Research Article

Performance Evaluation of the Air-Type Photovoltaic-Thermal Collector Combined with Transverse Triangle Obstacle

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An air-type photovoltaic-thermal collector (PVTC) is a solar collector that generates heated air and electricity from the incident solar energy in a single system. Adding a heat transfer device to this PVTC improves its performance but also increases the power consumption of the fan. Nevertheless, the additional energy consumption has not yet been adequately considered in previous related studies. The current study investigates the performance of a novel air-type PVTC combined with a transverse triangle obstacle (TTO) under various geometric conditions of TTO, considering the energy consumption of the PVTC. The TTO, suggested by the authors, improves the thermal energy recovery capability in PVTC by increasing the air velocity and promoting a mix of heated and relatively cold air in a fluid field, different from the traditional heat transfer devices used in previous PVTCs. The relative height, length, and pitch of the TTO are selected as geometric conditions. The effect of these geometric conditions on performance improvement and the increase in power consumption of the PVTC are discussed in detail. The result showed that the relative height has a dominant effect on the energy generation and consumption of the PVTC, while the other factors have an inconsiderable effect. In addition, this research evaluates the daily performance of the PVTC with and without TTO. The relative height of 0.45 results in the highest daily performance. Compared to PVTC without TTO, the PVTC using a TTO with a relative height of 0.45 provides an increase of 22.93% and 2.79% in the thermal output and electrical output, respectively, with daily average thermal and electrical efficiencies of 39.04% and 17.01%, respectively. The thermal equivalent net energy output, considering both the energy generation and consumption of the PVTC, could also be improved by 7.8%.

1. Introduction

A photovoltaic-thermal collector (PVTC) is one of the most promising renewable energy systems that can simultaneously generate heat and electricity from solar energy in a single system. This system can be used to heat space, dry crops, and supply hot water. There are various types of PVTC, including those that use air [1, 2], water [3, 4], and nanofluids [5–7], phase change materials [8, 9], and heat pipes [10, 11]. The air-type PVTC is a type of system that uses air to recover heat from the photovoltaic (PV) panel. This system is advantageous in terms of ease of construction and low maintenance costs caused by its simple design in comparison to other types of PVTCs [12]. However, the disadvantage of the air-type PVTC is the low thermal efficiency resulting from the low thermal conductivity and density of air. Hence, a lot of researchers have suggested various methods to enhance the thermal recovery capability of air-type PVTCs and studied their effectiveness.

One of the most general approaches to achieving better thermal performance of a PVTC is to expand the heat transfer area in the PVTC. Kumar and Rosen [13] investigated a double-pass PVTC with fin and compared it with a PVTC without fin. They reported that the addition of a fin could reduce the cell temperature by extending the heat transfer area. Fan et al. [14] established a dynamic model for PVTC combined with a solar air heater and fins. The lower height of the fin and an increase in the number of fins resulted in
higher thermal and electrical efficiencies. Ahmed and Mohammed [15] confirmed the influence of the porous medium installed in the lower air channel of the double-pass PVTC and confirmed that the total efficiency improved by 3%. Selimefendigil et al. [16] developed a dynamic model of the PV panel equipped with a porous fin using artificial neural networks. This research presented that the installation of the porous fin reduced the PV surface temperature and enhanced electrical efficiency. Fudholi et al. [17] analyzed the energy and exergy of a PVTC coupled with a V-groove heat exchanger. Total energy and exergy efficiencies varied from 38.8% to 81.8% and 12.44% to 13.26%, respectively. Singh et al. [18] compared the PVTC with a V-groove absorber to the PVTC with curved shape groove absorber. Their research found that the PVTC using a curved shape groove absorber had a better energy efficiency than those of the V-groove absorber. The use of cylindrical fins inside the air duct of a PVTC was proposed by Ozakın and Kaya [19, 20]. The results indicated that the configuration and material of the fin had a dominant effect on the PVTC’s performance. Franklin and Chandrasekar [21] installed longitudinal staves in the trailing portion of the air duct of the PVTC and analyzed its performance experimentally. This modification increases the maximum outlet air temperature from 30.7°C to 34°C. Tahmasbi et al. [22] analyzed the effect of porous metal foam on the electrical and thermal efficiencies of the PVTC. This study reported that the addition of porous metal foam could improve the electrical and thermal efficiencies by 3 to 4% and 10 to 40%, respectively. A vertical-type PVTC dryer with installed fins was developed and investigated by Çiftçi et al. [23]. They presented that the addition of fins and an increase in the air flow rate can enhance thermal efficiency and reduce drying time. Shrivastava et al. [24] investigated the PVTC with five different fin configurations and reported that longitudinal fins combined with inclined baffles yielded the most remarkable improvement in the PVTC’s thermal efficiency. Zhao et al. [25] evaluated the PVTC combined with three different fin numbers and indicated that both the electrical and thermal efficiencies were improved as the number of fins increased.

The other method is changing the PVTC’s configuration. Four different configurations of the PVTC, which have air flowing over the PV panel (Type I), under the PV panel (Type II), on both sides of the PV panel in the same direction (Type III), and on both sides of the PV panel in a double pass (Type IV), were investigated by Hegazy [26], and the Type III performed better than the other types. Amori and Abd-Alraheem [27] compared three different PVTCs. PVTCs with single-flow double pass, double-flow single pass, and single-flow single pass were experimentally compared, and the PVTC with double-flow single pass exhibited the highest energy efficiency. The annual energy outputs of four different configurations, PV panel (Model I), unglazed single-pass single-flow PVTC (Model II), glazed single-pass single-flow PVTC (Model III), and glazed double-pass PVTC (Model IV), were evaluated by Slimani et al. [28]. This research concluded that Model IV had the best annual energy output under Algiers climatic conditions. S Tiwari and GN Tiwari [29] suggested the PVTC partially covered by PV panels for a greenhouse solar dryer and evaluated its performance with the number of PVTCs under various air flow rates. They reported that the proposed system could increase the outlet air temperature up to 122.78°C. A comparative study on the four different PVTC configurations for building integration, which are (1) glass-glass PV with duct (Type I), (2) glass-glass PV without duct (Type II), (3) glass-Tedlar PV with duct (Type III), and (4) glass-Tedlar PV without duct (Type IV), was conducted by Tomar et al. [30]. Type I and Type II configurations produce the highest electrical and thermal outputs, respectively. Koşan and Aktaş [31] evaluated the performance of PVTC with double pass and PV panel combined with a heat pump. This study showed that the PVTC with double pass exhibited 1.31% higher electrical efficiency than the PV panel. Shahsavari and Acri [32] compared PVTC with and without glass combined with a sensible rotary heat exchanger and presented that the PVTC with glass could produce higher thermal energy and less electricity than PVTC without glass. Gupta et al. [33] suggested a PVTC dryer fabricated using a semitransparent PV panel and evaluated its performance. They reported that the proposed PVTC showed an energy efficiency of 69.27% for the forced convection drying mode and 43.58% for the natural convection drying mode. Vajedi et al. [34] evaluated a PVTC having an air channel with a decreasing hydraulic diameter and presented that the decreasing hydraulic diameter effectively increases the thermal and electrical efficiencies of a PVTC.

However, in the aforementioned literature, it was found that most studies on the thermal performance improvement of the PVTC focused on changing its configuration or using a fin, porous medium, and heat exchanger to extend the heat transfer area.

Thus, in the previous study, the authors proposed an air-type PVTC employing a novel heat transfer device, which improves the thermal performance of the PVTC in a different manner than previously used devices [35, 36]. In this PVTC, a transverse triangle obstacle (TTO) is located above the base surface of an air duct. This modification accelerates the air velocity in a fluid field and promotes the mixing of hot air near the heated wall and cold air at the bottom side of an air duct. Therefore, it can enhance the convective heat transfer coefficient, unlike other methods used in the previous literature for extending the heat transfer area. In addition, due to the simplicity of its design, it has merits in manufacturing and cost. However, the previous studies performed by the authors have been conducted with fixed geometric conditions of the TTO. As the geometric conditions of the TTO affect the thermal and electrical performance of the PVTC, it will be interesting to determine how the geometric conditions of the proposed heat transfer device influence these characteristics.

In addition, most previous studies have only reported the thermal and electrical efficiencies of the PVTC employing a heat transfer device, and very few studies consider an increase in power consumption due to the addition of the device. When the heat transfer device is inserted into the PVTC, the pressure drop in the air duct increases, increasing the power consumption of the fan that is used to operate an air-type PVTC. For this reason, it is necessary to consider…
both the increase in energy generation and the consumption of the PVTC to confirm the feasibility of the proposed heat transfer device. Moreover, the net energy output considering both the increase in performance and the power consumption of the PVTC should be compared with those of the PVTC without a heat transfer device. Nevertheless, there is very limited research in this regard.

This study investigated the energy generation and consumption of the PVTC combined with TTO under various geometric conditions. The relative height, length, and pitch of the TTO were chosen as the geometric conditions, and the effect of these geometric conditions was discussed in detail. In addition, this study evaluated the thermal equivalent net energy efficiency (TENEE) to confirm the feasibility of the proposed devices by simultaneously considering both the energy generation and the consumption of the PVTC. Finally, the daily performance of the PVTC with TTO was estimated and compared to the PVTC without TTO to determine how much more energy could be generated by using the TTO.

2. System Description

The PVTC with TTO was experimentally tested in a previous research by the authors [35]. Figure 1 shows the schematic of the PVTC system. As shown in the figure, the ambient air is drawn into the PVTC by a fan, and the heated air is blown into the space where it is needed. The PVTC is composed of a commercial PV panel, an air duct, and a TTO. The side and base surfaces of the air duct were insulated with insulators, and the TTO was located above the bottom insulator. The schematic view of the PVTC with TTO is shown in Figure 2. Table 1 summarizes the specification of the PVTC with TTO and the thermophysical properties of the materials.

3. Mathematical Modeling

Figure 3 shows the temperature of each component of the PVTC with TTO and the corresponding thermal resistance network. \( T_g \) indicates the top surface temperature of the glass. The other temperature is the average temperature of each component. The energy balance equations were derived from the thermal resistance network and used to develop the mathematical model. To simplify the mathematical model, the following assumptions were used [14, 28, 37].

(i) Steady-state condition was assumed for PVTC
(ii) Heat transfer takes place in one dimension perpendicularly to the air flow direction
(iii) The PV cell temperature is the same over the width and length of the PV panel
(iv) Heat transfer through ethylene vinyl acetate (EVA) is neglected
(v) The transmissivity of EVA was assumed to be 1

\[(vi) \text{ The convective heat transfer coefficient between the Tedlar and fluid is constant along the air flow channel}\]

\[(vii) \text{ The physical properties of the materials are constant}\]

\[(viii) \text{ Air leakage did not take place in the PVTC}\]

\[(ix) \text{ Heat losses at the side and base surfaces of the PVTC are ignored}\]

3.1. Energy Balance Equation for Each Component. This section describes the energy balance equation for each component. The energy balance equations derived from the thermal resistance network are as follows:

For the glass surface,

\[
\frac{T_g - T_{gs}}{R_{k,g}} = \frac{T_{gs} - T_{sky}}{R_{r,sky}} + \frac{T_{gs} - T_{amb}}{R_{r,amb}}. \tag{1}
\]

For glass,

\[
G \alpha_g + \frac{T_c - T_g}{R_{k,c} + R_{k,g}} = \frac{T_g - T_{gs}}{R_{k,g}}. \tag{2}
\]

For the PV cell,

\[
G \tau_g \alpha_c \lambda = \frac{T_c - T_g}{R_{k,c} + R_{k,g}} + \frac{T_c - T_{ted}}{R_{k,c} + R_{k,ted}} + \dot{w}_{pv}, \tag{3}
\]

where \( \dot{w}_{pv} \) is the electrical power generated by the PVTC and expressed as [38]

\[
\dot{w}_{pv} = G \eta_{el}. \tag{4}
\]

In Eq. (4), \( \eta_{el} \) indicates the electrical efficiency of the PV cell and is given as [38, 39]

\[
\eta_{el} = \eta_{el}[1 + \beta(T_c - 25)]. \tag{5}
\]
For Tedlar,

$$\text{Gr}_g \alpha_{\text{ted}} (1 - \lambda) + \frac{T_c - T_{\text{ted}}}{R_{k,c} + R_{k,\text{ted}}} = \frac{T_{\text{ted}} - T_f}{R_{k,\text{ted}} + R_{v,f}}. \tag{6}$$

For flowing air,

$$\frac{m_f C_{p,f} W_{ad}}{W_{ad} dx} dT = \frac{T_{\text{ted}} - T_f}{R_{k,\text{ted}} + R_{v,f}}. \tag{7}$$

By solving the above differential equation and using the boundary conditions, $T_f(0) = T_{f,\text{in}}$, Eq. (7) can be written as follows:

$$T_f(x) = T_{\text{ted}} - (T_{\text{ted}} - T_{f,\text{in}}) \exp \left[ - \frac{W_{ad}}{m_f C_{p,f} (R_{k,\text{ted}} + R_{v,f})} x \right]. \tag{8}$$

The outlet air temperature can be obtained by substituting the air duct length ($L_{ad}$) in $x$. Therefore, the heat gain of the PVTC can be written as follows:

$$\dot{q}_f = \frac{m_f C_{p,f}}{W_{ad} L_{ad}} (T_{f,\text{out}} - T_{f,\text{in}}) = \frac{m_f C_{p,f}}{W_{ad} L_{ad}} (T_{\text{ted}} - T_{f,\text{in}}) \cdot \left\{ 1 - \exp \left[ - \frac{W_{ad}}{m_f C_{p,f} (R_{k,\text{ted}} + R_{v,f})} L_{ad} \right] \right\}. \tag{9}$$

Using Eq. (9), Eq. (7) can be rewritten as follows:

$$\frac{m_f C_{p,f}}{W_{ad} L_{ad}} \left\{ 1 - \exp \left[ - \frac{W_{ad}}{m_f C_{p,f} (R_{k,\text{ted}} + R_{v,f})} L_{ad} \right] \right\} (T_{\text{ted}} - T_{f,\text{in}}) = \frac{T_{\text{ted}} - T_f}{R_{k,\text{ted}} + R_{v,f}}. \tag{10}$$

The different forms of these equations can also be found in previous research performed by the authors [36].

3.2. Thermal Resistance. Thermal resistances used in the energy balance equations are presented in this section. The convection resistance between the top surface of the glass and the ambient is defined as follows [40, 41]:

$$R_{v,\text{amb}} = \frac{1}{3 V_w + 2.8}. \tag{11}$$

The radiation resistance between the top surface of the glass and the sky is determined using the following relation [38, 42]:

$$R_{r,\text{sky}} = \frac{1}{\varepsilon_g \sigma (T_g^2 + T_{\text{sky}}^2) (T_g + T_{\text{sky}})}. \tag{12}$$
### Table 1: Specification of the PVTC with TTO and the thermophysical properties of the materials.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>PV panel</td>
<td></td>
</tr>
<tr>
<td>Model</td>
<td>LG360S2W-5K</td>
</tr>
<tr>
<td>Area (m²), $A_{pvt}$</td>
<td>2.027</td>
</tr>
<tr>
<td>Standard electrical efficiency (-), $\eta_{el}$</td>
<td>0.1737</td>
</tr>
<tr>
<td>Temperature coefficient (-/°C), $\beta$</td>
<td>-0.0041</td>
</tr>
<tr>
<td>Ratio of cell area to total area (-), $\lambda$</td>
<td>0.9288</td>
</tr>
<tr>
<td>Air duct</td>
<td></td>
</tr>
<tr>
<td>Height (m), $H_{ad}$</td>
<td>0.082</td>
</tr>
<tr>
<td>Width (m), $W_{ad}$</td>
<td>1</td>
</tr>
<tr>
<td>Length (m), $L_{ad}$</td>
<td>2.027</td>
</tr>
<tr>
<td>TTO</td>
<td></td>
</tr>
<tr>
<td>Height (m), $e_{tto}$</td>
<td>0.037</td>
</tr>
<tr>
<td>Length (m), $l_{tto}$</td>
<td>0.097</td>
</tr>
<tr>
<td>Pitch (m), $p_{tto}$</td>
<td>0.01265</td>
</tr>
<tr>
<td>Relative height (-), $e_{i}/H_{ad}$</td>
<td>0.4512</td>
</tr>
<tr>
<td>Relative pitch (-), $p_{i}/e_{tto}$</td>
<td>3.4189</td>
</tr>
<tr>
<td>Relative length (-), $l_{i}/e_{tto}$</td>
<td>2.6216</td>
</tr>
<tr>
<td>Air flow rate (kg/s), $m$</td>
<td>0.1123</td>
</tr>
<tr>
<td>Thickness, glass (m), $\delta_g$</td>
<td>0.003</td>
</tr>
<tr>
<td>Thickness, PV cell (m), $\delta_c$</td>
<td>0.0003</td>
</tr>
<tr>
<td>Thickness, Tedlar (m), $\delta_{ted}$</td>
<td>0.0005</td>
</tr>
<tr>
<td>Absorptivity, glass (-), $\alpha_g$</td>
<td>0.04</td>
</tr>
<tr>
<td>Thermal conductivity, glass (W/mK), $k_g$</td>
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</tr>
<tr>
<td>Emissivity, glass (-), $\varepsilon_g$</td>
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</tr>
<tr>
<td>Transmissivity, glass (-), $\tau_g$</td>
<td>0.95</td>
</tr>
<tr>
<td>Absorptivity, cell (-), $\alpha_c$</td>
<td>0.9</td>
</tr>
<tr>
<td>Thermal conductivity, cell (W/mK), $k_c$</td>
<td>0.036</td>
</tr>
<tr>
<td>Absorptivity, Tedlar (-), $\alpha_{ted}$</td>
<td>0.8</td>
</tr>
<tr>
<td>Thermal conductivity, Tedlar (W/mK), $k_{ted}$</td>
<td>0.033</td>
</tr>
</tbody>
</table>

The conduction resistance of the glass is given by

$$R_{k,g} = \frac{\delta_g}{2k_g}. \tag{14}$$

The conduction resistance of the Tedlar is given by

$$R_{k,ted} = \frac{\delta_{ted}}{2k_{ted}}. \tag{16}$$

The convection resistance between the Tedlar and flowing air can be attained using the following equation:

$$R_{v,fl} = \frac{1}{h_f} = \frac{D_h}{Nu_j k_f}, \tag{17}$$

where $Nu_j$ is the Nusselt number. The Nusselt number for the PVTC with TTO is given by Ref. [43]:

$$Nu = 0.2899 \, Re^{0.6828} \left( \frac{e_{tto}}{H_{ad}} \right)^{1.6939} \left( \frac{p_{tto}}{e_{tto}} \right)^{0.0563} \frac{H_{ad}^{0.0158}}{e_{tto}} \left( \frac{H_{ad}}{e_{tto}} \right)^{-0.0221} \exp \left[ 0.5604 \left( \ln \left( \frac{e_{tto}}{H_{ad}} \right)^2 \right)^2 \right] \exp \left[ -0.0122 \left( \ln \left( \frac{p_{tto}}{e_{tto}} \right)^2 \right)^2 \right] \exp \left[ -0.0159 \left( \ln \left( \frac{H_{ad}}{e_{tto}} \right)^2 \right)^2 \right]. \tag{18}$$

For the PVTC without TTO, the Nusselt number is given as [14]

$$Nu = 0.0158 \, Re^{0.8}. \tag{19}$$

The following empirical correlations were utilized to determine the specific heat, thermal conductivity, density, and dynamic viscosity of air for the calculation of the aforementioned equations [17, 37, 42]:

$$C_{p,f} = 1005.7 + 0.066(T_f - 27), \tag{20}$$

$$k_f = 0.02624 + 0.0000758(T_f - 27), \tag{21}$$

$$\rho_f = 1.1774 - 0.00359(T_f - 27), \tag{22}$$

$$\mu_f = 1.983 + 0.00184(T_f - 27) \times 10^{-5}. \tag{23}$$

3.3. Temperature Determination. Equations (1), (2), (3), (6), and (10) can be arranged in the form of a $5 \times 5$ matrix ($[X][T] = [Y]$) as follows:

$$
\begin{bmatrix}
X_1 & X_2 & 0 & 0 & 0 \\
-X_2 & X_3 & X_4 & 0 & 0 \\
0 & X_4 & X_5 & X_6 & 0 \\
0 & 0 & X_6 & X_7 & X_8 \\
0 & 0 & 0 & X_9 & X_{10}
\end{bmatrix}
\begin{bmatrix}
T_g \\
T_g \\
T_c \\
T_{ted} \\
T_f
\end{bmatrix}
= 
\begin{bmatrix}
Y_1 \\
Y_2 \\
Y_3 \\
Y_4 \\
Y_5
\end{bmatrix}.
$$
where

\[
X_1 = \frac{1}{R_{r,\text{sky}}} + \frac{1}{R_{r,\text{amb}}} + \frac{1}{R_{k,g}},
\]

\[
X_2 = -\frac{1}{R_{k,g}},
\]

\[
X_3 = -\left(\frac{1}{R_{k,c} + R_{k,g}} + \frac{1}{R_{k,c} + R_{k,\text{ted}}}\right),
\]

\[
X_4 = \frac{1}{R_{k,c} + R_{k,g}},
\]

\[
X_5 = -\left(\frac{1}{R_{k,c} + R_{k,g}} + \frac{1}{R_{k,c} + R_{k,\text{ted}}}\right),
\]

\[
X_6 = \frac{1}{R_{k,\text{ted}} + R_{r,f}},
\]

\[
X_7 = \frac{1}{R_{k,c} + R_{k,\text{ted}} + R_{k,c} + R_{k,\text{ted}} + R_{r,f}},
\]

\[
X_8 = \frac{1}{R_{k,\text{ted}} + R_{r,f}},
\]

\[
X_9 = \frac{m_f C_p f}{W_{\text{ad}} L_{\text{ad}}} \left(1 - e^{-\left(\frac{W_{\text{ad}} m_f C_p f (R_{k,\text{ted}} + R_{k,c})}{T_{f,\text{in}}}ight) t_{\text{ad}}}ight) - \frac{1}{R_{k,\text{ted}} + R_{r,f}},
\]

\[
Y_1 = \frac{T_{\text{sky}} - 273.15}{R_{r}} + \frac{T_{\text{amb}}}{R_{r}},
\]

\[
Y_2 = -G_{\alpha g},
\]

\[
Y_3 = -(G_{r,c} \alpha \lambda - \dot{w}_{\text{pv}}),
\]

\[
Y_4 = -G_{r,c} \alpha_{\text{ted}} (1 - \lambda),
\]

\[
Y_5 = \frac{m_f C_p f}{W_{\text{ad}} L_{\text{ad}}} \left(1 - e^{-\left(\frac{W_{\text{ad}} m_f C_p f (R_{k,\text{ted}} + R_{k,c})}{T_{f,\text{in}}}ight) t_{\text{ad}}}ight) T_{f,\text{in}}.
\]

To obtain the temperatures in Eq. (20), an initial temperature of 1°C above the ambient temperature was assumed for each component. The new temperatures were determined using the matrix inversion technique, \([T] = [X]^{-1}[Y]\). If the difference between the new and initial temperatures was above 0.01°C, the initial temperatures were replaced with the new temperature. Until the difference between the two sets of temperatures was less than 0.01°C, the above calculation was repeated.

3.4. Data Reduction. This section explains how the performance of the PVTC was calculated from the temperature determined by the matrix inversion technique. The heat gain of the PVTC was calculated based on Eq. (9) and the determined Tedlar temperature. The thermal efficiency, which represents the ratio between the heat gain of the PVTC and the incident solar irradiance, was obtained from the following equation:

\[
\eta_{\text{th}} = \frac{\dot{q}_f}{G}. \tag{26}
\]

The electrical power generation and efficiency were calculated from Eqs. (4) and (5), respectively, using the obtained cell temperature. The fan power consumption of the PVTC was calculated from the following equation [36, 44]:

\[
\dot{w}_{\text{fan}} = \frac{\dot{Q}_f \Delta P}{\eta_{\text{fan}} A_{\text{pv}}}, \tag{27}
\]

where \(\eta_{\text{fan}}\) is the fan efficiency, and the value of this parameter was assumed to be 0.5 [44]. The pressure drop (\(\Delta P\)) was determined by the following equation [45, 46]:

\[
\Delta P = f \frac{2p_f V_f^2}{(D_{h,f})^2}, \tag{28}
\]

where \(f\) is the friction factor. The friction factor for the PVTC with TTO is given by Ref. [43]:
\[ f = 41.4049 \times Re^{-0.1734} \left( \frac{e_{\text{tno}}}{H_{\text{ad}}} \right)^{7.5824} \left( \frac{p_{\text{tno}}}{e_{\text{tno}}} \right)^{0.3427} \left( \frac{I_{\text{tno}}}{e_{\text{tno}}} \right)^{-0.1015} \exp \left[ 2.5718 \left( \ln \left( \frac{e_{\text{tno}}}{H_{\text{ad}}} \right) \right)^2 \right] \exp \left[ -0.1177 \left( \ln \left( \frac{p_{\text{tno}}}{e_{\text{tno}}} \right) \right)^2 \right] \exp \left[ -0.0883 \left( \ln \left( \frac{I_{\text{tno}}}{e_{\text{tno}}} \right) \right)^2 \right]. \]

\[ f = 0.0791 \times Re^{-0.25}. \]

The friction factor for the PVTC without TTO is given as [18, 47]

The net electrical efficiency of the PVTC is expressed as the ratio of net electrical output recovered by the PVTC to the incident solar irradiance, and it can be written as follows [37]:

\[ \eta_{\text{el,net}} = \frac{\dot{w}_{\text{pr}} - \dot{w}_{\text{fan}}}{G}. \]

As the quality of thermal and electrical energies is different, many researchers introduced the thermal equivalent net energy output (TENEO) and thermal equivalent net energy efficiency (TENEE). The TENEO and TENEE are, respectively, given as [37, 48, 49]

\[ \dot{q}_{\text{net}} = \dot{q}_{\text{f}} + \frac{\dot{w}_{\text{pr}} - \dot{w}_{\text{fan}}}{C_f}, \]

\[ \eta_{\text{net}} = \eta_{\text{th}} + \frac{\eta_{\text{el,net}}}{C_f}. \]

where \( C_f \) is the conversion factor used to convert the electrical energy into the equivalent thermal energy. The value of \( C_f \) was assumed to be 0.38 [37, 49]. Figure 4 presents the solution procedure of the developed model. First, the design parameters and operating conditions were set to solve the model, and weather data were inputted into the model. The model’s input data are solar intensity, ambient temperature, and wind velocity. Second, temperatures for each component were assumed, and the air properties, thermal resistances, and matrix components were calculated. Third, the temperatures of each component were calculated using the matrix inversion technique described in Section 3.3. Once the temperatures were determined, the performance of the PVTC was calculated with the obtained temperatures. This study utilized MATLAB R2022a to develop and solve the simulation model.

3.5. Validation. In this section, the simulation results calculated by the mathematical model are compared with experimental results reported from a previous study of the same type of PVTC to ensure the accuracy of the developed model [35]. The parameters used for the mathematical model in the validation process were taken from Table 1, which are identical to those of the PVTC used in the experiment. The input values of the simulation model were solar intensity, ambient temperature, and wind velocity. The solar intensity and ambient temperature were taken from the previous experimental study, and the wind velocity was fixed at 0.5 m/s. Figure 5 shows the energy output of the PVTC predicted by the mathematical model with the values obtained from the previous experimental study. The mathematical model accurately predicted the experimental results of the proposed PVTC. It was also confirmed that the results of the steady-state model solved with the weather conditions given at each time point exhibited a similar change tendency to the experimental results obtained under transient conditions. The minimum and maximum deviations for the thermal output were 0.27 W/m² and 13.3 W/m², respectively, while those for electricity generation were 0.08 W/m² and 10.65 W/m², respectively. The mean absolute percent error (MAPE) for heat gain and electricity generation was 3.79% and 4.78%, respectively.

3.6. Methodology. This study investigated the thermal and electrical behaviors of the PVTC with various geometric conditions of TTO. In addition, the TENEE was also evaluated to consider both the energy generation and consumption of the PVTC. The relative height, length, and pitch were chosen as geometric conditions of the TTO. For the parametric study, the ambient temperature and wind velocity were fixed at 10°C and 0.5 m/s, respectively. The air flow rate and air duct height of 0.13 kg/s and 0.03 m, respectively, which were found to maximize the PVTC’s performance in a previous study [36], were used for the parametric study and daily performance evaluation. The other dimensions, design parameters, and material properties are the same as the values presented in Table 1. The daily performance of the PVTC with TTO was evaluated under the weather conditions of one of the winter days (21st November) for Ulsan in South Korea and compared with that of PVTC without TTO. The typical meteorological year data was adopted as weather data.

The heat gain, thermal efficiency, electrical power generation, electrical efficiency, power consumption of the fan, TENEO, and TENEE were chosen for performance evaluation. These performance indices were calculated from Eqs. (9), (26), (4), (5), (27), (32), and (33), respectively.

4. Results and Discussion

In this section, we investigate the performance of the PVTC under various geometric conditions of the TTO. The first three subsections discuss the effect of geometric conditions on the energy generation and consumption of the PVTC. In the fourth subsection, the daily performance of the PVTC is evaluated to find the geometric condition that maximizes the PVTC’s net energy output. The studied parameters and their levels are summarized in Table 2.
4.1. Effect of Relative Height. Figure 6 presents energy generation and consumption of the PVTC for various relative heights of the TTO with solar intensity. For a fixed solar intensity, the thermal and electrical efficiencies are raised with an increment in relative height. This is because the increment in height of the TTO causes a higher air velocity in the air duct, resulting in an increment in the heat transfer coefficient and a decrement in the cell temperature. From the figure, it was also observed that the changes in thermal and electrical efficiencies at a low solar intensity were less than those at a high solar intensity. The reason is that the recoverable heat is limited when the solar intensity is low,
even if the heat transfer coefficient increases with an increase in relative height, leading to little change in thermal efficiency. In the meantime, the electrical efficiency slightly increases because the cooling effect caused by the increase in relative height of the TTO is also inconsiderable at a low solar intensity. The fan’s power consumption also increases with a higher relative height because more interruptions in air flow within the air duct. For solar intensities less than 400 W/m², the TENEE significantly decreased with an increase in relative height. For solar intensities more than 600 W/m², the TENEE was slightly raised and then decreased as the relative height increased because of the considerable increase in the fan’s power consumption and a slight increase in thermal and electrical efficiencies. The maximum TENEE was found at the relative height of 0.25 when the solar intensity ranged from 200 to 400 W/m², 0.35 when the solar intensity ranged from 600 to 800 W/m², and 0.45 when the solar intensity was 1000 W/m². For all solar intensities, the TENEE at relative heights above 0.55 is lower than that at relative heights below 0.45 due to the significant increase in power consumption. Thus, the relative height of less than 0.45 was considered suitable for the PVTC.

4.2. Effect of Relative Length. Figure 7 shows the variation of energy generation and consumption of the PVTC for various relative lengths of TTO with relative heights. The increase in relative length reduced the thermal, electrical, and power consumptions of the PVTC. However, the relative length of the TTO did not significantly affect the PVTC’s thermal and electrical efficiencies. The TENEE decreased as the relative length increased when the relative height was less than 0.35 because the thermal and electrical efficiencies decreased with an increment in relative length while the power consumption remained at similar values. When the relative height was more than 0.45, the TENEE increased with an increase in relative length. This is because the fan’s power consumption declined more than thermal and electrical efficiencies.
The relative length, which maximizes the TENEE, was found to be 1 for the relative height of less than 0.35 and 6 for the relative height of more than 0.45.

4.3. Effect of Relative Pitch. The energy generation and consumption with various relative pitches of a TTO are presented in Figure 8. In the previous section, the relative length of TTO, which maximizes TENEE, was found to be 1 and 6 for the relative height of less than 0.35 and for the relative height of more than 0.45, respectively. Thus, the relative length was fixed as 1 for the relative height of less than 0.35 and 6 for the relative height of more than 0.45. A slight increase in thermal and electrical efficiencies was observed with an increase in relative pitch, and the influence of the relative pitch was insignificant compared to the effect of the relative height. This is because relative height has a more dominant impact on the increment in the heat transfer coefficient than relative pitch, as reported in previous research [43]. The power consumption slightly increased and then decreased with the increment in relative pitch but remained at similar values. The TENEE marginally increased with an increment in relative pitch when the relative height was less than 0.45. The TENEE decreased and then increased with increasing relative pitch when the relative height was more than 0.55, but the change in TENEE with relative pitch was inconsiderable. The maximum TENEE was found at a relative pitch of 9 for all relative heights. Among the investigated conditions, a relative height, length, and pitch of 0.45, 6, and 9 resulted in the highest TENEE value of 83.81%.

4.4. Comparison of Daily Performance. In the present work, the daily performance of the PVTC with TTO was evaluated with different geometric conditions of TTO and compared with those of PVTC without TTO. The weather conditions used in the daily performance simulation are presented in Figure 9. The simulation results at each time point were acquired using the solar intensity, ambient temperature, and wind velocity provided at each time point in Figure 9.
Thermal efficiency (–)

\[ \frac{\text{etto}}{\text{Had}} = 0.25, \frac{\text{etto}}{\text{etto}} = 1 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.45, \frac{\text{etto}}{\text{etto}} = 6 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.65, \frac{\text{etto}}{\text{etto}} = 6 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.75, \frac{\text{etto}}{\text{etto}} = 6 \]

(a)

Electrical efficiency (–)

\[ \frac{\text{etto}}{\text{Had}} = 0.25, \frac{\text{etto}}{\text{etto}} = 1 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.45, \frac{\text{etto}}{\text{etto}} = 6 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.65, \frac{\text{etto}}{\text{etto}} = 6 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.75, \frac{\text{etto}}{\text{etto}} = 6 \]

(b)

Power consumption (W/m²)

\[ \frac{\text{etto}}{\text{Had}} = 0.25, \frac{\text{etto}}{\text{etto}} = 1 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.45, \frac{\text{etto}}{\text{etto}} = 6 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.65, \frac{\text{etto}}{\text{etto}} = 6 \]

\[ \frac{\text{etto}}{\text{Had}} = 0.75, \frac{\text{etto}}{\text{etto}} = 6 \]

(c)

Figure 8: Continued.
The relative pitch of the TTO was fixed as 9 for all relative heights, while the relative length was fixed as 1 for the relative height of less than 0.35 and 6 for the relative height of more than 0.45. Figure 10 illustrates the thermal and electrical behaviors of a PVTC with and without TTO for different relative heights during the daily operation.

The thermal efficiency was slightly raised and then declined in accordance with the variation in solar intensity. The electrical efficiency declined with an increment in solar intensity and then increased with decreasing solar intensity. The reason is that the higher solar intensity increases the cell temperature, resulting in lower electrical efficiency. During the daily operation, the PVTC with TTO continuously showed higher thermal and electrical efficiencies than PVTC without TTO. In addition, higher thermal and electrical efficiencies were observed at a higher relative height of TTO. Figure 11 presents the daily energy output of the PVTC with and without TTO. The daily thermal and electrical energy outputs were raised with an increasing relative height of TTO. However, the higher relative height of TTO also resulted in a higher power consumption.

Thus, the TNEO with different relative heights of the TTO is shown in Figure 12. The TNEO was slightly raised with an increasing relative height of the TTO, but further increase in the relative height of 0.45 reduced the TNEO. This is because the PVTC’s power consumption has

\[
\begin{align*}
G &= 1000 \text{ W/m}^2 \\
0.60 &< H \leq 0.65 \\
0.65 &< H \leq 0.70 \\
0.70 &< H \leq 0.75 \\
0.75 &< H \leq 0.80 \\
0.80 &< H \leq 0.85
\end{align*}
\]
increased more than its thermal and electrical outputs. Indeed, the TENEO with a relative height of more than 0.65 was lower than that of the PVTC without TTO because of the considerable increase in power consumption. It implies that the improper geometric conditions of the TTO can reduce the net energy production, although thermal and electrical efficiencies increase. The highest TENEO was found at a relative height of 0.45, and it was considered as a proper geometric condition for TTO. At a relative height of 0.45, the daily average thermal and electrical efficiencies were 39.04% and 17.01%, respectively. From the results, it was also confirmed that the proposed PVTC performed better than other PVTCs, which is similar to the PVTC used in this study except for the TTO [50–54]. In addition, the PVTC with a relative height of 0.45 improved the thermal output by 22.93%, electrical output by 2.79%, and TENEO by 7.8% during the daily operation compared to the PVTC without TTO.

Figure 10: Thermal and electrical behaviors during the daily operation.

Figure 11: Daily energy output for different relative heights of TTO.
5. Conclusions

This study investigated the PVTC with TTO, which has different characteristics from the traditional heat transfer device used in previous PVTCs. The effect of the geometric condition of the TTO on the thermal, electrical, and power consumptions and TENE have been discussed. In addition, the daily performance of the proposed PVTC was evaluated and compared with that of the PVTC without TTO. The key findings are as follows:

(i) A higher relative height of TTO increases thermal efficiency, electrical efficiency, and the fan’s power consumption. The relative height that maximizes the TENE was varied from 0.25 to 0.45 depending on solar intensity.

(ii) There has been a decrease in thermal and electrical efficiencies, as well as the fan’s power consumption, with an increase in relative length.

(iii) Higher relative pitch increases thermal and electrical efficiencies while the power consumption remains similar. Consequently, the higher relative pitch results in higher TENE.

(iv) The relative height of the TTO strongly affected the energy generation and consumption of the PVTC, whereas the effect of the relative length and pitch of the TTO was relatively inconsiderable.

(v) The highest daily TENE was observed when the relative height of the TTO was 0.45. The installation of TTO with a relative height of 0.45 improved the thermal output by 22.93%, electrical output by 2.79%, and TENE by 7.8% compared to the PVTC without TTO during the daily operation.

(vi) The TTO with a relative height of more than 0.65 is considered improper because of the large power consumption of the fan, even though the higher relative height of the TTO resulted in higher thermal and electrical efficiencies.

This study found that installing the TTO with proper geometric conditions improved the energy output of the PVTC, even considering the increase in the fan’s power consumption. Hence, the feasibility of the proposed PVTC could be confirmed. On the other hand, the improper selection of geometric conditions resulted in a lower net energy output than the PVTC without TTO. These results indicate that power consumption should be considered when integrating the new heat transfer devices with PVTC.

The purpose of thermal performance improvement of the PVTC is to use recovered heat. Thus, future work is required to examine the performance and energy savings of conventional energy systems such as space heating and solar dryer coupled with the proposed PVTC. The findings of this study are anticipated to provide useful information in this regard.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>PV:</td>
<td>Photovoltaic</td>
</tr>
<tr>
<td>PVTC:</td>
<td>Photovoltaic-thermal collector</td>
</tr>
<tr>
<td>TENE:</td>
<td>Thermal equivalent net energy efficiency</td>
</tr>
<tr>
<td>TENO:</td>
<td>Thermal equivalent net energy output</td>
</tr>
<tr>
<td>TTO:</td>
<td>Transverse triangle obstacle</td>
</tr>
<tr>
<td>( A_{\text{pvt}} ):</td>
<td>Gross area of the PVTC (m²)</td>
</tr>
<tr>
<td>( C_f ):</td>
<td>Conversion factor</td>
</tr>
<tr>
<td>( C_p ):</td>
<td>Specific heat (J/kg·K)</td>
</tr>
<tr>
<td>( D_h ):</td>
<td>Hydraulic diameter (m)</td>
</tr>
<tr>
<td>( e_{\text{etto}} ):</td>
<td>Height of TTO (m)</td>
</tr>
<tr>
<td>( G ):</td>
<td>Solar intensity (W/m²)</td>
</tr>
<tr>
<td>( h ):</td>
<td>Convective heat transfer coefficient (W/m²·K)</td>
</tr>
<tr>
<td>( H_{\text{ad}} ):</td>
<td>Height of air duct (mm)</td>
</tr>
<tr>
<td>( k ):</td>
<td>Thermal conductivity (W/m·K)</td>
</tr>
<tr>
<td>( l_{\text{etto}} ):</td>
<td>Length of TTO (m)</td>
</tr>
<tr>
<td>( l_{\text{ad}} ):</td>
<td>Length of air duct (mm)</td>
</tr>
<tr>
<td>( m ):</td>
<td>Mass flow rate (kg/s)</td>
</tr>
<tr>
<td>( p_{\text{etto}} ):</td>
<td>Pitch of TTO (m)</td>
</tr>
</tbody>
</table>
Greek Letters

\( \alpha \): Absorptivity
\( \beta \): Temperature coefficient (\(^\circ\)C)
\( \delta \): Thickness (m)
\( \varepsilon \): Emissivity
\( \eta \): Efficiency
\( \lambda \): Ratio of cell area to total area
\( \mu \): Dynamic viscosity (kg/m\( \cdot \)s)
\( \rho \): Density (kg/m\(^3\))
\( \sigma \): Stefan-Boltzmann constant (W/m\(^2\)K\(^4\))
\( \tau \): Transmissivity.

Subscripts

amb: Ambient
\( c \): PV cell
elec: Electrical
\( f \): Fluid
gs: Glass surface
g: Glass
in: Inlet
k: Conductive
net: Net
out: Outlet
\( r \): Radiative
std: Standard condition
sky: Sky
\( t \): Tedlar
th: Thermal
\( v \): Convective
\( w \): Wind.

Data Availability

Data are available on request.

Conflicts of Interest

The authors declare that there is no conflict of interest regarding the publication of this article.

References


