ABSTRACT

Heat transfer enhancement in receivers of parabolic trough collectors offers several benefits including reduction in absorber tube circumferential temperature differences, reduced emissivity of the absorber tube selective coating, thus improved thermal and thermodynamic performance of the receiver. In this work, heat transfer enhancement in a parabolic trough receiver using perforated conical inserts was numerically investigated. The analysis was carried out for dimensionless insert’s cone angles in the range 0.40 - 0.90, dimensionless insert spacing in the range 0.06 - 0.18 and dimensionless insert size in the range 0.45 - 0.91. The flow was considered fully developed turbulent with Reynolds numbers in the range $1.02 \times 10^4 \leq Re \leq 7.38 \times 10^5$ depending on the temperature of the heat transfer fluid. The heat transfer fluid temperatures used were 400 K, 500 K, 600 K and 650 K. The numerical solution was obtained using the finite volume method together with the realizable $k$-$\epsilon$ model for turbulence modeling. From the study, there is a range of Reynolds numbers and geometrical parameters for which the gain in performance is more than the increase in pumping power due to heat transfer enhancement. The use of perforated conical inserts in the receiver’s absorber tube increases the thermal efficiency in the range 3-8% for some range of geometrical parameters.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Area, $m^2$</td>
</tr>
<tr>
<td>$A_a$</td>
<td>Collector’s projected aperture area, $m^2$</td>
</tr>
<tr>
<td>$A_r$</td>
<td>Projected absorber tube area, $m^2$</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat capacity, $J kg^{-1} K^{-1}$</td>
</tr>
<tr>
<td>$C_{2p}$</td>
<td>Inertial resistance factor, $m^{-1}$</td>
</tr>
<tr>
<td>$C_R$</td>
<td>Concentration ratio</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Turbulent model constant</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat capacity, $J kg^{-1} K^{-1}$</td>
</tr>
<tr>
<td>$d_{gi}$</td>
<td>Glass cover inner diameter, m</td>
</tr>
<tr>
<td>$d_{go}$</td>
<td>Glass cover outer diameter, m</td>
</tr>
<tr>
<td>$d_{ai}$</td>
<td>Absorber tube’s inner diameter, m</td>
</tr>
<tr>
<td>$d_{ao}$</td>
<td>Absorber tube’s outer diameter, m</td>
</tr>
<tr>
<td>$f$</td>
<td>Darcy friction factor</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient, $W m^2K^{-1}$</td>
</tr>
<tr>
<td>$h_a$</td>
<td>Glass cover heat transfer coefficient, $Wm^2K^{-1}$</td>
</tr>
<tr>
<td>$HTF$</td>
<td>Heat transfer fluid</td>
</tr>
<tr>
<td>$I_0$</td>
<td>Direct solar radiation, $W m^{-2}$</td>
</tr>
<tr>
<td>$k$</td>
<td>Turbulent kinetic energy, $m^2 s^{-2}$</td>
</tr>
<tr>
<td>$k_p$</td>
<td>Pressure coefficient</td>
</tr>
<tr>
<td>$L$</td>
<td>Length, m</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass flow rate, $kg/s$</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>$p$</td>
<td>Insert spacing, m</td>
</tr>
<tr>
<td>$p_s$</td>
<td>static pressure, Pa</td>
</tr>
<tr>
<td>$p_v$</td>
<td>Velocity pressure, Pa</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure, Pa</td>
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<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$q''$</td>
<td>Heat flux, $W m^{-2}$</td>
</tr>
<tr>
<td>$Q_u$</td>
<td>Useful heat gain, $W$</td>
</tr>
<tr>
<td>$r_p$</td>
<td>Insert size</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
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</table>
The increasing world population and increasing urbanization rates are continually putting pressure on available resources. One of such resources is energy. The demand for modern energy services has continued to increase in both developed and developing countries. In addition, more than 1.3 billion people are still without access to the much-needed electricity and 2.6 billion people are still relying on traditional biomass to meet their cooking needs [1]. Therefore, there is need to meet the increasing demand for modern energy services and provide energy to those who do not yet have access.

The continued reliance on fossil-based energy has led to increased emission of harmful substances and greenhouse gases, which have accelerated global warming and climate change [2, 3]. Thus, to meet the increasing demand for energy and provide energy to those without access requires development and deployment of clean and renewable energy technologies.

Solar resource is one of the clean and widely available sources of energy with potential to supply a significant portion of the world’s total energy needs. The sun’s total energy output is about $3.8 \times 10^{20}$ MW, equivalent to 63 MW/m² on the sun’s surface, of which about $1.7 \times 10^{14}$ kW is intercepted by the earth’s surface [4]. Even with this small fraction received on the earth’s surface, about 30 minutes can supply the world energy demand for one year [4]. However, significant research and development is still needed to harvest all this potential energy from the sun.

Solar radiation can be converted into electricity by the use of photovoltaic systems as well as solar thermal systems. The ease of storage of thermal energy from concentrated solar thermal systems for later conversion to electricity when the sun is not available makes solar thermal systems attractive [5]. The commonly used concentrated solar thermal technologies include the parabolic trough systems, parabolic dish systems, linear Fresnel systems and power tower systems.

Parabolic trough systems are the most commercially and technically developed solar thermal systems. However, some challenges still have to be overcome to further reduce the cost of electricity from these systems. Some of the challenges include the presence of circumferential temperature gradient in the receiver’s absorber tube, higher losses at elevated temperatures and degradation of the heat transfer fluid that occurs as temperatures increase above 673.15 K leading to formation of hydrogen in the receiver’s annulus space [6]. The presence of hydrogen in the receiver’s annulus space significantly increases the receiver’s thermal loss and subsequently reduces the thermal efficiency of the receiver [6]. Vacuum conditions will be lost as hydrogen builds up in the annulus space. Moreover, hydrogen has a significantly higher thermal conductivity than air. These are also some of the current focus of research and development regarding parabolic trough systems.

Furthermore, reducing the cost of electricity from these systems to make it cost competitive with electricity from fossil
based plants is one of the drivers of research regarding parabolic trough systems. With availability of lightweight materials, increasing concentration ratios is seen as one of the ways to further reduce the cost of electricity generated from these systems [7]. As higher concentration ratios are used, the circumferential temperature differences and peak temperatures will increase. Additionally, increasing concentration ratios will increase the entropy generation rates in the receiver due to higher finite temperature differences as the concentration ratios increase [8].

Enhancement of the convective heat transfer in the receiver’s absorber tube is one of the ways considered useful in minimising the absorber tube’s circumferential temperature differences, absorber tube peak temperatures, reducing entropy generation rates thereby increasing the life of the system and improving the receiver’s thermal and thermodynamic performance. As such, heat transfer enhancement in parabolic trough receivers has received considerable attention in the recent past. Recent studies on heat transfer enhancement in parabolic trough receivers include Ravi Kumar and Reddy [9] for a receiver with porous fins, Ravi Kumar and Reddy [10] for a receiver with a porous disc and Muñoz and Abánades for a receiver with internal helical fins [11]. The studies showed improved performance of the receiver with heat transfer enhancement. However, the studies considered a uniform heat flux boundary condition on the receiver’s absorber tube. In actual systems, the heat flux distribution is non-uniform. The use of realistic non-uniform heat flux boundary condition is essential in determining the actual temperature gradients, peak temperatures, as well as entropy generation rates in the receiver.

The advances in computational methods and computing power have made it possible to model complex engineering systems under actual operating conditions. In a recent study, Cheng et al. [12] analysed the heat transfer enhancement of a parabolic trough receiver using unilateral longitudinal vortex generators with a realistic non-uniform heat flux boundary condition. In another study, Wang et al. [13] investigated heat transfer enhancement using metal foams in a parabolic trough receiver for direct steam generation with a realistic non-uniform heat flux boundary condition. These studies show that, representation and modelling of the actual thermal performance of a parabolic using realistic heat flux distribution is possible.

Though significant amount work has been done on heat transfer enhancement in several applications as shown in reviews by Manglik [14, 15], heat transfer enhancement in parabolic trough receivers has not been widely studied. Moreover, most studies on parabolic trough receiver’s thermal performance assume an approximate and uniform heat flux profile on the receiver’s absorber tube.

In this study, a numerical investigation of heat transfer enhancement in a parabolic trough receiver with perforated conical inserts is presented. Moreover, unlike most studies on parabolic trough thermal performance, the actual heat flux distribution on the receiver’s absorber tube is used in this work.

**PHYSICAL MODEL**

Figure 1 shows the physical model of the receiver with perforated conical inserts in the absorber tube. The perforated conical inserts are supported on a thin axially placed rod as shown in Fig. 1(a).

![Physical model and computational domain of a parabolic trough receiver with perforated conical inserts](image)

(c) Computational domain

The geometrical parameters defining the placement of the inserts are: the spacing between two consecutive inserts (p), the size of the insert (r_p) and the cone angle of the insert (β). For this study, a simplified model of the parabolic trough receiver is considered. The effect of the central rod and other supports is
considered negligible. Far from the entrance, flow becomes periodically fully developed after about five perforated conical inserts regardless of the spacing. Therefore, a periodic module of the receiver’s absorber tube was considered. Furthermore, due to the symmetrical nature of the model, only half of the receiver tube is considered. The computational domain used in this study is shown in Fig. 1(c).

Similar to actual receivers, the space between the absorber tube and the glass cover is considered evacuated to low vacuum pressures (about 0.0001 mm Hg or 0.013 Pa) [16], such that only radiation heat loss takes place. The receiver tube dimensions used are similar to those of the SEGS LS-2 receiver [17]. The absorber tube has an inner diameter \(d_{ri}\) of 6.6 cm, outer diameter \(d_{ro}\) of 7.0 cm while the receiver’s glass cover has an inner diameter \(d_{gi}\) of 11.5 cm.

From Fig. 1, the following non-dimensional variables are defined:

\[
\hat{p}_c = \frac{p}{L}; \quad \hat{\beta}_c = \frac{\beta}{\beta_{\text{max}}}; \quad \tilde{r}_c = \frac{2r_c}{d_{ri}}
\]

(1)

where \(L = 1\) m, \(\beta_{\text{max}} = 90^\circ\), \(\hat{p}_c\) is the dimensionless insert spacing, \(\hat{\beta}_c\) is the dimensionless insert cone angle and \(\tilde{r}_c\) is the dimensionless insert size.

The simulation and geometrical parameters used in this work are shown in Table 1.

### NUMERICAL ANALYSIS

**Governing Equations**

For high heat fluxes, high flow rates are necessary for better heat transfer and reduced absorber tube’s circumferential temperature difference. Therefore, the flow inside the receiver’s absorber tube is considered fully developed turbulent. Further, we assumed flow and heat transfer to be steady-state. Therefore, the governing equations are the Reynolds averaged Navier-stokes (RANS) equations given as [18]

**Continuity**

\[
\frac{\partial(\rho \vec{u}_i)}{\partial x_i} = 0
\]

(2)

**Momentum equation**

\[
\frac{\partial}{\partial x_j} \left( \frac{\rho \vec{u}_i \vec{u}_j}{x_j} \right) = -\frac{\partial \vec{p}}{\partial x_j} + \frac{\partial}{\partial x_j} \left[ \mu_t \left( \frac{\partial \vec{u}_i}{\partial x_j} + \frac{\partial \vec{u}_j}{\partial x_i} \right) - \frac{2}{3} \mu_t \frac{\partial \vec{u}_k}{\partial x_k} \delta_{ij} - \rho \vec{u}_i \vec{u}_j \right] + S_m
\]

(3)

**Energy equation**

\[
\frac{\partial}{\partial x_j} \left( \rho_p \vec{u}_i \vec{u}_j T \right) = \frac{\partial}{\partial x_j} \left( \lambda \frac{\partial T}{\partial x_j} \right) + \frac{\partial}{\partial x_j} \left( \sigma_{ht} \frac{\partial T}{\partial x_j} \right) + \frac{\partial}{\partial x_j} \left( \frac{\partial(c_p T)}{\partial x_j} \right) + \sum_{n=1}^{R} \frac{\partial}{\partial x_j} \left( \frac{\partial T}{\partial x_j} \right) + \sum_{n=1}^{R} \frac{\partial}{\partial x_j} \left( \frac{\partial T}{\partial x_j} \right)
\]

(4)

### Table 1 Geometrical and simulation parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absorber tube diameter, (d_{ri}) (m)</td>
<td>0.066</td>
</tr>
<tr>
<td>Absorber tube thickness, ((d_{ro}-d_{ri})/2), (m)</td>
<td>0.002</td>
</tr>
<tr>
<td>Glass cover inner diameter (m)</td>
<td>0.115</td>
</tr>
<tr>
<td>Inlet temperatures, (T_{inlet}) (K)</td>
<td>400, 500, 600 and 650</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>1.02 × 10^4 – 7.38 × 10^5</td>
</tr>
<tr>
<td>Direct normal solar irradiance (W/m²)</td>
<td>1000</td>
</tr>
<tr>
<td>Geometrical concentration ratio, (C_R)</td>
<td>86</td>
</tr>
<tr>
<td>Rim angle, (\phi_r)</td>
<td>80°</td>
</tr>
<tr>
<td>Absorber tube absorptivity, (\alpha_{abs})</td>
<td>0.96</td>
</tr>
<tr>
<td>Collector reflectivity, (\rho)</td>
<td>0.96</td>
</tr>
<tr>
<td>Glass cover transmissivity, (\tau_g)</td>
<td>0.97</td>
</tr>
<tr>
<td>(\hat{p}_c)</td>
<td>0.06 – 0.18</td>
</tr>
<tr>
<td>(\tilde{r}_c)</td>
<td>0.45-0.91</td>
</tr>
<tr>
<td>(\hat{\beta}_c)</td>
<td>0.40 - 0.90</td>
</tr>
</tbody>
</table>

In Eqs. (3) and (4), \(-\rho \vec{u}_i \vec{u}_j\) are the Reynolds stresses, \(\vec{u}_i\) and \(\vec{u}_j\) are the time-averaged velocity components in the \(i\)- and \(j\)-directions respectively, \(\vec{T}\) is the time-averaged temperature and \(\vec{p}\) the time averaged pressure.

The Reynolds stress components in the RANS equations are determined through the eddy viscosity model, which uses the Boussinesq approach to relate stress to strain as [18]

\[
-\rho \vec{u}_i \vec{u}_j = \mu_t \left( \frac{\partial \vec{u}_i}{\partial x_j} + \frac{\partial \vec{u}_j}{\partial x_i} \right) - \frac{2}{3} \mu_t \frac{\partial \vec{u}_k}{\partial x_k} \delta_{ij}
\]

(5)

In Eq. (6) \(k\) is the turbulent kinetic energy per unit mass given by [18]

\[
k = \frac{1}{2} (\vec{u}_i^2 + \vec{v}_i^2 + \vec{w}_i^2)
\]

(6)

And \(\mu_t\) is the eddy viscosity given by [18]

\[
\mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}
\]

(7)

In Eq. (7), \(\varepsilon\) is the turbulent dissipation rate and \(C_{\mu}\) is an empirical relation/constant depending on the model used.
For turbulent closure, the realizable $k-\varepsilon$ model was used. The $k-\varepsilon$ turbulence modes are the most widely used and validated models for most flows [18, 19]. The realizable model is an improvement of the standard $k-\varepsilon$ with superior performance compared to all $k-\varepsilon$ models for separated flows and flows with secondary complex features. In the realizable $k-\varepsilon$ model, two additional equations for turbulent kinetic energy and turbulent dissipation rate are solved together with the RANS equations. Detailed description of the realizable $k-\varepsilon$ model is given in Ref. [18].

The source term ($S_{m}$) in the momentum equation (Eq. (3)) represents the pressure drop across the perforated conical insert. The perforated conical insert is modeled as porous media of finite thickness with directional permeability over which there is a pressure drop. The pressure drop across the perforated insert is defined as a sum of the viscous term according to Darcy’s law and an inertial loss term [20] as:

$$\nabla P = -\left(\frac{\mu}{\alpha_p} \frac{\nu}{2} \rho |\nabla u| \right) \Delta m$$

(8)

where $\alpha_p$ is the permeability of the porous medium, $C_{2p}$ is the inertial resistance factor, $\Delta m$ is the thickness of the porous media. For perforated plates, it has been shown that the first term is negligible such that only the inertial loss term should be considered [20, 21]. The coefficient $C_{2p}$ has been determined from data presented by Weber et al. [22] for perforated plates and flat bar screens. In the stream wise direction $C_{2p} = 853 \text{ m}^{-1}$ for the considered porosity of 0.65, and insert thickness of 0.0015 m, in the other directions inertial resistance factors of much higher magnitudes are specified to restrict flow in those directions.

**Boundary Conditions**

The boundary conditions used in this study are: (1) a non-uniform heat flux on the outer wall of the absorber tube. The heat flux distribution on the absorber tube circumference is shown in Fig. 2 at rim angles of 80° and 120°, which was obtained using SoiTrace [23]. In Fig. 2, the receiver’s absorber tube circumference is spread and the angles of 0° and 360° shown in Fig. 2 correspond to top most part of the tube i.e. an angle 90° in Fig. 1(b). For this study, the rim angle ($\phi_r$) used was 80° and the aperture width was 6 m or a concentration ratio of 86. The receiver angle $\theta$, is the receiver’s circumferential angle as shown in Fig. 1(b). A direct normal irradiance (DNI) of 1000 W/m² was used. (2) Periodic boundary conditions are used for the absorber tube’s inlet and outlet. (3) The inner absorber tube walls are considered no-slip and no-penetration. (4) For the inlet and outlet of the receiver’s annulus space, symmetry boundary condition is used such that the normal gradients for all flow variables are zero. (5) For the outer glass cover, a mixed boundary condition is used to account for both radiation and convection heat transfer. For radiation heat transfer from the receiver, the receiver exchanges heat by radiation with the larger enclosure, the sky.

The sky temperature is determined as a function of the ambient temperature from [24]:

$$T_{sky} = 0.0552 T_{amb}^{1.5}$$

(9)

The ambient temperature used is 300 K. The convection heat transfer coefficient for heat transfer between the glass cover and the ambient is given by [25]

$$h_w = V_w^{0.58} d_{go}^{-0.42}$$

(10)

where $V_w$ is the wind speed, taken as 2 m/s in this study and $d_{go}$ is the glass cover outer diameter.

On the symmetry plane, the normal velocity and the normal gradients of all flow variables are zero.

**Solution Procedure**

The numerical solution was obtained using a commercial software package ANSYS® 14.5. The governing equations together with the boundary conditions were solved using a finite-volume approach implemented in a computational fluid dynamics code, ANSYS FLUENT [20]. The computational domain was discretized using tetrahedral mesh elements with a structured mesh in the absorber tube’s inner wall normal direction. The coupling of pressure and velocity and was done using the SIMPLE algorithm. Second-order upwind schemes were employed for integrating the governing equations together with the boundary conditions over the computational domain. To capture the high resolution of gradients in the near wall regions, the $y'$ value of about 1 was used for all simulations. $y' = y u_t / \nu$, where $\nu$ is the fluid’s kinematic viscosity, $y$ is the distance from the wall and $u_t$ is the friction velocity. The enhanced wall treatment was used for modeling the near-wall phenomena for such low values of $y'$. The solution was considered converged when the scaled residuals were in the order of less than $10^{-4}$ for the continuity equation, less than $10^{-5}$ for the momentum equation, and less than $10^{-6}$ for the energy equation.

Fig. 2. Heat flux distribution on the receiver’s absorber tube at different concentration ratios.
for velocity, turbulent kinetic energy and turbulent dissipation rate and less than $10^{-7}$ for energy. Convergence was further monitored using the convergence history of volume-averaged entropy generation. The solution was considered converged when the volume-averaged entropy generation remained constant for more than 200 successive iterations. The entropy generation model is similar to one used in our previous work [8].

The heat transfer fluid used was SYLTHERM800 [26]. Constant thermo-physical properties were used at each temperature considered as derived from the product’s technical data [26]. Table 2 shows the physical properties of the heat transfer fluid at the different temperatures.

### Table 2 Syltherm 800 thermal properties at $T_{inlet} = 400$ K, 550 K and 650 K

<table>
<thead>
<tr>
<th>Thermal property</th>
<th>400 K</th>
<th>550 K</th>
<th>650 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific heat capacity ($c_p$), J/kg K</td>
<td>1791.43</td>
<td>2047.32</td>
<td>2218.26</td>
</tr>
<tr>
<td>Density ($\rho$), kg/m$^3$</td>
<td>840.06</td>
<td>696.01</td>
<td>577.70</td>
</tr>
<tr>
<td>Thermal conductivity ($\lambda$), W/m K</td>
<td>0.114845</td>
<td>0.086661</td>
<td>0.067833</td>
</tr>
<tr>
<td>Viscosity ($\mu$), Pa.s</td>
<td>0.002163</td>
<td>0.000555</td>
<td>0.000284</td>
</tr>
</tbody>
</table>

Stainless steel (321H) was used as the absorber tube material with temperature dependent thermal conductivity. Pyrex® was used as the glass cover material [27].

Grid dependence tests were carried out for representative cases at all Reynolds numbers considered in the study. The solution was considered grid independent when the maximum change of Nusselt number and friction was less than 1% as the mesh element size was changed. The sample grid dependence studies are shown in Table 3. The indices $i$ and $i+1$ indicate the mesh before and after refinement respectively.

### Table 3 Sample mesh dependence studies at $T_{inlet} = 400$ K

<table>
<thead>
<tr>
<th>Mesh</th>
<th>$f$</th>
<th>$Nu$</th>
<th>$f^{i+1} - f^{i-1}$</th>
<th>$Nu^{i+1} - Nu^{i-1}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>114,510</td>
<td>0.689</td>
<td>2074.4</td>
<td>0100</td>
<td>0.087</td>
</tr>
<tr>
<td>226,331</td>
<td>0.691</td>
<td>1908.6</td>
<td>0.021</td>
<td>0.012</td>
</tr>
<tr>
<td>319,413</td>
<td>0.690</td>
<td>1885.9</td>
<td>0.002</td>
<td>0.003</td>
</tr>
<tr>
<td>434,744</td>
<td>0.690</td>
<td>1880.6</td>
<td>0.002</td>
<td>0.003</td>
</tr>
</tbody>
</table>

### DATA REDUCTION

In this study, friction factor, Nusselt number and absorber tube’s circumferential temperature difference were used to characterize the thermal performance of receiver.

The friction factor is calculated according to

$$f = \frac{\Delta P}{\frac{1}{2} \rho \cdot u_m^2 \cdot \frac{L_p}{d_i}}$$

where $\Delta P$ is the pressure drop, $\rho$ is the density of the heat transfer fluid, $d_i$ is the inner diameter of the absorber tube, $u_m$ is the mean velocity in the absorber tube, $L_p$ is the length of the periodic module. Where $L_p = p/2 + p/2$.

The heat transfer is given in terms of the average heat transfer coefficient, $h$ and the average Nusselt number, $Nu$. The average heat transfer coefficient is given by

$$h = \frac{q^*}{(T_a - T_b)}$$

Where $T_a$ is the average temperature of the absorber tube’s inner wall and $T_b$ is the bulk temperature of the fluid at the periodic boundaries. This is the average temperature of the inlet and outlet temperatures of the periodic module. From Eq. (12), the average Nusselt number is given by

$$Nu = h \cdot d_i / \lambda$$

In which, $\lambda$ is the thermal conductivity of the heat transfer fluid.

The flow Reynolds number is given as

$$Re = \frac{u_m \cdot d_i}{\nu}$$

Where, $\nu$ is the kinematic coefficient of viscosity of the heat transfer fluid.

The absorber tube’s circumferential temperature difference is defined as the difference between the average temperature of the lower and upper halves of the absorber tube according to

$$\varphi = T_{r,ul} - T_{r,up}$$

To investigate the actual collector thermal performance, the actual gain in collector performance due to heat transfer enhancement should be compared with the corresponding increase in pumping power. Collector performance can be characterized in terms of collector’s thermal efficiency which is a function of heat transfer rate and incident solar radiation. For comparison of a receiver with perforated inserts with a non-enhanced receiver, the thermal efficiency has been modified to include the pumping power. The modified thermal efficiency is function of the heat transfer rate, pumping power and incident solar radiation according to

$$\eta_{th,m} = \frac{\dot{Q}_a - W_p}{A_a \cdot I_b}$$

In Eq. (16), $\dot{Q}_a = m \cdot c_p \cdot (T_{outlet} - T_{inlet})$ is the heat transfer rate; $W_p = \dot{V} \Delta P$ is the pumping power; $A_a$ is collector’s aperture area and $I_b$ is the incident solar radiation.
RESULTS AND DISCUSSION

Code Validation

Our study was validated in a number of steps. First we validated the receiver model using test data from Dudely et al. [17] for temperature gain, and then using experimental data from Burkholder and Kutscher [28] for heat loss validation. This validation of the receiver model was presented in our earlier work in Ref. [8].

The perforated insert model was validated using data from Guohui and Saffa [29]. The variation of the pressure coefficient \( k_p = p_s/p_v \) with distance from the perforated plate is shown in Fig. 3. The same trend as obtained by Guohui and Saffa [29] was obtained in the current work, with maximum deviation of less than 3%. (\( p_s \) is the static pressure and \( p_v \) is the velocity pressure = \( 1/2 \rho v^2 \)).

Heat Transfer Results

The circumferential temperature difference is expected to increase as the concentration ratios increase. Figure 4 shows the variation of absorber tube’s circumferential temperature difference with Reynolds number and concentration ratio for a receiver with a plain absorber tube. As expected in the figure, the temperature difference will reduce as the Reynolds numbers increase.

Figure 4 further shows that, at a given Reynolds number, an increase in the concentration ratio increases absorber tube’s circumferential temperature difference due to the resulting higher heat fluxes. Therefore, as the concentration ratios increase, higher flow rates become necessary to reduce the temperature difference in the absorber tube. Heat transfer enhancement is the other option to improve the receiver’s performance and reduce absorber tube’s temperature difference. In this work, the use of perforated conical inserts for heat transfer enhancement in the receiver’s absorber tube is considered. The results for heat transfer enhancement using perforated conical inserts are presented in the following sections.

Figures 5 and 6 show the variation of the Nusselt number with Reynolds number for different values of insert spacing when \( \bar{\tau} = 0.91 \) for \( \bar{\beta} = 0.70 \) and \( \bar{\beta} = 0.90 \) respectively, for an inlet of temperature 600 K. While Fig. 7 shows the variation of Nusselt number with Reynolds number for different values of insert spacing when \( \bar{\tau} = 0.91 \) for \( \bar{\beta} = 0.70 \) for an inlet temperature of 650 K. As shown, the Nusselt number continues to increase as the Reynolds number increases due to the thinner boundary layer at higher Reynolds numbers.

The Nusselt number is also shown to increase as the insert spacing reduces due to increased flow impingement and turbulent intensity resulting from the use of many inserts per unit meter. For the range of parameters considered, the change in Nusselt number as the insert spacing reduces is not very significant as can be seen in Figs. 5-7. The heat transfer rate at higher temperatures is higher than that at lower temperatures. At high temperatures, the Reynolds numbers are higher for the same flow rate thus thinner boundary layer.

Fig. 4. Variation of absorber tube’s circumferential temperature difference with Reynolds number and concentration ratio.

Fig. 5. Variation of Nusselt number with Reynolds number at various values of insert spacing for \( \bar{\beta} = 0.71 \) and \( \bar{\tau} = 0.91 \) at 600 K
increases, the free flow area reduces resulting in increased turbulence intensity and flow impingement on the absorber tube’s walls. As shown in Fig. 8, the Nusselt number increases as the insert’s cone angle increases. Significant increase in Nusselt number are shown at angles ($\beta_c$) higher than 0.7. For the range of parameters considered, the heat transfer is enhanced in the range 1.05-2.24 times compared to that of a plain receiver tube for $0.45 \leq \tilde{r} \leq 0.91$.

### Friction Factor Results

As expected, while heat transfer enhancement increases performance, it also increases fluid friction. Figure 9 shows the variation of friction factor with insert size. As shown, fluid friction significantly increases with the size of the insert. Also the fluid friction increases significantly as the insert’s cone angle increases. Insert size and insert’s cone angle significantly affects fluid friction as compared to heat transfer performance.

Figure 10 - 11 show the variation of friction factor with Reynolds number at different values insert spacing.

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At a given cone angle, the Nusselt numbers are shown to increase with increase in insert size. As the insert size increases, the free flow area reduces resulting in increased turbulence intensity and flow impingement on the absorber tube’s walls. As shown in Fig. 8, the Nusselt number increases as the insert’s cone angle increases. Significant increase in Nusselt number are shown at angles ($\beta_c$) higher than 0.7. For the range of parameters considered, the heat transfer is enhanced in the range 1.05-2.24 times compared to that of a plain receiver tube for $0.45 \leq \tilde{r} \leq 0.91$.

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As shown, reducing the insert spacing significantly increases the friction factor. Thus, the insert spacing, insert size and insert cone angle should be chosen carefully to increase heat transfer performance while keeping fluid friction low. The general trend of friction factor in turbulent flow is also shown to exist, i.e. friction factor reduces slightly as Reynolds numbers increase.

![Fig. 11. Variation of friction factor with Reynolds number at different values insert’s spacing for \( \beta_c = 0.90, \tilde{r}_c = 0.61 \) and 500 K](image)

**Absorber Tube Temperature Difference**

The improved heat transfer performance in the receiver’s absorber tube is expected to reduce the absorber tube’s circumferential temperature difference. Figures 12 and 13 show the variation of absorber tube temperature difference with Reynolds number at different values of insert spacing. As shown in the figures substantial reductions in the absorber tube’s circumferential temperature difference can be achieved.

![Fig. 12. Variation of absorber tube circumferential temperature difference with Reynolds at different values insert’s spacing for \( \beta_c = 0.60, \tilde{r}_c = 0.90 \) and 650 K](image)

The reductions are significant and crucial at low Reynolds numbers where the temperature difference is higher than 50 K, the temperature difference for safe working of the receiver. For example, at \( \beta_c = 0.60, \tilde{r}_c = 0.90, \tilde{p}_c = 0.06 \) and an inlet temperature of 650 K, the circumferential temperature difference in the absorber tube is about 89.5 K compared to 171. 5 K in a non-enhanced receiver tube when the Reynolds number is 54,890. For the same geometrical parameters, the circumferential temperature difference is 31.54 K compared to 59.7 K of a non-enhanced receiver tube as the Reynolds number increases to 202,688.

For the range of parameters considered, the absorber tube’s circumferential temperature difference reduced by 3.40 – 55.6%.

Apart from ensuring safe operation of the receiver, any reduction in the absorber tube’s circumferential temperatures or actual temperatures of the receiver’s absorber tube will lead to lower absorber tube temperatures and this lowers radiation losses. Moreover, at low absorber tube temperatures, the coating emissivity reduces. This will reduce the receiver’s thermal loss and improve the performance of the receiver.

![Fig. 13. Variation of absorber tube circumferential temperature difference with Reynolds at different values insert’s spacing for \( \beta_c = 0.90, \tilde{r}_c = 0.91 \) and 650 K](image)

**Thermal Performance Evaluation**

Equation 16 provides a means of comparing an enhanced receiver tube with a plain receiver tube. The collector thermal efficiency is a measure of how much of the incident energy does useful work. In Eq. (16), the pumping power has been included in the equation for collector thermal efficiency to account for increase in pumping power due to heat transfer enhancement.

Figures 14(a&b) show the variation of collector thermal efficiency with insert spacing. The thermal efficiency of an enhanced receiver tube is shown to be higher than that of a plain receiver tube below some Reynolds number. The reduction in absorber tube temperatures, resulting reduction in absorber tube coating emissivity and improved heat transfer rates are the main factors leading to the increase in thermal efficiency. The thermal efficiency is shown to become lower than that in a collector with a non-enhanced receiver above some Reynolds number. This is because at values of Reynolds number higher than this, the achieved gain in performance is lower than the increase in pumping power. Similar trends can
be obtained for other inlet temperatures and different combination of insert orientation and insert size.

The Reynolds number at which the thermal efficiency becomes lower than that in a plain receiver depends on the orientation angle, insert size and insert spacing. In general, increase in efficiency in the range 3-8% are obtainable in the range

$$0.10 \leq \tilde{\beta}_c \leq 0.18, \quad 0.45 \leq \tilde{\tau} \leq 0.76 \quad \text{and} \quad 0.40 \leq \tilde{\beta} \leq 0.70$$

provided the flow rates are lower than 0.0103 m$^3$/s (about 8.6 kg/s at 400 K, 7.66 kg/s at 500K, 6.56 kg/s at 600 K and 5.92 kg/s at 650K). Outside the ranges mentioned above, efficiency increase may be achieved if the flow rate and geometrical parameters are properly matched. For example, at flow rates higher than the one mentioned above, lower insert sizes, lower orientation angles and larger spacing might provide an increase in efficiency. The maximum increase in collector thermal efficiency is shown at the lowest Reynolds number.

**CORRELATIONS**

Based on the results from our numerical investigation, correlations for Nusselt number and friction factor were derived using regression analysis.

For a smooth absorber tube, the Nusselt number correlation was obtained as

$$Nu = 0.0104 Pr^{0.374} Re^{0.885} \quad (17)$$

$$R^2 = 1.0$$ for this correlation and the correlation predicts the Nusselt number within ±4%.

The friction factor correlation for a plain receiver is

$$f = 0.173 Re^{-0.1974} \quad (18)$$

$$R^2 = 0.994$$ for the friction factor correlation and the correlation is valid within ±3.5%

Equation (17) and Eq. (18) are valid for

$$1.02 \times 10^4 \leq Re \leq 7.38 \times 10^5$$

$$9.29 \leq Pr \leq 33.7$$

$$400 \text{ K} \leq T \leq 650 \text{ K}$$

For the receiver tube with perforated conical inserts, the Nusselt number is correlated by

$$Nu = 0.005746 \left( \tilde{\beta}_c \right)^{-0.1840} \left( \tilde{\tau} \right)^{0.3348} \left( 1 + 0.02759 \tan \beta_c \right) \left( Re \right)^{0.9355} \left( Pr \right)^{0.3968} \quad (19)$$

The coefficient of determination ($R^2$) is about 0.991 for this correlation. Figure 15 shows the parity plot for the Nusselt number. The correlation is valid within ±12% for the range of parameters considered (tested for over 552 data points).

**CONCLUSION**

In this study, heat transfer enhancement in parabolic trough receiver using perforated conical inserts was investigated. The use of perforated conical inserts is shown to improve the heat transfer performance, reduce absorber tube circumferential temperature differences, improve receiver thermal performance and increase fluid friction. Increase in insert size, increase in insert cone angle and reduction in insert spacing are shown to improve the heat transfer performance but at the expense of increased fluid friction.

The use of perforated conical inserts increases the thermal efficiency in the range 3-8% provided the flow rate does not...
exceed 0.0103 m³/s (about 8.6 kg/s at 400 K, 7.66 kg/s at 500K, 6.56 kg/s at 600 K and 5.92 kg/s at 650K) and for geometrical parameters in the range
\[0.10 \leq \tilde{\rho} \leq 0.18, \quad 0.45 \leq \tilde{\varepsilon} \leq 0.76\]
and \[0.40 \leq \tilde{\beta} \leq 0.70\]. Above this flow rate, the increase in pumping power is much more than the increase in thermal performance.

The increase in thermal efficiency is due to increased heat transfer performance, reduction in absorber tube temperatures, and reduced coating emissivity at these lower temperatures. Reduced absorber tube temperatures and low coating emissivity reduces the receiver’s radiation loss.

![Fig. 16: Nusselt number parity plot](image)

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