NUMERICAL AND CFD BASED ANALYSIS OF POROUS MEDIA SOLAR AIR HEATER

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ABSTRACT

In the present work a CFD mathematical modeling and CFD study was conducted to investigate the temperature rise in a solar air heater using porous media. In a mathematical modeling emphasis is given to study the behavior of solar air heater without porous media. To compare the performances under different set of conditions, obtained by changing various governing parameter like air mass flow rate, inlet temperature and intensity of solar radiation. Work has been carried out using commercial code of FLUENT version 6.3.26. Solar air heater with porous media gives higher thermal performance than without porous media.

Keyword- Solar air heater, porous media, CFD.

1.0 INTRODUCTION

The thermal efficiency of solar air heater has found to be generally poor because of their inherently low heat transfer capability between the absorber plate and air flowing in the duct. In order to make the solar air heater economically viable, their thermal efficiency needs to be improved by enhancing the heat transfer parameters. Jiang et al. [1] had studied forces convection heat transfer from cylinder embedded in a packed bed numerically. The local volume averaged conservation equations are used to examine the effect of the effects of the Reynolds ($R_e$), the Darcy ($D_a$), Forchheimer ($F_p$) and effective Prandtl numbers. An increase in either $R_e$ or $F_p$ results in the heat transfer enhancement. This enhancement is found to be consistent with that obtained from the prediction of boundary layer theory, which shows the Nusselt number ($N_u$) dependence on the $R_e$ to the one half powers. The effect of decreasing $D_a$ is an increase in $N_u$ and increasing $D_a$ decreases the heat transfer. The effect of $F_p$ is found
to depend on the product \((D_a R_e)\). A comparison between the numerical predictions and experimental data gives the values of effective thermal conductivities and quantified the average thermal dispersion. They utilized the finite element for the computation of the various parameters. The packing surrounding cylinder is discretized into triangular elements to form the computational domain. In the paper they defined the effective thermal conductivity \((K_{eff})\) as the superposition of stagnant conductivity (independent of flow) and flow-dependent dispersion conductivity. If the thermal dispersions are present the value of \(K_{eff}\) is found to increase. Thermal dispersion is result of simultaneous existence of temperature and velocity gradient within the pores of porous medium. Also, an attempt is made to quantify its contribution to heat transfer from an embedded cylinder is presented in this paper. They found out that the value of \(K_{eff}\) is about ten times the thermal conductivity of air, \((0.3 \text{ W/mK})\). Elradi et al. [2] had described the correlation of transient heat transfer and pressure drop which has been developed for air flowing the porous media the various porous media are arranged in different porosities to increase heat transfer, area density and the total heat transfer rate. In the double-pass solar collector, the mass flow rate has more effect on the temperature rise. The solar radiation has more effect on temperature rises at low porosity. In addition, the Reynolds number has more effect than the Nusselt number at low porosity. Heat transfer coefficient increases by using more porous media in the lower channel of the double-pass solar collector. Higher porosities in the porous media improved the thermal efficiency. Experimental analysis of the data suggests that due to higher mass flow rate, the thermal efficiency decreases. Pressure drop study indicated that lower pressure losses are encountered with low porosity than with high porosity of the saturated porous media. Mittal et al. [3] in his paper studied the effect of roughness elements on the effective efficiency of the solar air heater. As we know the solar air heater is widely used collection devices but the thermal efficiency of solar air heaters has found to be generally poor because of their inherently low heat transfer capability between the absorber plate and air flowing in the duct. This heat transfer capability can be improved by using the roughness elements on the absorber plate. In his paper the research has been done on the various roughness elements and their effect on the thermal efficiency of the solar air heaters proposed by different investigators. To compare the effective thermal efficiency of the solar air heaters having different types of geometry of roughness elements on the absorber plate a expression has been developed

\[
\eta_u = \frac{q_u - P}{I A_o}
\]

Where \(\eta_u\) = effective efficiency
\(q_u\) = useful thermal energy gain
\(P\) = mechanical power
\(C\) = conversion efficiency
\(I\) = Solar radiation
\(A_o\) = Area of absorber plate

Karwa et al. [4] in his paper has critically reviewed the various correlations employed by researchers to predict the heat transfer coefficient for air flow in solar air heater duct in the range \(2300 < Re_a < 15000\). Some of the correlations have been basically developed for circular section ducts or for ducts with high aspect ratio. All of them do not take into account the
effect of the combined thermal and hydrodynamic development length and also do not cover the complete range of Reynolds number of interest. The Nusselt number values predicted by this correlation differ by as much as 30 to 40%. The study established the need of a reliable heat transfer coefficient correlation applicable from transition to early turbulent flow regime in asymmetrically heated rectangular section ducts of solar air heaters, which includes the effect of development length and aspect ratio on the heat transfer coefficient. Muluwork et al. [5] in his paper presented experimental results for air flow in the rectangular duct with repeated discrete rib roughness on the upper broad wall, which is subjected to uniform heat flux. The effect of relative roughness length ratio on tendon number and friction factor for V-down and V-up and traverse discrete rib roughened surfaces have been studied for Reynolds number about $2000 < Re < 15500$. The use of artificial roughness results in considerable enhancement in the heat transfer. The Stanton number ratio enhancement obtained is between 1.32 to 2.47 in the range of system and operating parameters covered in the investigation. Mittal et al. [6] investigates the thermo hydraulic performance on a packed bed solar air heater having its duct packed with blackened wire screen matrices of different geometrical parameters (wire diameter and pitch) as shown in fig. 2.1. To obtain the effective efficiency a mathematical model has been developed on the basis of energy transfer mechanism in the bed. The following assumptions have been made during the development of the mathematical model.

- Edge and back losses have been neglected.
- Environmental temperature and wind velocity have been assumed to be Constant.

![Figure 1. Packed bed solar air heater](image)

Pradhap Raj et al. [7] had analyzed the performance of the non-porous and porous flat plate solar air collector with mirror enclosure. The addition of side mirror enclosures is to increase the amount of solar radiation absorption at the collector plate so that the collector increase the yield and operate at high temperature range. Therefore with the addition of side mirror one can able to maximize the output of fixed flat plate collectors. A flat plate air collector will be more efficient if it is made up of porous medium when comparing it with the non-porous collectors. The performance of a solar air heater without any cover is very poor and hence at least one cover should be used for better performance. The performance of the air heater is dependent on the number of covers used and the temperature difference between the inlet air to the ambient air. Therefore, the efficiency will be maximum when the inlet air temperature
is more than the ambient air temperature. Even plastic covers can be used where the inlet temperature rise over ambient air temperature is small. The highest output obtained from the inclined side mirror when compared to the vertical side mirror. Since, the double exposure solar collector unit cost is estimated to be only 70% greater than a conventional air collector. It is efficient to get for the double exposure solar collector.

2.1 MATHEMATICAL MODELING

In the present study, at first mathematical model is obtained by the application of the governing conservation laws. The heat balance is accomplished across each component of given solar air heater i.e., the glass covers, the air stream and the absorber plate. The heat balance for the air stream yields the governing differential equations and the associated boundary conditions. The heat and fluid flow are assumed steady and one dimensional. It is because of the radiation heat exchange terms that render the problem non-linear hence making the exact solution cumbersome. So a numerical approach is applied which would give a solution with a fairly good accuracy. The finite difference method (FDM) will be used to solve the differential equations and hence to simulate a given solar air heater.

2.1.1. Double-pass flat-plate collector without porous media

For top glass cover:

G.E: \[ I_{\text{G}} = h_a(T_{c1} - T_a) + h_{fc1}(T_{c1} - T_{fl}) + h_{rc1}(T_{c1} - T_{c2}) \] ...\( (1) \)

F.D.E: \[ I_{\text{G}} = h_a(T_{c1}[i] - T_a) + h_{fc1}(T_{c1}[i] - T_{fl}[i]) + h_{rc1}(T_{c1}[i] - T_{c2}[i]) \] ...\( (2) \)

\[ T_{c1}[i] = \frac{I_{\text{G}} + h_a T_a + h_{fc1} T_{fl}[i] + h_{rc1} T_{c2}[i]}{h_a + h_{fc1} + h_{rc1}} \] ...\( (3) \)

For down flow air stream:

G.D.E.: \[ m_c \frac{dT_{fl}}{dx} = h_{fc2}(T_{c1} - T_{fl}) + h_{rc2}(T_{c2} - T_{fl}) \] ...\( (4) \)

F.D.E: \[ m_c \frac{T_{fl}[i+1] - T_{fl}[i]}{dx} = h_{fc2}(T_{c1}[i] - T_{fl}[i]) + h_{rc2}(T_{c2}[i] - T_{fl}[i]) \] ...\( (5) \)

\[ T_{fl}[i+1] = \left(1 - \frac{(h_{fc2} + h_{rc2})\Delta x}{m_c}\right) T_{fl}[i] + \frac{(h_{fc2} T_{c1}[i] + h_{rc2} T_{c2}[i])\Delta x}{m_c} \] ...\( (6) \)
For second glass cover
G.E.: 
\[ i \alpha e c_0 = h_{\text{efcoc}}(T_{c2} - T_{c1}) + h_{\text{f2oc}}(T_{c2} - T_{f2}) + h_{\text{f2oc2}}(T_{c2} - T_{f2}) + h_{\text{epo}}(T_{c2} - T_p) \] ...
(7)

F.D.E
\[ i \alpha e c_0 = h_{\text{efcoc}}(T_{c2}[i] - T_{c1}[i]) + h_{\text{f2oc}}(T_{c2}[i] - T_{f2}[i]) + h_{\text{f2oc2}}(T_{c2}[i] - T_{f2}[i]) + \]
\[ \frac{\partial}{\partial x} \left( c_0 \frac{\partial T_{c2}[i]}{\partial x} \right) \] ...
(8)

\[ T_{c2}[i] = \frac{i \alpha e c_0 + h_{\text{f2oc2}} T_{f2}[i] + h_{\text{f1oc2}} T_{f1}[i] + h_{\text{epc}} T_{p}[i]}{h_{\text{efcoc}} + h_{\text{f2oc2}} + h_{\text{f1oc2}} + h_{\text{epc}}} \] ...
(9)

For up flow air stream:
G.D.E.: 
\[ \frac{\partial}{\partial x} \left( c_p \frac{\partial T_{f2}[i]}{\partial x} \right) = h_{f2cp}(T_{c2} - T_{f2}) + h_{f2cp}(T_p - T_{f2}) \] ...
(10)

F.D.E.: 
\[ \frac{\partial}{\partial x} \left( c_p \frac{\partial T_{f2}[i]}{\partial x} \right) = h_{f2cp}(T_{c2}[i] - T_{f2}[i]) + h_{f2cp}(T_p[i] - T_{f2}[i]) \]
\[ \frac{\partial}{\partial x} \left( c_p \frac{\partial T_{f2}[i]}{\partial x} \right) \] ...
(3.11)

\[ T_{f2}[i+1] = \left( 1 + \frac{(h_{f2cp} + h_{f2cp}) \Delta x}{c_p} \right) T_{f2}[i] + \frac{(h_{f2cp} T_{c2}[i] + h_{f2cp} T_{p}[i]) \Delta x}{c_p} \] ...
(11)

For absorber plate:
G.E.: 
\[ i \alpha p c_0 = h_{f2ap}(T_p - T_{f2}) + h_{f2ap}(T_p - T_{f2}) + h_{f2ap}(T_p - T_{f2}) \] ...
(3.13)

F.D.E.: 
\[ i \alpha p c_0 = h_{f2ap}(T_p[i] - T_{f2}[i]) + h_{f2ap}(T_p[i] - T_{f2}[i]) + u_b(T_p[i] - T_a) \]
\[ i \alpha p c_0 = h_{f2ap}(T_p[i] - T_{f2}[i]) + h_{f2ap}(T_p[i] - T_{f2}[i]) + u_b(T_p[i] - T_a) \]
\[ T_p[i] = \frac{i \alpha p c_0 + u_b(T_a + h_{f2ap} T_{f2}[i] + h_{f2ap} T_{p}[i] + h_{f2ap} T_{c2}[i])}{u_b + h_{f2ap} + h_{f2ap}} \] ...
(12)

2.1.2. Double-pass flat-plate collector with porous media
For top glass cover:
G.E: 
\[ i \alpha e c = h_{e}(T_{c1} - T_a) + h_{f2cl}(T_{c1} - T_{f1}) + h_{f2oc}(T_{c1} - T_{c2}) \] ...
(14)

F.D.E: 
\[ i \alpha e c = h_{e}(T_{c1}[i] - T_a) + h_{f2cl}(T_{c1}[i] - T_{f1}[i]) + h_{f2oc}(T_{c1}[i] - T_{c2}[i]) \]
\[ i \alpha e c = h_{e}(T_{c1}[i] - T_a) + h_{f2cl}(T_{c1}[i] - T_{f1}[i]) + h_{f2oc}(T_{c1}[i] - T_{c2}[i]) \]
\[ T_{c1}[i] = \frac{i \alpha e c + h_{e} T_a + h_{f2cl} T_{f1}[i] + h_{f2oc} T_{f2}[i]}{h_{e} + h_{f2cl} + h_{f2oc}} \] ...
(15)

\[ \frac{\partial}{\partial x} \left( c_p \frac{\partial T_{f2}[i]}{\partial x} \right) \] ...
(16)

![Figure 3. Computational domain for Double-pass flat-plate collector with porous media](image-url)

For down flow air stream:
For second glass cover

G.E.:  
$$\alpha_v c_v = h_{nco}(T_{c2} - T_{c1}) + h_{flo2}(T_{c2} - T_{f2}) + h_{fco2}(T_{c2} - T_{f2}) + h_{npo}(T_{c2} - T_{p})$$  \hspace{1cm} (20)

F.D.E  
$$\alpha_v c_v = h_{nco}(T_{c2} - T_{c1}) + h_{flo2}(T_{c2} - T_{f2}) + h_{fco2}(T_{c2} - T_{f2}) + h_{npo}(T_{c2} - T_{p})$$  \hspace{1cm} (21)

For up flow air stream:

G.E.:  
$$mc \frac{dT_{f2}}{dx} = K_{eff} \frac{dT_{f2}}{dx} + h_{flo2}(T_{c2} - T_{f2}) + U_b(T_a - T_{f2}) + \alpha_p \tau_0 \tau_c$$  \hspace{1cm} (23)

F.D.E:  
$$mc \frac{T_{f2b+1} - T_{f2b-1}}{\Delta x} = K_{eff} \frac{T_{f2b+1} - T_{f2b-1}}{2\Delta x} + h_{fco2}(T_{c2}[i] - T_{f2}[i])$$  \hspace{1cm} (24)

$$T_{f2b+1} = T_{f2b} \left[ \left( \frac{2mc}{2mc + K_{eff}} \right) - \left( \frac{2 \Delta x}{2mc + K_{eff}} \right) (h_{flo2} + U_b) \right] - \frac{T_{f2}(i-1)}{2mc + K_{eff}}$$  \hspace{1cm} (25)

Boundary conditions:

B.C:  $$T_{f2}[1] = T_1$$  \hspace{1cm} (27)

B.C:  $$T_{f2}[0] = T_1 + \lambda$$  \hspace{1cm} (28)

Where  \( \lambda \)  is temperature rise in the down of air stream during the first pass between the two glass plates. For the sake of convenience the heat transfer coefficients between the air stream and the covers and between the air stream and the absorber plate are assumed equal and can be calculated as follows:

$$h_{flo1} = h_{flo2} = h_{fco2} = h_{floP} = h_f$$  \hspace{1cm} (29)

The air density:  $$\rho = \frac{p_{atm}}{K_{T_{atm}}}$$  \hspace{1cm} (30)

Kinematic viscosity:  $$\nu = \frac{f}{\rho}$$  \hspace{1cm} (31)

Thermal diffusivity:  $$\alpha = \frac{k}{\rho c_p}$$  \hspace{1cm} (32)

Prandtl number:  $$P_r = \frac{\nu}{\alpha}$$  \hspace{1cm} (33)
Hydraulic diameter: \( D_h = \frac{4A}{P} = 2D \) \( \ldots(34) \)
Reynolds number: \( Re = \frac{\rho UD_h}{\mu} = \frac{2m}{\mu} \) \( \ldots(35) \)
Nusselt number: \( Nu = 0.0333Re^{0.8}Pr^{0.36} \) \( \ldots(36) \)
Convective heat transfer coefficient between any two surfaces \( h_{12} = \frac{\sigma(T_1 + T_2)(T_1^2 + T_2^2)}{L_1 + L_2 + T} \) \( \ldots(37) \)

2.2 Computational Fluid Dynamics (CFD) Modeling

Computational fluid dynamics (CFD) is concerned with the efficient numerical solution of the partial differential equations that describe fluid dynamics. CFD techniques are used in many areas of engineering where fluid behavior is the main element. Numerical analysis applied to fluid flow and heat transfer problems.

All CFD codes contain three main elements:
1. A pre-processor which is used to input the problem geometry, generate the grid and define the flow parameters and the boundary conditions to the code.
2. A flow solver which is used to solve the governing equations of the flow subject to the conditions provided. There are four different methods used as a flow solver:
   (i) Finite difference method,
   (ii) Finite element method,
   (iii) Finite volume method,
   (iv) The spectral method.
3. A post-processor which is used to display the data and show the results in graphical and easy to read format.

A three-dimensional numerical model was developed using the CFD numerical package FLUENT. An experimental model was used to evaluate the flow patterns. Flow visualization was used to investigate the flow structure.

The full geometry and computational mesh for CFD simulations were created in the Gambit software, which is used as a pre-processor for the CFD solver and post-processor, namely FLUENT version 6.3.26.

3.0 Result and Discussions

The variations of outlet temperature with different mass flow rate for non-porous and porous media are shown in 4.25 for different depth, inlet temperature and solar radiation. It is concluded that outlet temperature is decreasing with increase in mass flow rate. It is found that the use of porous media in lower channel increases the outlet temperature. The effect of depth on the outlet temperature for both non-porous and porous solar air heater is displayed in figures. The graph illustrates that, with increase of depth the outlet temperature decreases. This is due to the low convective heat transfer coefficient between the absorber plate and air.
Figure 4. Variation of outlet fluid temperature with mass flow rate

4.26 shows the variation of maximum temperature of the absorber plate temperature and air stream temperature \((T_p)\) with mass flow rate. As like \(\Delta T_p\) it also reduces rapidly for mass flow rates 0.01 kg/m.s-0.05 kg/m.s the reason is same as discussed above for \(\Delta T_p\). As we increase the duct depth from 10 cm to 14 cm and 14 cm the value of \(T_p\) increases for a given mass flow rates due to low velocities and consequently convective heat transfer coefficient decreases.

Figure 5. Variation of Plate temperature with mass flow rate

4.27 shows the variation of the maximum temperature difference between the top glass cover temperature and ambient temperature \((\Delta T_p)\) with mass flow rates. \(\Delta T_p\) reduces rapidly for mass
flow rates between 0.01 kg/m.s to 0.05 kg/m.s after that gradual reduction is there for further mass flow rates. As we know the heat transfer coefficient ($h_p$) is the function of fluid velocity, so it increases with higher mass flow rates and consequently higher heat transfer is there for higher velocities. Hence, the rise in ($\Delta T_g$) is small at higher mass flow rates. For a given flow rate, increasing the spacing between the glass covers and the absorber plate decreases the flow velocity. Hence, the heat transfer coefficient decreases. Therefore, where the glass temperature increase due to radiation exchange with the absorber and heat convection from glass to the airstream decreases by increasing the depth of solar air collector.

![Graph showing the variation of temperature difference between top cover and ambient air with mass flow rate](image)

**Figure 6.** Variation of temperature difference between top cover and ambient air with mass flow rate

### 4.0 Analysis of CFD modeling results

The figure 4.1 shows the modeling, meshing and constraints of flat plate solar air heater. The iteration is carried out for laminar flow .i.e. 0.01 Kg/sec. The figure 4.2 shows the grid formation of solar air heater.
The analysis of the CFD model results, clearly demonstrated that the application of a metallic mesh insertion in the heating channels of passive solar collectors is an efficient way to intensify heat transfer from the heating surface to the working fluid and consequently improving the thermal performance of solar collectors. The output temperatures obtained from simulations for the geometry with the metallic mesh and for the conventional were validated with literature which showed the accuracy of the CFD model.
Table 1. Initial boundary conditions

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Solar air heater without porous media</th>
<th>Solar air heater with porous media</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area</td>
<td>0.065m$^2$</td>
<td>0.065m$^2$</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>0.01Kg/sec</td>
<td>0.01Kg/sec</td>
</tr>
<tr>
<td>Solver</td>
<td>Pressure based</td>
<td>Pressure based</td>
</tr>
<tr>
<td>Formulation</td>
<td>Implicit</td>
<td>implicit</td>
</tr>
<tr>
<td>Flow Zone</td>
<td>Laminar zone</td>
<td>Porous zone &amp; laminar zone</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>288K</td>
<td>288K</td>
</tr>
<tr>
<td>Density of air</td>
<td>1.225kg/m$^3$</td>
<td>1.225 kg/m$^3$</td>
</tr>
<tr>
<td>Thermal conditions</td>
<td>Convection and radiation</td>
<td>Convection and radiation</td>
</tr>
</tbody>
</table>

The figure 4 and 5 shows Temperature variation and Temperature variation of solar air heater without porous media and with porous media.

Figure 9. Temperature variation of solar air heater without porous media
Figure 10. Temperature variation of solar air heater with porous media

5.0 CONCLUSIONS

The present study shows mathematical modeling for predicting the heat transfer characteristics and the performance of solar air heater with and without porous media. The solar air heater with porous media gives higher thermal performance than without porous media. The thermal conductivity of porous media has significant effect on the thermal performance of the solar air heater. Numerical results and CFD analysis has predicted that the effect of depth on outlet temperature for both porous and non porous solar air heater and illustrates that with increase of depth the outlet temperature decreases. This is due to the low convective heat transfer coefficient between the absorber plate and air. Increasing the depth results increase in $\Delta T_p$ for given mass flow rates.

REFERENCES