Influence of Rim Angles and Heat flux Distribution Boundary on the Heat Transfer of an Absorber Tube for a Parabolic Trough Solar Collector

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Abstract. One of the most important solar concentrator among other concentrating collectors is the parabolic trough solar collector because of its successful application in solar electricity generation. This study investigated the influence of collector rim angles and non-uniform heat flux distribution boundaries on the inner heat transfer coefficients of an absorber tube for a parabolic trough solar collector. Laminar flow steady-state condition was considered where buoyancy effects were significant. Numerical simulation was conducted in ANSYS Fluent version 14.5. Sinusoidal non-uniform heat flux distributions boundary conditions were implemented via a user defined function in Fluent. It was established that the absorber tube-wall temperature rise with a rise in the collector rim angle and the circumferential span of non-uniform heat flux allocations limit. It was furthermore revealed that when buoyancy-driven secondary flow is present, the internal heat transfer coefficient increased more than twice advanced than the pure forced-convection case (i.e. no buoyancy effects), and representing heat transfer improvement due to secondary flow effects. Also, when buoyancy effects are present, the mean internal heat transfer coefficient improved as the heat flux intensity is increased, the collector rim direction and the circumferential span of the non-uniform heat flux distributions boundary. This study further discovered that the mean internal heat transfer coefficient became higher with a rise in the temperature of the absorber tube inlet flowing fluid.

Keywords: collector rim angle, heat flux, heat transfer, secondary flow effects

1.0 Introduction

The parabolic trough solar collector is a very important solar concentrator due to its successful applications in the solar electricity generation in the late 1980s in the Mojave Desert of southern California. Parabolic trough solar collectors which track the sun in a particular axis are made up of a parabolic produced glass reflector and a recipient tube enclosed in a glass tube to decrease heat losses [1]. As shown in Figure 1 (a) it concentrates the direct solar radiation on to a linear receiver tube positioned along its focal line.
Figure 1 Parabolic trough solar energy collector [2]  
Figure 2 Parabolic trough solar energy collector receiver [3]

The solar heat flux incident on the receiver is absorbed by a working fluid flowing in the collector absorber tube [4]. It can heat fluid up to temperatures range of 50°C and 400°C. It has concentration ratios ranging from 10 to 85 [5]. The glass cover of the receiver has a limitation that the reflected rays must pass through glass cover before impinging on the tube with transmittance loss of about 0.9, when the glass is clean [5]. On the other hand, the glass cover is generally treated with an anti-reflective coating to decrease loss due to the transmittance, thus improving the performance of the collector. The optical effectiveness of a classic parabolic trough solar concentrator ranges from 75 to 80% [6]. Figure 2 show a parabolic trough solar collector receiver which is made of a steel absorber tube mounted inside an anti-reflective evacuated glass tube [3]. The selective coating on the steel absorber tube has a multilayer cermet coating to increase solar absorptivity and decrease thermal emissivity to low radiation loss. The glass-to-metal seals and metal bellows are used to sustain the vacuum in annular zone and permit thermal expansion between the steel and glass tubes. The vacuum inside the annulus decreases convective heat loss. [7] Disclosed that normal convective heat loss in the annulus may be insignificant as long as the Rayleigh number is less than 1000. [8] Numerically studied the heat loss from the receiver tube and discovered that the heat losses rise with the length of the collector receiver and the difference between the temperature of the fluid and the surroundings.

In parabolic trough solar collectors, the concentrated heat flux hits on a horizontal absorber tube from underneath thus creating circumferential non-uniform heat flux distributions in the region of the tube-wall and hence non-uniform heating of the fluid. In a laminar or weak turbulent flow system, the buoyancy induced secondary flow effect resulting from the non-uniform heating of the fluid can influence harnessing of solar heat flux incident on the receiver. The induced secondary flow effect improves the thermal mixing rate of the fluid, in this manner increasing the inner heat transfer rate.

This study investigates the influence of the collector aperture rim angles and non-uniform heat flux distribution boundaries on the absorber tube inner-wall heat transfer coefficients for a parabolic trough solar concentrating collector. The absorber tube solar collectors play an
important role in translating incident solar heat flux into absorbed thermal energy and transmitting it to a heat transfer fluid.

2.0 Model Description and numerical formulation

Figure 3 reveals a schematic depiction of a parabolic trough solar collector design with a reflector that focuses the solar radiation from the base segment of an absorber tube. The absorber tube has an external diameter of $D_o$, an internal diameter of $D_i$ and length of $L$. Figure 4 gives a computation domain used in this study, which consists of an absorber model in a horizontal orientation with liquid water as heat transfer fluid. The pipe wall is divided into $N \times M$ number of segments, where $N = 36$ is used to explain the external tube-wall heat flux distribution boundary. The pipe has a wall thickness of $t$, internal diameter $D_i$, external diameter of $D_o$, and $L_{TOT}$ as total length.

Figure 3 Parabolic trough solar collector designs. Figure 4 shows an absorber tube model.

Considering the element $(m, n)$ in Figure 4, under steady state conditions, Eq. 1 can be obtained based on energy balance principle.

$$q_{o,(m,n)} = q_{i,(m,n)} + q_{x,(m,n)} + q_{x,(m+1,n)} + q_{\phi,(m,n)} + q_{\phi,(m,n+1)} + q_{\text{conv},(m,n)} + q_{\text{rad},(m,n)}$$  \hspace{1cm} (1)

where $q_{o,(m,n)}$ is the heat transfer rate on the external wall plane at position $(m, n)$ and $q_{i,(m,n)}$ gives heat transfer rate to the fluid in position $(m, n)$. The mean inner heat transfer coefficient, $\bar{h}_i$ considering the total length of the pipe model in terms of the internal-wall surface temperature, $T_{w,i}$ is represented in Eq. 2:

$$\bar{h}_i = \frac{\sum_{m=1}^{M} \sum_{n=1}^{N} q_{i,(m,n)}}{2\pi R_i L_{TOT} (T_{w,i} - T_b)}$$  \hspace{1cm} (2)
with Eq. (1), $q_{x,(m,n)}$ and $q_{x,(m+1,n)}$ describe the axial conductive heat transfers, while $q_{\phi,(m,n)}$ and $q_{\phi,(m,n+1)}$ describe the tangential conductive heat transfers, which are modelled based on Fourier law [9]. $q_{\alpha,\text{conv.}(m,n)}$ is the heat transfer loss by forced-convection from the external-wall plane at $(m, n)$ to the surroundings, modelled from Newton’s law of cooling, while $q_{\alpha,\text{rad.}(m,n)}$ is the radiation heat transfer loss to the surroundings, modelled based on Stefan-Boltzmann law of the emissive power of a surface under thermodynamic temperature condition [9].

3.0 Numerical model solution procedure and model validation
A steady-state temperature within the fluid and tube wall was determined by numerical simulations conducted for laminar flow in ANSYS Fluent version 14.5 [10]. Reynolds numbers range of 120 to 1,800 is considered. Heat transfer fluid was assumed incompressible. Governing equations are the momentum, continuity, and energy equations. The equations are solved by employing finite volume method described by Patanker [11]. No-slip condition is applied at the interior wall surface of the pipe. Uniform tube outlet pressure boundary was used. Sinusoidal non-uniform external pipe-wall heat flux boundary conditions were implemented via a user defined function in Fluent. A grid independence study was conducted. The model validation was conducted by based on the Nusselt number, $Nu = 4.36$ [9] for the inlet Reynolds number of 202. The axial local Nusselt number figures obtained from the numerical model agreed very well and only deviated by 7% in terms of $Nu = 4.36$, at $Gz^{-1} \approx 0.0828$, as the flow developed down the exit of the pipe and reduced to 4% at $Gz^{-1} \approx 0.128$.

4.0 Results and Discussions
4.1 Non-uniform solar radiation heat flux distributions for different collector rim angles

![Figure 5](image-url) Temperature contours at different collector aperture rim angles ($\varphi_r$) and non-uniform radiation heat flux distributions ($\alpha$)
Figure 5 indicates variations temperature contours for the collector aperture rim angles, $\varphi_r = 70^\circ$, $90^\circ$, $110^\circ$ and $130^\circ$ and for sinusoidal non-uniform heat flux distribution boundaries for $\alpha = 140^\circ$, $180^\circ$, $220^\circ$ and $260^\circ$ respectively. The arrows indicated the flow path. The pipe inlet fluid temperature of 300 K was used. The pipes considered have external diameter of 73 mm and wall thickness of 5.2 mm. It can be noticed that the external-wall surface temperatures increased in the flow path. The underneath region of the pipe was warmest, as revealed by the red shade. As the collector aperture rim angle and the span of the heat flux boundary increased, a bigger segment of the tube’s external surface was at temperatures close to the maximum temperature. The temperatures at the upper region of the pipe were small since there was a minute incident heat flux in those surfaces. These differences in the temperature result in density differential in the heat transfer fluid, which results in buoyancy-driven secondary flow, and thereby improves the thermal mixing rate of the fluid in the laminar or weak turbulent flow region.

The variation of the non-uniform pipe-wall temperature on the outer surface of the absorber pipes in Figure 5 is presented in Figure 6 for an inlet Reynolds number of 1100. Figure 6 shows the length-wise mean external surface temperature plots for the section numbers $n = 1$ to 36, for the sinusoidal non-uniform heat flux boundaries considered and for the collector aperture rim angles, $\varphi_r = 70^\circ, 90^\circ, 110^\circ$ and $130^\circ$ in that order. It reveals that the external-wall temperature increased with a rise in the collector aperture rim angle and the circumferential span of the heat flux distributions on the surface of the absorber pipe receiving the incident heat flux. The highest external-wall temperature occurred at the underneath region of the pipe, which corresponded to where the heat flux boundary reached its maximum and decreased to the top portion of the tube that received small amount of incident heat flux.

Figure 6 Non-uniform pipe-wall temperature variation over the circumferential surface at different collector aperture rim angles
4.2 Heat transfer coefficients for collector rim angles and circumferential non-uniform heat flux distributions boundary

The effects of buoyancy-driven secondary flow on the internal convective heat transfer for different collector rim angles and non-uniform heat flux boundaries for inlet the Re = 300 to 1300 and heat flux intensity of 7.1 kW/m² are plotted in Figures 7 and 8. As expected, the mean internal heat transfer coefficients increased with Re. It can be seen in Figure 7 that at different inlet Re, there was no appreciable increase in the internal heat transfer coefficients as well as the collector rim angles, when buoyancy effect with respect to temperature dependent of the fluid density was not considered.

![Figure 7](image1.png)

**Figure 7** Average internal heat transfer coefficient with the collector rim angle without secondary flow effects

![Figure 8](image2.png)

**Figure 8** Internal heat transfer coefficient for collector rim angles with buoyancy effects present
Unlike in Figure 7, it was found that in Figure 8, there was appreciable rise in the mean internal heat transfer coefficient where buoyancy-induced secondary flow effects were present, as the collector aperture rim angle was increased. This is attributed to an improved heat input rate (W) into the absorber tube due to the increase in collector aperture rim angle and the circumferential surface of the tube receiving the incident heat flux. It was establish that between Re = 300 and Re = 1300, the mean internal heat transfer coefficients increased up to 12% and 8% respectively, from $\varphi_r = 70^\circ$ to $130^\circ$.

Figure 9 shows the variations of average internal heat transfer coefficient with the absorber tube inlet fluid temperature at different collector aperture rim angles and heat flux distributions boundaries. For collector aperture rim angles cases investigated, in which buoyancy effect was considered, the internal heat transfer coefficients were 56%, 59% and 61% respectively more than where it was not considered. This shows appreciable increase in internal heat transfer as results of buoyancy effects resulting from the non-uniform heat flux. Furthermore, the internal heat transfer coefficient increased to about 17% and 10%, where secondary flow effects were available and where it was not considered, by raising the fluid inlet temperature between 293 K and 340 K, at the same ambient temperature. This reveals the effects of the inlet heat transfer fluid temperature on the internal heat transfer coefficient of the absorber tube.

5.0 Conclusion
This study investigated the influence of collector aperture rim angles and non-uniform solar heat flux distribution boundaries on internal heat transfer where buoyancy-induced secondary flow was present on an absorber pipe for a parabolic trough solar collector under steady-state laminar flow conditions. A sinusoidal non-uniform heat flux distribution boundary was implemented at the inlet Re. range of 130 to 1,800. It was established that where buoyancy effects is available, the internal heat transfer coefficients for the absorber pipe increased with the span of the non-
uniform heat flux distributions boundary and for the collector aperture rim angles cases considered. With the buoyancy effect, the internal heat transfer coefficient was about 61% more than where there were no secondary flow effects. This indicates high internal heat transfer improvement because of buoyancy effect due to the non-uniform heat flux distributions boundary. Thus, the buoyancy induced secondary flow effects occurring in laminar flow cases or weak turbulent flow cases could be very useful to improve absorption of solar heat flux to reduce heat loss and improve thermal performance of the absorber pipe. It was also discovered that at the same ambient temperature condition, the internal heat transfer coefficient rise with a rise in the absorber pipe inlet fluid temperature.

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