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A state of art review on methodologies for heat transfer and energy flow characteristics of the active building envelopes

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ABSTRACT

Significant share of total final energy use is accounted by the building sector in most of the countries around the world. One way to reduce building energy consumption is to adopt energy efficiency technologies and strategies. Due to environmental concerns and high cost of energy in recent years there has been a renewed interest in building energy efficiency and integration of renewable energy technologies. Active building envelope technology, i.e. transpired solar collectors (TSCs), provides a cost-efficient way of minimizing energy demand of buildings in accordance with global principle of sustainability, which has also proven reliable for diverse applications such as preheating fresh air delivery into the buildings and supplying domestic hot water in summer etc. The objective of this paper is to review the heat transfer and energy flow characteristics of the active building envelopes, particularly focusing on various types of TSCs. Present work consists of background and concept of TSCs, research literature for thermal performance, theoretical modelling, experimental study and numerical simulation investigation. Diverse mathematical models, including thermal models, air flow models, porosity models, and turbulence models etc., have also been presented and compared. Following that, more than 20 parameters affecting TSC performance have been analyzed and evaluated. The literature has illustrated that the best overall performance of turbulence model is RNG k-ε; the effects of those parameters on TSC efficiency are completely different, depending on local climatic conditions, time and site constraints, and the interaction between different factors.

1. Introduction

Nowadays, the building sector accounts for approximately 40% of the total world final energy consumption and for about one third of CO₂ emissions into the atmosphere, which can result in global warming and climate changes [1,2]. Similarly, in the UK, the domestic sector is responsible for almost 40% of national carbon emissions [3]. The “Climate Change Act 2008” has already set a target of 80% reduction in CO₂ emissions in the UK (relative to 1990 emissions) to be achieved by 2050 [4]. The increasing trend towards building consumption will persist in the coming years due to the extension of built regions and related energy demands [2,5].

The building envelope plays a vital role not only in thermal comfort but also in building energy efficiency. There are significant opportunities for the building envelope to solve the aforementioned problems, which could exploit solar energy through integrating solar thermal technologies into the buildings. Transpired solar collector (TSC) is one of most popular solar thermal technologies, which is also called as active building envelope [6,7] from the perspective of TSC’s active generation of solar thermal energy.

Research on TSCs started in the early 1990s [8,9] and was focused on feasibility studies and testing [10,11]. Authors (Shukla et al., 2012) have reviewed the performance of TSCs from the literature before 2012 [12]. In this work, the objective mainly focuses on the heat transfer and energy flow characteristics of TSCs for the recent five years, i.e., from 2012 to 2016, and those literatures were not reviewed by current authors before 2012 [14–43,46,49–54,56–62]. In Section 2, background and concept of active building envelopes will be introduced; in Section 3, the heat transfer and energy flow characteristics of the active building envelopes from the perspective of mathematical modelling.
Nomenclature

\(A_c\) effective area of the collector in m\(^2\)
\(A_p\) area of the holes in m\(^2\)
\(A_{PV}\) total PV area in m\(^2\)
\(C_f\) conversion factor (thermal to mechanical) and given as 0.2
\(C_p\) specific heat capacity of air in J/kgK
\(D\) perforation/hole equivalent diameter in m
\(F_{pb}\) view factor between the UTC plate and the back wall
\(F_R\) heat removal factor
\(G\) incident solar irradiance on the UTC plate per unit area in W/m\(^2\)
\(h\) convective heat transfer coefficient in W/m\(^2\)K
\(H\) total height of UTC-2stage in m
\(H_{as}\) UTC height in m
\(i\) control volume index
\(k\) air thermal conductivity in W/mK
\(N_h\) holes number
\(N_{Nu}\) Nusselt number (= \(h\times k\))
\(P\) hole pitch in m, i.e., distance between center of hole and center of next closest hole
\(\Delta P\) pressure drop in Pa
\(P_{PV}\) electrical energy generated by PV panels in W/m2
\(Pr\) Prandtl number (=\(\nu\times k\))
\(Q\) air volume flow rate in m\(^3\)/h
\(Q_{conv}\) convective heat flux between difference surfaces (or air) in W/m2
\(Q_{gain}\) thermal heat gain from the collector in W
\(Q_{rad}\) radiative heat flux between different surfaces in W/m2
\(Q_{solar}\) solar radiation absorbed by the plate (or PV module) in W/m2
\(q_\text{in}\) heat collected from the collector in W
\(Re\) Reynolds number (= \(\nu\times k\)) where \(\nu\) is relevant velocity
\(T_{amb}\) ambient temperature in K or °C
\(T_c\) cavity exit air temperature in K
\(T_{col}\) collector temperature in °C
\(T_{in}\) inlet temperature in K or °C

Greek symbols

\(\alpha\) air thermal diffusivity in m\(^2\)/s
\(\beta\) temperature coefficient of the PV module in %/K.
\(\eta\) effectiveness ratio of collector
\(\epsilon_b\) emissivity of the back wall
\(\epsilon_p\) emissivity of the UTC plate (PV panel)
\(\eta_t\) efficiency of the TSC in %
\(\eta_{inst}\) instantaneous efficiency of the BPSC in %
\(\eta_{eff}\) effective efficiency in %
\(\eta_e\) equivalent thermal efficiency in %
\(\eta_f\) first law efficiency in %
\(\eta_m\) fan motor efficiency in %.
\(\eta_{PV}\) electrical efficiency for PV panels in %
\(\eta_{pv/T}\) combined thermal and electrical efficiency for BIPV/T systems in %
\(\eta_{sec}\) second law efficiency in %
\(\eta_{rev}\) electrical efficiency of PV module under STC in %
\(\tau_a\) effective transmittance-absorbance factor
\(\sigma\) Stefan-Boltzmann constant (5.67×10 \(^{-8}\) W/(m\(^2\)K\(^4\)));
\(\beta_p\) plate porosity
\(\nu\) air kinematic viscosity in m\(^2\)/s
\(\epsilon\) porosity of the TSC plate

experimental study, numerical simulations particularly CFD, and parametric sensitivity analysis including more than 20 parameters affecting TSC performance are indicated in detail, respectively; finally, conclusions and future work are drawn in Section 4.

2. Background and working principle for TSC

Active building envelopes, i.e., TSCs derived from Canada and USA and research on TSCs started in approximately early 1990s. TSCs have been already widely employed in USA and Canada and the technology has been extensively monitored by their government agencies. In addition, Natural Resources Canada has developed a feasibility tool called RETScreen to model the energy savings from TSCs [13]. However, TSCs are relatively new for other regions e.g. Europe and China.

Authors already described the historical development and working principle of TSCs, collector construction and its parameters, diverse types of TSCs in detail [12]. In this work, all those aforementioned contents will be not introduced again. The concept of TSCs is just presented using schematics below (Fig. 1(a)). There have already been some TSCs (active building envelop) applied in the existing buildings below (Fig. 1(b-d)).

Solar energy is employed by TSCs to heat the perforated absorber surface, which could transfer thermal energy to the ambient air for preheating fresh air delivered into buildings/room. Generally, the perforated absorber plate is a metallic sheet e.g. steel or aluminium, which could be integrated to the building façade and PV panels, and could generate solar thermal energy actively. Therefore, TSCs are also called as active building envelopes as mentioned before [6,7].

3. Heat transfer and energy characteristics of TSCs

Investigations on TSCs for the heat and airflow mass transfer, thermal efficiency, heat exchange efficiency, exergy efficiency, energy characteristics have been carried out for lots of researchers in last approximately 30 years, i.e., since 1991. Authors have already reviewed thermal performance of TSCs from the literature before 2012 [12]. In this work, a detailed summary for various models and study in recent 5 years has been presented in chronological order as shown in Table 1.

There are four different investigation methodologies for thermal performance of TSCs as follows:

1) Mathematical modelling study;
2) Physical experimental study including PIV, IR, prototype experiments and monitoring;
3) Numerical simulations including computational fluid dynamics (CFD); and
4) Parametric sensitivity studies.
Seen from the Table 1, there are mainly two types of transpired solar collectors (TSCs) classified by their materials including unglazed transpired collector (UTC), and glazed transpired collector (GTC) or called transparent transpired collector (TTC). According to the metal sheets pattern of solar collector system, it includes flat and corrugated TSCs. In addition, TSCs could be also categorized by their configuration as follows: stand-alone TSC, hybrid and building integrated TSC [12].

3.1. Mathematical models for heat transfer study

3.1.1. Thermal modelling of TSC system

Several mathematical/thermal model for TSC has been currently reported in literature. This section categories them in three main areas (i) energy balance (ii) efficiency calculations/models (iii) heat exchange effectiveness and (iv) Nusselt Number. A developed optimal BIPV/T system with UTC has been investigated analytically, experimentally and numerically [14,18,22,23]. Its thermal network of UTCs with and without PV modules is illustrated schematically in Fig. 2.

The cavity underneath the UTC plate is separated into several control volumes (e.g. i, i+1 etc.). One corrugation is corresponded to each control volume. The energy (heat) balance analysis was carried out on each temperature node of every control volume, which will be discussed in the following Section Energy balance.

(i) Energy balance (Heat balance)

Energy balances at the UTC surface (heat transfer surface), the air node in the cavity and the surface of the back wall were defined by Li et al. [14], which also reflect the heat balances of the UTC plate and the air fluid [16], using the Gauss-Seidel method as

\[ Q_{\text{total}} = Q_{\text{rad, p, sky}} + Q_{\text{rad, p, b}} + Q_{\text{conv, p, a}} + Q_{\text{conv, p, c}} \]  

\[ Q_{\text{conv, p, c}} = (i - 1) \rho V_{\text{air}} C_{p} (T_{i,j} - T_{i-1,j}) + \rho V_{\text{air}} C_{p} (T_{i,j} - T_{i}) + Q_{\text{conv, c, b}} \]  

\[ Q_{\text{rad, p, sky}} = \epsilon \sigma (T_{\text{air}}^{4} - T_{\text{sky}}^{4}) \]  

\[ Q_{\text{rad, p, b}} = \epsilon \sigma (T_{\text{air}}^{4} - T_{\text{wall}}^{4}) \]  

where \( Q_{\text{total}} \) is the solar radiation absorbed by the UTC plate in W/m², \( Q_{\text{rad, p, sky}} \) and \( Q_{\text{rad, p, b}} \) represent the radiation exchange between the plate and the sky, and the back wall, respectively, which can be calculated as in Eqs. (4) and (5):

\[ Q_{\text{rad, p, sky}} = \epsilon \sigma (T_{\text{air}}^{4} - T_{\text{sky}}^{4}) \]  

\[ Q_{\text{rad, p, b}} = \epsilon \sigma (T_{\text{air}}^{4} - T_{\text{wall}}^{4}) \]  

\[ Q_{\text{conv, p, a}}, Q_{\text{conv, p, c}} \] and \( Q_{\text{conv, c, b}} \) are the convective heat flux between the plate and ambient air, and cavity exit air, between cavity exit air and the back wall, respectively, in W/m². \( V_{\text{air}} \), \( C_{p} \), \( T_{\text{air}}, T_{\text{sky}}, T_{\text{wall}} \), \( \epsilon_{p}, \epsilon_{b} \) are the mean suction velocity in m/s or m³/(s·m²), control volume index, specific heat capacity of air in J/kg·K, cavity exit air temperature, air temperature through the perforation, UTC plate surface temperature and sky temperature in K, Stefan-Boltzmann constant (5.67×10⁻⁸ W/(m²·K⁴)), view factor between the UTC plate and the back wall, emissivity of the
<table>
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<tr>
<th>Publications/References</th>
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<tbody>
<tr>
<td>Li et al. 2014 [14]</td>
<td>To predict cavity exit air temperature, plate surface temperature with Weather and design parameters; Gauss-Seidel method for energy prediction of UTCs with and without PV modules</td>
<td>Corrugated UTCs/UTCs integrated with BIPV/T</td>
<td>CFD, Correlation prediction, Physical experiment, design concept of PV module dimensions</td>
<td>Regression predictions illustrate a good agreement with the CFD simulation results, with 90% of the prediction points within ± 20% error band for both Nu correlations and within ± 10% error band for the effectiveness.</td>
<td>The portion of the exterior heat flux recaptured by suction should be considered.</td>
</tr>
<tr>
<td>Li et al. 2016 [15]</td>
<td>To investigate the heat transfer and flow resistance characteristics of a vacuum glazed TSC with slit-like perforations; To analyze the effects of key parameters in relation to effective thermal efficiency considering fan power</td>
<td>GTC with slit-like perforations</td>
<td>Mathematical and Physical Experiment, Matlab–7: temperature of each control volume; Thermal modelling and pressure drop model</td>
<td>Perforation diameter and pitch have a minor impact on the heat collected compared to the pressure drop for the ranges of D and P.</td>
<td>Effective efficiency improves with increasing of perforation diameter and ambient temperature, decreasing of the pitch, plenum thickness and inlet air temperature; the effect of coating emissivity is less obvious than that of absorptivity.</td>
</tr>
<tr>
<td>Belusko et al. 2008 [16]</td>
<td>To investigate the performance of jet impingement in USC of roof (Not reviewed by Shukla et al. 2012)</td>
<td>USC</td>
<td>Mathematical and Physical Experiment</td>
<td>Jet impingement can enhance the thermal efficiency of collector by 21% and increase pressure losses a little. The efficiency can enhance with increasing of the hole spacing.</td>
<td>Jets’ Flow distribution along the collector is the most important parameter to determine the efficiency.</td>
</tr>
<tr>
<td>Vasan et al. 2014 [17]</td>
<td>To study wind effects on UTCs</td>
<td>UTCs</td>
<td>Physical Experiment</td>
<td>Assumption of reference uniform wind speed through the UTC will cause up to 50% overestimation of heat exchange effectiveness (HEE) and up to 20% underestimation of convective heat transfer coefficients (CHTC), compared to realistic non-uniform distribution.</td>
<td>Winds with 45° incidence angle to the UTC have most significant impact on CHTC and HEE.</td>
</tr>
<tr>
<td>Li et al. 2014 [18]</td>
<td>To investigate important factors which affect the performance of corrugated UTCs with and without PV panels</td>
<td>Corrugated UTCs/UTCs integrated with BIPV/T</td>
<td>CFD</td>
<td>Horizontal installations of UTCs are proposed for low-rise buildings and vertical for high-rise buildings. The heat significant factor on energy performance is amplitude, then crest length and cavity width. The wavelength and slope length have the most obvious effect on the collector temperature.</td>
<td>Thermal efficiency and cavity exit air temperature will decrease with increasing TL particularly when increasing wind speed and decreasing suction velocity.</td>
</tr>
<tr>
<td>Croitoru et al. 2016 [19]</td>
<td>To prove the efficiency increase of lobed geometries, underlining the benefit of passive solar systems</td>
<td>UTCs</td>
<td>Physical Experiment (PIV, IR)</td>
<td>The more complex dynamics of the lobed flows, it causes a better heat transfer rate. The comparison between a conventional USC and a new geometry with innovative perforation will result in interesting results, with nearly 40% increase in thermal efficiency over the same perforated cladding and the exhausted airflow.</td>
<td>The stream wise vortices induced by the lobed cross shaped perforation help in the stabilization of the jet flows inside the ventilated cavity.</td>
</tr>
<tr>
<td>Tajdaran et al. 2016 [20]</td>
<td>To develop a comprehensive CFD model for TSC systems</td>
<td>TSCs</td>
<td>CFD (STAR-CCM+) and experiment</td>
<td>Wind angles affect the impact of solar radiation on the operating temperature of TSCs. Effect of wind angle and solar radiation on Heat Change Effectiveness is minor.</td>
<td>Heat Exchange Effective nes is at its highest value when system works at the lowest suction ratio with lowest wind speed.</td>
</tr>
<tr>
<td>Badache et al. 2012 [21]</td>
<td>To model and optimize thermal performance of a UTC prototype using a full factorial experiment with four factors at three levels</td>
<td>UTC</td>
<td>Physical experiment</td>
<td>The resulting regression model shows the impact of hole diameter is not statistically significant (with &gt; 95% certainty), while that of irradiation is.</td>
<td>The response surfaces made it possible to identify the optimal set of four parameters for which the UTC efficiency ranges between 70% and 80%.</td>
</tr>
<tr>
<td>Li et al. 2013 [22]</td>
<td>To analyze the convective heat transfer processes for both flat and corrugated UTCs</td>
<td>Corrugated UTCs</td>
<td>CFD</td>
<td>The suction velocity has the most obvious impact on the boundary layer development and thermal level of turbulence intensity is less important than the presence of perforations in the approach flow for flat UTCs.</td>
<td>(continued on next page)</td>
</tr>
</tbody>
</table>
Radiation heat transfer from the heated corrugation wall to the construction wall not only increased the construction wall temperature but also produced 15% to 25% of the total heat gain of the air. Increasing coverage percent of PC has a positive impact on the second law efficiency but a negative impact on the first law efficiency.

A weak heat exchange process appears in the plenum: the heat gain of the air. It is not determined to obtain a unique optimum value for suction velocity or PV coverage percent because of their values relying on the solar radiation.

Table 1 (continued)

<table>
<thead>
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<th>Reference</th>
<th>Type of TSCs</th>
<th>Software/Model Used</th>
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<tr>
<td>Gholampour et al. 2016</td>
<td>UTC and GTC</td>
<td>Mathematical modelling, Matlab</td>
<td>To investigate experimentally thermal performance of UTCs</td>
</tr>
<tr>
<td>Rad et al. 2016</td>
<td>UTC</td>
<td>Physical experiment</td>
<td>To develop a novel wall-integrated and unglazed solar air collector i.e. back-pass non-transpired solar collectors (BPSC)</td>
</tr>
<tr>
<td>Gao et al. 2013</td>
<td>UTC</td>
<td>Physical experiment</td>
<td>To analyze heat transfer characteristic of GTC using CFD</td>
</tr>
<tr>
<td>Greig et al. 2014</td>
<td>UTC and GTC</td>
<td>Mathematical modeling, Matlab</td>
<td>To investigate experimentally thermal performance of UTCs</td>
</tr>
<tr>
<td>Badache et al. 2016</td>
<td>UTC</td>
<td>Physical experiment</td>
<td>To develop a mathematical model for predicting thermal performance of UTC</td>
</tr>
<tr>
<td>Collins et al. 2014</td>
<td>UTC</td>
<td>CFD, Physical experiment (PIV)</td>
<td>To develop a predictive model, validate, and investigate the PV/Thermal flat transpired plate prototype capabilities</td>
</tr>
<tr>
<td>Love et al. 2012</td>
<td>UTC</td>
<td>CFD, Physical experiment</td>
<td>To examine the air flow behavior in a channel with a corrugated surface in a TSC</td>
</tr>
<tr>
<td>Badache et al. 2016</td>
<td>TSC</td>
<td>CFD, Physical experiment</td>
<td>To examine experimentally flow dynamics in a channel with a corrugated surface in a TSC</td>
</tr>
<tr>
<td>Badache et al. 2016</td>
<td>TSC integrated with PVT</td>
<td>Physical experiment</td>
<td>To develop a predictive model, validate, and investigate the TSCs thermal performance of a TSC experimentally and numerically</td>
</tr>
<tr>
<td>Gao et al. 2014</td>
<td>UTC</td>
<td>CFD, Physical experiment</td>
<td>To develop a mathematical model for predicting thermal performance of UTC</td>
</tr>
<tr>
<td>Greig et al. 2013</td>
<td>TSC</td>
<td>CFD, Physical experiment</td>
<td>To develop a mathematical model for predicting thermal performance of UTC</td>
</tr>
<tr>
<td>Gao et al. 2013</td>
<td>UTC</td>
<td>CFD ( Fluent 6.3)</td>
<td>To investigate experimentally thermal performance of UTCs</td>
</tr>
</tbody>
</table>

There is no effect on the performance of BPSC when wind speeds are up to 4 m/s across the collector metal plate. There is no effect on the performance of BPSC when wind speeds are up to 4 m/s across the collector metal plate.
For the case of with PV panels, it is similar to the above case of without PV panels. Energy balances at the surface of the PV module, the air node in the cavity and the surface of the back wall are written as

\[ Q_{\text{rad}} = Q_{\text{rad.PV}} + Q_{\text{rad.PV c}} + Q_{\text{conv.PV}} + Q_{\text{conv.PV c}} + Q_{\text{conv.PV b}} + Q_{\text{conv.PV b c}} \]  

\[ Q_{\text{rad.PV c}} + Q_{\text{rad.PV b c}} = 0 \]  

where \( P_{\text{PV}} \) is the electrical power generated by the PV panels in W/m². Each term calculation is similar to Eqs. (1)–(3), just using the thermal characteristics of PV module instead of the characteristics of the UTC plate. \( Q_{\text{rad.PV c}} \) and \( Q_{\text{rad.PV b c}} \) represent the radiation exchange between the PV module and UTC plate, as well as between the UTC plate and the back wall, respectively. Here noted that the temperature of UTC plate \( T_p \) is considered to be equal to the cavity air temperature \( T_c \) [40].

(ii) Efficiency

a) Collector efficiency

The thermal efficiency of TSC [27,30,32,38], which is used as a measure of heat transfer performance for the TSC, is calculated from the ratio of the useful heat gain from the collector to the incident irradiance on the collector surface and could be determined as

\[ \eta = \frac{[\eta_{\text{C}}(T_{\text{out}}-T_{\text{amb}})/A_c]G}{\dot{m}_{\text{out}} C_p (T_{\text{out}}-T_{\text{amb}})/A_c G} \]  

where \( \dot{m}_{\text{out}} \) is the mass flow rate through the TSC in kg/s, \( T_{\text{out}} \) is the outlet air temperature in °C, \( T_{\text{amb}} \) is the ambient temperature in °C, \( A_c \) is the effective area of the collector in m², \( G \) is the incident solar irradiance on the collector surface in W/m².

Badache et al. [32] put forward a novel formula of the collector efficiency as a function of air mass flow rate in the range of 2.2–6.3 g/s, which is written as:

\[ \eta = 80.31 + 14.89 m_{\text{out}} - 7.66 m_{\text{out}}^2 \]  

where \( \dot{m}_{\text{out}} \) is the heat collected from the collector in W, \( W_{\text{fan}} \) is the fan power in W, \( C_f \) is the conversion factor (thermal to mechanical) and given as 0.2, \( Q \) is the air volume flow rate in m³/h, \( \Delta P \) is the pressure drop in Pa, \( \eta_{\text{fan}} \) is the fan motor efficiency in % and given as 0.85. The effective efficiency presents the ratio of the net heat gain to the total solar energy input on the collector surface.

b) Effective efficiency

A novel efficiency defined by Gupta et al. [15,41] namely effective efficiency \( \eta_f \) taking into account the fan power requirement arising from the airflow resistance via solar collector, which could be calculated by Eq. (11).

\[ \eta_f = \frac{q_C - \frac{W_{\text{fan}}}{C_f}}{A_c G} = \frac{Q\Delta P}{\eta_{\text{fan}} C_f} \left( \frac{G}{3600 A_c G} \right) \]  

where \( q_C \) is the heat collected from the collector in W, \( W_{\text{fan}} \) is the fan power in W, \( C_f \) is the conversion factor (thermal to mechanical) and given as 0.2, \( Q \) is the air volume flow rate in m³/h, \( \Delta P \) is the pressure drop in Pa, \( \eta_{\text{fan}} \) is the fan motor efficiency in % and given as 0.85. The effective efficiency presents the ratio of the net heat gain to the total solar energy input on the collector surface.

c) First law efficiency (energy efficiency)

The instantaneous first law or energy efficiency \( \eta_f \) for TSC with PV/T system was defined by Gholampour et al. [26] as a ratio of the useful energy delivered to the received solar energy by the collector:
\[ \eta_p = \frac{\text{useful energy}}{\text{received energy}} = \frac{m_{\text{out}} C_p (T_{\text{out}} - T_{\text{in}}) + W_{\text{pv}} - W_{\text{fan}}}{A_c} \]  
where \( m_{\text{out}} \) is the power of PV panel in W, \( W_{\text{fan}} \) is the power of fan in W.

\[ \eta_{se} = \frac{m_{\text{out}} C_p (T_{\text{out}} - T_{\text{in}})}{A_c A} = \frac{F_{T_a} - [F_{T_b} (T_{\text{in}} - T_{\text{amb}})]/G}{1 - (4 T_{\text{amb}}/3 T_{\text{sun}}) + (1/3) T_{\text{amb}}/T_{\text{sun}}^4} \]  
where \( T_{\text{sun}} \) is the sun temperature, which is equal to approximately 6000 K. The second law efficiency represents a kind of the quality of useful energy gains, since the energy quality is very important for TSC with PV/T system [26].

\[ \eta_{et} = \frac{m_{\text{out}} C_p (T_{\text{out}} - T_{\text{in}})}{A_c A} = \frac{F_{T_a} - [F_{T_b} (T_{\text{in}} - T_{\text{amb}})]/G}{1 - (4 T_{\text{amb}}/3 T_{\text{sun}}) + (1/3) T_{\text{amb}}/T_{\text{sun}}^4} \]  
where \( m_{\text{out}} \) is \( F_p \) is the collector efficiency factor, \( r_a \) is the effective transmittance-absorptance factor, \( U_i \) is the heat loss coefficient in W/m²K, \( T_{\text{in}} \) is the inlet temperature in °C. For the term of the effective transmittance-absorptance product in the above Eq. (15) \( F_p U_i \) and the term of overall heat transfer coefficient for losses from the collector \( F_p U_i \) are constant over the whole collector plane [38].

(i) **Instantaneous efficiency of the BPSC**

The instantaneous efficiency of the BPSC \( \eta_{et} \) is defined by the equation of Hottel-Whillier-Bliss equation [38,45] and Eq. (14):  
\[ \eta_{et} = \frac{m_{\text{out}} C_p (T_{\text{out}} - T_{\text{in}})}{A_c A} = \frac{F_{T_a} - [F_{T_b} (T_{\text{in}} - T_{\text{amb}})]/G}{1 - (4 T_{\text{amb}}/3 T_{\text{sun}}) + (1/3) T_{\text{amb}}/T_{\text{sun}}^4} \]

The HEE of the TSC could be defined as the air temperature rise in the collector to the maximum temperature rise [24,27,46]. There is another name for HEE i.e. the effectiveness ratio of collector [14,47], which was defined by Van Decker based on the plate surface temperature \( (T_p) \), the ambient temperature \( (T_{\text{amb}}) \), and the air temperature through the perforation \( (T_s) \), which maintains relatively a constant along the plate, as follows:

\[ \varepsilon = (T_s - T_{\text{amb}})/(T_p - T_{\text{amb}}) \]  

HEE could be separated into three partials effectiveness \( (\varepsilon_p, \varepsilon_h \) and \( \varepsilon_s \) combined each region in order to compare the temperature rise in the perforation/hole, the front and back sides of the UTC:

\[ \varepsilon_f = (T_{\text{amb}} - T_{\text{amb}})/(T_p - T_{\text{amb}}) \]  

\[ \varepsilon_h = (T_{\text{amb}} - T_{\text{amb}})/(T_p - T_{\text{amb}}) \]  

\[ \varepsilon_s = (T_{\text{amb}} - T_{\text{amb}})/(T_p - T_{\text{amb}}) \]
\[ \epsilon_b = (T_{out,h} - T_{in,h})/(T_p - T_{amb}) \]  
(18)

\[ \epsilon_b = (T_{out} - T_{in,h,b})/(T_p - T_{amb}) \]  
(19)

where \( T_{in,h} \) is the average air temperature at the slot inlet in °C, \( T_{out,h} \) is the average air temperature at the slot outlet in °C. Following that, the total effectiveness could be expressed by:

\[ \epsilon = \epsilon_f + \epsilon_b \]  
(20)

(ii) Nusselt number (\( Nu \))

In heat transfer at boundary (surface) within fluid, Nusselt Number is the ratio of convective to conductive heat transfer across (normal to) the boundary and this makes this parameter very relevant for TSC as it further includes diffusion within the surface/holes. \( Nu \) number could be estimated by an empirical correlation [15,48] as follows:

\[ Nu_{D} = 2.75 \{(P/D)^{-1.2} Re_f^{0.43} + 0.011 \sigma_p Re_f (V_{win}/V_{suc})^{0.08}\} \]  
(21)

where \( P \) is the pitch in m, \( D \) is the perforation equivalent diameter in \( \text{mm} \), \( Re_f \) is Reynolds number, \( \sigma_p \) is the plate porosity, \( V_{win} \) is the air free-stream velocity or wind speed in m/s, \( V_{suc} \) is the surface suction velocity in m/s.

However, \( Nu \) of convection between fluid and plate could be expected [16] as follows at very small hole diameters:

\[ Nu_{P} = 0.17 + 0.725 P^{0.38} (Re_f)^{0.43} \leq Re_f \leq 2000 \]  
(22)

where \( Pr \) is Prandtl number, \( Re_f \) is based on the gap between the surfaces.

Collins et al. [24] obtained the \( Nu \) number for all flow based on fit analysis of the numerical data using a least square power law, which is written as:

\[ Nu_{P} = 0.0081 Re_f^{-0.029} \]  
(23)

where \( Re_f \) is the suction parameter and is equal to \( V_{win}/V_{suc} \). The aforementioned correlation is just suitable for wind velocities in the range of 0.5–2 m/s, and suction velocities of 0.01–0.04 m/s.

Gholampour et al. [26] estimated \( Nu \) number for TSC plate and PV panel based on parameter studies as follows.

\[ Nu_{P} = 0.7112 R^{0.711 (a/b)^{-0.033}} (P/D)^{0.0218} \]  
(24)

and

\[ Nu_{PV} = 0.7145 R^{0.5243 (a/b)^{-0.05003}} (P/D)^{-0.1079} \]  
(25)

where \( a/b \) is the width to length of PV panel ratio. The above correlation results for \( Nu \) number of TSC plate and PV panel have been compared with CFD numerical results.

3.1.2. Combination with electrical modelling of PV

The electrical efficiency of PV modules with UTC is generally defined as a function of its surface temperature and written as [14,49]:

\[ \eta_{PV} = \eta_{ref} [1 - \beta_{ref} (T_{PV} - T_{ref})] \]  
(26)

where \( \eta_{ref} \) is the electrical efficiency of module under STC (standard test condition) in %, \( T_{ref} \) is the surface temperature of PV module under STC in °C, \( \beta_{ref} \) is the temperature coefficient of the PV module in %/K. Following that, the electrical energy generated \( P_{PV} \) can be obtained as Eq. (27), which is a function of the incident solar irradiance \( G \) in W/m². The combined thermal and electrical efficiency for BIPV/T systems \( \eta_{PV/T} \) can be written as Eq. (28).

\[ P_{PV} = G \eta_{PV} \]  
(27)

\[ \eta_{PV/T} = \frac{(Q_{gain} + P_{PV} A_{PV})}{A_c} \% \]  
(28)

\( Q_{gain} \) is the thermal heat gain from the collector in \( W \), \( A_{PV} \) and \( A_c \) are the total PV and collector effective area in m², respectively.

3.1.3. Resolution algorithm Vs CFD

Semenou et al. [39] presented a simple model for TSC to be employed in the design steps of an installation instead of CFD simulations at each step of the design, which calculates very fast and could reduce strongly the costs of expensive standard CFD simulation. Fig. 3 illustrates the resolution algorithm developed by them in detail in order to determine the temperature vectors.

Authors claimed that predicted thermal efficiencies of TSC are within 2% of discrepancy as provided by Solar Rating and Certification Corporation (SRCC). However, it is worth to note that results were validated only for small collector size of 4.5 m² and model may show further variations for large size TSC and other BIPV/TSC systems, which further need to be investigated.

3.1.4. Porosity model

In the study of TSC performance effective absorber area is calculated using porosity model. Porosity of any TSC might depend on hole patterns, distance between holes and size (diameter) of the holes. This factors plays a very important role as it directly determines the useful solar energy and effective absorber area and indirectly impacts on flow rate, plenum size, air flow and heat losses [12]. The porosity \( \xi \) could be defined as the ratio of the area of the holes \( A_h \) to the area of collector plate \( A_c \) [26]:

\[ \xi = \frac{N_h A_h}{A_c} \]  
(29)

where \( N_h \) is the holes number depending on the hole patterns. When the hole pattern is square, \( \xi \) could be formulated and simplified as [26,39,47]:

\[ \xi = \left( \frac{\pi}{4} \right) \left( \frac{D}{P} \right)^2 = 0.78 \left( \frac{D}{P} \right)^2 \]  
(30)

When the hole pattern is triangle, \( \xi \) could be formulated and simplified as [26,48]:

\[ \xi = 0.907 \left( \frac{D}{P} \right)^2 \]  
(31)

3.2. Experimental study

3.2.1. Experimental apparatus/setup

Corresponding to the configuration of active building envelope and its thermos-physical characteristics, the experimental system mainly involved the TSC system element: collector (absorber plate), metal profile, air duct, fan, PV modules [14], PCM; Thermal apparatus: pyranometer [27,29,38], IR [19], radiometer [15]; Sensors or transduce e.g. Pressure transducer/pressure transmitter [38], sensors of temperature: thermocouples (PT 100 probes [38]), thermistors [29,38]; Air flow apparatus including PIV [19,25,28], velocity meter, TSI wind speed tester [15], laminar flow element, hot-wire anemometer/anemometry [14,27,38,47] wind tunnel [17]; Electrical devices: solar simulator (incandescent lamps [16], halide lamp [19]), power resistor; Data acquisition and analysis instrument: Fluke [15] and Hobo data logger [29].

3.2.2. Aims of experimental study

The aim of experimental investigation is mainly to validate mathematical theoretical models and numerical models (discussed in Section 3.3.3). Another aim is to provide experimental support for parametric analysis study (introduced in Section 3.4). Since 2012, many experimental studies on the heat transfer and energy flow characteristics of TSCs have already carried out [14–17,19–21,25–29,32,33,38,39].

Belusko et al. [16] built an indoor experimental testing rig for the UTC with jet impingement, and carried out the thermal experiments involved subjecting the collector to the various flow rate and a constant irradiance. The maximum increase in thermal efficiency of UTC was
The optimized parameters included the hole diameter, absorber coatings method to optimize thermal performance of an UTC prototype. The parameter for circular and lobed jets, and compare the baseline case with round perforations and the innovative cladding with lobed perforations, respectively.

Badache et al. [21] employed a full 3^4 factorial design-of-experiments method to optimize thermal performance of an UTC prototype. The optimized parameters included the hole diameter, absorber coating, irradiation and mass flow rate at three levels. The optimal combination of levels for the aforementioned four factors was able to provide the efficiency of collector up to 70–80%.

Greig et al. [25] conducted a detailed experimental investigation on the flow behavior characteristics over the corrugated geometry for different Reynolds numbers in a TSC system using PIV measurements. It was found that the strong turbulence occurs even at the Reynolds number lying within the conventional laminar domain.

Gholampour et al. [26] performed a detailed experimental study under different operating conditions and evaluated the prototype performance of PV/Thermal flat TSC system according to the energetic, equivalent thermal and exergetic efficiencies. This system was designed, constructed and tested under weather conditions in Kerman, Iran.

3.3. Numerical simulations

3.3.1. CFD simulation tools

Computational Fluid Dynamics (CFD) is the most popular methodology and advanced computational technique in recent two decades to predict 2D/3D airflow and heat transfer in and around active building envelopes with high temporal resolution. CFD is a powerful alternative tool for studying convective heat transfer on TSCs, eliminating difficulties combined with the precise control of the experimental conditions [22]. CFD models can provide the thermal and airflow field details as well as valuable insights for novel TSC designs which optimize the heat extraction [22].

The major CFD simulation tools available in the market are ANSYS/Fluent and ANSYS/CFX. Other CFD simulation tools include TASCflow, STAR-CCM+ etc [12,20]. GAMBIT and ICEM are employed for all geometry models and meshing. In addition, TRNSYS is another major simulation tool applied widely [50] for the simulation investigation of TSC. Lots of researchers have already applied commercial CFD code to investigate thermal performance of TSC in last five years [14,18,20,22,24,26,27,31].

Li et al. [14] predicted cavity exit air temperature, plate surface temperature with weather and design parameters and made energy prediction of UTCs with and without PV modules. Li et al. [18] in 2014 also investigated various important factors which affect the performance of corrugated UTCs with and without PV panels (BIPV/T). Tajdaran et al. [20] developed a comprehensive CFD model for TSC systems using STAR-CCM+ commercial software. It was found that HEE is at its highest value when TSC system operates at the lowest suction ratio with lowest wind speed. Li et al. [22] in 2013 analyzed the convective heat transfer processes for both flat and corrugated UTCs. It was concluded that the level of turbulence intensity is less significant than the presence of perforations in the approach flow for flat UTCs. But, for corrugated UTCs, the perforation dimensions play the less significant role in the system performance than incident turbulence intensity. Collins et al. [24] evaluated the effectiveness in the asymptotic region and heat loss due to wind in the starting region of an unglazed transpired solar collector with a trapezoidal corrugation. Gholampour et al. [26] developed a predictive model and validated and investigated the PV/Thermal flat transpired plate capabilities. Badache et al. [27] investigated thermal efficiency of a UTC for two plenum thickness (5 cm and 15 cm). The conclusion was drawn that both thermal efficiency of UTC and absorber plate effectiveness are relatively insensitive to the thickness of the plenum. Gao et al. [31] analyzed the heat transfer characteristic in UTC using CFD methodology. It is found that the hole pitch effect on heat transfer in UTC is important, while the hole diameter effect is less noticeable.

3.3.2. Turbulence models

It is very significant to select a suitable turbulence closure model to predict specific flow features and thermal fields for active building envelopes using CFD methodology. Li et al. evaluated the performance of five potentially suitable turbulence closure models including...
Renormalization Normal Group $k$-$ε$ (RNG $k$-$ε$), Standard $k$-$ε$, Shear Stress Transport $k$-$ω$ (SST $k$-$ω$), Realizable $k$-$ε$ and Reynolds Stress Model (RSM) applied in simulating both flat and corrugated UTCs [22]. Table 2 summarized the turbulence models for TSC simulation investigation since 2012. Observed from Table 2, the RNG $k$-$ε$ is the most popular turbulence model selected by all those researchers.

Table 2 illustrates the comparison of all those five turbulence models for simulating UTCs. Seen from Table 3, it is found that the RNG $k$-$ε$ has the best overall performance, which not only provides the most accurate and consistent results with economic computing cost, but also has more stable convergence. The Standard $k$-$ε$ and SST $k$-$ω$ models could have moderate accuracy, however, which are not as good as the RNG $k$-$ε$ model. It is not recommended to use Realizable $k$-$ε$ and RSM models for simulating UTCs due to their unsatisfactory overall performance e.g. inaccurate predicting results and higher computing cost [22].

### 3.3.3. Validation of numerical models

It is very necessary to validate the CFD numerical models and programs before it is extensively adopted for the scientific investigation [51–54]. Normally, the CFD simulation is validated against with experimental data under the same meteorological and operating conditions [26]. Li et al. [22] validated the CFD model based on two aspects including the airflow and thermal field for the flat UTC model and corrugated UTC model. The non-isothermal condition was considered for the flat UTC model because of limited available experimental data. The agreement of the plate surface temperature between CFD simulation results and experimental data are satisfactory due to error within 2 K; however, for the corrugated UTC model, both non-isothermal and isothermal cases have been taken into account and modeled. The agreement of stream-wise mean velocity and turbulent kinetic energy between the CFD results and experimental data is very good in the near wall region i.e. within $0.03\,m$ [22].

Tajdaran et al. [20] performed several varieties of validation cases (15 cases) including diverse ambient air temperature, wind speed and angle, suction ratio, and vertical solar radiation, which imply that the sophisticated CFD model used in their work is able to deliver a realistic and accurate representation of the TSC system performance. In addition, it is found that CFD model is flexible in nature and well suitable to capture the effect of different climatic conditions [20]. (Fig. 4).

Authors used PIV data acquisition software to study complex fluid dynamics. It has been claimed that using CFD model and experimentation, a proposed geometry can increase the performance of conventional UTSC with almost 40% increase in thermal efficiency.

Hall et al. [50] carried out a two-fold model (TRNSYS) validation process for a novel flat cassette-panel TSC (CP-TSC) according to available experimental data from 95 days monitoring. The absorber plate temperature and plenum outlet temperature have been compared between simulation results and measured values. It was concluded that CP-TSC TRNSYS component is able to predict the daytime outlet air temperature (active outlet) with a high accuracy. The model is also good to predict the metal surface temperature when there is no flow through the perforations (bypass metal).

Collins et al. [24] validated their numerical model through comparing the simulation results of the absorber plate effectiveness with experimental values produced by Kutscher [9] and Van Decker et al. [47]. Their numerical simulation results slightly overestimate the effectiveness of the absorber plate at around 7%, although the difference appears in perforation shape.

Gholampour et al. [26] performed the validation of their numerical model for two PV/Thermal flat transpired plate systems (Fig. 5). One is with a PV panel (46% PV coverage) placed at the beginning of the transpired plate; another one is with two PV panels (92% PV coverage) located at the beginning and the end of the transpired plate. It is concluded that, for each validation case, there is a good agreement between the simulation and measured results for the mean PV panel temperature, outlet air temperature, and resistive load power.

Badache et al. [27] carried out various experimental measurements under different conditions including three air mass fluxes (0.0133, 0.0266 and 0.0411 kg/s/m²), three irradiance levels (300, 450 and 600 W/m²), and two plenum thicknesses (5 and 15 cm) in order to validate the CFD numerical models. In addition, the uncertainty of the experimental measurements have been also considered and investigated. The predicted collector efficiency from the numerical results lied within the uncertainty range for most of the measured efficiency. Particularly, for the cases of two plenum thicknesses, there was a good agreement between the measured values and simulation results at medium and high mass fluxes (0.0266 and 0.0411 kg/s/m²). But, for the case of low mass flux (0.0133 kg/s/m²), the numerical model overpredicted the experimental results. In addition, a minor difference was found for low (300 W/m²) and medium (450 W/m²) irradiance levels, however, an obvious difference appeared at high irradiance level (600 W/m²). The reason of the disagreement is mainly due to a fact that the sky temperature is very hard to know accurately. Other disagreement reasons include temperature variations in the laboratory (estimated at ±1.5 °C) and air movement resulted from the ventilation system.

Li et al. [14] measured the following parameters for the numerical model validation: the air temperature at the cavity exit, the surface temperature of four PV panels, and the representative temperature of the absorber plate at around 7%, although the difference appears in perforation shape. For the case of low mass flux (0.0133 kg/s/m²), the numerical model overpredicted the experimental results. In addition, a minor difference was found for low (300 W/m²) and medium (450 W/m²) irradiance levels, however, an obvious difference appeared at high irradiance level (600 W/m²). The reason of the disagreement is mainly due to the fact that the sky temperature is very hard to know accurately. Other disagreement reasons include temperature variations in the laboratory (estimated at ±1.5 °C) and air movement resulted from the ventilation system.

<table>
<thead>
<tr>
<th>Turbulence model</th>
<th>RNG $k$-$ε$</th>
<th>Standard $k$-$ε$</th>
<th>SST $k$-$ω$</th>
<th>Realizable $k$-$ε$</th>
<th>RSM</th>
<th>Others</th>
</tr>
</thead>
<tbody>
<tr>
<td>Li et al. [14]</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Li et al. [18]</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tajdaran et al. [20]</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Li et al. [22]</td>
<td>×</td>
<td>×</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Collins et al. [24]</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gholampour et al. [26]</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Badache et al. [27]</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gao et al. [31]</td>
<td>×</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2: Summary of turbulence models for TSC simulation investigation since 2012.
3.4. Parametric study

The aim of parametric sensitivity analysis is to assess the dependence of diverse parameters in the heat transfer and energy characteristics of TSC [12]. The parametric sensitivity study has been already carried out by lots of researchers in past 20 years. Table 4 illustrates the various parametric study and the corresponding investigation results in recent five years.

There are many very significant parameters including perforation/hole diameter, pitch and pitch-to-diameter ratios, plate thickness, plenum thickness, width and depth, ambient and inlet air temperature, absorber porosity, solar radiation, airflow rate, approach flow velocity, (absorber coating) absorptivity and emissivity, wind angle and speed, incident turbulence intensity, corrugation collector wavelength, amplitude, crest and slope length, cavity width, PV panel width and height, distance between PV panels, time of the day, suction ratio, suction velocity (m$^3$/s*m$^2$) (or air flow rate), PV coverage percent, the location of PV panel, and air mass flux.

It is found that the effect of hole diameter and pitch on the thermal performance of GTC is less noticeable [30]. On the contrary, the effect of hole diameter and pitch on the HEE, effective efficiency and pressure drop is significant [15,33]. Increasing hole diameter and pitch will decrease HEE [33]. The reason resulting in completely different effects is depending on local climatic conditions, time and site constraints, and the interaction between different factors. Seen from Fig. 6, for an airflow rate of 72 m$^3$/h*m$^2$ and solar radiation of 900 W/m$^2$, changing the pitch from 12 to 24 mm (its corresponding change in perforation diameter from 0.80 to 1.55 mm) leads to a drop of 5.5 °C in the air temperature rise. The similar air temperature drops also appear in other solar radiation including 650 and 400 W/m$^2$ with a constant airflow rate [46]. Fig. 7 illustrates HEE as a function of collector pitch and perforation diameter. As mentioned before, observing from Fig. 7, increasing perforation diameter and collector pitch will decrease HEE [33].

Thermal efficiency and the absorber plate effectiveness in UTC

<table>
<thead>
<tr>
<th>Turbulence model</th>
<th>Accuracy</th>
<th>Computing cost</th>
<th>Performance</th>
<th>Convergence</th>
<th>Recommendation for simulating UTCs</th>
</tr>
</thead>
<tbody>
<tr>
<td>RNG k-ε</td>
<td>Most accurate</td>
<td>Economic</td>
<td>Best overall</td>
<td>More stable</td>
<td>Yes</td>
</tr>
<tr>
<td>Standard k-ε</td>
<td>moderate</td>
<td>–</td>
<td>Good</td>
<td>More stable</td>
<td>Yes</td>
</tr>
<tr>
<td>SST k-ω</td>
<td>moderate</td>
<td>–</td>
<td>Acceptable</td>
<td>–</td>
<td>Yes</td>
</tr>
<tr>
<td>Realizable k-ε</td>
<td>inaccurate</td>
<td>–</td>
<td>Unsatisfactory</td>
<td>–</td>
<td>No</td>
</tr>
<tr>
<td>RSM</td>
<td>Bad</td>
<td>Much higher</td>
<td>Unsatisfactory</td>
<td>No</td>
<td>No</td>
</tr>
</tbody>
</table>

Fig. 4. PIV data acquisition software used by Tajdaran et al. [20].

Fig. 5. Energy flow diagram of PV/Thermal flat transpired plate collector [26].
### Table 4
Comparison of parametric study and the corresponding investigation results.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Study/Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effect of perforation/hole diameter and pitch</td>
<td>Effective efficiency increases with increasing perforation diameter and with decreasing pitch [15].</td>
</tr>
<tr>
<td>Effect of plenum thickness and depth</td>
<td>Effective efficiency increases with shrinking plenum thickness [15].</td>
</tr>
<tr>
<td>Effect of ambient and inlet air temperature</td>
<td>Effective efficiency increases with decreasing inlet air temperature and with increasing ambient temperature [15].</td>
</tr>
<tr>
<td>Influence of solar radiation</td>
<td>Solar radiation ranges from 150 W/m² to 600 W/m² at 0° incidence angle (parallel flow) to the TSC is more effective than that at 67.5° incidence angle [20].</td>
</tr>
<tr>
<td>Effect of airflow rate</td>
<td>Airflow rate has strong effect on exit air temperature [30]. Air mass flow rate (kg/s) has the greatest impact on the efficiency of TTC [32].</td>
</tr>
<tr>
<td>Effect of (absorber coating) absorptivity and emmissivity</td>
<td>Effect of coating absorptivity is stronger than that of emissivity [15].</td>
</tr>
<tr>
<td>Effect of wind angle and speed</td>
<td>Winds with 45° incidence angle to the UTC have most significant impact on CHTC and HEE [17]. Increasing wind speeds from 1 m/s to 4 m/s will decrease both ‘temperature rise’ and ‘surface temperature’ [20]. Greater effect of wind speed on Heat Exchange effectiveness occurs/is found at higher suction ratios [20]. In theory, wind speed should have no influence on the calculated heat exchanger effectiveness [24]. Increasing wind velocity will increase heat loss and then decrease equivalent thermal efficiency and second law efficiency [33]. There is no effect of wind speed on the temperature rise per unit of solar radiation for the tested BPSC (Back-pass solar collector) within the range of 0.3 and 4 m/s [38].</td>
</tr>
<tr>
<td>Effect of incident turbulence intensity (TI)</td>
<td>Exterior Nu will increase with the turbulence intensity (TI), however, cavity exit air temperature and thermal efficiency will increase with increasing TI, particularly when increasing wind speed and decreasing suction velocity [18].</td>
</tr>
<tr>
<td>Effect of corrugation collector wavelength</td>
<td>Increasing wavelength from 0.2 m to 1.2 m will increase ( T_{\text{col,exit}} ) up to 10°C and enhance energy efficiency up to three times higher [18].</td>
</tr>
<tr>
<td>Effect of corrugation collector amplitude</td>
<td>Effect of amplitude is hard to determine, since the collateral response to the slope length change is not able to be distinguished [18]. Decreasing amplitude will increase thermal efficiency for a reasonable assumption, since decreasing amplitude will decrease slope length [18].</td>
</tr>
<tr>
<td>Effect of corrugation collector crest and slope length</td>
<td>Effect of crest length on ( T_{\text{col,exit}} ) is very minor [18]. Increasing crest length from 0.03 m to 0.11 m will enhance ( T_{\text{col,exit}} ) within 0.2 °C [18]. Slope length has the most important effect on the thermal performance [18].</td>
</tr>
<tr>
<td>Effect of cavity width</td>
<td>Decreasing cavity width will cause a moderate increasing of ( T_{\text{col,exit}} ) [18].</td>
</tr>
<tr>
<td>Effect of PV panel width and height</td>
<td>Effect of PV panel height is largest for UTCs with PV panels [18]. TSCs deliver different operating temperatures dependent on time of day [20].</td>
</tr>
<tr>
<td>Effect of suction ratio ( (V_{\text{ext}}/V_{\text{air}}) )</td>
<td>Effect of suction ratio increases at lower wind speeds [20]. Increasing both suction ratio and wind speed result in decreasing heat exchange effectiveness [20].</td>
</tr>
<tr>
<td>Effect of suction velocity in m³/s·m² (or airflow rate?)</td>
<td>Temperature behind the absorber decreases as the suction velocity increases [24]. For higher suction velocities, lower temperatures will cause higher heat transfer rates behind the absorber and lower heat exchanger effectiveness numbers for the absorber [24]. Increasing suction velocity will improve the first law efficiency but decrease the second law efficiency [26]. For GTC, the effect of suction velocity on thermal efficiency is not as noteworthy as that for UTC [30].</td>
</tr>
</tbody>
</table>

(continued on next page)
depends significantly on its performance from both the PV/T TSC system, increasing solar radiation has a desirable impact on the efficiency of the TTC [32]. Sensitivity for the plenum depth to affect the efficiency of TSC is very significant. Increasing the plenum depth will improve equivalent thermal efficiency, thermal efficiency and second law efficiency [33].

Increasing inlet air temperature will reduce effective efficiency, equivalent thermal efficiency and second law efficiency [15,33]. Similar conclusions are also drawn that there is an unfavorable impact on the PV/T TSC system performance for both first and second law efficiency when increasing ambient temperature [26].

The effect of solar radiation on the collector system performance depends significantly on different types of TSC and incidence angle. For the PV/T TSC system, increasing solar radiation has a desirable impact on its performance from both first and second law efficiencies [26]. Similar conclusions are also drawn that the equivalent thermal efficiency, second law efficiency and temperature gain will enhance with increase in solar radiation [33,38,39]. The efficiency of UTC is relatively insensitive to the irradiance [27]. A minor effect of solar radiation on thermal efficiency of GTC is found [30], however, increasing irradiation has a negative effect on the collector efficiency for another TTC [32]. Fig. 8 shows exit air temperature as a function of solar radiation and airflow rate [46]. Seen from Fig. 8, increasing solar radiation for various airflow rates, all of the exit air temperature increase linearly.

Airflow rate has a strong effect on exit air temperature, the efficiency of TTC and temperature rise [30, 32, 38 and 46]. Fig. 9 shows collector efficiency as a function of airflow rate. Observing from Fig. 9, the collector efficiency increases exponentially with the airflow rate for various models [63].

With increase in wind speed, temperature rise and surface temperature will decrease [20], meanwhile, heat loss, equivalent thermal efficiency and second law efficiency increase [33]. On the contrary, there is no influence of wind speed on the calculated HEE in theory [24] and temperature rise per unit of solar radiation for the tested BPSC [38].

Increasing suction velocity will enhance the first law efficiency but decrease the second law efficiency [26], meanwhile, improve mass flow rate and heat transfer [31,33]. For GTC, the impact of suction velocity on thermal efficiency is not as significant as that for UTC [30].

The aforementioned effects of various parameters can be also categorized based on their similar characteristics as follows [12,18]. Among these, the most important significant parameters are the wind speed and suction velocity in TSCs [9,12,22,17,55–57], except the ambient air temperature and solar radiation [18].

- **Effect of perforation geometry**: perforation/hole diameter and pitch, pitch-to-diameter ratios, hole shape and pattern [15,21,30,31,33].
- **Effect of TSC configuration and its physical characteristics**: plate thickness, plenum thickness, width and depth [15,26,33,34], absorber porosity, (absorber coating) absorptivity and emissivity [15], PV coverage percent [26], corrugation collector wavelength, amplitude, crest and slope length [18], cavity width [18], PV panel width and height [18], distance between PV panels, color of coating.
- **Effect of climatic conditions**: ambient and inlet air temperature [15,26,33,39], solar radiation [20,26,27,30,32,33,38,39], approach flow velocity, wind angle and speed [17,20,24,33,38], incident turbulence intensity [18], air mass flux [27].
- **Effect of time and site constraints**: time of the day [20], the location of PV panel [26].

### Table 4 (continued)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Study/Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effect of PV coverage percent</td>
<td>There is a significant impact of the PV coverage percent on the systems’ (PV/T) performance [26]. Increasing coverage percent of PC has a positive impact on the second law efficiency [20].</td>
</tr>
<tr>
<td>Effect of the location of PV panel</td>
<td>There is a very small impact of the location of PV panel on the systems’ (PV/T) performance [26].</td>
</tr>
<tr>
<td>Effect of air mass flux (kg/s/m²)</td>
<td>With increasing air mass flux, the efficiency of UTCs improves [27]. With increasing air mass flux, the heat transfer predominance shifts progressively to the back-side of the absorber plate [27]. The net heat loss from the absorber plate to the surroundings increases with decreasing mass flux [27].</td>
</tr>
</tbody>
</table>
Since decreasing temperature difference between outlet and ambient air temperature or increasing air mass flux, when maintaining solar irradiance as a constant, will improve TSC thermal efficiency. Meanwhile, TSC thermal efficiencies reach the maximal value 89% when the temperature difference is minimal (8 K or 12 K for solar irradiance with 300 or 450 W/m², respectively) or air mass flux is maximal 0.0411 kg/s.

4. Conclusions and future work

In recent five years, various efficiency models of TSCs combined with PV modules have been developed not only from the perspective of energy, but also exergy. As advanced CFD numerical techniques develop, diverse turbulence models have been employed in analysis, evaluation and optimization on heat and airflow mass transfer, thermal efficiency, heat exchange efficiency, exergy efficiency, energy characteristics of TSCs integrated into other building services. The objective of this work has been met via following main valuable and scientific results found by intensive review of literature in past five years:

1) The most important significant parameters affecting TSC efficiency are the wind speed and suction velocity in TSCs. However, effects of various parameters on TSC efficiency are completely different, depending on local climatic conditions, time and site constraints, and the interaction between different factors.

2) Exergy analysis becomes more significant and helpful for a better design of TSC with PV/T system, since PV/Thermal flat transpired collectors generate both thermal and electrical energy. The second law efficiency represents the quality of useful energy gains, since the energy quality is very important for TSC with PV/T system.

3) The turbulence model of RNG k-ε has the best overall performance, which not only provides the most accurate and consistent results with economic computing cost, but also has more stable convergence, compared to standard k-ε, SST k-ω, Realizable k-ε and RSM.

4) The TSCs could be integrated with other building technologies e.g., PV modules, heat pump, mechanical ventilation with heat recovery, PCM etc., which will significantly enhance whole building energy efficiency.

5) There are many concepts for illustrating different point of view in TSC efficiency including the collector efficiency, effective efficiency, first law efficiency (energy efficiency), second law efficiency (exergy efficiency), equivalent thermal efficiency and instantaneous efficiency of the BPSC.

Current project of Active-LIVing Envelopes (ALIVE) from EPSRC [6,7] is now in the process of WP1 (Work Package) for investigating energy generation, storage and background research. WP2 for mathematical modelling using CFD and MATLAB and WP3 for experimental testing and electrical simulation will be carried out later on.

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References


