PARAMETRIC STUDY OF SOLAR AIR HEATER WITH DIFFERENT RIB GEOMETRIES

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ABSTRACT
In the present work, a 2D CFD simulation using ANSYS FLUENT 15.0 has been performed to investigate the performance of flat plate solar air heaters with different rib geometries (artificial roughness) such as semi-circular, circular, rectangular and triangular (wedge) and hyperbolic rib with flat tip. In fact, the hyperbolic rib profile is the novelty in this work. The CFD model of smooth duct is validated with the Dittus-Boelter Nusselt number correlation and the Blasius equation for heat transfer and frictional pressure drop respectively. RNG k-ε model is used for turbulent modelling. The proposed hyperbolic rib profile gives best performance at low Re whereas rectangular profile is found to be best at high Re.

INTRODUCTION
In the present scenario, there is a need to exploit the renewable energy resources to meet the growing demand for energy to sustain the growth and development of nations. Solar energy is one such resource which could be tapped to fulfil this growing energy demand. Solar collectors have been used to convert solar energy into thermal energy since last few decades and efforts are being made to improve their performance. Numerous investigations have been performed worldwide to enhance thermo-hydraulic performance of solar air heaters [1-8]. Recently Suman et al. [9] published a comprehensive review of literature pertaining to solar collectors and their performance evaluation. In general, artificial roughness is created on the absorber plate to enhance surface area and turbulence near its surface which in turn enhances heat transfer rate. The ribs break laminar sublayer causing an intense mixing near the surface. As a result, there is an increase in heat transfer rate. However, in the process frictional pressure drop also gets increased.

On the basis of review, shape of rib, its height, pitch and their arrangement (inline or staggered) on the absorber plate do affect heat transfer and pressure drop characteristics significantly. With this in mind, a novel design of hyperbolic rib with flat tip has been evolved, which not only eliminates the entrapment of upstream eddies but also results in better thermo-hydraulic performance compared with rectangular, triangular and semicircular rib geometries.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( c_p )</td>
<td>[J/kgK]</td>
<td>Specific heat</td>
</tr>
<tr>
<td>( H )</td>
<td>[m]</td>
<td>Duct height</td>
</tr>
<tr>
<td>( L )</td>
<td>[m]</td>
<td>Length</td>
</tr>
<tr>
<td>( W )</td>
<td>[m]</td>
<td>Width of duct</td>
</tr>
<tr>
<td>( p )</td>
<td>[Pa]</td>
<td>Pressure</td>
</tr>
<tr>
<td>( q' )</td>
<td>[W/m²]</td>
<td>Heat flux</td>
</tr>
<tr>
<td>( P )</td>
<td>[m]</td>
<td>Pitch</td>
</tr>
<tr>
<td>( T )</td>
<td>[K]</td>
<td>Temperature</td>
</tr>
<tr>
<td>( \dot{Q}_u )</td>
<td>[W]</td>
<td>Energy gain by air</td>
</tr>
</tbody>
</table>

Special characters
- \( \alpha \) [deg] Attack angle
- \( \phi \) [deg] Wedge angle
- \( \eta \) [-] Thermo-hydraulic performance
- \( \mu \) [N-s/m²] Viscosity of air

Subscripts
- \( i \) Inlet
- \( o \) Outlet
- \( m \) Arithmetic mean of inlet and outlet
- \( f \) Roughened surface
- \( s \) Smooth surface
- \( w \) Absorber wall
- \( 0 \) Ambient or reference

MODEL DESCRIPTION
A 2D CFD analysis has been performed using ANSYS FLUENT on hyperbolic, rectangular, triangular and semicircular rib geometries, shown in Figure 1.
The solar air heater geometry used in the present CFD simulation is shown in Figure 2. It consists of absorber plate having length \( L_2 \), subjected to constant heat flux of 1000 W/m\(^2\). The entrance length \( L_e = 5\sqrt{WH} \) and exit length \( L_s = 2.5\sqrt{WH} \) have been provided as per ASHRAE standard 93 [10]. The idea is to have fully developed flow all along the length of absorber plate and proper mixing of cold and hot air streams at its downstream.

**Figure 2** Schematic diagram of solar air heater

Following are the assumptions made:
- 2D, steady-state and fully developed turbulent flow
- Duct wall and rib material is aluminium, which is homogeneous and isotropic
- Thermal conductivity of absorber plate is constant
- Radiation and conduction heat losses are negligible

Values of geometrical used in the present simulation are shown in Table 1.

**Table 1** Geometrical parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Entrance length of duct</td>
<td>225</td>
</tr>
<tr>
<td>Absorber plate length</td>
<td>275</td>
</tr>
<tr>
<td>Exit length of duct</td>
<td>121</td>
</tr>
<tr>
<td>Width of duct</td>
<td>100</td>
</tr>
<tr>
<td>Depth of duct</td>
<td>20</td>
</tr>
<tr>
<td>Hydraulic diameter</td>
<td>33.33</td>
</tr>
</tbody>
</table>

**Governing Equations**

Mass conservation or continuity equation

\[
\frac{\partial}{\partial x_j}(u_j) = 0
\]  

(1)

Momentum conservation or Navier-Stokes equation

\[
\frac{\partial (\rho u)}{\partial t} + \frac{\partial}{\partial x_j}\left( \rho u u_j \right) - \frac{\partial p}{\partial x_j} + \mu \frac{\partial}{\partial x_j}\left( \frac{\partial u}{\partial x_j} + \frac{\partial u}{\partial x_j} \right) = 0
\]  

(2)

Energy conservation equation

\[
\frac{\partial (u T)}{\partial t} = \frac{\partial}{\partial x_j}\left( \frac{\partial T}{\partial x_j} \right) + \mu \frac{\partial}{\partial x_j}\left( \frac{\partial u}{\partial x_j} \right)
\]  

(3)

The boundary conditions employed in the code are shown in Table 2.

**Table 2** Boundary conditions

<table>
<thead>
<tr>
<th>Location</th>
<th>Boundary condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet</td>
<td>Velocity ((u))</td>
<td>1.72-7.88 m/s</td>
</tr>
<tr>
<td>Temperature</td>
<td></td>
<td>300 K</td>
</tr>
<tr>
<td>Absorber plate</td>
<td>Velocities</td>
<td>(u = v = 0) m/s</td>
</tr>
<tr>
<td>Heat flux</td>
<td></td>
<td>1000 W/m(^2)</td>
</tr>
<tr>
<td>Other walls (all insulated)</td>
<td>Velocities</td>
<td>(u = v = 0) m/s</td>
</tr>
<tr>
<td>Heat flux</td>
<td></td>
<td>0 W/m(^2)</td>
</tr>
<tr>
<td>Outlet</td>
<td>Pressure</td>
<td>(p_{atm})</td>
</tr>
</tbody>
</table>

**Numerical Methodology**

Simulations have been performed using finite volume based commercial CFD software ANSYS FLUENT 15.0. Steady pressure based solver has been used to solve the governing equations in a sequential manner. The momentum and energy equations have been discretized with second order upwind scheme. For pressure-velocity coupling SIMPLE (Semi Implicit Pressure Linked Equation) algorithm has been used. For turbulent modelling, RNG k-\(\varepsilon\) model with enhanced wall treatment has been employed. Sum of residual mass and residual energy must be less than 10\(^{-6}\) and 10\(^{-9}\) respectively for the convergence of solution.

Grid independence test has been conducted by varying mesh size from 13781 to 313245. The results have been found to be independent of grid at 220147. However, to capture the laminar sublayer near wall a finer mesh of size 313245. For the meshing we have used structured non-uniform mesh with quadrilateral element.

**RESULTS AND DISCUSSION**

This section has been divided into three sections, viz. model validation, parametric study and flow visualization.

**Model Validation**

The proposed CFD model is validated with the following well-established smooth duct correlations:

Dittus-Boelter Nusselt number correlation [11] for heat transfer:

\[
N_u = 0.023 \, R \, e^{0.8} \, Pr^{0.4}
\]  

(4)

Blasius friction factor correlation [11] for frictional pressure drop:

\[
f = 0.316 \, R e^{-0.25}
\]  

(5)

Heat transfer and pressure drop characteristics are obtained by evaluating Nusselt number and friction factor for the flow inside the duct.

Using the temperature data obtained from CFD simulation, the average heat transfer coefficient \(h\) is computed by applying energy balance:

\[
Q_o = \dot{m} c_p (T_o - T_i) = h A (T_w - T_w)
\]  

(6)

The average Nusselt number is given by
In order to evaluate friction pressure data (inlet and outlet of duct) from the CFD model is extracted and Darcy-Weisbach equation is used to evaluate Darcy friction factor $f$, i.e.

$$\Delta p = f \frac{L}{D_h} \left( \frac{1}{2} \rho V^2 \right)$$

where, $V$ is average fluid velocity inside the duct

$\Delta p$ is difference between inlet and outlet fluid pressure

The computed values of friction factor and Nusselt number have been compared with the corresponding values obtained from Blasius and Dittus-Boelter correlations in Figure 3. The percentage mean deviations in friction factor and Nusselt number are 1.59 and 2.94 respectively. With deviations being small, it can be concluded that the proposed numerical model is in good agreement with the well-established correlations for friction factor and Nusselt number.

### Parametric Study

As a matter of fact that provision of ribs increases both heat transfer and frictional pressure drop. Higher heat transfer is desirable whereas higher pressure drop entails higher pumping power. Thus, Webb and Eckert [1] defined a parameter known as thermo-hydraulic performance parameter $\eta$ which relates heat transfer rate with pressure drop in the following manner:

$$\eta = \frac{Nu_j / Nu_s}{(f_j / f_s)^{1/3}}$$

It relates combined effect of heat transfer and pressure drop in a frictional duct with that in a smooth duct. Higher the value of thermo-hydraulic performance parameter better is the performance.

The parameters such as rib height and pitch for novel hyperbolic rib geometry with flat tip have been varied to optimize the heat transfer and pressure drop characteristics. For rib height $e = 1$ mm and pitch $P = 10$ mm, the thermo-hydraulic performance is found to be the best. Figure 4 shows the comparison of performance of novel hyperbolic rib geometry with rectangular, triangular and semicircular rib geometries for

![Figure 3 Validation of model](image)

**Figure 3** Validation of model

![Figure 4 Performance comparison (a) $f$ versus Re (b) $Nu$ versus Re (c) $\eta$ versus Re.](image)

**Figure 4** Performance comparison (a) $f$ versus Re (b) $Nu$ versus Re (c) $\eta$ versus Re.
Figure 4(a) shows the variation of friction factor $f$ with $Re$ for different rib geometries. For any rib shape, friction factor decreases with the increase in $Re$. In fact, the rectangular rib has the highest value of friction factor for any $Re$ followed by hyperbolic, triangular and semicircular ribs. Semi-circular ribs have the best performance as far as frictional pressure drop is concerned. Figure 4(b) shows the variation of heat transfer coefficient or $Nu$ with $Re$ for different rib geometries. The hyperbolic rib geometry has the highest heat transfer coefficient at low $Re$ flows. However, heat transfer coefficient is highest for rectangular geometry at high $Re$ flow. Heat transfer characteristics of other geometries are inferior with semicircular rib has the least heat transfer coefficient. Thus the performance of semicircular rib as far as heat transfer is concerned is the worst. To have the true picture of performance, thermohydraulic parameter for each geometry has been evaluated and plotted in Figure 4(c). The thermohydraulic performance is best for hyperbolic rib geometry at low $Re$ whereas it is best for rectangular rib at high $Re$. Thermohydraulic performance is found to be worst for semicircular rib geometry for any $Re$.

To understand the physics behind such behaviour, the flow patterns for each rib geometry have been investigated and shown in Figures 5 and 6.

Streamline contour plots have been drawn in Figure 5 to show the effect of $Re$ on the recirculation zones near each rib. A big recirculation zone appears at the downstream of each rib due to the separation of boundary layer leading to the formation of eddies (recirculation zone). The separation causes an increase pressure drop whereas eddies flow or recirculation causes an increase in heat transfer rate as it causes mixing. The pitch should be large enough to avoid entrapment of eddies between the adjacent ribs. Local heat transfer coefficient varies along the surface of absorber plate. Local heat transfer coefficient as well as local skin friction coefficient is maximum at the point of reattachment of boundary layer between the ribs whereas they decrease as the boundary layer develops in the direction of flow. It can be seen clearly from Fig. 5 that recirculation zone decreases with the increase in $Re$ due to which pressure drop across each rib decreases resulting in an overall drop in pressure across the duct, which is evident from reducing friction factor with $Re$ (Figure 4(a)). Further a small recirculation zone starts appearing in the upstream of rib with increase in $Re$. This would result in further increase heat transfer rate.

A comparison of flow patterns at $Re = 6000$ for hyperbolic, rectangular and triangular ribs is shown in Figure 6. The semicircular ribs have not been shown here. Sizes of recirculation zones in hyperbolic and rectangular ribs are comparable whereas that of triangular is larger and that may be the reason why the pressure drop effects become more dominant than heat transfer effects resulting in its poor thermohydraulic performance.

The isothermal contours for hyperbolic, rectangular and triangular rib geometries have been shown in Figure 7 for $Re = 6000$, $e = 1$ mm and $P = 10$ mm. Temperature distribution is near the surface tells the local variation in heat transfer coefficient. Lower local plate surface temperature is the proof of higher local heat transfer or vice versa. Hotspots (red in colour) can be seen in the root of rib profile. It may be due to the entrapment of eddies in the corners. Since heat flux is const at the surface, the local variation in surface temperature is attributed to the local variation in heat transfer coefficient, which is dependent upon the flow patterns formed near the surface. Each rib geometry has recirculation zones of different sizes and shapes. Accordingly the pressure drop varies from rib to rib.
Figure 7 temperature contours for different rib (a) triangular rib, (b) rectangular rib and (c) hyperbolic rib.

Figure 8 and 9 show the variations of surface velocity gradient (a measure of local wall shear stress or frictional pressure drop) and local heat transfer coefficient along the absorber plate length for hyperbolic rib profile with Re = 6000, e = 1 mm and P = 10 mm.

The velocity gradient at the wall helps in identifying whether the flow is attached or detached. The flow over the rib accelerates till its peak and should decelerate beyond that. However, due to existence of adverse pressure gradient in the downstream, boundary layer could not follow the rib contour and flow separation takes place. The separation is marked by local recirculation of flow in the form of eddies. The flow reattaches itself after a certain distance from the rib. Peaks and valleys show the separation and reattachment of flow across different ribs.

Figure 8 Variation of surface velocity gradient along the absorber plate length

Figure 9 Variation of local heat transfer coefficient along the absorber plate length

Figure 9 shows the corresponding variation in local heat transfer coefficient along the absorber plate. Heat transfer coefficient strongly depends upon the variations in local velocity near the absorber plate surface. The local heat transfer coefficient is maximum at the point of reattachment in the inter-rib spacing already mentioned before.

CONCLUSION

In this paper, a numerical investigation of novel hyperbolic rib profile with flat tip has been carried out using ANSYS FLUENT 15.0. For optimized rib height and pitch (i.e. e = 1 mm and P = 10 mm), the comparison of the performance of this novel rib profile has been done with rectangular, triangular and semicircular rib profiles in terms of friction factor, heat transfer coefficient and thermohydraulic performance parameter. The proposed hyperbolic rib profile gives best performance at low Re whereas rectangular profile is found to be best at high Re.
REFERENCES


