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Computational Fluid Dynamics Analysis of Linear Solar Collector by using Semicircular Absorber Tube

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ABSTRACT: In the present study, thermal analysis of a linear parabolic solar collector has been studied by using circular and semicircular absorber tube. Computational fluid dynamics analysis has been carried out by using non uniform heat flux distributions on the surface of the absorber tube. The non uniform heat flux distribution on the absorber tube is calculated by using Monte Carlo ray tracing method in the SolTrace software. The results are presented for the conventional absorber tube receiver and semicircular absorber tube receiver, for the range of inlet temperature 300 K to 650 K and Reynolds numbers 0.25×10^5 to 2.82×10^5. The maximum gain in the collector efficiency of the parabolic trough collector is found to be 5% at the Reynolds number equal to 25000.
KEYWORDS: Computational fluid dynamics, solar thermal power, concentrated solar power, concentrated solar collector, linear solar collector, Finite volume method, Monte Carlo ray tracing method, Semicircular absorber tube

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I. INTRODUCTION

Parabolic trough collector is an industry scale solar power generation technology. In the past few decades, many authors presented various experimental and numerical studies to improve the performance of the PTC. The studies include improvements in design, receiver thermal performance, selective coating performance, the effect of optical and geometrical errors on the performance, improvements in the measurement of optical errors and heat transfer fluid performance. One of the most important challenges is to increase the heat to be transferred from the absorber tube to the heat transfer fluid. An improvement in heat transfer rate reduces the total number of receiver elements required in a solar field. The less number of receiver elements in a solar field reduces the installation and operation cost thus makes the PTC system more economical. In active and passive methods, the passive method is widely accepted and it is used to increase the heat transfer rate by increasing the fluid contact area using fins. However, the increase in contact area using fins increases the pressure drop which further leads to more requirement of pumping power. Therefore, it is important to study the heat transfer enhancement and pressure drop in a receiver to optimize the system.

In recent years, researchers have presented different techniques to study the heat transfer characteristics of the PTC receiver. Forristall [1] has studied the heat transfer in PTC receiver using Engineering Equation Solver (EES) software and found a close agreement with the experimental results of Dudley, Kolb, Mahoney, Mancini, Sloan and Kearney [2]. Cheng, He, Xiao, Tao and Xu [3] has analyzed heat transfer in the PTC receiver by coupling MCRT with FVM and compared the results with the experimental results of Dudley, Kolb, Mahoney, Mancini, Sloan and Kearney [2] and found a good agreement with an average relative error in collector efficiency below ±2%. Later, a similar technique has been used by many researchers to study the heat transfer in PTC receiver [4-11]. Since this technique has been widely used and well validated, it has been employed in the present study.

Reddy, Kumar and Satyanarayana [12] studied heat transfer characteristics of PTC receiver by introducing porous fins inside the receiver and reported that inclusion of porous fins in receiver enhances the heat transfer on account of significant pressure drop. Gong, Wang, Wang, Tan, Lai and Han [13] had carried out a numerical study to enhance the heat transfer in a PTC receiver with internal pin fin arrays by using MCRT coupled with FVM technique. They have reported an improvement in the average Nusselt number and overall heat transfer performance factor. The studies [12, 13] suggest that pin fin can enhance the heat transfer but on the cost of significant pressure drop. In an array of pin-fins, a single pin-fin act as a bluff body shedding vortices thus increases the pressure drop. In the current study, a semicircular absorber tube is introduced inside the PTC receiver to enhance the heat transfer.

II. GEOMETRY

The PTC system mainly contains parabolic reflector and receiver assembly as shown in Fig. 1. Cylindrical parabolic concentrator reflects the incident Sun rays on to a receiver located at its focal line. Receiver assembly contains a metal absorber tube enclosed in an evacuated glass tube to allow the concentrated rays to reach the absorber tube. The cross-sectional view of the conventional receiver and receiver with semicircular absorber tube is shown in Fig. 2. The Parabolic concentrators follow the curvature generated by Eq. (1), where 'f' is focal length. The rim angle 'ψ' can be calculated by using Eq. (2) [11], where 'w' is aperture width. The dimensions of the PTC system is presented in Table 1

$$x^2 = 4fy \quad (1)$$

$$\psi = 2\tan^{-1}(a/4f) \quad (2)$$

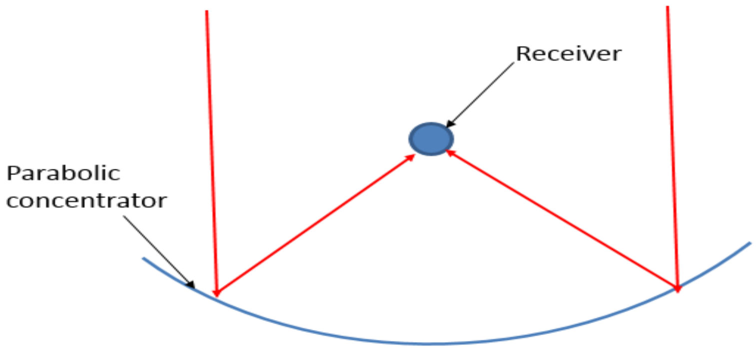


Fig. 1 Schematic of the parabolic trough collector

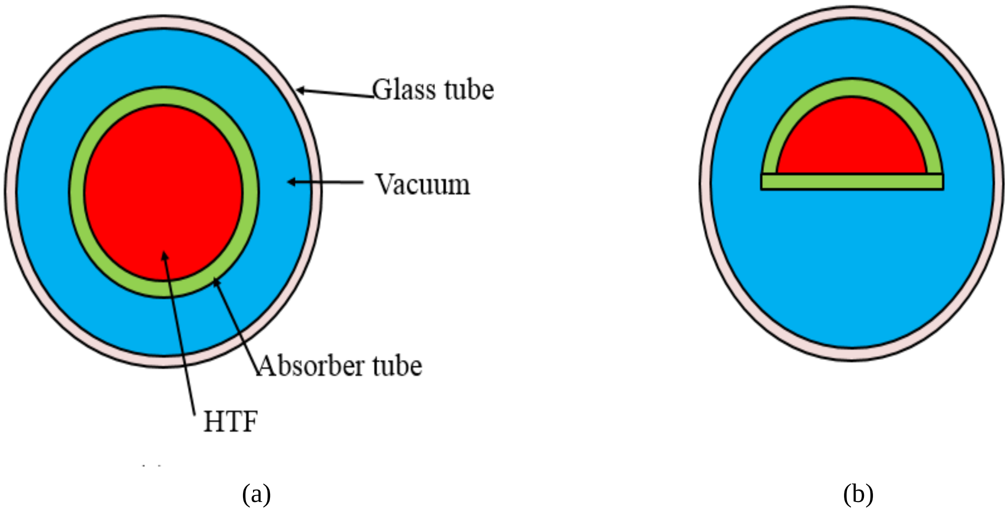


Fig. 2 Cross sectional view of receiver (a) circular absorber tube (b) semicircular absorber tube

Table 1: Geometrical properties of PTC system [2]

Parameter	Value
a	5 m
L	4 m
d_{gti}	109 mm
d_{gto}	115 mm
d_{ati}	66 mm
d_{ato}	70 mm
f	1.84 m

III. RAY TRACING

The distribution of heat flux over the absorber tube is vital for heat transfer analysis. Some studies have presented that heat flux distribution over the absorber significantly affects the thermal performance of the PTC receiver [10]. Monte Carlo Ray Tracing method in SolTrace software is used to calculate the solar flux distribution over the absorber tube[14]. Ray tracing simulations were carried out for the PTC system for the geometrical parameters given in Table 1. Sun has been modeled as a Gaussian distribution with a cone angle of 2.6 mrad [11]. The optical properties of the PTC system are presented in Table 2. An ideal tracking system of the PTC system was assumed where the direct normal irradiation was taken as 1000W/m^2 for all ray tracing simulations. Both the slope and specularity error of the mirror were considered as 3 mrad. A sample ray tracing in SolTrace is shown in Fig. 3. The solar flux distribution obtained from the ray tracing is introduced as a boundary condition in the thermal analysis discussed in the subsequent section.

Table 2: Optical properties of the PTC system [2]

Parameter	Value
α_r	0.92
ρ_r	0.08
τ_g	0.935
ρ_g	0.045
ρ_m	0.93
τ_m	0

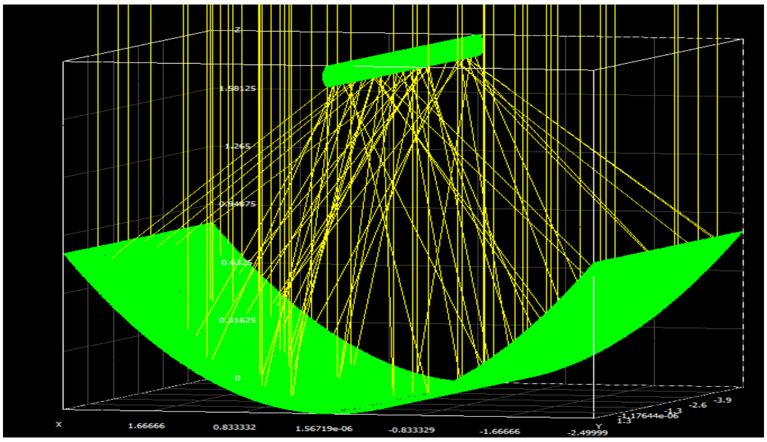


Fig. 3 Sample ray tracing in SolTrace

IV. CFD ANALYSIS

Finite volume method in Fluent software is used to carry out the thermal analysis of the PTC receiver assembly. The realizable k-ε turbulent model is used for the flow simulations. The vacuum space between the absorber tube and glass tube is modeled using discrete ordinates (DO) radiation model. The heat transfer fluid Syltherm 800 is used and its physical properties are specified by a polynomial function of temperature [6]. The steady state flow is assumed and the boundary conditions defined are as follows. (i) Inlet: inlet velocity = 0.3 to 2.52 ms^{-1}, temperature (Tin) = 323 to 650 K. (ii) Outlet: Pressure outlet. (iii) Absorber tube outer surface: distribution of heat flux obtained from ray tracing (iv) Absorber tube inner surface: wall with no slip (v) glass tube outer wall: convection and radiation.

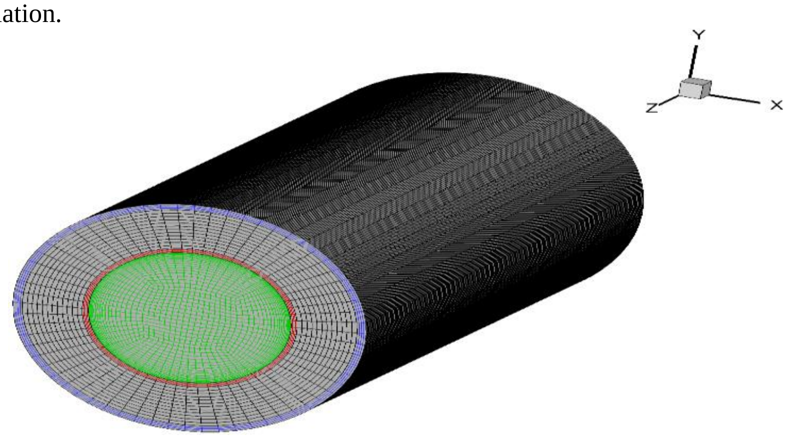
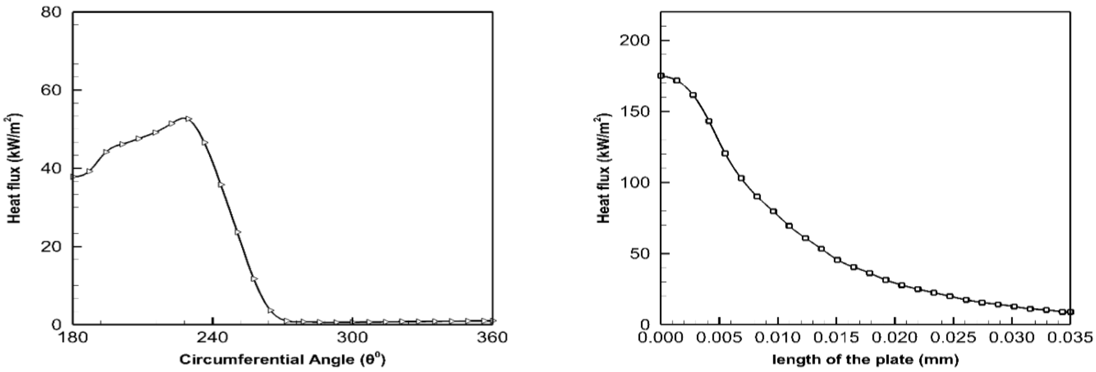


Fig. 4 Mesh of PTC receiver

The geometry of the receiver assembly was created in SolidWorks software and imported to ICEM CFD software [15] for generating structured mesh as shown in Fig. 4. A very fine mesh has been generated near the wall to capture the high velocity and temperature gradients. The generated mesh was further used in Fluent to carry out FVM analysis. Second order upwind scheme was used for discretizing the momentum and energy equation. Turbulent kinetic energy, turbulent dissipation rate, and discrete ordinate equations are discretized using first order upwind scheme. The convergence criteria for scaled residuals are taken as 10^{-5} for all the simulations. A grid independent test was carried out to determine the optimal mesh size with gives 1,252,452. The simulated results were validated with experimental results of Dudley, Kolb, Mahoney, Mancini, Sloan and Kearney [2] and a very good agreement is obtained.

V. RESULTS AND DISCUSSION

This section presents the heat flux and temperature distributions over the absorber tube and collector efficiency at different operating conditions for the semicircular and circular receiver. Fig. 5 shows the heat flux distribution on the circumference of absorber tube obtained by ray tracing. The value of heat flux is high on the lower portion of the absorber tube due to concentrated solar flux and low on the upper portion due to direct sunlight. A non-linear heat flux distribution is observed over the absorber, which leads to a high temperature gradient. This high temperature gradient causes a thermal stress, hence a distortion in the absorber tube.



(a) **(b)**

Fig. 5 Distribution of heat flux over absorber tube (a) Circular (b) Semicircular

The temperature distribution in the semicircular absorber tube and circular absorber tube are compared in Fig. 6 for the inlet mass flow rate of 5.5 kg/s and inlet temperature 600K. In comparison to the upper portion, the temperature at the lower portion is high for both the cases.

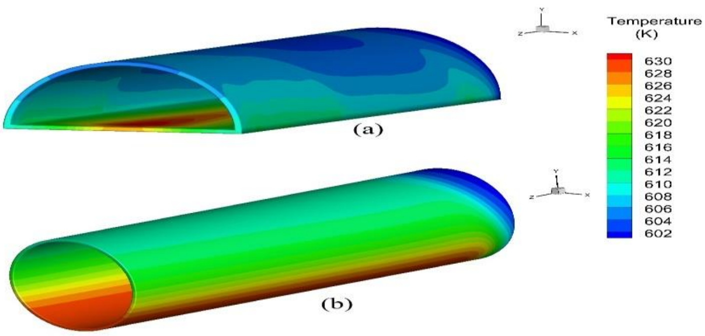


Fig. 6 Temperature distribution over (a) semicircular absorber tube (b) circular absorber tube

Fig. 7 shows the variation of collector efficiency and pressure drop with respect to Reynolds number for semicircular tube and circular tube receiver at inlet temperature 600K. The collector efficiency is calculated by using Eq. (3). Heat gained by the heat transfer fluid is obtained by using Eq. (4).

$$\eta_c = \frac{Q_c}{I.a.L} \tag{3}$$

Where 'I' is direct normal irradiation.

$$Q_c = \dot{m}(\overline{T_{out}C_{p,out}} - \overline{T_{in}C_{p,in}}) \tag{4}$$

As observed in Fig. 7, higher collector efficiency is observed with semicircular tube absorber in comparison to circular tube absorber for the range of Reynolds numbers $0.25x10^5$ to $2.82\ x10^5$. The gain in collector efficiency with the semicircular tube is relatively more at low Reynolds number. The maximum gain in the efficiency is found to be 5% at the Re = 25,000. As expected, the pressure drop in the semicircular tube is higher in comparison to the circular tube. The causes may be that fins increases the fluid contact area and decreases cross sectional area. The difference in the pressure drop between the semicircular tube and circular tube varies from 2 Pa to 166 Pa for Reynolds number range of Re = $0.25x10^5$ to $2.82\ x10^5$ respectively.

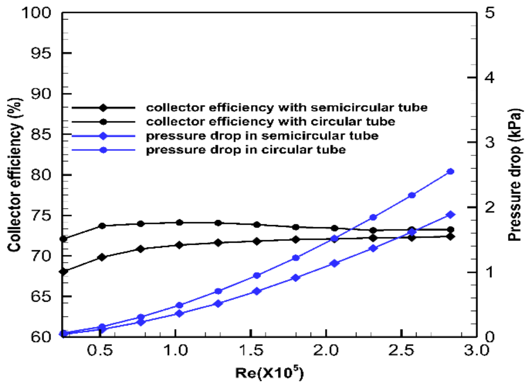


Fig. 7 Distribution of heat flux over absorber tube

VI. CONCLUSIONS

In this paper, the performance of parabolic trough solar collector has been studied with semicircular tube absorber and circular tube absorber. The simulations have been performed by coupling MCRT and FVM for different inlet temperature (300K-650K) and Reynolds number ($0.25x10^5$ to $2.82\ x10^5$). An improvement in the collector efficiency is obtained with semicircular tube absorber at low Reynolds numbers on account of slightly higher pressure drop. Semicircular tube absorber may be employed at low fluid velocities preferably between 0.3 to 0.6 ms^{-1} to improve the thermal performance of PTC.

Nomenclature

a : aperture width (m)
c_p : specific heat of heat transfer fluid($J\ kg^{-1}\ K^{-1}$)
d_{gti} : glass tube inner diameter (mm)
d_{gto} : glass tube outer diameter (mm)
d_{rti} : absorber tube inner diameter (mm)
d_{rto} : absorber tube outer diameter (mm)
f : focal length (m)
I : direct normal irradiance ($W\ m^{-2}$)
L : length of receiver tube (m)
$\dot{m}$: mass flow rate of heat transfer fluid ($kg\ s^{-1}$)
q : local heat flux on absorber tube ($W\ m^{-2}$)
Re : Reynolds number
T : temperature (K)
T_{in} : inlet temperature of heat transfer fluid (K)
T_{out} : outlet temperature of heat transfer fluid (K)
$\bar{T}$: average temperature of heat transfer fluid at a cross-section (K)
Q_c : heat gained by heat transfer fluid (W)

Greek Symbols

α_r : absorptivity of the absorber tube
ρ_m : mirror reflectivity
ρ_r : absorber tube reflectivity
ρ_g : glass tube reflectivity
ψ : rim angle (degrees)
τ_g : glass tube transmissivity
τ_m : mirror transmissivity
η_c : collector efficiency

Abbreviations
CSP: concentrated solar power
FVM: finite volume method
MCRT: Monte Carlo ray tracing
PTC: parabolic trough collector

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