–copper heat pipes. The specifications of the two types of heat pipes are given in Table 2.

The specifications of the two HP-PV/T systems considered in the simulation are given as follows:

- HP-PV/T system with auxiliary heating equipment: when the daily solar energy is insufficient to cover the daily hot water load (in our simulation, a water temperature higher than 45 °C can be considered available), auxiliary energy is required to cover the hot water load.
- HP-PV/T system without auxiliary heating equipment: when a daily solar energy is insufficient to heat the water to reach the available temperature of 45 °C, the water is sequentially heated in the following day until the available temperature is reached.

In the present simulation, the circulation pump of the HP-PV/T system starts up at the solar time of 8:00 (start time) and shuts down at the solar time of 16:30 (end time). Data used in the simulation are given as follows:

1. Each HP-PV/T system includes four solar collectors with the total collector area of 3.868 m² and total PV cell area of 2.22 m².
2. The collector slope is equal to latitude +5°.
3. Storage tanks with storage capacities of 250, 300, 350, and 400 L are chosen as the water tank for each HP-PV/T system.

<table>
<thead>
<tr>
<th>Type</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water–copper heat pipe</td>
<td>Pipe wall: copper; dimension: evaporator section, φ10 × 1 × 1300 mm, condenser section, φ24 × 1 × 90 mm; working fluid: water; normal boiling point of the working fluid (1 atm): 99.97 °C.</td>
</tr>
<tr>
<td>Ammonia–aluminum heat pipe</td>
<td>Pipe wall: aluminum; dimension: evaporator section, φ10 × 1 × 1300 mm, condenser section, φ24 × 1 × 90 mm; working fluid: ammonia; normal boiling point of the working fluid (1 atm): −33.33 °C.</td>
</tr>
</tbody>
</table>
4. The mass flow rate of the circulating water is 0.1 kg/s, flowing through each collector at 0.025 kg/s.

5. The initial water temperature of the storage tank is set equal to \([T_a, 5 \degree C]\), where \(T_a\) is the ambient temperature at the local solar time of 8:00, and the double bracket, \([[]]\), indicates that the greater of the two values is to be used.

To predict the annual performance of the HP-PV/T system, a numerical simulation program written in C++ language has been developed. Figs. 8a and b, respectively show the flow charts of the computation process for the HP-PV/T systems with and without auxiliary heating equipment. Fig. 9 shows the flow chart of the computation process of the collector model. In the present simulation, the HP-PV/T systems operate to collect the solar energy only from 8:00 to 16:30 (solar time). For the system with auxiliary heating equipment, if the water is heated to the available temperature at the end time of 16:30, it will be supplied to the user. Otherwise, the water will be sequentially heated to the available temperature using an auxiliary heater before being supplied to the user. The total daily auxiliary energy required for the hot water is calculated by

\[
Q_{aux, total} = \begin{cases} 
M_w \cdot \Delta T_w \cdot \left( T_{ava} - T_{w,in} \right), & T_{w,15} < T_{ava} \\
0, & T_{w,15} \geq T_{ava}
\end{cases}
\]

where \(T_{ava}\) is the available water temperature, \(T_{ava} = 45 \degree C\).

For the system without auxiliary heating equipment, if the water is heated to the available temperature at the end time of

\[
Q_{aux} = \begin{cases} 
M_w \cdot \Delta T_w \cdot \left( T_{ava} - T_{w,in} \right), & T_{w,15} < T_{ava} \\
0, & T_{w,15} \geq T_{ava}
\end{cases}
\]

where \(T_{ava}\) is the available water temperature, \(T_{ava} = 45 \degree C\).
16:30, it will be supplied to the user. Otherwise, the water will be sequentially heated in the following day until it reaches the available temperature. The heat loss from the storage tank during the night is considered in this simulation.

5. Results and analysis

5.1. HP-PV/T system with auxiliary heating equipment

The monthly thermal and electrical performances of the HP-PV/T system with auxiliary heating equipment are shown in Figs. 10 and 11, respectively. The useful thermal and electrical energy obtained by the system primarily depend on the solar radiation in the location. In Lhasa, where the available solar radiation is high (Fig. 7), a large amount of thermal and electrical energy can be obtained with the thermal energy varying between 6720 kJ/(m² d) and 11,610 kJ/(m² d) and the electrical energy between 1030 kJ/(m² d) and 1630 kJ/(m² d). In Hong Kong, where the available solar radiation is low (Fig. 7), the thermal and electrical energy obtained by the system are only 3650 kJ/(m² d) to 6380 kJ/(m² d) and 600 kJ/(m² d) to 880 kJ/(m² d), respectively. Fig. 10 also gives the thermal performance of the HP-PV/T system with different
Fig. 11. Monthly electrical performance of the HP-PV/T system with auxiliary heating equipment.

Fig. 12. Solar contribution ratio of the HP-PV/T system with auxiliary heating equipment.
water storage capacities (250, 300, 350, and 400 L). The system with small water storage capacity obtains less thermal energy than that with large water storage capacity, as shown in Table 4, because the system with small water storage capacity experiences a faster rise of water temperature in the tank, thereby increasing the temperature of the collector and the heat loss of the system. For the electrical energy output of the HP-PV/T system, the effect of water storage capacity is inconspicuous (Fig. 11).

As shown in Fig. 10, for Hong Kong, Lhasa, and Beijing, auxiliary energy is still required to cover the hot-water load. In Hong Kong, the available solar radiation is low, so some auxiliary energy is required in the period from November to the following April. The ambient temperature is higher from May to October (Fig. 7), requiring little auxiliary energy. In Lhasa, although the available solar radiation is high, and a substantial amount of thermal energy can be obtained all year round, the ambient temperature is low, still requiring auxiliary energy. In Beijing, the available solar radiation is not very high, and the ambient temperature is low in the period from October to the following April, requiring a large amount of auxiliary energy to cover the hot-water load in this period.

Fig. 12 shows the solar contribution of the HP-PV/T system. The solar contribution of the HP-PV/T system depends on the location and the water storage capacity of the system (or the hot-water load per unit collecting area, \( M_{w} / A_{c} \)). For Lhasa, the solar contribution of the HP-PV/T system exceeds 70% when the total water volume is less than 250 L. The solar contribution of the system from July to October in Hong Kong also exceeds 70%. In Beijing, the solar contribution from May to August can also exceed 60%.

Table 3 gives the number of days that the water of the HP-PV/T system can be heated to higher than 45 °C using only solar energy. In Hong Kong, when the hot-water load per unit collecting area \( (M_{w}/A_{c}) \) is lower than 90.3 kg/m², the water can be heated with a temperature higher than 45 °C in most days from May to October. This finding indicates that, in most days, solar energy can cover the total hot water load without any auxiliary thermal energy. In Lhasa, the available solar radiation is high but the ambient and initial water temperatures are low, so there are only few days during which the water can be heated higher than 45 °C. When \( M_{w}/A_{c} \) is equal to 64.5 kg/m², the water can be heated to more than 45 °C in Beijing for over 12 days per month; however, water cannot be heated to more than 45 °C from November to the following February.

Table 4 gives annual results of the HP-PV/T system with auxiliary heating equipment. For Hong Kong, the annual available solar radiation is 4408.58 MJ/m², and the HP-PV/T system can obtain 1665.05–1872.77 MJ/m² thermal energy and give 261.32–264.98 MJ/m² electrical energy output, when the \( M_{w}/A_{c} \) value varies between 64.5 kg/m² and 103.2 kg/m². For Lhasa, the results are: annual available solar radiation is 2939.67–3328.25 MJ/m², and electrical energy output is 462.14–471.46 MJ/m². For Beijing, the results are: annual available solar radiation is 2530.28 MJ/m², useful thermal energy obtained by the system is 2111.07–2352.95 MJ/m², and electrical energy output is 322.84–3211.15 MJ/m².

### Table 3

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<th>( V_{w} ) (L)</th>
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<th>Mar</th>
<th>Apr</th>
<th>May</th>
<th>June</th>
<th>July</th>
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\( M_{w}/A_{c} \) is the hot-water load per unit collecting area and \( V_{w} \) is the water volume of the water tank.

### Table 4

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<tr>
<th>Location</th>
<th>( V_{w} ) (L)</th>
<th>( M_{w}/A_{c} ) (kg/m²)</th>
<th>( Q_{sol} ) (MJ/m²)</th>
<th>( Q_{e} ) (MJ/m²)</th>
<th>( Q_{aux} ) (MJ/m²)</th>
<th>( f_{s} )</th>
<th>( E ) (MJ/m²)</th>
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</table>

\( Q_{sol} \) is the total solar irradiation energy per unit collecting area; \( Q_{e} \) is the useful thermal energy per unit collecting area; \( Q_{aux} \) is auxiliary thermal energy required to cover the hot-water load per unit collecting area; \( f_{s} \) is the solar fraction, denoting the percentage of hot-water load covered by solar energy; and \( E \) is the electrical energy per unit collecting area (the PV cell covering factor of the collector is 0.674).
Table 4 shows that although increasing the value of \(M_w/A_c\) can help obtain more useful thermal energy and electrical energy, the temperature-rise of the water will decrease, and auxiliary thermal energy required to cover the hot-water load will increase. For example, in Lhasa, when the value of \(M_w/A_c\) is equal to 64.5 kg/m², only 714.21 MJ/m² of auxiliary thermal energy is required, and the solar contribution of the system can reach 0.805. However, when the value of \(M_w/A_c\) reaches 103.2 kg/m², a large auxiliary thermal energy of 2463.49 MJ/m² will be required, and the solar contribution of the system is only 0.575.

5.2. HP-PV/T system without auxiliary heating equipment

Fig. 13 shows the monthly useful thermal and electrical energy obtained by the HP-PV/T system without auxiliary heating equipment in different areas. The largest thermal and electrical energy obtained by the HP-PV/T system in Hong Kong are both from July to November. In Lhasa, the largest electrical energy output appears in the period of October to February of the following year. However, given the effect of low ambient temperature, the largest thermal energy output only appears in October and November.
Beijing has a uniform monthly thermal energy output, except in January and December.

Table 5 gives the number of days that the water of the HP-PV/T system without auxiliary heating equipment can be heated to more than 45 °C using solar energy. In Lhasa, when the hot-water load per unit collecting area ($M_w/A_e$) is lower than 90 kg/m², more than 10 days of the hot water load per month can be covered by the solar energy, satisfying the domestic hot water demand. Although the available solar radiation is lower in Hong Kong and Beijing, when $M_w/A_e$ is lower than 90 kg/m², there are also over 9 days per month that the hot water load is totally covered by solar energy from April to October.

Table 6 gives the annual results of the HP-PV/T system without auxiliary heating equipment. When $M_w/A_e$ is equal to 64.5 kg/m², in Hong Kong, there are 172 days a year that the hot water can be heated at more than 45 °C using solar energy. In Lhasa and Beijing, the results are 178 days and 158 days, respectively. However, with the $M_w/A_e$ increasing, the days will decrease. Compared with the system with auxiliary heating equipment (Table 4), useful thermal and electrical energy obtained by the system without auxiliary heating equipment are lower. The reduction in the useful thermal energy varies between 23% and 38%, and that of the electrical energy varies between 1.5% and 4.3% because in the system without auxiliary heating equipment, the water is heated only by the solar energy. When the water temperature does not reach the available temperature (45 °C), it will be sequentially heated in the following day until it reaches the available temperature, and the heat loss from the storage tank will occur in the nighttime.

### Table 5
Number of days in each month the water can be heated at more than 45 °C using only solar energy (system without auxiliary heating equipment).

<table>
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<th>February</th>
<th>March</th>
<th>April</th>
<th>May</th>
<th>June</th>
<th>July</th>
<th>August</th>
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### Table 6
Annual performances of the HP-PV/T system without auxiliary heating equipment.

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<th>$M_w/A_e$ (kg/m²)</th>
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<th>$Q_o$ (MJ/m²)</th>
<th>$E$ (MJ/m²)</th>
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$N$ is the total number of days that the water can be heated to an available temperature (higher than 45 °C) for the domestic hot water only by the solar energy for the entire year.

### Table 7
Analysis of energy and environment effects of the HP-PV/T system.

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<td>–</td>
<td>283.71</td>
</tr>
<tr>
<td>Running cost, $ per annum</td>
<td>143.67</td>
<td>289.46</td>
<td>39.26</td>
</tr>
<tr>
<td>Lhasa</td>
<td>6187.59</td>
<td>5445.08</td>
<td>2246.27</td>
</tr>
<tr>
<td>CO₂ emission, tons per annum</td>
<td>1.176</td>
<td>3.866</td>
<td>0.427</td>
</tr>
<tr>
<td>Electricity output, kWh per annum</td>
<td>–</td>
<td>–</td>
<td>504.03</td>
</tr>
<tr>
<td>Running cost, $ per annum</td>
<td>143.67</td>
<td>289.46</td>
<td>39.26</td>
</tr>
<tr>
<td>Beijing</td>
<td>5379.03</td>
<td>4733.55</td>
<td>2617.24</td>
</tr>
<tr>
<td>CO₂ emission, tons per annum</td>
<td>1.022</td>
<td>3.361</td>
<td>0.497</td>
</tr>
<tr>
<td>Electricity output, kWh per annum</td>
<td>–</td>
<td>–</td>
<td>351.15</td>
</tr>
<tr>
<td>Running cost, $ per annum</td>
<td>204.40</td>
<td>411.82</td>
<td>68.91</td>
</tr>
</tbody>
</table>
cause of the higher water temperature in the following day, the collector temperature (temperatures of the PV cell and absorb plate) is higher, resulting in a larger heat loss from the collector and the storage tank. Moreover, if the weather is bad in the following day, the system cannot obtain any energy, and only the heat loss exists.

5.3. Analysis of the energy and environmental effects

A simple analysis of the primary energy savings and carbon reduction of the HP-PV/T system is conducted by comparing the HP-PV/T system with the conventional systems (electric and gas water heaters). The results are given in Table 7. The investigation assumes that the daily hot water demand is 350 L, and the energy efficiencies of the electric and gas water heater are 100% and 88%, respectively. CO₂ intensities of the grid (China) and gas are assumed to be 0.71 kg CO₂/kWh and 0.19 kg CO₂/kWh, respectively. The cost of electricity from the national grid is assumed to be $0.087/kWh, and the gas price is $0.038/kWh.

The analysis indicates that, for Hong Kong, the HP-PV/T system will save primary energy of 2098.12 kWh per annum compared with the gas water heater and will also have a PV electricity output of 283.71 kWh per annum (equal to a reduction of CO₂ emission of 0.201 tons), resulting in a reduction of total CO₂ emission of approximately 0.599 tons. For Lhasa, the primary energy saving of the HP-PV/T system is 3941.32 kWh per annum when compared with the gas water heater, and electricity output is 504.03 kWh per annum. The CO₂ reduction can reach 1.106 tons. For Beijing, the annual primary energy savings, electricity output, and CO₂ reduction are 2761.79 kWh, 351.15 kWh, and 0.774 tons, respectively. The initial cost of the HP-PV/T system proposed in the current paper is approximately $2834.6, which is higher than those of the gas and electric water heaters (approximately $315.0). However, the annual running cost of the HP-PV/T system is only approximately $68.91 when used in Beijing, which is significantly lower than those of the gas ($204.40) and electric water heaters ($411.82).

6. Conclusion

HP-PV/T systems can be used in cold regions without being frozen. A detailed simulation model for the HP-PV/T system was presented. Using this model, the annual electrical and thermal behavior of two HP-PV/T systems, namely, systems with and without auxiliary heating equipment, were predicted and analyzed when used in three typical climate areas of China, namely, Hong Kong, Lhasa, and Beijing. The following are the results:

1. The useful thermal and electrical energy obtained by HP-PV/T systems primarily depend on the available solar radiation in the area. For the system with auxiliary heating equipment, when used in Hong Kong, Lhasa, and Beijing, the annual thermal energy are 1665.05–1872.22 MJ/m², 2939.67–3328.25 MJ/m², and 2111.07–2352.95 MJ/m², respectively; the annual electrical energy produced are 263.32–264.98 MJ/m², 462.14–466.1 MJ/m², and 322.84–328.15 MJ/m², respectively. For the system without auxiliary heating equipment, the annual thermal and electrical energy are lower than those of the system with auxiliary heating equipment.

2. For the HP-PV/T system with auxiliary heating equipment, the solar thermal contribution mainly depends on the available solar radiation and the hot-water load per unit collecting area (Mₒ/Aₒ). When Mₒ/Aₒ is equal to 64.5 kg/m², the annual solar thermal contribution of the HP-PV/T system in Hong Kong, Lhasa, and Beijing are 68.5%, 80.5%, and 64.7%, respectively. However, with Mₒ/Aₒ increasing, the solar thermal contribution decreases.

3. For the HP-PV/T system without auxiliary heating equipment, when Mₒ/Aₒ is equal to 64.5 kg/m², in Hong Kong, there are 172 days a year that the hot water can be heated to more than 45 °C using solar energy. In Lhasa and Beijing, the results are 178 days and 158 days, respectively.

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References