Thermal performance enhancement in parabolic trough solar collectors by using an absorber tube with fins

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ABSTRACT

The main objective of this paper is to investigate the effect of fins inside the absorber tube of parabolic trough collector (PTC). The analysis compares two cases: one without fins and one with four fins inside the absorber tube. The study is divided into two parts. In the first part, meteorological data is used to calculate the temporal numerical values of solar heat flux for the city of Laghouat, Algeria. Solar flux density values for different seasons (A summer , B autumn , C spring ,D winter) have been computed, and a ray-tracing method is applied to determine the distribution of thermal flux on the lateral surface of the receiver tube. In the second part, a simulation is performed to model conjugate heat transfer within the receiver tube, with water as the heat transfer fluid. The simulation examines the case where four fins are added inside the absorber tube, along with a secondary reflector. The results for the first case (simple receiver tube) show that the highest fluid temperatures and system efficiency are reached during summer (73°C, 40%), while the lowest values occur in winter (45°C, 32%). Intermediate values are observed in B.autumn and C.spring (58°C, 32%). For the second case (receiver tube with fins), higher performance is recorded during D.summer, with maximum fluid temperatures of 86.8°C and an efficiency of 49%. The lowest values in D.winter are 50.5°C and 39%, while B.autumn and C.spring show values of 77°C and 48%. Additionally, system performance is enhanced by incorporating fins inside the tube, leading to an improvement ratio of approximately 1.21 to 1.49.

1. INTRODUCTION

Solar energy has seen remarkable growth in recent years, driven by technological advancements that have lowered costs and supported by government policies promoting renewable energy. Theoretically, the potential of solar energy far exceeds the global energy demand. It is harnessed using photovoltaic (PV) solar panels or concentrating thermal power plants. Concentrated solar power (CSP) technology, still under development, promises to offer nations a reliable source of clean energy, decrease reliance on non-renewable resources, and help mitigate climate change[1-2]. CSP technology generates electricity by concentrating solar rays onto a heat absorption receiver. The four most common CSP technologies available in the market are:

Parabolic Trough Systems: These systems use curved mirrors to focus sunlight onto a receiver tube, where a heat-transfer fluid is heated to produce steam that drives a turbine and generates electricity.Solar Power Towers: Utilizing a field of mirrors known as heliostats, these systems focus sunlight onto a central receiver mounted on a tower. The intense heat warms a fluid, usually molten salt, which is then used to produce steam and generate electricity via a turbine. Linear Fresnel Reflectors: These systems employ flat, long mirrors to concentrate sunlight onto elevated tubes. The fluid inside the tubes is heated by the concentrated light, creating steam that powers a turbine to generate electricity. Dish/Engine Systems: Parabolic dish mirrors concentrate sunlight onto a receiver at the focal point of the dish. The receiver transfers the heat to a working fluid, which is then used to generate electricity through a Stirling engine or a similar device.[3] The solar Parabolic Trough Collector consists of a trough-shaped reflector with a parabolic profile. Solar radiation falling on the trough is reflected and concentrated onto the focal line of the parabola, where an absorber tube containing a heat transfer fluid is positioned. The radiation heats the fluid as it flows through the tube.

PTC technology can be utilized in a wide range of applications, including solar cooling, refrigeration, industrial heat, desalination, chemical processes, and electricity production.[4]. In recent years, numerous studies have been conducted to enhance the performance of conventional Parabolic Trough Collector (PTC) systems. These studies have primarily focused on improving the concentration of solar radiation, optimizing the absorber tube, and enhancing the heat transfer fluid. To improve the concentration of solar radiation and ensure uniform heat flux distribution over the absorber tube's surface, several researchers have proposed the addition of a secondary reflector. This secondary reflector is positioned opposite the primary reflector, with its focal line aligned with the axis of the absorber tube. Regue et al.[5] demonstrated that incorporating a secondary reflector into the system increased thermal efficiency by approximately 12% compared to a conventional PTC design. Ali Marzban[6] Studies suggest utilizing compound turbulators and magnetic hybrid nanofluids. Faisal A et al [7] proposed a cascaded absorption cooling (CAC) system to improve solar cooling efficiency. By coupling NH_3–H_2O and H2O–LiBr systems, it reduces thermal energy use by 7.1%. Using a performance-enhanced parabolic trough collector (PEPTC) cuts the solar field area by 14% and boosts overall cooling efficiency by 22%. This combined approach enhances solar cooling performance and supports sustainable cooling solutions. Oveepsa Chakraborty et al [8] analyzed the performance of a parabolic solar trough collector by adding rotating star inserts in the receiver to induce rotational motion in the heat-carrying fluid. Using hybrid nanofluids (1% copper oxide and aluminum oxide), four cases were simulated in ANSYS Fluent. The key findings are:Case 3 (with 25 mm star inserts rotating at 15 rad/s) showed the best performance, improving thermal efficiency by 38.03% and Nusselt number by 50.90% compared to the receiver without inserts.This improvement came with a 20.41% increase in pump work demand at the same flow rate (0.033 kg/s). Ipsita Mishraa et al. [9] investigated how shielded twisted tape inside the absorber tube of a parabolic trough collector (PTC) impacts performance. The optimal configuration, found using simulations, includes a 0.002 m tape thickness, 0.001 m gap, 0.020 m inner cylinder center distance, and 50 twists. This setup boosts the thermal efficiency to 86.5%, or 2.063 times that of a conventional PTC. Shielding around the inserts significantly improves thermal performance. A. Y. Al-Rabeeah et al [10] studied experimentally how different copper absorber tube designs enhance the thermal efficiency of (PTC). Four designs were tested: single, double, loop-evacuated tubes, and a double-evacuated tube with a flat plate. Under Hungarian weather conditions, the double-evacuated absorber with a flat plate showed the best performance, with a maximum thermal efficiency of 52.7% and 59.05% at mass flow rates of 60 and 120 L/h, respectively. The study concludes that the double-evacuated and loop-evacuated tubes with flat plates significantly outperform traditional designs in terms of thermal efficiency.The presence of fins inside the absorber tube of a parabolic trough collector (PTC) has a significant effect on its heat transfer efficiency. For example, research has indicated that incorporating fins enhances the thermal performance by increasing the heat exchange surface area.

Mohamed et al [11] examined the thermal performance of a full-sized parabolic trough collector using oil-based nanofluid (Al_2O_3-Syltherm 800) with varying volume fractions and internal longitudinal fins. Combined methods improve performance by as much as 7.25%. In order to decrease the thermal stress on the receiver and enhance the total efficiency of (PTC) Agagna et al [12] studied the use of integrated spherical pins in the receiver design of PTC . Non-uniform heat flux leads to overheating and thermal stress, affecting performance. By using a numerical model that combines optical and thermal analyses, the study shows that spherical pins can increase the heat transfer coefficient by 27.2% and reduce heat loss by 9.32%. The optimal configuration with five pins enhances overall efficiency by 4.1% and exergetic efficiency by 2.2%. This solution effectively improves PTC performance while reducing thermal stress and overheating risks. Guerraiche et al [13] analyzed materials (aluminum, copper, stainless steel) of the absorber in PTC and found copper to be best at minimizing stress. Additionally, using Al_2O_3 nanoparticles in the heat transfer fluid improved efficiency by up to 14%. J.S. Akhatov et al. [14] carried out the thermal performance of a (PTC) with an absorber tube featuring inner helical axial fins. Three-dimensional numerical simulations were performed using COMSOL Multiphysics, with water as the heat transfer fluid and copper as the tube material. The simulations consider constant solar radiation (Io = 1000 W/m²) and a concentration ratio of C = 60, varying the inlet fluid velocity between 0.2 m/s and 0.24 m/s. The electrical analog method is used to model heat losses, and the study evaluates temperature variations to assess receiver performance under specific conditions. Waleed Al-Aloosi et al. [15] investigated the enhancing heat transfer in a (PTC) by incorporating fins into the absorber tube. Key outcomes include:Various fin configurations, increasing in number from 2 to 6, were analyzed using ANSYS FLUENT simulations.and The model with six fins demonstrated the highest thermal efficiency, outperforming the smooth reference model by 40%.Different fin shapes—circular, elliptical, and square—were also tested, with circular fins achieving the highest performance (1.4), followed by elliptical (1.31) and square (1.26) at a Reynolds number (Re) of 4000.A 51% increase in heat transfer (Nusselt number) was observed when Re was increased from 4000 to 8000.Overall, the six-fin model with circular fins provided the most significant performance improvement. Moreover, Faruk Yeşildal et al [16] reviewed the methods to enhance the thermal efficiency of (PTC), including: Increasing absorber surface area for better solar energy absorption,

Using tube inserts to create turbulence and improve heat transfer,Applying selective coatings to reduce reflection.

Enhancing the thermal conductivity of the working fluid. Modifying the shape of the absorber tube or reflective surface to minimize energy losses,While these strategies improve efficiency, they may increase pressure drops and system costs. Mohammed Reda Haddouche et al .[17] evaluated the absorber tube's thermal performance in a parabolic trough solar collector with a new type of inserts to enhance heat transfer between the absorber tubes and the heat transfer fluid. The inserts reduce the temperature differential in the absorber tube. Additionally, the use of nanofluids and different fin arrangements has been found to further enhance the absorber tube's heat transfer capabilities, leading to higher outlet temperatures and improved overall performance . The design of the absorber tube with fins plays a crucial role in maximizing heat absorption from solar radiation and increasing the efficiency of parabolic trough solar collectors .

Mohamed H. Yehia et al [18] conducted a design to assess the thermal performance of a full-sized parabolic trough collector using oil-based nanofluid (Al_2O_3-Syltherm 800) at various volume fractions and internal longitudinal fins. Internal fins with nanofluid can increase thermal performance by up to 7.25%, and nanofluid can increase thermal performance by 0.1-1.16%.andrew.s .et al [19] explored the impact of axial rotation and internal fins on the thermal performance of (PTC) receiver tubes. Axial rotation slightly reduces the outlet temperature by up to 6%, but improves temperature homogeneity between fluid layers, eliminating stratification. An optimal increase of 110% in the Nusselt number is observed at a rotation rate of N=21. However, this comes with a significant pressure drop of 81.6% due to increased turbulence. Overall, combining axial rotation with internal fins enhances heat transfer, improving the thermal performance of the PTC receiver tube. Sezer Sevim et al. [20] Studied how adding curvilinear fins to the receiver pipe increased the fluid's heat transfer characteristics. The analysis's Reynolds number range was found to be between 3000 and 21000. In a six-finned pipe, the Nusselt number increased by 1.34 times. In a twelve-finned pipe, the Nusselt number increased by 3.06 times. Mujahid K et al.[21] Investigated numerically of three-dimensional turbulent flow in PTC collector , using the finite volume method in Ansys Fluent to simulate the equations and the MCRT method to model solar radiations as a heat flux. Fins and nanoparticles improve absorber tube heat transmission.and configuration "b" with six fins shows higher heat transfer performance.

Nourhan B et al. [22] Assessed the heat transfer characteristics of the absorber tube with the module of LS-2 parabolic trough collector with internal longitudinal fins with rectangular crosssection are investigated using ANSYS CFD and oil syltherm 800 type as a heat transfer fluid. They Found that the Round edge fins have higher Nusselt number and lower pressure losses and Thermal efficiency and enhancement index are calculated at different conditions. Subhas Chandra et al. [23] created a new type of solar air heater with louvered fins has been created, and tests have been done to assess its thermal performance. The results show that thermal efficiency increases with flow rate; at a mass flow rate of 0.007 kg/s, outlet temperature and thermal efficiency reached a maximum of 58.66 °C and 70%, respectively, for a spacing of 2 cm. Louvered fins improve the solar air heater's thermal performance. Heat transfer efficiency and rate have increased. Mohammad Zaboli et al. [24] Utilized a finite volume approach is utilized to analyze a (PTC) collector with inner helical axial fins as a swirl generator or turbulator. Enhancement of thermal performance of 23.1% achieved utilizing innovative collector design and Minimum improvement of 14.1% and maximum improvement of 21.53% observed with different fin pitches. Laaraba et al. [25]examined numerically the impact of finned absorbers on a parabolic trough collector's thermal performance, the length of the fin improves the thermal efficiency, Nusselt number, and friction factor of finned absorbers. The addition of fins improves the parabolic collector's thermal performance by 8.45%.Fins should have a length of 15 mm and a thickness of 6 mm. Palacios et al. [26] Applied the CFD Solidworks® flow simulation software for the thermal analysis results of a parabolic trough collector with various receiver internal fin shapes were examined and compared with those of a conventional parabolic collector with a cylindrical receiver. Fractal fins receiver increased temperature by 27%.and Fractal Descartes configuration improved temperature by 29%.

This comparative study analyzes the performance of parabolic trough solar collectors with secondary reflecter across one day in four seasons (A. summer, B. autumn , C. spring , D. winter) for two cases one without fins and other with integration of four fins within the absorber tube. By evaluating the thermal efficiency and heat transfer characteristics during different seasonal conditions, the research aims to identify four fin configurations that enhance energy capture and minimize heat losses. The study employs numerical simulations data to assess variations in temperature, heat transfer fluid performance, and overall collector efficiency throughout the year. The findings provide valuable insights into how fin design influences the operational effectiveness of PTC collectors in diverse climatic scenarios.

The investigation is divided into two main sections: optical and thermal analyses. The optical analysis, conducted using the optical software, utilizes the MCRT method to determine the heat flux distribution around the reciver . This distribution is then approximated using Gaussian functions and integrated into the CFD solver . Subsequently, The conjugate heat transfer and HTF flow equations within the reciver are solved to determine the change in the heat transfer fluid (HTF) outlet temperature.we used the meteorological conditions of four seasons in 2023, in Laghouat city, located in southern Algeria (coordinates: 2°56' longitude, 33°46' latitude, and 700 meters altitude).

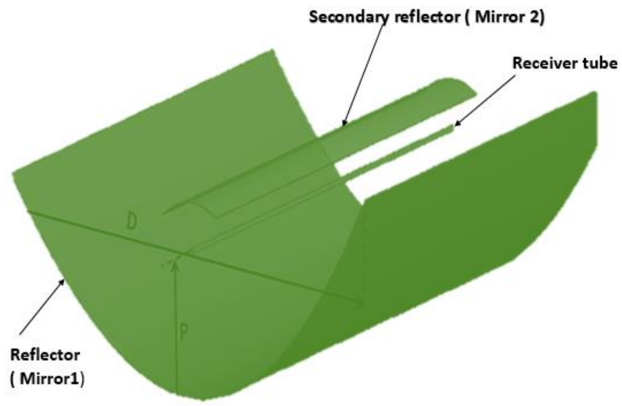

Figure 1. Sketch of a PTC with a secondary reflector

2. Optical analysis

This section focuses on analyzing the distribution of heat flux along the lateral surface of the receiver tube . The geometrical and optical properties of the PTC system examined in this study are illustrated in Figure 2 and detailed in Tables 1 . The solar irradiance data (refer to Figure 3) was taken for four days in four seasans , 2023, in the city of Laghouat, southern Algeria (coordinates: 2°56' longitude, 33°46' latitude, and 700 meters altitude). The SolTrace software, utilizing the Monte Carlo method for ray tracing (depicted in Figure 4), was employed to calculate the heat flux distribution around the absorber tube (shown in Figure 5).

The optical properties of (PTC) are crucial for their performance in concentrating solar power (CSP) systems. These properties include:**Reflectivity:** This pertains to the capacity of the parabolic mirror to reflect solar radiation. . High reflectivity ensures that more sunlight is directed towards the absorber tube.**Absorptivity:** This measures how well the receiver tube absorbs the concentrated solar sunlight . A high absorptivity indicates that more of the incident solar radiation is collected by the receiver tube, converting it to heat.**Transmissivity:** If the receiver tube is enclosed in a glass cover, the transmissivity refers to the ability of the glass to transmit solar radiation with minimal losses.**Specularity:** This property describes the mirror's ability to reflect light in a concentrated beam rather than scattering it. High specularity is essential for focusing sunlight accurately onto the absorber tube.**Optical performance** : This is the overall efficiency with which the PTC system converts incident solar sunligh into usable heat energy. It is influenced by all the other optical properties and the geometric design of the collector. **Incidence Angle Modifier (IAM)**: This factor accounts for the variations in the optical efficiency because of variations in the sunlight's angle of incidence . As the angle of incidence increases, the efficiency typically decreases. These properties are essential for designing and optimizing PTC systems to maximize their efficiency and effectiveness in capturing and converting solar energy.

The study uses optical software for determining the distribution of the heat flux on the receiver tube utilizing the MCRT method. The following are the conditions for the ray tracing simulation: A Gaussian distribution with a cone angle of 2.73 mrad is used to model the Sun. Mirror is chosen for the reflector plate material, featuring a reflectivity of 0.95, a shape error of 3 mrad, and a specular reflection error of 0.5 mrad. For the secondary reflector, the metal collector tube has a reflectivity of 0.05, a shape error of 0.0001 mrad, and a specular reflection error of 0.0001 mrad. Figure 2 displays a sample ray tracing . The solar irradiance at the start of each hour, obtained from meteorological software, is entered into the programming code used in SolTrace .[27]

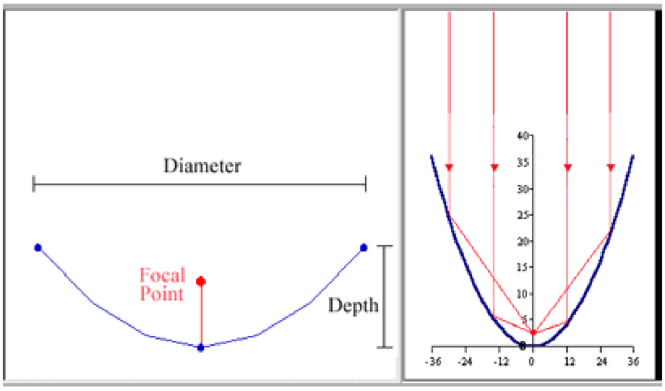

Figure 2. Geometry of the PTC

Table 1: Tube and reflectors geometric parameters

Primary reflector opening	1m
Primary reflector focal length	0 .2083 m
Collector length	1 m
Inner receiver diameter	0.008 m
Outer receiver diameter	0.01 m
Secondary reflector opening	0.25 m
Secondary reflector focal length	0.0781 m

Table 2 : Solar irradiance for four days for diffrents seasons

Time (h)	I(W/m²) A.summer	I(W/m²) B.autumn	I(W/m²) C.spring	I(W/m²) D.winter
09.00	646	490	441	182
10.00	815	674	636	357
11.00	942	808	785	489
12.00	1018	885	877	566
13.00	1037	898	906	583
14.00	999	846	869	539
15.00	906	734	770	437
16.00	764	569	614	285

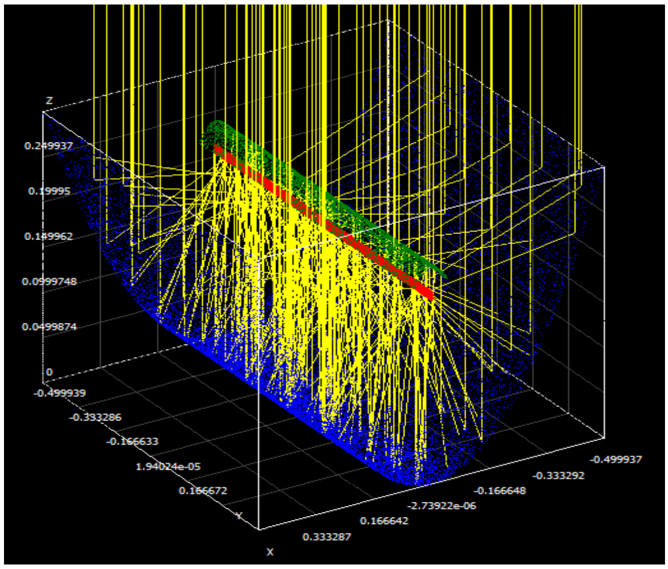

Figure 4. Ray tracing

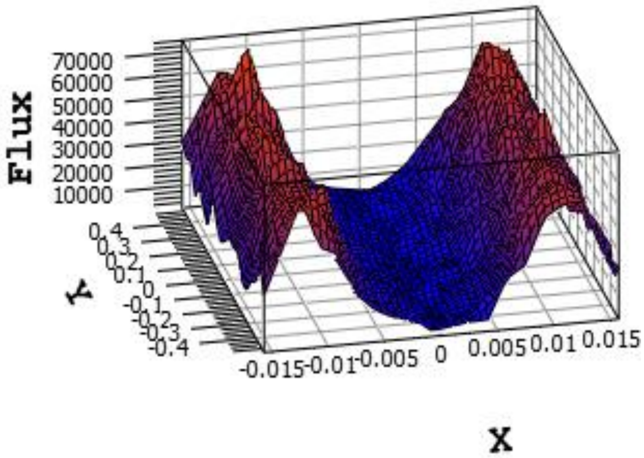

Figure 5. Contour plot of heat flux on the absorber tube (surface plot)

The surface plot of the concentrated solar heat flux distribution is presented in Fig. 5. It is clearly shown that the distribution is symmetric along the plan X=0 and the plan Y=0. the heat intensity distribution over the reciver is not the same at every point. The flux distribution can be divided into three parts with different levels of heat flux. In the first part, the heat flux reaches a peak value of 79054 W/m^2. The second part shows a rapid decrease in the heat flux, reaching a minimum peak value of 47432.4 W/m^2. This decrease can be explained by fewer incident rays toward the parabolic solar collector in that specific area. In the third part, the heat flux reaches a maximum peak similar to that in the first part. These findings suggest that connecting a PTC to the reciver with secondery reflector can improve its performance. Additionally, according to the results given by the Soltrace simulation, the distribution of the heat flux over the absorber tube shows an average flux of 27818.7 W/m^2 +/- 0.437617 %. distribution is shown to reach the focal point rapidly. Outside.the central region, the concentrated solar radiation is very low because the receiver tube receives only a direct radiation.The concentrated solar heat flux distribution calculated by Optical software is shown in Figure. 4.

The surface plot of the concentrated solar heat flux distribution is presented in Fig. 5. It is clearly shown that the distribution is symmetric along the plan X=0 and the plan Y=0.

The variation of the heat flux around the receiver tube over the four days in four seasons(A.summer,B.autumn ,C.spring ,D.winter) 2023, obtained by optical code, is interpolated using the following Gaussian function:[28]

$$Q = a_1 \exp[-((t-b_1)/c_1))^2] + a_2 \exp[-((t-b_2)/c_2))^2] \quad (1)$$

Where, t is the time (s) and the coefficients ai, bi, ci are given in Table 3.

Table 3. Coefficients of the Gaussian function in equation (1) for the four days in diffrents seasons (A.summer,B.autumn ,C.spring ,D.winter)

- **A.Summer**

I	1	2
ai[W/m^2]	2.30 e+04	2.183 e+04
bi [s]	3.571 e+04	2.009 e+04
Ci[s]	1.177 e+04	1.136 e+04

- **B.autumn**

I	1	2
ai[W/m^2]	1.997e+04	1.937 e+04
bi [s]	3.433 e+04	2.08 e+04
Ci[s]	9929	9706

- **C.Spring**

I	1	2
ai[W/m^2]	2.011e+04	1.965 e+04
bi [s]	3.52 e+04	2.172 e+04
Ci[s]	9794	9646

- **D.Winter**

I	1	2
ai[W/m^2]	1.123 e+04	1.009 e+04
bi [s]	3.307 e+04	2.204 e+04
Ci[s]	8218	7577

2. CFD Analysis

The purpose of the CFD analysis is to solve the conjugate heat transfer and HTF flow equations in the absorber tube. Fluid flow and heat transfer in both domains solid and fluid are described by a system of five equations to be simultaneously solved.

3.1 Governing Equations

where is the static pressure, is the stress tensor (described below), and and are the gravitational body force and external body forces (for example, that arise from interaction with the dispersed phase), respectively. also contains other model-dependent source terms such as porous-media and user-defined sources.

The stress tensor is given by:

- **Continuity** [27]

$$\nabla(\rho\vec{v}) = 0 \quad (2)$$

- **Momentum**

$$\frac{\partial}{\partial t}(\rho\vec{v}) + \nabla.(\rho\vec{v}\vec{v}) = -\nabla p + \nabla.(\overline{\overline{\tau}}) + \rho\vec{g} + \vec{F} \quad (3)$$

where is the static pressure, $\overline{\overline{\tau}}$ is the stress tensor (described below), and $\rho\vec{g}$ and $\vec{F}$ are the gravitational body force and external body forces (for example, that arise from interaction with the dispersed phase), respectively. also contains other model-dependent source terms such as porous-media and user-defined sources.

The stress tensor $\overline{\overline{\tau}}$ is given by

$$\overline{\overline{\tau}} = \mu\left[\left(\nabla\vec{v} + \nabla\vec{v}^{T}\right) - \frac{3}{2}\nabla.\vec{v}I\right] \quad (4)$$

where μ is the molecular viscosity, I is the unit tensor, and the second term on the right hand side is the effect of volume dilation.

- **Thermal energy**

$$\frac{\partial(\rho h)}{\partial t} + \nabla(\rho U h) = \nabla.(\lambda\nabla T) \quad (5)$$

Where, U is the velocity vector, p is the fluid pressure, T is the temperature and h is the enthalpy.

Note that equation (5) is applied for both domains fluid and solid (for conduction in the solid, the left term in equation 3 is zero).

The net gain in thermal energy in the absorber tube is given by the following equation:

$$\dot{Q}_{net} = \dot{m}C_p(T_{out} - T_{in}) \quad (6)$$

Where, $\dot{m}$ is the mass flow rate (kg/s), Cp is the specific heat capacity (J/kg K); Tin and Tout represent the HTF temperatures at the absorber tube inlet and outlet respectively.

The instantaneous PTC thermal efficiency is given by the following equation:

$$\eta = \frac{\dot{m}C_p(T_{out} - T_{in})}{A_c I_b(t)} \quad (7)$$

Where, I_b is the direct solar radiation (W/m^2) and Ac is the aperture area of the reflector.

3.2 Solving method

The geomtries are crated by the desgin modeler for the two cases :case 1 (without fins) and case 2 (with 4 fins) as showen in figure 6.

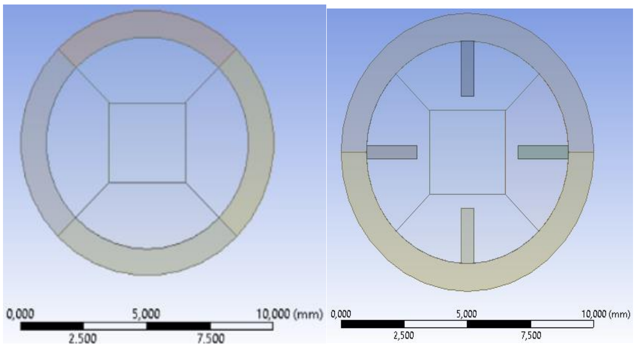

Figure 6 :view of the geomerty of the receiver of the PTC without and witth fins

The physical domain is composed of two sub-domains solid and fluid. As shown in Figure 7,two blocks are merged. The structured block represents the solid domain (receiver tube and fins) and the O-grid block represents the fluid domain, the following distribution of the nodes is chosen: 110 nodes / meter in z direction; 36 nodes in x and y directions; and 5 nodes along the thickness of the tube.

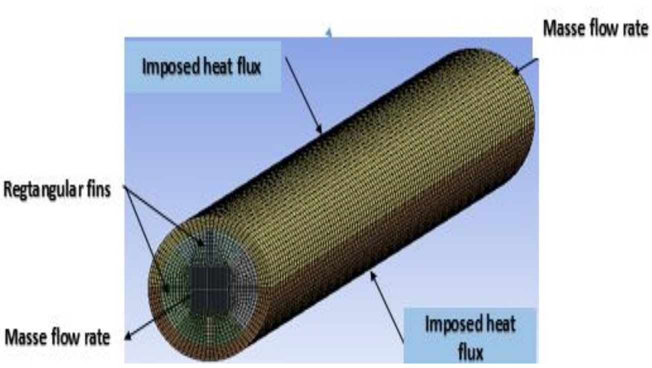

Figure 7: receiver tube meshing and boundary conditions

In this section, we examine the fluid flow and temperature distribution within the absorber tube, as well as the collector efficiency to make a comparison between two cases without fins (simple tube) and with four fins inserted in the tube.

The first step to solving the problem is to implement the heat flux, expressed by equation (1), into the ANSYS-FLUENT solver. The second step is to initialize the solver and specify the boundary conditions. The domain is initialized as follows: At time t=0s, the fluid and the solid are considered to be at the same temperature T_0; the velocity and pressure of the fluid are set to zero. The boundary conditions are illustrated in Figure 7. The mass flow rate of the heat transfer fluid (HTF) through the tube is set at 0.0018 kg/s. The mass flow rate is chosen to ensure laminar flow inside the tube. A laminar flow is determined by the condition (Re < 2000), where Re is the Reynolds number, given by the following expression.

$$\mathrm{Re} = \frac{\rho V d_{in}}{\mu} \qquad (8)$$

Where V is the average flow velocity through the tube; μ is the dynamic viscosity of the fluid; and d_{in} is the inner diameter of the receiver tube.

In addition to the initial conditions and boundary conditions, certain solver configurations are required. The laminar model is selected to solve the fluid flow, and the thermal energy model is chosen to solve the conjugate heat transfer. The time step selected is 15 minutes, while the total simulation time is 16 hours.

4. Results and discussion

4.1 Validition :

To analyze the performance of a PTC using the proposed model, it is important to validate it. For this purpose, the model is applied to the system described in reference [30]. This system is a small PTC equipped with a parabolic secondary reflector. The simulation results are compared with the experimental results reported in this reference. The results to be compared are the temperature of the fluid (water) at the outlet of the receiver tube. The test data and results are showed in figure 8, Note that these tests were conducted on 03/05/2018.

The results obtained for the fluid outlet temperature are presented in Figure 9. As can be seen, the temperature values obtained by simulation are approximately close to the experimental values. There is a small difference between the two curves. This is due to experimental conditions, and energy storage. However the average absolute error is about 2.83 °C, and the standard deviation is 1.35. Based on these values, we can conclude that the current model is in good agreement with the experimental data. Therefore, the model can be used to analyze the optical and thermal performance of the PTC.

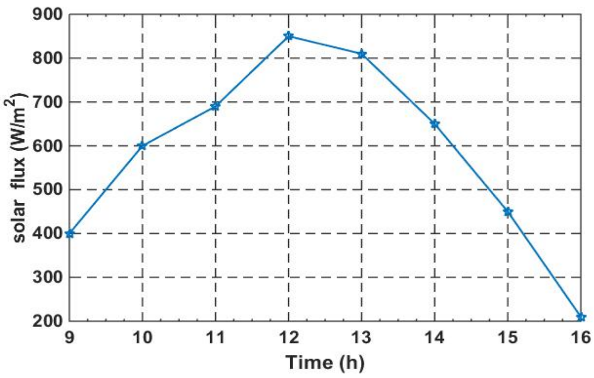

Figure.8 Experimental solar flux as a function of time for the day 03/05/2018[27]

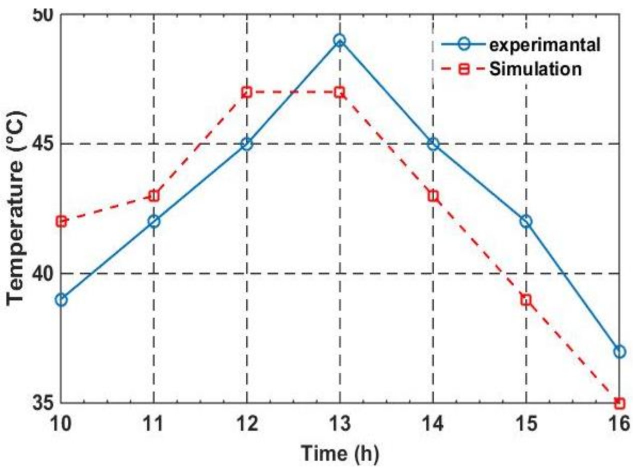

Figure .9 Fluid outlet temperature as a function of time

4.2 HTF outlet temperature

The figure 10 shows the temperature contour throughout the domain (on summer season at 1:00 PM and winter) for a CCP system with a secondary reflector. As can be seen, the temperature gradually increases along the tube, reaching avergage value (about 370.436 K in summer and 329.57 K in winter).

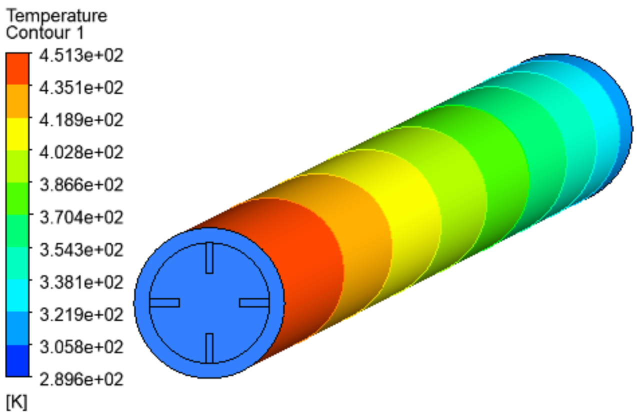

Figure.10 Distribution of heat flux by concentrated solar radiation on the outer surface of the absorber tube at 1:00 PM on summer .

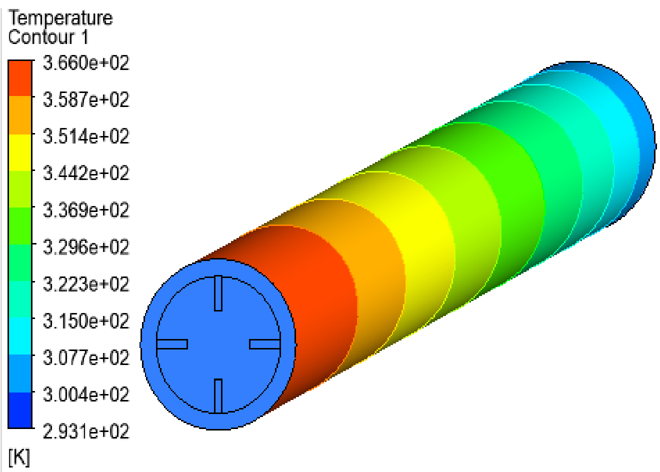

Figure.11 Distribution of heat flux by concentrated solar radiation on the outer surface of the absorber tube at 1:00 PM on winter

However, it gradually decreases in the radial direction to reach a minimum at the center of the tube (approximately 441 K on summer and 360 K on winter).

In this study, we consider the water temperature at the outlet in the center of the tube. Figure 11 represents the evolution of the water outlet temperature of the absorber tube for four days: summer , autumn , spring, and winter . The temporal profile of the temperature is similar to that of the concentrated heat flux on the surface of the tube, with the maximum being reached at noon. We also note that the maximum temperatures obtained in the first case (simple tube) are approximately 73°C on summer and 50°C in winter , while in autumn and spring they are equal to 58°C.

We also note that the maximum temperatures obtained in the second case (tube with fins) are approximately 86°C on summer 21 and 50°C in winter , while in autumn and spring they are equal to 77°C.

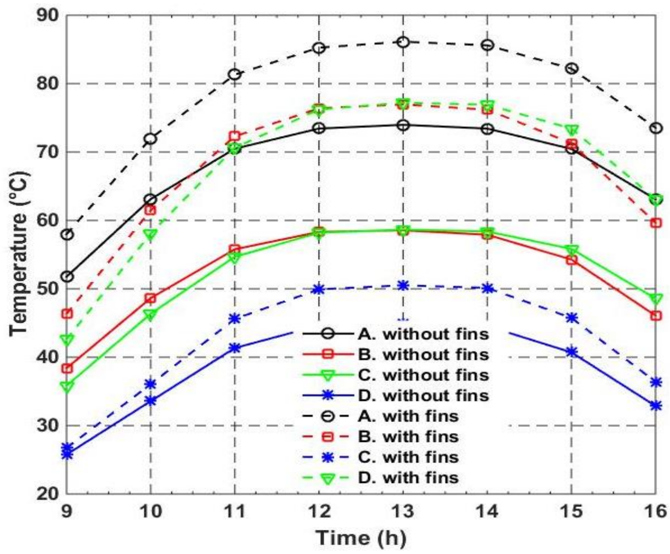

Figure.11 Variation in fluid temperature with the addition of four fins at the outlet as a function of time on four days in diffrents seansons

4.3 Useful thermal energy

Once the temperatures at the tube outlet are determined, the heat flux absorbed by the tube is calculated according to equation (6). The mass flow rate of water through the tube is approximately 0.0018 kg/s, corresponding to a fully laminar flow (see equation 7). Figures .12 show the variations of useful heat versus time. The maximum energy gained is recorded in summer (400 W), while the minimum absorbed energy is recorded in winter (188 W). For spring and autumn , the maximum absorbed energy at noon is the same (approximately 292 W). We also note for the second case(absorber with four fins) The maximum energy gained is recorded in summer (499 W), while the minimum absorbed energy is recorded in winter (230 W). For autumn and spring , the maximum absorbed energy at noon is the same (approximately 431 W)

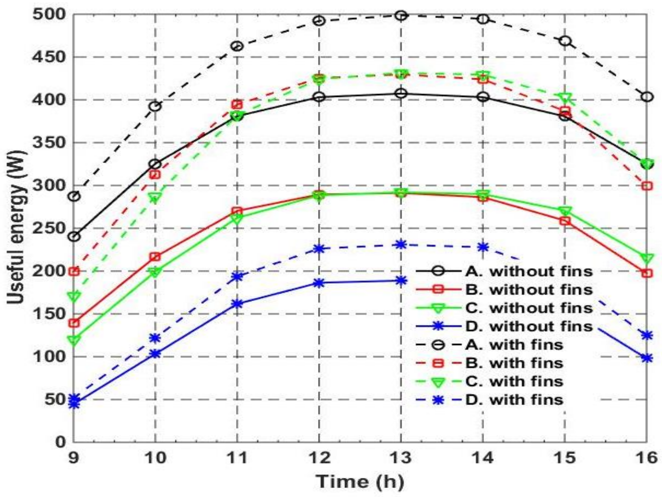

Figure 12: Variation of useful thermal energy with respect to time

4.4 Thermal Efficiency

the thermal efficiency of the system throughout the day from 9 AM to 4 PM shows that efficiency varies slightly over time.The thermal efficiency of the system is determined from equation (7). The calculated values for the four months (A.summer ,B. autumn, C.spring, and D.winter) are shown in Figure .13. The highest efficiency is recorded in summer (around 40% throughout the day), while the minimum value is obtained in winter (around 32%). For autumn and Spring , the efficiency is the same (32%).for the second case (receiver tube with the four fins)The efficiency obtained is recorded in summer (around 49% throughout the day), while the minimum value is obtained in winter (around 39%). For spring and autumn , the efficiency is the same (48%). A gain in efficiency of approximately 9% to 15% can be achieved by inserting four fins into the absorber tube for the four days of different seasons (see Figure 13).

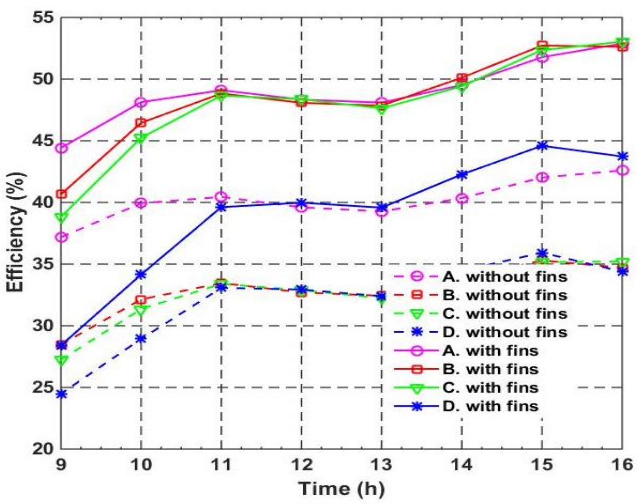

Figure 13: Variation in thermal efficiency as a function of time for two cases: a simple tube and a tube with the insertion of four fins, over four days in diffrent seasons

5.Conclusion

The aim of this article is to assess and compare the thermal performance of parabolic trough solar collectors with a secondary reflector across four seasons (A.summer, B.autumn, C.spring, and D.winter), This comparative study examines the performance of parabolic trough solar collectors with a secondary reflector over the course of a day across four seasons. The analysis covers two cases: one without fins and the other with four integrated fins in the absorber tube, evaluating the impact of fins on the system's efficiency and performance in different seasonal conditions. The key findings are as follows:

- The benefit of using a secondary reflector in PTC is that it helps achieve a more uniform distribution of heat flux around the receiver tube.
- The results obtained for the first case (simple receiver tube) using water as a heat transfer fluid Illustrate the lowest efficiency observed at the seasons of B.autumn ,C.spring,D.winter with average value 32% for a flow rate of 0.0018 kg/s.For the second case, with the tube having four fins inserted, the highest efficiency is approximately 48% for the same flow rate at seasons A.summer ,B autumn ,C spring
- The use of fins inserted in the absorber tube significantly improves the performance of the PTC system.
- An efficiency gain of about 9% to 15% can be achieved by inserting four fins into the receiver tube.
- The system's performance can be increased by using fins inside the tube, with an efficiency ratio of approximately 1.21 to 1.49.

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NOMENCLATURE

Ac aperture area of the concentrator, m^2
pspecific heat of heat transfer fluid, J/kg.K
D asorber tube diameter, m
I_b solar irradiance, W/m2
L length of absorber tube, m
p presure, Pa
T temperature, K
Tin inlet temperature of heat transfer fluid, K
Tout outlet temperature of heat transfer fluid, K
U velicoty, m/s
W primary reflector width, m
$\dot{m}$ mass flow rate of heat transfer fluid, kg/