Effect Of Enclosed Air Gap And Empty Gap For Thermal Screens In PV Cells Isolation Inside The Hybrid Photovoltaic-Thermal Channel On A Rooftop Designed For Natural Ventilation In Bioclimatic Buildings

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Abstract: The present paper reports numerical investigation results of mixed convection in the inclined Hybrid Photovoltaic-Thermal (PV/T) channel using the empty or enclosed air gaps for the solar PV cells isolation. The aim of this work is to evaluate streamlines and isotherms patterns in or between the top and the bottom of the chimney. The forces friction and end losses at the top and bottom of the chimney.

1 INTRODUCTION

In a low energy building design, Photovoltaic (PV) modules are nowadays integrated in the fabric of the buildings, functioning as Hybrid Photovoltaic/Thermal systems to provide natural ventilation and generate better and more electrical energy. The multi-functional component consists of a PV panel with a cavity (air gap) between the PV panel and the fabric of the buildings. The air gap is a natural draft channel and flow in the gap is driven by buoyancy (heat) and the induced wind pressure difference between the top and bottom of the channel. The forces opposing flow are friction and end losses at the top and bottom of the chimney.

by integrating the PV cells in a titled open channel; in which the air gap beneath the PV cells provides natural cooling of the PV cells and reduces its operational temperature. In fact, the PV cells’ efficiency is greatly dependent on solar cell temperature, and if solar cell temperature increases, solar cell efficiency decreases [1, 2, 3]. There is growing interest in studying heat and mass flow in air gaps behind photovoltaic panels [4, 5], and in the channels. Many studies have shown that, a Hybrid Photovoltaic/Thermal solar chimney is an excellent passive ventilation strategy to use in bioclimatic buildings for enhancing natural ventilation and providing thermal comfort for occupants. But, the cooling process due to the air flow rate in natural convection is directly correlated with the solar radiation intensity that controls air circulation inside the channel. In high solar radiation regions, air flow rate inside the channel is weak due to the dominated natural and recirculation cells above the open lines. Reverse flow near the outlet reduces mass flow rate through the air gap. Indeed, many works have shown that air circulation inside the channel is very weak at low Grashof number and a further increase of Grashof number increases the air recirculation zone and backflow inside the channel. Hence Raji et al.[6], in a numerical study of a mixed convection problem in a ventilated cavity submitted to a constant heat flux on one of its vertical walls, have observed that more increase of Raleigh number (Ra) leads to more significant cells above the open lines causing a visible tightening of the lines and showing a net domination of the natural convection effect. Then E. Bilgen et al. [7] observed in their study of natural convection heat transfer in a partially open inclined square cavity, that flow rate increases rapidly with increasing Ra up to $10^8$. They concluded that at low Ra, heat transfer is conduction dominated and becomes convection dominated for Ra $> 10^6$. Tan Boom et al.[8] observed in a solar roof collector that backflow at the outlet of the air gap reduces the total mass transfer of air through the air gap. Hence S.L. Sinha et al.[9] reported in the numerical simulation of a two-dimensional room air flow with and without buoyancy that the intensity of the recirculation zone in the cavity increases as Grashof function.
number Gr increases to $10^8$. On the basis of the literature review, it clearly appears that the decrease in mass flow rate is a result of increasing backflow at the outlet opening of the air gap observed at high Raleigh or Grashof numbers due to high solar radiation. Consequently, in high solar radiation regions, Photovoltaic cells in a Hybrid Photovoltaic/Thermal Solar Chimney have to be cooled and/or thermally isolated. An innovative design process recommends the use of an isolated material. Enclosed air gap, empty gap, and a phase change material appear as effective isolation technologies for PV cell systems. However, no work was reported on mixed convection in the hybrid photovoltaic-thermal chimney with the empty gap or with the enclosed air gap integrated on the roof of buildings. None of the surveyed papers however showed the interrelated influence of thermal efficiency or electrical efficiency in the hybrid solar chimney using the empty gap and enclosed air gap isolation process. Thereafter, due to the practical interest of this problem in a wide variety of engineering applications of passive cooling, the subject needs further effort to improve knowledge in this field. The objective of the present study is to analyze the effect of empty gap or the effect of the enclosed air gap for PV cell isolation on the heat and flow patterns in a rooftop hybrid photovoltaic-thermal channel for natural ventilation in bioclimatic buildings. The present paper consists in numerically studying, a mixed convection problem in a ventilated heated inclined photovoltaic-thermal chimney with the empty isolated gap or the enclosed air gap at the top front covered glass plate with a constant heat flux. In this analysis the forced air flow enters the solar chimney through an inside opening size and leaves from the outlet opening size. Then it is very important to know the air movement or temperature distribution inside these proposed solar chimneys, and the thermal and electrical efficiencies of the devices.

2 PROBLEM FORMULATION

2.1 Physical Model and Governing Equations

In the present work, a scale analysis and a numerical study of the combined convective and radiation heat transfer in an inclined two-dimensional channel with constant heating flux are presented for the range of conditions applicable to temperature regulation of rooftop mounted PV panels. The two-dimensional analysis is restricted to the channel for which the range of the Aspect ratio is $2 \leq A \leq 10$. The governing two-dimensional conservation equations for laminar flow in an open ended inclined channel with isolated empty gap or enclosed air gap subject to the Boussinesq approximation. Under these assumptions, the dimensionless governing equations, written in terms of vorticity and stream function formulation, in fluid flow and in an enclosed air gap are as follows:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (1)$$

$$\frac{\partial \omega}{\partial \tau} + U \frac{\partial \omega}{\partial X} + V \frac{\partial \omega}{\partial Y} = \text{Ri}(\sin \alpha \frac{\partial \theta}{\partial X} + \cos \alpha \frac{\partial \theta}{\partial Y}) +$$

$$\left(\frac{1}{Re}\right)(\frac{\partial^2 \omega}{\partial X^2} + \frac{\partial^2 \omega}{\partial Y^2}) \quad (2)$$

$$\frac{\partial \theta}{\partial \tau} + U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} =$$

$$\left(\frac{1}{Re.Pr}\right)(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}) \quad (3)$$

$$\omega = -\left(\frac{\partial^2 \Psi}{\partial X^2} + \frac{\partial^2 \Psi}{\partial Y^2}\right) \quad (4)$$

The stream function and vorticity are related to the velocity components by the following expressions:

$$U = \frac{\partial \psi}{\partial Y} \; , \; V = -\frac{\partial \psi}{\partial X} \; \text{and} \;$$

$$\omega = \frac{\partial V}{\partial X} - \frac{\partial U}{\partial Y} \quad (5)$$

Where the scales are defined:

$$(X, Y) = \left(\frac{x}{\epsilon}, \frac{y}{\epsilon}\right) ; \; (U, V) = \left(\frac{u}{v_0}, \frac{v}{v_0}\right)$$

$$\Psi = \frac{\psi}{\nu_0 \epsilon} \; ; \; \omega = \frac{\Omega e}{v_0} \; ; \; \theta = \frac{\lambda(T - T_u)}{\phi e} \quad (6)$$
\[ \theta_s = \frac{\lambda(T_s - T_a)}{\phi e}; \quad \text{Re} = \rho v_0(2e_f) / \mu; \]

\[ Gr = \rho^2 g \beta \phi e^4 / \lambda \mu^2 \]

and the aspect ratio expressed as:

\[ A = L / e; \quad A_f = e_f / e; \quad B = e_0 / e \]

### 2.2 Boundary Conditions

- Initial conditions: at \( \tau = 0 \):

\[ \theta = \omega = U = V = \Psi = 0 \]

(7)

The boundary conditions, associated with the problem are as follows:

- At the inlet of the channel: \( Y = 0, \; B \leq X \leq 1 \)

\[ \theta = \omega = U = 0 ; \quad V = 1; \quad \Psi = -X + 1 \]

(8)

- At the outlet of the channel: \( Y = A, \; B \leq X \leq 1 \)

\[ U = 0; \quad (\partial \omega / \partial Y)_{Y=A} = 0; \quad (\partial V / \partial Y)_{Y=A} = 0; \]

(9)

\[ (\partial \Psi / \partial Y)_{Y=A} = 0 \]

If \( V \geq 0 \) then \( (\partial \theta / \partial Y)_{Y=A} = 0 \)

If \( V < 0 \) then \( \theta = 0 \)

- At the bottom of the enclosed air gap: \( Y = 0, \; 0 \leq X \leq B \)

\[ (\partial \theta / \partial Y)_{Y=0} = 0, \quad \omega = -(\partial^2 \Psi / \partial Y^2) \bigg|_{Y=0}; \quad V = U = \Psi = 0 \]

(10)

- At the top of the enclosed air gap: \( Y = A, \; 0 \leq X \leq B \)

\[ (\partial U / \partial Y)_{Y=A} = 0; \quad (\partial V / \partial Y)_{Y=A} = 0; \]

(11)

\[ (\partial \omega / \partial Y)_{Y=A} = 0; \quad (\partial \Psi / \partial Y)_{Y=A} = 0 \]

On rigid plates of the empty gap: \( 0 \leq Y \leq A \) At \( X = 0 \):

\[ \theta_{gl} = (\lambda \alpha_{gl} / e + \lambda q_{rsl, PV} (\phi e) + h_r \theta_s) / (h_{CV} + h_r) \]  

(13)

At \( X = B : U = 0; \; V = 0; \; \Psi = -B + 1 \)

\[ \omega = -(\partial^2 \Psi / \partial X^2) \bigg|_{b} \]

(14)

(\partial \theta / \partial X)_{X=b} = (\eta_{sl} - \tau_s \alpha_{PV}) - (q_{rsl, PV} + q_{s, PV} / \phi) \phi \]  

At \( X = 1; \) \[ \omega = -(\partial^2 \Psi / \partial X^2) \bigg|_{X=1} \]

\[ U = V = \Psi = 0 - (\partial \theta / \partial X) \bigg|_{X=1} = q_{s, PV} / \phi \]  

(15)

With:

\[ q_{rsl, PV} = \sigma(T_{PV}^4 - T_{sl}^4)/(1/\varepsilon_{PV} + 1/\varepsilon_{sl} - 1); \]

\[ q_{PV, s} = \sigma(T_{PV}^4 - T_{PV}^4)/(1/\varepsilon_{PV} + 1/\varepsilon_{P} - 1) \]

(16)

\[ T_S = 0.0552T_a^{1.5}; \; h_r = 4 \alpha e \gamma (T_{sl} + T_S / 2)^3 \]

A wind convection coefficient \( h_{CV} \) is estimated from wind velocity by Eq. (17).

\[ h_{CV} = 5.67 + 5.86V_a \]

(17)

### 2.3 Evaluation of Model Characteristics

From the engineering viewpoint, the most important concern is heat transfer through the PV cell modules, and the heated plates. These are best represented by the Nusselt number, which is a measure of the ratio of heat transfer by conduction to the flux convected by fluid flow. The local Nusselt number on the back plate and the inner side PV cells plate of the channel are given by:

\[ Nu_{pv} = \alpha e / (\lambda(T_p - T_a)) = 1/\theta_{pv} \]

(18)

The thermal and the electrical efficiencies of the hybrid photovoltaic-thermal collector are respectively given as follows:

\[ \eta_{sl} = \eta_{ref} + \beta \varepsilon_{PV}(T_{PV} - 298) + \gamma \log(\phi / 1000) \]

(19)

\[ \eta_{th} = D_m C_p(T_{out} - T_{in}) / (\phi L) \]

The mass flow rate in the channel is expressed

\[ \dot{m} = (1/\rho)(T_{PV} / y) dy \]  

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\[ D_m = (\rho v_0 / (1 - B)) \int_B^1 V(X, A) dX \]  
(20)

Where: \( T_{\text{pv}} \) is the average absolute temperature of the PV plate, \( T_{\text{out}} \) and \( T_{\text{in}} \) are respectively the outlet and the inlet absolute temperature of the air in the channel.

3 NUMERICAL PROCEDURE

3.1 Method of solution

The governing nonlinear partial differential equations, (1-3), were discretized using a finite difference technique. The first and second derivatives of the diffusive terms were approached by central differences while a second order upwind scheme was used for the convective terms to avoid possible instabilities frequently encountered in mixed convection problems. The integration of equations (2) and (3) was assured by the Thomas algorithm. At each time step, the Poisson equation, Eq. (4), was treated by using the Point Successive Under-Relaxation method (PSUR) with an optimum under-relaxation coefficient equal to 0.8 for the grid (51 x 101) adopted in the present study. Convergence of iteration for stream function solution is obtained at each time step. The following criterion is employed to check for a steady-state solution.

\[
\sum \left( \phi_{i,j}^{n+1} - \phi_{i,j}^{n} \right) / \phi_{i,j}^{n} < 10^{-5}
\]

where \( \phi \) stands for \( \Psi, \theta, \omega, n \) refers to time and i and j refer to space coordinates. The time step used in the computations is \( \Delta \tau = 10^{-5} \). Grid independency solutions are assured by comparing different grid meshes for the highest Grashof and Reynolds numbers used in this work (Gr = 10^3 and Re = 300). It was found that the differences between meshes of 121 x 121 and 141 x 141 were not significant for all variables. The results obtained with these grids were comparable to those obtained with a non-uniform grid 51 x 101. Thus, a non-uniform mesh of 51 x 101 was selected. For the computing, the input parameters listed in Table 1 were used. The vortices computational formula of Woods [10] for approximating wall vorticity was used:

\[ \omega_p = (1/2) \omega_{p-1} - 3(\Psi_{p+1} - \Psi_p) / \Delta \eta^2 \]

where \( \Psi_p \) and \( \Psi_{p+1} \) are stream function values at points adjacent to the boundary; \( \eta \) the normal abscise on the boundary wall.

3.2 Validation

The numerical code was validated against the results of Orhan Aydin et al. [11] obtained in the case of a square cavity differentially heated. For comparisons, made in terms of streamline and isotherm patterns, explored in the cavity for the Raleigh number \( Ra = 103 \) or \( Ra = 104 \), a stretched 41 x 41 grid was used with \( Pr = 0.71 \) for pure natural convection (Re=1) showed fairly good agreement, Fig. 2.

4 RESULTS AND DISCUSSION

4.1 Method of solution

In the solar chimney, temperature distribution and flow fields of the inlet jet and buoyancy force which induce natural convection have been analyzed numerically. As reviewed in the literature and shown by S.L. Sinha et al. [9], the results in fig. 3 show that when Reynolds number Re=20, the recirculation cells or the back flow phenomenon appears more important in the channels. This tendency is more significant under natural convection and leads to the intense heating of the PV cells in the PV/T collector reducing its electrical efficiency. As shown by the streamlines in Fig. 3, air circulation inside the channel is very weak at the low Grashof number and increases with Grashof number. High insulation effect increases air circulation inside the channel and the isotherms become more distorted. Fig. 4 indicates fluid flow patterns inside the channel. As shown, the streamline contours exhibit the circulation patterns as the fluid motion which is affected by PV cell sources and the adiabatic plate.

![Comparison of streamlines and isotherms](image)

### TABLE 1

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absorptivity of the glass, ( \alpha_g )</td>
<td>0.05 (-)</td>
</tr>
<tr>
<td>Transmittance of the glass, ( \tau_g )</td>
<td>0.95 (-)</td>
</tr>
<tr>
<td>Absorptivity of PV cells surface, ( \alpha_{PV} )</td>
<td>0.89 (-)</td>
</tr>
<tr>
<td>Transmittance of PV cells surface, ( \tau_{PV} )</td>
<td>0.09 (-)</td>
</tr>
<tr>
<td>Absorptivity of back plate, ( \alpha_p )</td>
<td>0.05 (-)</td>
</tr>
<tr>
<td>Width of the channel, ( e )</td>
<td>0.05 (m)</td>
</tr>
<tr>
<td>Empty gap/enclosed air gap width, ( e_0 )</td>
<td>0.02 (m)</td>
</tr>
<tr>
<td>Length of the channel, ( L )</td>
<td>0.30 (m)</td>
</tr>
<tr>
<td>Fluid flow duct width, ( e_i )</td>
<td>0.03 (m)</td>
</tr>
<tr>
<td>Slope of collector, ( \alpha )</td>
<td>( \pi/6 ) (rad)</td>
</tr>
</tbody>
</table>

The flow characteristics show the simultaneous existence of the natural convection back flow cells at the outlet of the air flow duct and the open lines of the forced convection for the low values of the Reynolds number. The direction of flow, due to the thermal buoyancy force, is clockwise in this geometry as observed in the channels. The plot shows that from the bottom...
of the channel, the inlet air flow converges especially towards the active PV cells which are heated by a uniform flux, and the buoyancy forces drive the heated air to the outlet. This situation indicates that mixed convection is the best transfer mode which is able to extract the maximum excess heat of the solar PV cells in the hybrid Photovoltaic-Thermal chimney perhaps for the

Fig. 3. Distribution of the flow and the temperature for various Grasof number

Fig. 4. Comparison of streamline patterns in the channels. Simple collector (a), Enclosed air gap thermal screen collector (b), Empty gap thermal screen collector (C). $\Theta = 3000\, W.m^{-2}$, $Gr = 10^6$, $A = 6$, $B = 40\%$. 
Heating or the passive cooling of the buildings. Naturally, when the Reynolds number increasing the open lines appear indicating the establishment of the forced flow as shown by A. Raji, et al [6]. Otherwise, the outlet airflow velocity increases and reaches a maximal value at the middle of the fluid flow duct before decreasing to attain the minimal value near the insulated back wall as shown in Fig.5. Particularly, the results in figs.4 and 6 show the recirculation cells corresponding to the back flow phenomenon in most parts of the channel; which revealed itself to be more intense in the simple collector than in the isolated collectors. As shown in fig.6, the empty and enclosed air gaps significantly improve air draught by reducing backflow in the channel. The corresponding isotherm plots are presented in fig. 7. The isotherm patterns are much bifurcated and illustrate the effects of mixed convection in the channel. This tendency justifies the fact that mixed convection is the real mode of heat transfer which can decrease the temperature of the PV cell in the hybrid Photovoltaic-Thermal channel; and can provide the best electrical efficiency. Figs. (8-9) plot the dimensionless temperature variation along the PV cell plate and the variation of the temperature gap between the active plates along the channel. The dimensionless temperature is relatively low near the channel entrance where the thermal boundary layer is thin and the dimensionless temperature increases quickly as the boundary layer thickness increases along the PV cell plate fig.8. In the case of the simple collector, the dimensionless temperature gap between the two active plates increases along the channel whereas it decreases after reaching a maximum value because of the greater heated air flow removal out of the channel.

The empty gap as well as the enclosed air gap screens significantly affects the thermal parameter of the proposed Hybrid Photovoltaic-Thermal collector. In fact, fig.9 shows that the temperature along the heated PV cells plate is a decreasing function of the enclosed air gap width. These results show that the initial temperature along the heated PV cells plate in the simple collector (B = 0) is reduced to 40% in the filled air gap thermal screen collector (B =25%), while it is reduced to 22% in the empty gap thermal screen collector for the same width. The Local Nusselt number is another characteristic parameter in convective heat transfer problems. Its variation along the active walls is plotted in Figs.10 and 11 respectively along the PV cells plate and the back heated plate. In general, the Nusselt number at the PV cells plate and the hot plate starts with a high value at a point close to the bottom and decreases monotonically to a small value at the top. The local Nusselt number is higher near the channel entrance where the boundary layer thickness is very small and temperature difference between surface and inlet air temperature is very small. If the boundary layer thickness
increases, the temperature difference increases, which causes the local Nusselt number to decrease.

Fig. 7. Comparison of isotherm pattern in the channels. Simple collector (a), Enclosed air gap thermal screen collector(b), Empty gap thermal screen collector (c). $\phi = 3000 \text{ W} \cdot \text{m}^{-2}$, $Gr = 10^6$, $A = 6$, $B = 40\%$.

Fig. 8. Distribution of the temperature along the PV Cells plates.

Re=5 $Gr=10^6$

Fig. 9. Distribution of the temperature gap between the PV cells and the back plates inside the channel.

Re=20

Fig. 10. Variation of the local Nusselt number along the PV cells plate.
4.2 Electrical and thermal performances
As shown in fig.12, the electrical efficiency of the solar PV cells is an increasing function of the Reynolds number in the enclosed air gap collector as well as in the empty gap collector. Similar results were obtained by K. Sopian et al.[12] and Garg, H. P. et al.[1]. The empty gap as well as the enclosed air gap screens significantly affects the thermal and electrical efficiencies of the proposed Hybrid Photovoltaic-Thermal collector. The mass flow rate, fig.5, and the thermal and electrical efficiencies increase more in the isolated collectors than in the simple collector. For the same value of B (B= 40%), the thermal efficiency of the empty gap collector is better than the thermal efficiency of the enclosed air gap collector, while the electrical efficiency in the enclosed air gap screen collector is higher than the electrical efficiency in the empty gap screen collector and in the simple collector (B=0%). In fact, as shown in fig. 8, the temperature along the heated PV cells plate is reduced to 40% using the enclosed air gap thermal screen collector, while it is reduced to 22% in the empty gap thermal screen collector for the same width (B =25%). The solar PV cells are less heated, consequently the electrical efficiency is better in the channels with enclosed air gap width or empty gap width than the electrical efficiency in the simple collector, whereas thermal efficiency is better in the empty gap collector than in the enclosed air gap thermal screen collector.

5 Conclusion
The numerical investigation in this study allowed the authors to know the mode of air flow in the hybrid photovoltaic-thermal channel with the empty gap or with the enclosed air gap for the lowest or the highest value of Reynolds number. The air flow role is to extract excess heat from the solar PV cells in the channel, to provide both thermal and electrical energy simultaneously. The empty gap or the enclosed air gap insulates the solar PV cells in these collectors. Within the investigated parameter ranges, the following conclusions can be drawn:
- The mass flow rate, the thermal and the electrical efficiencies increase in the isolated collectors than in the simple collector.
- The back flow phenomenon is more intense in the simple collector than in the isolated collectors.
The empty and the enclosed air gaps improve air draught significantly by reducing backflow in the channels.

Electrical efficiency and mass flow rate are increasing functions of enclosed air gap width in the proposed collectors.

Future work will be to integrate the empty gap or the enclosed air gap collector in the building fabric and use such a modelling to estimate simultaneously thermal comfort in the room, the thermal and electrical efficiencies.

**NOMENCLATURE**

- $C_p$: Specific heat (J. kg$^{-1}$.K$^{-1}$)
- $e_0$: Empty gap width / enclosed air gap width (m)
- $e_1$: Fluid flow duct width (m)
- $E$: Collector width ($e_0 + e_1$)(m)
- $A$: Aspect ratio of the channel ($A = L/e$)
- $A_F$: Aspect ratio of the fluid flow duct ($A_F = e_0/e$)
- $B$: Aspect ratio of the empty gap/enclosed air gap ($B = e_0/e$)
- $D_m$: Mass flow rate (Kg.s$^{-1}$)
- $De$: Aspect ratio of the empty gap/enclosed air gap ($B = e_0/e$)
- $q_{r1}$: Net radiative flux between PV plate and glaze (W.m$^{-2}$)
- $q_{r2}$: Net radiative flux between adiabatic plate and PV plate (W.m$^{-2}$)
- $g$: Gravitational acceleration (m.s$^{-2}$)
- $h_{Lv}$: Local convective heat transfer coefficient (W. m$^{-2}$.K$^{-1}$)
- $h_r$: Local radiative heat transfer coefficient (W. m$^{-2}$.K$^{-1}$)
- $L$: Length of the channel (m)
- $n$: Coordinate in normal direction
- $t$: Time (s)
- $T$: Temperature (K)
- $ΔT$: Difference of temperature (K)
- $T_a$: Ambient air temperature (K)
- $V_a$: Wind velocity (m.s$^{-1}$)
- $U$, $V$: Velocity component in x and y directions (m.s$^{-1}$)
- $U$, $V$: Dimensionless velocity component in X and Y directions; $U = u/u_v$, $V = v/v_v$
- $v_0$: Air inlet velocity (m.s$^{-1}$)
- $W_s$: Air outlet velocity (m.s$^{-1}$)
- $x$, $y$: Coordinates defined in fig. 1 (m)
- $X$, $Y$: Dimensionless spatial coordinates; $X = x/e$, $Y = y/e$
- $Re$: Reynolds number: $Re = \frac{\rho u_e (2x)}{\mu}$
- $Pr$: Prandtl number: $Pr = \mu C_p / \lambda$
- $Nu$: Nusselt number: $Nu = \frac{\theta_e}{\theta_{gl}} = \frac{1}{\eta}$
- $Gr$: Grashof number: $Gr = \frac{\beta \Delta T e^3}{\nu^2}$
- $Ri$: Thermal Richardson number ($Ri = Gr / Re^2$)

**Greek symbols**

- $θ$: Dimensionlesstemperature = $\frac{\Delta(T-T_{atm})}{\varepsilon_0}$
- $θ_θ$: Dimensionles sky temperature $θ_θ = \frac{\Delta(T-T_{atm})}{\varepsilon_0}$
- $τ$: Dimensionless time $τ = \frac{v_{e1}}{e}$
- $ψ$: Dimensionless stream function: $ψ = \frac{\psi}{v_e}$
- $ω$: Dimensionless vorticity: $ω = \frac{dv_e}{v_e}$
- $Ω$: Vorticity (s$^{-1}$)
- $ψ$: Stream function (m.s$^{-2}$)
- $\beta$: Thermal expansion coefficient (K$^{-1}$)
- $η_{th}$: Thermal electrical efficiency (-)
- $η_{pr}$: PV cell electrical efficiency at standard conditions(-)
- $η_{pv}$: PV cell electrical efficiency coefficient (-)
- $β_{pv}$: PV cell temperature coefficient (K$^{-1}$)

**REFERENCES**
