INVESTIGATION OF WIND HEAT LOSS FROM UNGLAZED TRANSPIRED SOLAR COLLECTORS WITH CORRUGATION

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ABSTRACT

Unglazed transpired solar collectors (UTSCs) preheat makeup ventilation air for buildings, and thereby reduce energy used for heating. The performance of the UTSC is dependent on by the heat loss from the absorber plate, which is strongly affected by the wind speed. To date, correlations to determine UTSC heat loss are limited to sinusoidal corrugations. In reality, trapezoidal corrugations have been added to UTSCs to provide them with structural stiffness. These corrugations prevent the attachment of the flow to the absorber plate and cause flow separation which increases the heat loss. In this study, a numerical simulation of a UTSC with trapezoidal shape corrugation has been performed to investigate the effects of wind and suction velocity on the UTSCs performance. It has been found that heat loss from new configuration is up to three times greater than the heat loss from a perforated flat absorber plate. A correlation of heat loss due to the wind speed and suction velocity was developed.

Keywords: solar collector, Transpired, Unglazed and Heat transfer.

1. INTRODUCTION

Unglazed transpired solar collectors (UTSCs) have been a topic of many investigations as they prove to be an effective way of reducing HVAC heating loads. Because the collector is unglazed, it is relatively cheap, easy to manufacture and is light weight. These collectors are used in many commercial, industrial and institutional applications intended to reduce the HVAC loads in buildings, and in crop drying applications.

Commercially available products have a trapezoidal design which provides structural stiffness. The main goal of this study is to investigate the heat loss from an absorber plate of this geometry. Evaluating heat loss for this absorber plate is important to calculate the collector efficiency, and to find the optimum operating condition.

A number of research initiatives have studied UTSCs over the last twenty years. Researchers have made efforts in estimating the heat transfer coefficient and effectiveness. However, most of the studies focused on flat absorber plates perforated with circular holes arranged in triangular and hexagonal arrays. Kutscher (1992) experimentally studied the heat loss of flat absorber plates with triangular arrays. He derived a correlation for Nusselt number as a function of wind speed, suction velocity, pitch and perforation diameter. Following the Kutscher work, Cao et al.(1993) numerically studied plates which have long narrow rectangular slots. His model neglect the heat transfer from the back of the absorber plate, and the heat transfer form the back of the absorber plate, and the heat transfer to the air was assumed to be transferred only by the front and the sides of the perforations. Golneshan (1994) studied the same absorber plates analytically by performing a two dimensional momentum integral analysis, and experimentally by testing four different absorber plates in the asymptotic region. He found a relation between the plate effectiveness and six dimensionless parameters. Arulanandam et al. (1999) performed a CFD model on plates with circular holes in a square array with zero wind condition. Gawlik et al. (2002) studied the plate with sinusoidal corrugation in the asymptotic region .They found that flow over the corrugation will be attached or separated, and they drove a relation expressing the wind heat loss in terms of wind speed, suction velocity and plate geometry. Gawlik et al. (2005) tested the effect of low conductivity of the absorber material on the plate performance. He found that the material conductivity has a small effect on thermal performance of the absorber. Hence, the plate could be assumed as an isothermal especially for low prosperity plate.
Characterization of heat loss from a corrugated plate is limited in the literature. Past work considered only the heat loss from the flat plate geometry. Iglish (1944) gives an explanation of the flow behavior of a flat plate with homogenous suction. His model was improved by Maddaeus (1983) who developed a 3-D solution for the boundary layer, and a series solution for the boundary layer in the asymptotic region. Simple and exact solutions of the thermal and momentum boundary layer were performed by Arpaci (1984) and Kutscher (1992). In the case of sinusoidal corrugated plate, Gawlik (2002) found that asymptotic region was reached after a longer starting length compared with the flat one.

2. NUMERICAL MODEL

2.1 Plate Geometry

The geometry studied here reflects the Solarwall™ product produced by Conserval engineering Inc (2007). Figure 1 shows a section and dimensions of this absorber plate. The perforations consist of a semi-circular piece that is open on both ends. Perforations were placed in an aligned position; three on the bottom plate and three on each side of the corrugation. Figure 2 shows the perforations and their arrangement on the plate. Because of this arrangement of the perforations, a 3D CFD model was necessary.

Figure 1: Solar wall dimensions

Figure 2: Solarwall™ holes shape and arrangement

Figure 3 illustrates the solution domain of the numerical simulation. To get an accurate indication of the flow behavior before and after the corrugation, one of the perforations was placed before the corrugation and the other two are placed after. The domain of the numerical simulation has two regions. Region (1) is the upper region where the plate is exposed to wind, while Region (2) (Plenum) is located under the plate. In this region,
air is sucked through the perforations. Both regions were divided into set number of volumes to facilitate the use of structured mesh. Between these two regions, the absorber plate was placed. This plate was designed to be a solid region of 1 mm thickness and with heat generation equivalent to absorbed solar radiation of 500W/m².

2.2 Governing Equations

The governing equations used in the numerical simulation are the conservation of mass (Eqn. 1), momentum (Eqns. 2-4), and energy (Eqn. 5) equations. It is assumed that flow is laminar, air properties are constant and there is no viscous dissipation term.

\[
\frac{\partial}{\partial x} (\rho u) + \frac{\partial}{\partial y} (\rho v) + \frac{\partial}{\partial z} (\rho w) = 0
\]  

(1)

\[
\frac{\partial}{\partial x} (\rho u^2) + \frac{\partial}{\partial y} (\rho uv) + \frac{\partial}{\partial z} (\rho uw) = -\frac{\partial p}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right)
\]  

(2)

\[
\frac{\partial}{\partial x} (\rho uv) + \frac{\partial}{\partial y} (\rho vv) + \frac{\partial}{\partial z} (\rho vw) = -\frac{\partial p}{\partial y} + \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right)
\]  

(3)

\[
\frac{\partial}{\partial x} (\rho uw) + \frac{\partial}{\partial y} (\rho vw) + \frac{\partial}{\partial z} (\rho ww) = -\frac{\partial p}{\partial z} + \mu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right)
\]  

(4)

\[
\frac{\partial}{\partial x} (\rho C_p u T) + \frac{\partial}{\partial y} (\rho C_p v T) + \frac{\partial}{\partial z} (\rho C_p w T) = k \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right)
\]  

(5)

Details of the boundary conditions area are as follows:
At plane x=0 in Region 1: inlet boundary with a specified wind speed
At plane x=0.2 in Region 1: outlet boundary condition and it is specified as zero pressure outlet
At planes z=0, z=0.024: these two planes are specified as symmetry
At Planes x=0, x=2.4, z=0, z=2.4 in region 2: these are specified as symmetry
At plane y=5, in region 1: this plane is modeled as symmetry and it was extended up so that it will not affect the velocity near the wall
At plane y=-5: this plane is modeled as pressure outlet with a specified pressure value relevant to suction face velocity.

2.3 Mesh

The domain was broken into set of control volumes. An unstructured mesh was used with an implantation of size function that makes the grid fine near the holes and course far away from the hole with a degree of increasing. This procedure resulted in many irregular volumes and highly skewed elements.

The other procedure used was an application of a structured mesh with some expansion factors to make the mesh fine near the holes and walls and course in other regions. Figure 4 shows a mesh design for the inlet velocity boundary.

![Figure 4: mesh design for the inlet boundary](image)

A mesh refinement study was performed on this simulation to test the appropriate mesh element number that is needed for this simulation. For the three dimensional model, a course mesh was first created. The number of mesh elements was doubled in three dimensions near the holes and the walls, since these regions accounts for the most variation in the flow field. One of the runs has been chosen to test the meshing refinement convergence. This case was with 2 m/s wind velocity and 0.01 m/s suction face velocity. This run was chosen since it has high wind velocity with minimum suction flow rate where the flow has high potential of becoming turbulent. For all of the meshing schemes, of the outlet temperature in the first step was measured. Table 1 shows the variation of temperature with number of grid elements.
Table 1: Number of nodes with outlet temperature

<table>
<thead>
<tr>
<th>Number of grid Elements</th>
<th>Outlet Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>956432</td>
<td>300.422</td>
</tr>
<tr>
<td>1319168</td>
<td>300.392</td>
</tr>
<tr>
<td>2040388</td>
<td>300.390</td>
</tr>
</tbody>
</table>

Since the presented geometry have many of irregular volumes, number of mesh elements does not follow a certain ratio. All of the presented simulations were performed using 13191688 elements.

3. RESULTS AND DISCUSSION

In the case of the no wind condition, corrugations did not have any effect on the flow behavior on the upper side of the plate, and the correlations that have been used in flat plate scenarios can be applied to the corrugated one.

In the application of UTSCs, suction velocity is much higher than minimum velocity. When the flow reaches the asymptotic region, Reynolds number will be constant along the absorber plate. For a plate that has corrugations, flow behavior is different. Although the corrugations will increase the probability of the flow to be turbulent, low speed velocities with relativity high suction flow rate gives good indication of the flow behavior and results in some recirculation zones down steam of the corrugation. These separations and recirculation zones were observed by Gawlik et al. (2002). Modeling using a turbulent assumption does not show these separations and recirculation zones. The model results were restricted to small size absorber since the absorber plate geometry has some areas where suction does not has any effect on the flow. This enhances the chance of the flow to become turbulent.

3.1 Attached flow

The definition of the heat loss from the absorber plate is that heat loss from the entrance region of the plate where the velocity boundary layers reach 99% of the free stream velocity. Flow over the corrugated plate could be attached or separated. For the attached flow, the flow is moving along the absorber plate like in flat plate. The heat loss correlation will be same as the flat plate one. However, the geometry shape that plate contributed to increase the heat loss from the absorber. Figure 5 shows the velocity vectors of the attached flow. The analytical equation for the heat loss from the flat plate $Q_{flat}$ is given by Kutscher (1992).

$$Q_{flat} = \rho C_p \Delta T U_a \left[ \frac{v}{P_{fr + Pr^2}} \right]$$  \hspace{1cm} (7)

Where $\rho$ is the density of air, $C_p =$ specific heat of air, $U_a =$ the wind velocity $\Delta T$ is the temperature difference between surface of corrugated plate and free stream, $V_o$ is the suction face velocity, $v$ is kinematic viscosity of air, and $Pr$ is the Prandtl number.
By assuming a linear relation between the attached flow for the corrugated plate and the flat one, the equation for the heat loss becomes:

\[
Q_{\text{att}} = C Q_{\text{flat}}
\]  

(8)

The heat loss for the attached plate can be also expressed in terms of Nusselt number, which is based on the heat loss from the plate to the crosswind in the portion of the plate that consists of the developing region of boundary layer. The total heat loss from the plate to the crosswind in this developing region is equal to total heat
loss from the plate since the temperature profile keeps constant along the plate. The heat transfer coefficient in terms heat loss in the developing region is:

\[ h = \frac{Q_{\text{dev}}}{A_s \Delta \theta} \quad (9) \]

Where \( A_s \) is the plate absorber area in the developing region, \( Q_{\text{dev}} \) is the heat loss in the developing region of the boundary layer from the numerical simulation.

\[ A_s = W L_s \quad (10) \]

\( W \) is the width of the plate and \( L_s \) is the starting length (length of the developing region). Since the geometry of the plate has a width of 2.4 cm, it is significant to present the attached heat loss in terms of unit width plate absorber

\[ Q_{\text{att}} = \frac{Q_{\text{dev}}}{W} \quad (11) \]

Then Nusselt number could be expressed as the following equation:

\[ Nu_{\text{att}} = \frac{Q_{\text{att}}}{k \Delta \theta} \quad (12) \]

Thus, the correlation is of the form:

\[ Nu_{\text{att}} = 1.145 Nu_{\text{flat}} \quad (13) \]

Figure 7: Relationship between the flat plate Nusselt number and for the attached flow
3.2 Separated flow

Separated flow usually occurs at high wind velocities. Numerical model shows that flow separates when it reaches the peak of the corrugation, and results in a recirculation zone after the crest of the plate. Then, reattaches after the recirculation zone Figure 9 shows the recirculation zone that occurs after the corrugation. Due to this phenomenon, heat loss becomes higher when compared to attached flow. The Reynolds number based on the suction velocity and the starting length is:

$$Re_s = \frac{u_{el}L_s}{v}$$  (14)

And the Reynolds number based on the wind velocity and the starting length is

$$Re_W = \frac{u_{el}L_s}{v}$$  (15)

By dividing these two Reynolds numbers, a new Parameter will appear. This parameter is called the suction parameter.

$$Re = \frac{Re_W}{Re_s} = \frac{u_w}{v_0}$$  (16)
Figure 10: Wind heat loss for flat and corrugated plates a) at 1m/s wind speed b) at 2 m/s wind speed

Same procedure is used to evaluate the separated wind heat loss from the plate. This heat loss will be expressed in terms of this Nusselt number to be as a function of Reynolds number that based on the wind and suction velocity:

\[ N_{u_{sep}} = 26.3 Re^{0.47} \]  \hspace{1cm} (17)

Figure 11: Correlation for the wind heat loss in separated flow

3.3 Wind Speed for the Separated Flow

The previous simulations show that there are two equations expressing the heat loss in the corrugated absorber: one for the attached and the other one is for separated. It is important to know the critical point that separates the flow. In spite of applying suction at wall which decreases the chance of separation, numerical model shows that the flow is strongly affected by the wind speed. This happens since the plate has sharp edges which enhance the chance of separation. Therefore, a first step has been done to evaluate the critical wind speed velocity. It has been found that the flow start to separate from the plate at a wind speed of 0.65 m/s.

3.4 Validation

To validate the presented results, this model was compared with Gawlik (2002). It was the only study that tested the wind heat loss from corrugated plates. For the attached flow, at wind speed of 0.5 m/s and suction velocity of 0.01m/s, wind heat loss from the given correlation is 19.24 W/m, while it is equal to 21.9 W/m form Gawlik (2002) model. For the separated flow, at wind speed of 2 m/s and suction velocity of 0.01 m/s, wind heat loss from the presented plate is 410 W/m, while it is equal to 474 W/m from Gawlik (2002) model. This big difference could be a result of the flow behavior difference over the corrugation.

4. CONCLUSIONS

Heat loss from UTSCs was calculated at low wind speed. The flow is either attached or separated over the plate. Correlation of both types were developed using the numerical simulation. The heat loss for equation for the attached flow is :

\[ N_{u_{att}} = 1.45 N_{u_{flat}} \quad \text{for} \quad U_\infty < 0.65 \text{ m/s} \]

And for the separated flow
\[ \text{Nu}_{\text{sep}} = 26.3 \text{Re}^{0.47} \quad \text{for} \quad U_{\infty} > 0.65 \text{ m/s} \]

The critical wind velocity of the flow to be separated is 0.65 m/s. The previous results are applicable for the corrugated UTSC at low wind speeds.

**Nomenclature**

- \( A_s \) surface area of the corrugated plate, \( m^2 \)
- \( C_p \) specific heat of air, \( J/kg.K \)
- \( h \) heat transfer coefficient, \( W/m^2K \)
- \( k \) thermal conductivity, \( W/mK \)
- \( L_s \) starting length, \( m \)
- \( \text{Nu} \) Nusselt number for corrugated plate
- \( \text{Nu}_{\text{att}} \) Nusselt number for corrugated plate in attached flow
- \( \text{Nu}_{\text{sep}} \) Nusselt number for corrugated plate in separated flow
- \( Pr \) Prandtl number
- \( Q_{\text{att}} \) convective heat loss from corrugated plate in attached flow \( W/m \)
- \( Q_{\text{flat}} \) convective heat loss from corrugated plate in attached flow \( W/m \)
- \( Q_{\text{dev}} \) heat loss in the developing region
- \( Re \) Reynolds number, corrugated plate
- \( Re_s \) Reynolds number based on suction velocity
- \( Re_w \) Reynolds number based on wind velocity
- \( \Delta T \) temperature difference between surface of corrugated plate and free stream, \( K \)
- \( U_{\infty} \) wind velocity \( m/s \)
- \( V_o \) suction face velocity \( m/s \)
- \( W \) width of the test plate

**Greek**

- \( \nu \) kinematic viscosity of air \( m^2/s \)
- \( \rho \) air density \( kg/m^3 \)

**REFERENCES**


Unglazed transpired solar collectors (UTSCs) preheat makeup ventilation air for buildings, and thereby reduce energy used for heating. The performance of the UTSC is dependent on the heat loss from the absorber plate, which is strongly affected by the wind speed. To date, correlations to determine UTSC heat loss are limited to sinusoidal corrugations. In reality, trapezoidal corrugations have been added to UTSCs to provide them with structural stiffness. These corrugations prevent the attachment of the flow to the absorber plate and cause flow separation which increases the heat loss. In th