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Research Article

Improvement of the thermal performances of a parabolic trough solar concentrator with concentric receiver tube and nanofluid

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ABSTRACT

Parabolic trough solar concentrators (PTCs) encounter significant challenges arising from Non-Uniform Heat Flux (NUHF) around the receiver, which causes local overheating and pronounced circumferential temperature gradients. This study proposes the solution to this problem by uniforming the heat flux distribution, minimizing the temperature gradient, and improving performance. A numerical analysis is carried out on a novel small-scale PTC model featuring a concentric absorber tube design. The study compares the heat recovery performance of conventional, concentric, and eccentric absorber configurations using two heat transfer fluids enhanced with nanofluids: water and Syltherm-800 oil. A three-dimensional simulation model was presented by coupling Monte Carlo Ray-Tracing (MCRT) with (CFD) in ANSYS Fluent to predict the thermo-hydraulic response of parabolic trough receivers. Furthermore, Particular attention is given to the effects of Particular attention is given to the effects of critical operational factors, namely inlet fluid temperature, mass flow rate, and nanofluid concentration on heat transfer and overall efficiency. The results confirm that the concentric parabolic trough receiver (PTR) operates with lower peak wall temperatures and smaller circumferential thermal gradients, which lessen thermal stresses and supports higher overall efficiency. When compared to the reference design, the concentric and eccentric receiver tube configurations lower the absorber tube's temperature gradient by roughly 45% and 60.6%, respectively. These new configuration enhance heat collecting efficiency by up to 2.62% and 3.26% relative to the smoothe PTC tube. Moreover, the use of nanofluids added significant enhancements of efficiency, by 6.12% and 8.23% for the concentric and eccentric

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arrangements, respectively. This work highlights the promise of novel receiver designs and the use of nanofluids to boost both thermal and mechanics performance of PTCs and contributes knowledge beyond earlier work in the literature and serves toward improving solar therma.

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INTRODUCTION

Worldwide energy demand has risen significantly, resulting in numerous plans aimed at the usage of both renewable and nonrenewable energy sources. Solar energy is the most remarkable feature from the list of alternatives that could be the most widely used and rapidly growing in the future. For this purpose, the parabolic trough collector (PTC) is a reliable and useful method for collecting solar rays and turning it into heat energy. Recently, the development of Parabolic Trough Concentrator (PTC) has focused on achieving higher concentration ratios and operating temperatures to reduce investment costs and enhance thermal efficiency, particularly at high temperatures [1]. This objective is essential to guarantee the energetic and financial viability. PTC possess numerous drawbacks, such as low efficiency, maintenance demands, high costs, complexity, energy losses, durability issues, and the necessity for solar tracking [2]. However, PTCs still face a great number of problems and limitations. NUHF generates excessively localized heating as well as steep thermal gradients into the receivers which results in increased thermal stress. One of the effects is that the absorber tube goes outside the focal plane leading to quite significant optical losses due to the loss of the particular coating being very rapid at high local temperatures, hence, the maximum operating temperature is limited [3]. These aforementioned barriers baffle the efficiency of PTC systems significantly over time. Moreover, NUHF is the leading cause of the large decrease in the thermal efficiency of PTC from where the energy conversion performance is adversely affect. Implementing the most efficient measures to ensure a preferable uniform heat flux distribution, should be the main target. All the above mentioned are first of all ways of accomplishing this: temperature distribution being more even, the peak temperatures being decreased and the gradients of the temperature being lessened. There are several ways of improving solar collector efficiency, which, in fact, can be mostly divided into three main aspects: raising the heat conductivity of the receiver tube, changing the collector design in order to have better distribution of solar flux and to give the receiver tube the possibility of making heat transfer by convection easier [4].

Improving heat transfer in solar parabolic trough receivers is gradually gaining importance as a major measure to mitigate NUH. A primary goal of current research is to devise novel receiver configurations that would allow better fluid flow patterns and the utilization of new methods for heat transfer enhancement. A wide range of both passive and active methods have been considered for their thermal performance improvement and energy conversion efficiency increase [5]. The application of passive techniques broadly involves the use of the fins or turbulence promoters, changing the shape or the surface properties of the receiver, raising the capacity of the absorber tube for heat retention, and the addition of more material. Through these methods, the heat transfer to the fluid from the receiver tube can be elevated by up to 50% [6]. Efficient use of the concentrated solar power (CSP) system with improved parabolic trough solar receiver can be the source of a steep drop in solar power plant costs. While active methods do consume more energy, they still rely on additional heat transfer from external sources inside the absorber tube. Researchers must develop technologies that have the dual effect of lowering the expense and being more sustainable in the use of renewable energy. One of the promising options for increasing heat transfer is the use of turbulators along with nanofluids [7]. As an example, turbulence promoters or inserts have been studied to create turbulence and increase convective heat transfer. The inserts disturb the boundary layer and facilitate heat transfer by attaining greater mixing.

Different methods have been explored to increase heat transfer in the receiver tube. One such method is the insertion of twisted tapes inside the receiver tube where Chang et al. [8], found heat transfer rates to be significantly enhanced. Moreover, oscillatory or pulsating flow has also been investigated to raise heat transfer coefficients. These techniques involve periodically varying flow rates or flow directions, resulting in enhanced heat transfer characteristics. Afsharpanah et al. [9] performed a computational research which was mainly focused on raising the heat exchange rate in a solar parabolic trough collector by the usage of modified dual twisted-tape inserts. The authors considered several tape shapes such as V-cut, square-cut, and perforated designs, and used water under pressure as the working fluid. Their findings showed that the dual V-cut design led to the greatest heat performance, raising the average Nusselt number by 19.58% and 17.44% at Reynolds numbers 10,000 and 20,000, respectively. The study, by and large, acknowledged that all the twisted tape modifications had higher efficiencies than the smooth tube.

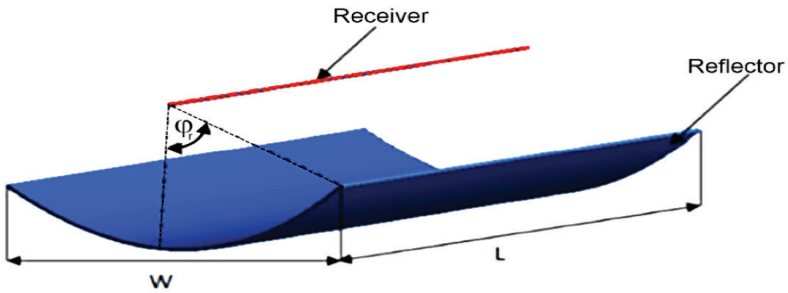
The authors Fatouh et al. [10], examined the usage of the extended surfaces on the receiver tube with the main idea to double the heat transfer area and discovered the considerable increase of the convective heat transfer coefficients. DeSá et al. [11] doped a computational analysis to check the influence of pulsating flow on parabolic trough solar receivers and displayed that the cooing effectiveness is better by an oscillatory flow than a flow of steady state. Nevertheless, the need of the new ways of heat transfer promotion in parabolic trough solar receivers as well as the designing and operational optimization of parabolic trough solar receivers continues to exist.

Nevertheless, limited research have emphasized on the improvement of cascade solar energy conversion systems by modifying the absorption tube geometry to make it more efficient. Zhang et al. [12], developed a novel convergent-divergent tube concept for parabolic trough solar collectors (PTR) that was targeted to raise the heat transfer and the temperature uniformity getting by more rapid heat exchange. Research evidence highlights that the innovative tube is a substantial heat transfer enhancer that the Nusselt number under certain conditions may reach 66% more when compared to a standard PTR system. Several paper have examined the use of concentric reveiver tubes to enhance heat transfer for PTR. Liu et al. [13], introduced an innovative solar cascade heat collection configuration involving double-tube for the parabolic trough receiver (PTR) utilizing two distinct heat transfer fluids (HTF). The results demonstrate an improvement in the parabolic trough receiver's performance due to the novel solar cascade heat collection arrangement. The novel solar cascade heat collection arrangement could elevate the overall thermal efficiency by a maximum of 1.5%. Liu et al. [14], provide a new design with an inner tube and wing-like fringe and employ two heat transfer fluids (water and thermal oil) to increase heat-collecting efficiency while delivering various grades of thermal energy. The model shows the new system reduces heat loss by 33.1% to 50.1% and improves efficiency by 0.61% to 7.67%.

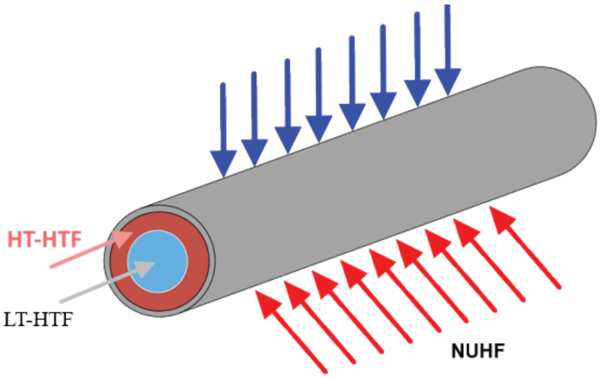
Benabderrahmane et al. [15] showed that the heat transfer along the absorber tube of a PTC was enhanced by the use of a corrugated insert of the center leading to an increase of the Nusselt number of up to 3.7 times with respect to a plain tube. The total efficiency factor was in the range of 1.3 to 2.6, and it increased with the decrease of the corrugation pitch and the increase of the twist ratio. Guerraiche et al. [16], studied the operation of an integrated new prototype of PTC integrating latent heat storage system using a concentric tube setup for a parabolic trough solar receiver. The authors informed that the method could achieve high water outlet temperatures along with better thermal efficiency. Imtiaz and Lee et al. [17] conducted an experimental and numerical evaluation of the thermal performance of a newly designed concentric receiver tube. Their results revealed that the concentric tube arrangement led to the even distribution of the flow and the maintenance of the temperature symmetry along the receiver length. Moreover, the local temperature and heat flux on the inner and outer tube surfaces showed a good angular uniformity, and the simulated data matched well with the experimental measurements. Acuna et al. [18] presented the concept of a concentrator system with a compound parabolic concentrators and a concentrator receiver tube. The findings revealed that the proposed design had a potential of reaching higher outlet temperature and efficiency by 4% and 10%, respectively, compared to the traditional single-tube designs. Wang et al. [19] offers a tube receiver whose eccentricity is oriented so as to lessen the thermal stresses in the receiver tube. Here, the effects of eccentricity and the angle of orientation on the receiver tube were investigated. The results suggest that the eccentric design is very effective in lowering thermal stress, and most of the variations of the eccentricity and orientation angle lead to the change of the stress level. Through the use of inner and outer pipe inserts that are concentric and eccentric Chang and co-workers [20] have found that heat transfer in a PTC absorber can be greatly improved. Their research suggests that heat transfer with these types of inserts could be increased up to almost 1.64 times of the original value. According to Pérez-Álvarez et al. [21] the change of orientation angle and eccentricity of the eccentric receiver are factors that determine the thermal stress experienced by the receiver tube. The research results demonstrate that the combination of an eccentric tube receiver with parabolic trough concentrators leads to a significant reduction in thermal stresses that the system reliability can be enhanced by the proper design of the eccentric tube besides a successful reduction of thermal stress in parabolic troughs. In their work, Bellos et al. [22] came up with the idea of using a flow of a cylindrical insert in a parabolic trough solar collector. They studied the heat efficiency of inserts both placed centrally and eccentrically to decide the configuration that gives the highest thermal gain.

However, researchers still encounter difficulties in maximizing the efficiency of PTCs, specifically in improving the ability of the absorber tubes to absorb heat flux. Introducing nanoparticles into the fluid is one potential solution to alleviate this problem. Nanoparticles are frequently mentioned in various literary works. A comprehensive review of the literature identifies many research works that emphasize finding new methods to increase heat transfer in parabolic trough solar receivers. Guerraiche et al. [23], propose methods to enhance heat transfer and homogenize heat flux distribution to improve PTC performance. According to their results, adding Alumina (Al_2O_3) nanoparticles to water is beneficial to the heat transfer efficiency of the system and the improvement ranges from 3% to 14%, for volume concentration varying from 2% to 6%. Meanwhile, the study suggests that copper is the best material for the tube because it undergoes the least thermal stress among aluminum and stainless steel. Wang et al. [24] examined the impact of various nanofluids as heat transfer fluids (HTFs), the results revealed that incorporating nanoparticles greatly enhanced the efficiency. Current research aims to improve collection methods through the use of nanoparticles in the HTF and the optimal design of parabolic trough collectors. Benabderrahmane et al. [25], used single-phase and two-phase models to simulate the turbulent forced convection of a hybrid nanofluid in a non-uniformly heated PTC receiver. The hydrodynamic results of both models were similar, but the thermal results were distinct. The study found that a hybrid nanofluid with 1.5% copper and 0.5% alumina in water provided the highest thermal performance. Khan et al. [26] performed computational experiments on three different designs of absorber tubes for the commercial LS-2 collector: a smooth tube, a tube with twisted tape insert, and a tube with longitudinal fins. Their results showed that the use of twisted tape inserts together with nanofluids drastically enhanced the thermal efficiency and gave a higher heat transfer coefficient than the smooth tube. Besides, the work of cascade solar energy conversion systems with double tube absorbers and nanofluids has been scarcely addressed.

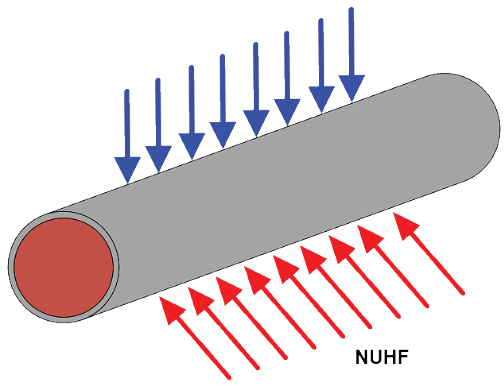
In order to increase energy efficiency, Mustafa et al. [27] present a new PTC system that uses a base fluid in the external glass cover and a double-fluid absorber tube inside the absorber tube. It was discovered that employing a double-fluid absorber tube with a nanofluid rather than a single-fluid system can increase the PTC's energy efficiency. In addition, it can impact the minimization of the heat loss of the commercial LS-2 collector. Khan et al. [28] explored the effect of employing a new receiver tube with a nanofluid on the energy performance of PTC with two heat transfer fluids. Also, it can impact the reduction of heat loss of the commercial LS-2 collector. They found that using this new absorber tube makes the single and two-fluid PTC system much more energy efficient. Moreover, they indicated that the collector efficiency can be enhanced when a smooth tube is employed by using a two-fluid system. Abbasian et al. [29] investigate the performance of the two-fluid PTC with a wavy grooved absorber tube. Based on their findings, using two heat transfer fluids, the thermal efficiency was substantially enhanced,



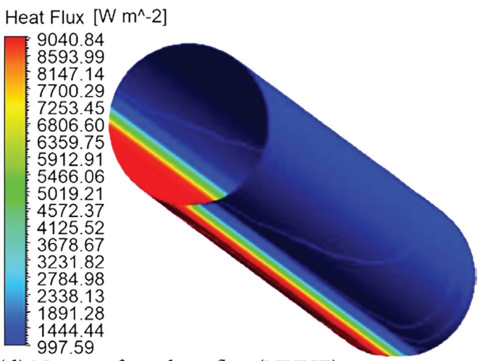
(a) The PTC Components



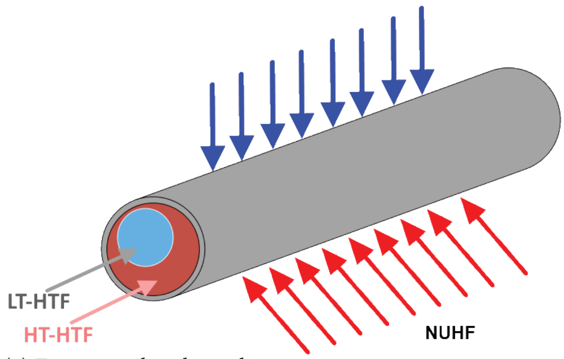
(b) Concentric absorber tube



(c) Conventional absorber tube



(d) Non-uniform heat flux (NUHF).



(e) Eccentric absorber tube.

Figure 1. Physical model.

from 38% up to 49%. Interestingly, the highest efficiency was obtained by the collector with a dual-fluid configuration with an exterior corrugated absorber tube and interior grooves and absorber roof and is about 70% when compared with smooth collectors at about 40%.

This work addresses the challenge of non-uniform thermal flux in parabolic trough collectors and seeks to improve thermal efficiency while providing two thermal energy levels. The study proposes a new concentric receiver tube of a cascade system type using two heat transfer fluids (HTFs) and nanofluids. The application of two HTFs allows conversion of solar energy at several ranges of temperature, optimizing the inner absorber tube temperature gradient, and improves the distribution of heat flux. This leads to increased heat transfer and minimized thermal stresses on the absorber. All these bring about improved performance and design of parabolic trough solar collectors (PTCs). Highlighted is the proposed methodology with the practical advantage of using renewable energies, promising higher performance, lower solar power station operating costs, and longer system collector durability, and these combine to ensure long-term solar energy sustainability.

PHYSICAL MODEL

A new parabolic trough concentrator (PTC) is presented along with a concentric receiver tube. The initial idea of a parabolic trough solar concentrator by Guerraiche et al. [16] is the underlying concept for the physical model. As depicted in Figures (1.a, b, c and e). The rim angle (φ_r) is 90°, the total aperture area (A_a=W×L) is 1.80 m^2 with an eccentric offset (e) of 10 mm along the y-axis; the inner tube is the one that is placed inside the absorber tube. Figures 1b, 1c, 1d, and 1e depict the details of the tube, showing the inner and outer diameters as 27 and 28 mm, respectively. The numerical investigation identifies a difference between a smooth absorber tube and a concentric receiver tube which has two separate HTFs that merge both in the concentric and eccentric configurations so as to get heat sources of high and low temperatures, respectively.

In this system, Water is selected as the low-temperature heat transfer fluid (LT-HTF) that flows through the inner tube and provides low-grade thermal energy. The high-temperature fluid (HT-HTF) adopted is Syltherm-800 oil which flows across the space between the inner surface of the absorber tube and the outer surface of the inner tube and serves as a carrier for high-grade thermal energy. The aim of this study is to propose a solution to offer significant insights and recommendations for enhancing heat transfer within the receiver tube, this will be achieved through the following approaches: Investigating the effects of concentric and eccentric pipe inserts in PTC. Comparing three system types: a conventional receiver with a single absorber tube and a single heat transfer fluid, a configuration with concentric absorber tubes, and a configuration with an eccentric receiver tube. Study the impact of various operational factors on the performance of the recently proposed system, with the aim of identifying the most optimal design.

Table 1. Properties of Syltherm-800 (HT-HTF) and water (LT-HTF) [30]

HT-HTF (Syltherme-800 oil)		***LT-HTF (water)***
ρ (kg/m³)	$aT^2 + bT + C\ /c = -6.0616 \times 10^{-2}$, $a = 1.1057 \times 10^3$, $b = 4.1535 \times 10^{-1}$	$aT^3 + bT^2 + cT + d/$ $a = 1.772 \times 10^{-5}$, $b = 2.067 \times 10^{-2}$, $c = 7.355$, $d = 1.71956 \times 10^2$
Cp (J/kg K)	$aT^2 + bT/a = 1.7080$, $b = 1.1078 \times 10^3$	$aT^4 + bT^3 + cT^2 + d/$ $a = 1.471 \times 10^{-6}, b = -1.973 \times 10^{-3}$, $c = 1.005 , d = 2.2965 \times 10^2$
K (W/m·K)	$aT^2 + bT + C/\ a = -5.7534 \times 10^{-10}$, $b = 1.8752 \times 10^4, c = 1.9002 \times 10^{-1}$	$aT^3 + bT^2 + c\ /a = 3.419 \times 10^{-8}$, $b = 4.581 \times 10^{-5}, c = +2.229$
μ (kg/m·s)	$aT^4 + bT^3 + cT^2 + dT + e/$ $a = 6.6720 \times 10^{-13}$, $b = -1.5660 \times\times 10^{-9}$, $c = 1.3882 \times 10^{-6}$, $d = -5.5412 \times 10^{-4}$, $e = 8.4866 \times 10^{-2}$	$aT^4 + bT^3 + cT^2 + dT + e/$ $a = 4.078 \times 10^{-11}$, $b = -5.502 \times 10^{-8}$, $c = 2.789 \times 10^{-5}$, $d = 6.302 \times 10^{-3}$, $e = 0.536574$

Table 2. Properties of the nanofluid (Al_2O_3/ Syltherm-800 oil) [30]

Properties	Formula
ρ (kg/m³)	$\rho_{nf} = (1-\varphi)\rho_f + \varphi\rho_s$
Cp (J/kg·K)	$(\rho C_p)_{nf} = (1-\varphi)(\rho C_p)_f + \varphi(\rho C_p)_s$
K (W/m·K)	$K_{nf} = \frac{K_s + 2K_f - 2\varphi(K_f - K_s)}{K_s + 2K_f + \varphi(K_f - K_s)} K_f$
μ (kg/m·s)	$\mu_{nf} = \frac{\mu_f}{(1-\varphi)^{2.5}}$

temperature and flow properties are stable over time. It is assumed that the particles of the nanofluid are dispersed evenly in the fluid so that the thermal properties are uniformly enhanced. Additionally, this model makes an assumption that there is no heat loss through the surrounding environment so that it can put its focus on the internal heat transfer processes. The flow inside the tubes is treated as fully developed, and the heat transfer correlations are used based on the standard Nusselt number formulas. Moreover, for the sake of computational stability, the material properties are considered to be constant throughout the temperature.

Table 3. Properties of (Al_2O_3) solid particles used in nanofluid [30]

Material	*ρ (kg/m³)*	*Cp (J/ (kg·K))*	*K (W/m·K)*
Alumina (Al_2O_3)	3970	765	40

Table 4. Properties of solid materials [30]

Material	*ρ (kg/m³)*	*Cp (J/kg·K)*	*K (W/m·K)*
Steel	8030	502.48	16.27

Efficacy investigation of nanofluid (*AL_2O_3/Syltherm-800 oil*) usage as a HT-HTF. Table 1 shows the properties of Syltherm-800 (HT-HTF) and water (LT-HTF) [30]. Tables 2, Table 3 and Table 4 give the properties of the fluids.

NUMERICAL AND CFD MODELING

The numerical modeling was accomplished with the finite volume method. The SIMPLE algorithm was used to handle the pressure-velocity coupling. The second-order upwind scheme was employed for the advection terms treatment in the momentum and energy equations. The local heat flux pattern in the PTR was obtained by coupling the CFD analysis with a Monte Carlo ray-tracing (MCRT) code. We use SolTrace open-source software to calculate the incident heat flux on the receiver tube by the MCRT method [12]. Figure 1.a depicts that the receiver bottom wall is having the highest flux; the top part is the least exposed to the flux. These NUHF distributions were implemented in the Fluent CFD model by polynomial correlations of the flux data. The correlations were devised by a User-Defined Function (UDF), which sets the actual boundary conditions on the outer surface of the absorber tube.

Somme assumptions are applied in the numerical and CFD modeling to optimize computational efficiency and ensure consistency in results. The model presumes that the conditions are at steady state, which implies that both

Boundary conditions used in the simulations included an ambient temperature fixed at 300 *K*, a NUHF on the absorber's outer wall (see Fig. 1.d), and zero velocity at the interface for the absorber's inner surfaces. The efficiency of the investigated systems (concentric and eccentric) is examined by varying the inlet temperatures and the mass flow rates for both heat transfer fluids a presented in Table 5.

Table 5. Temperature and mass flow values studied.

ṁ_water (Kg/m³)	0.006, 0.007, 0.008, 0.009
T in_water (K)	300, 308, 318, 328
ṁ_oil and nanofluid (Kg/m³)	0.06, 0.07, 0.08, 0.09
Tin_oil and nanofluid (K)	500, 550, 600,650
Volume concentration ϕ (%)	1%, 2%, 3%, 4%

Parameter Used in Analysıs

These factors are very important for figuring out how well the system under study transfers heat and moves fluids, as shown by the equations:

$$Re = \frac{\rho u d}{\mu} \tag{1}$$

$$Nu = \frac{hd}{K} \quad (2)$$

$$h = \frac{q_w}{T_w - T_m} \quad (3)$$

$$f = \frac{2\,\Delta P\left(\frac{d}{L}\right)}{\rho u^2} \quad (4)$$

Where: *d, L, ρ, μ, K, Tm, Tw, qw, ΔP*, are the inner diameter, Length of the receiver tube, density, viscosity, thermal conductivities of the HTF, average temperature of the Heat Transfer Fluid (HTF), average temperature on the inner surface of the inner tube or absorber tube, heat flux, pressure drop, u: the average velocity of the HTF.

The efficiency of heat collection (η) measures how effectively the collector converts the incident solar energy into thermal energy [29]. It can be expressed by,

$$\eta = \frac{Q_C}{A_a \cdot DNI} \quad (5)$$

In which, DNI is direct normal irradiance, A_a corresponds to the aperture area of the collector, Qc denotes the total collected heat, accounting for both high-temperature and low-temperature contributions from the LT-HTF and HT-HTF.

- For a concentric receiver tube

$$Q_c = (\dot{m}_{oil} \times Cp_{oil} \times \Delta T_{oil}) - (\dot{m}_{water} \times Cp_{water} \times \Delta T_{water}) \quad (6)$$

- For a conventional receiver tube

$$Q_c = \dot{m}_{water} \times Cp_{water} \times \Delta T_{water} \quad (7)$$

Grid Independency Test and Validation

Highly refined meshes are used to maintain computational accuracy and make sure that y^+ stays below 1. To achieve accurate numerical results, a refined mesh was applied, ensuring that y^+ remained under 1 as presented in Figure 2.

An independence study of a grid was performed to check for accuracy and credibility of results computed. Three grids were employed in simulation by Fluent to evaluate the Nusselt number and friction of a conventional PTR at a Reynolds number of 2.104. On comparing the results, the grid having 3522375 nodes was identified as the ideal grid to carry out the necessary simulations, as presented in Table 6.

Figure 2. Receiver tube mesh.

To ensure the validity of the developed numerical model derived in this study, a comparison was conducted with the calculation carried out by Bellos et al. of the numerical model suggested in this paper was checked by comparing it to results from Bellos et al. [22], who looked at how well circular and eccentric inserts worked in a PTC receiver tube (Fig. 3). Their study analyzed the performance of concentric and eccentric pipe inserts in the PTC receiver tube. Figure 3, shows a strong agreement with the maximum deviation in Nusselt number being below 8.95% and the maximum variation in the friction factor (*f*) remaining under 4.67% between the result of Bellos et al. [22] and the findings of the present study.

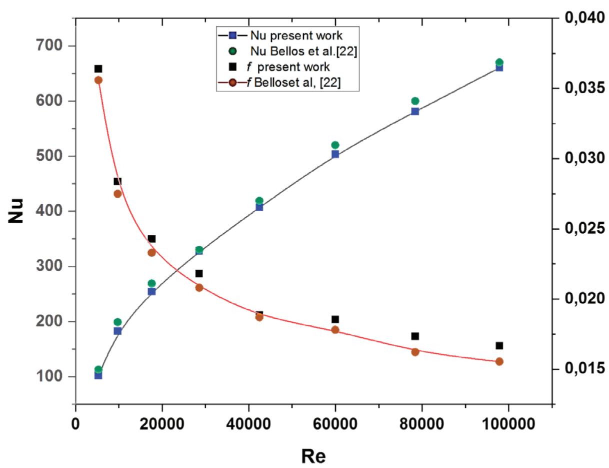


Figure 3. Validation: Nusselt number and the friction factor Vs Reynolds number.

Table 6. Grid independence study

Grid number	*Nu* (outer tube)	*Nu* (Inner tube)	*f* (outer tube)	*f* (inner tube)
1128625	218.85	46.23	0.035	0.06243
2259875	218.90	45.41	0.042	0.06251
3522375	218.82	45.32	0.032	0.06250
3657845	219.02	45.30	0.031	0.06248

RESULTS AND DISCUSSION

The results of numerical simulations for different input temperatures and mass flow rates in the conventional, concentric, and eccentric receiver tubes are shown, and the effects of the three geometries on the PTC›s heat transfer rate are examined.

Flow Structure

Effects of concentric and eccentric configuration

This section explores the impact of the concentric and eccentric configurations on the temperature and velocity fields inside the absorber tube and the heat transfer fluid. Fig.4 and Fig.5, showing temperature and velocity profiles for specific parameters: Tin_oil = $500K$, $\dot{m}\_oil$ = $0.7kg/s$, Tin_water = $300K$, and $\dot{m}\_water$ = 0.006 kg/s, and $\dot{m}\_water$ = 0.006 kg/s across three different models.

Figure 4 shows the temperatures around the circumference of the solar receiver tube, which is the metal surface

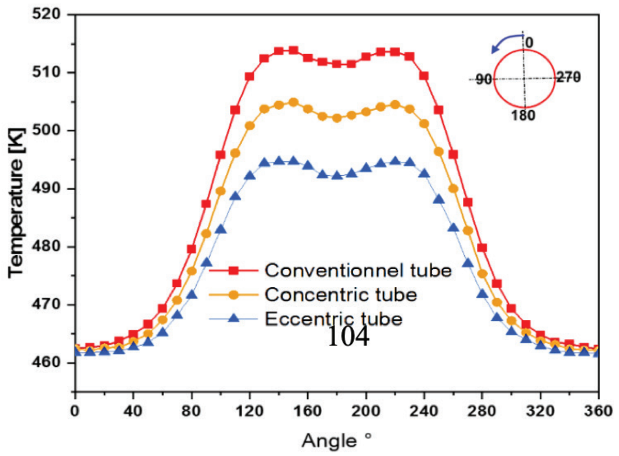


Figure 4. Temperature contour for the three PTC absorber models.

in contact with the heat transfer fluid, for the three PTC configurations. The figure makes it very clear that the temperature differences (ΔT) between the various points are becoming less and less when moving from the standard to the concentric and eccentric designs. The conventional tube exhibits a temperature gradient of 55 K, providing a baseline for assessing the improved thermal behavior of tubes with concentric or eccentric inserts. Which is reduced to $42K$ and $34K$ for the concentric and eccentric receiver tube arrangements, respectively. This corresponds to reductions of 45% and 60.60% compared to the single-tube configuration. The high peak temperatures and significant circumferential temperature gradients observed in the conventional tube can have several adverse effects on the absorber tube's performance and efficiency, such as excessive temperature stress and potential deformation. On the other hand, the concentric receiver tube exhibits lower peak temperatures and reduced circumferential temperature gradients.

Figure 5 demonstrates a significant increase in the velocity gradient in the lower section of the absorber tube. This leads to a higher local heat transfer factor in this particular region. In addition, the solar radiation in this area causes a high heat flux that can be more effectively dissipated by the heat transfer fluids. Improved dissipation capability significantly enhances the degree to which temperatures fluctuate around the absorber tube, especially compared with a simple, single-tube configuration. Adding concentric and eccentric pipe inserts, each containing an inner tube, is an effective means of reducing the cross-sectional area available for oil flow. This reduction in the flow area results in a significant acceleration of the oil, even though the overall mass flow rate matches that of an equivalent single-tube configuration. Consequently, the convective heat transfer from the oil to the inner wall of the absorber tube is significantly more efficient.

Effects of using nanofluid

In this section, investigates the impact of concentric and eccentric configurations on the temperature distributions

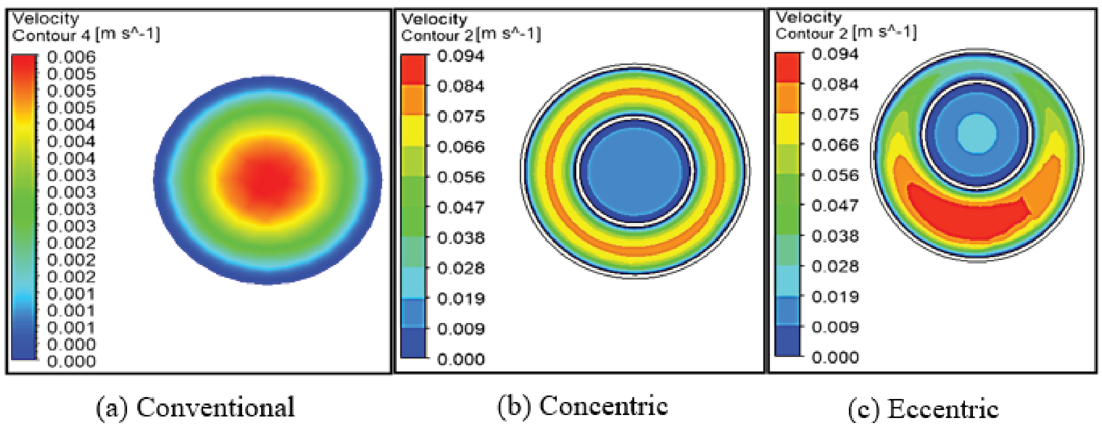


Figure 5. Velocity Contours.

within the absorber tube when using a nanofluid (*AL_2O_3/Syltherm-800 oil*) as *HT - HTF* and the water as *LT-HTF* for different operating conditions.

Effects of HT-HTF and LT-HTF mass flow rates on temperature gradients

This section illustrates the impact of changes in the mass flow rates of HT-HTF and LT-HTF on Tmax and the temperature gradient (ΔT) of the absorber tube, given specified parameters (*Tin_oil* = 600*K*, *Tin_water* = 300*K*, and *ṁ_oil* = 0.06 *kg/s*).

Figure 6 illustrates that increasing the mass flow rate of the *LT-HTF* in the inner tube has negligible effects on (T_{max}) and gradient temperature (ΔT) for both concentric and eccentric receivers respectively. However, the use of nanofluid significantly reduces ΔT by 5*K* in the concentric tube and 30*K* for the concentric and the eccentric tube respectively which present an improvement of 5% and 51.56% compared with a smooth configuration. This confirms that nanofluid enhances heat transfer and leading to a lower temperature gradient.

Figure 7 demonstrates that increasing the mass flow rate of the *HT - HTF* fluid results in a decrease in both T_{max} and ΔT for both concentric and eccentric receiver tubes. This is attributed to the improved convective heat transfer performance at higher *HT - HTF* flow rates. Notably, the application of nanofluid further reduces ΔT by 5*K* in the concentric tube and 30*K* in the eccentric tube respectively which presented 14.82% and 50.57% compared by system without nanofluid, highlighting its potential for enhancing thermal performance in solar receiver's tubes.

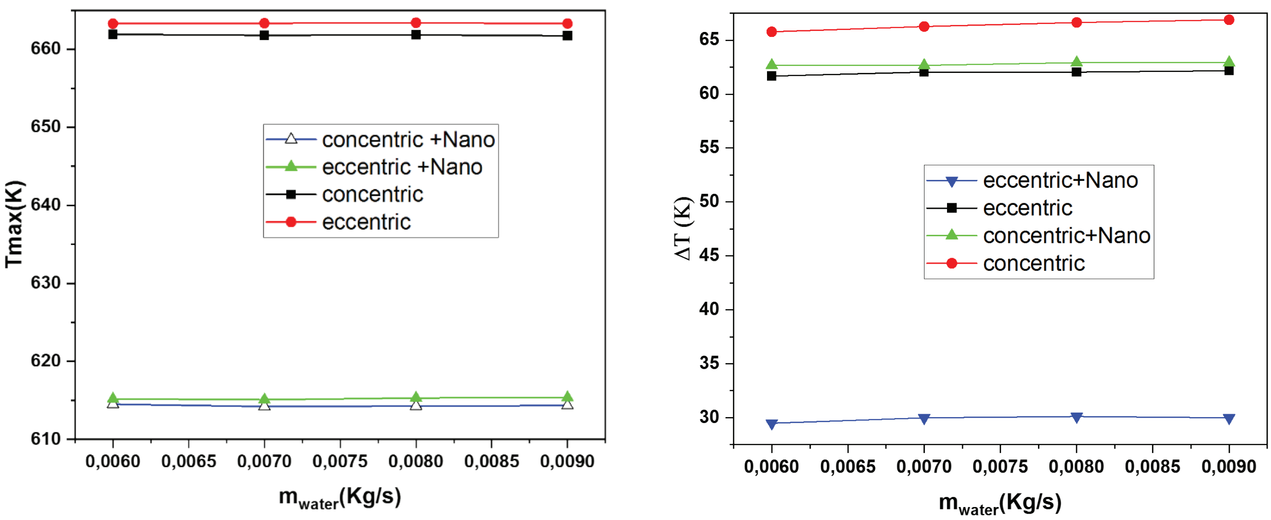


Figure 6. T_{max} and ΔT vs water (*LT - HTF*) mass flow rates.

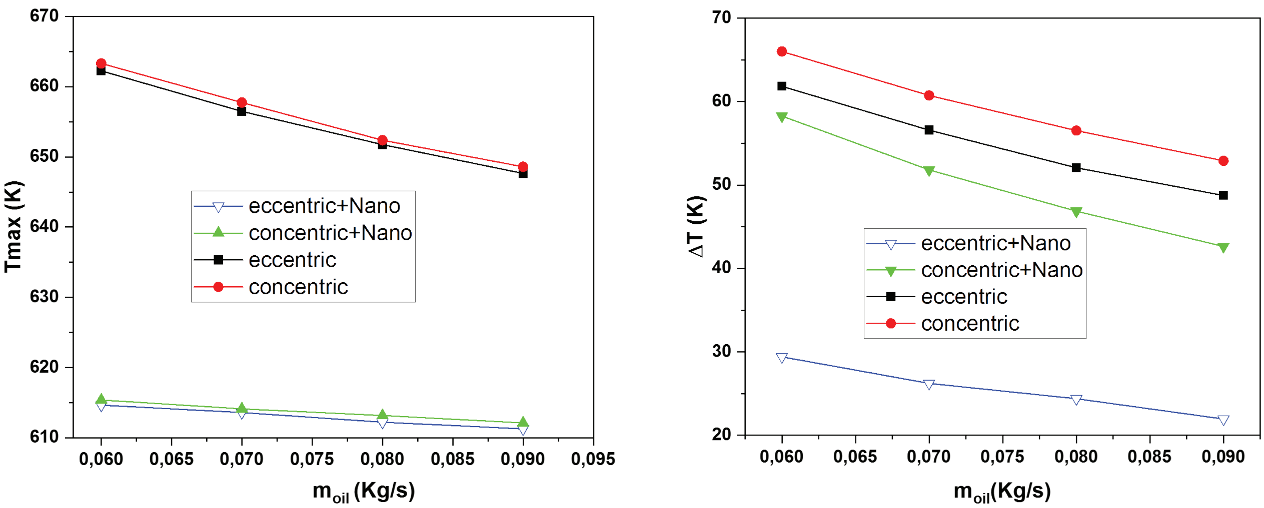


Figure 7. T_{max} and ΔT vs oil (*HT - HTF*) mass flow rates.

Effect of Lt-Htf, Ht-Hth inlet temperatures

We illustrate the impact of the low-and high-temperature heat transfer fluids inlet temperatures on the Tmax and the ΔT of the absorber tube under specified conditions $Tin\_oil = 500K$, $\dot{m}\_oil = 0.7 kg/s$, $Tin\_water = 300K$, and $\dot{m}\_water = 0.006\ kg/s$. In Fig. 9, adjustments in the inlet temperature of water (low-heat transfer fluids) within the inner tube exhibit minimal effects on the (T_{max}) and the ΔT for both concentric and eccentric receiver's tubes. However, employing nanofluid results in an average ΔT decrease of 13*K* in the concentric tube and *34K* in the eccentric tube. Figure 8 illustrates that a higher inlet temperature of the high-temperature heat transfer fluid results in a rise in Tmax for both concentric and eccentric receivers. Consequently, the ΔT of both receivers becomes smaller as the inlet temperature of the *HT-HTF rises*, with nanofluid lowering ΔT by *4 K* in the concentric tube and by *30 K* in the eccentric tube.

Effect of nanofluid concentration

Figure10 illustrates the effect of nanofluid concentration on Tmax and ΔT with: Tin_oil=600K, Tin_water=300K,$\dot{m}\_water = 0.006\ kg/s$, $\dot{m}\_oil = 0.006\ kg/s$. In The results indicate that both the Tmax and ΔT decrease as the concentration of nanofluid increases. The temperature gradient ΔT in an eccentric arrangement, is significantly greater than that in a concentric design, the latter having an average of 14K.

Heat transfer improvement

To better assess the concentric and eccentric cases and the effect of flow parameters on thermal performance, we define these two reports (dimensionless numbers): heat transfer improvement ratio by the relation (Nu_{co}/Nu_{ec}) and ratio of the increase in resistance to flow by (f_{co}/f_{ec}). Where: Nu_{co} and Nu_{ec}: Nusselt number of absorber tube inner surface for the concentric and the eccentric receiver tube respectively. f_{co} and f_{ec}: the frictional force factor of absorber

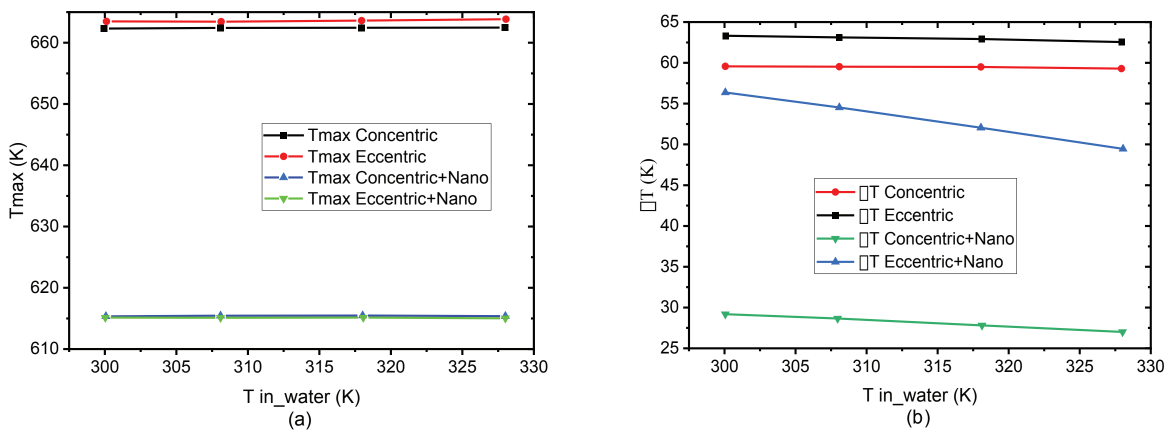
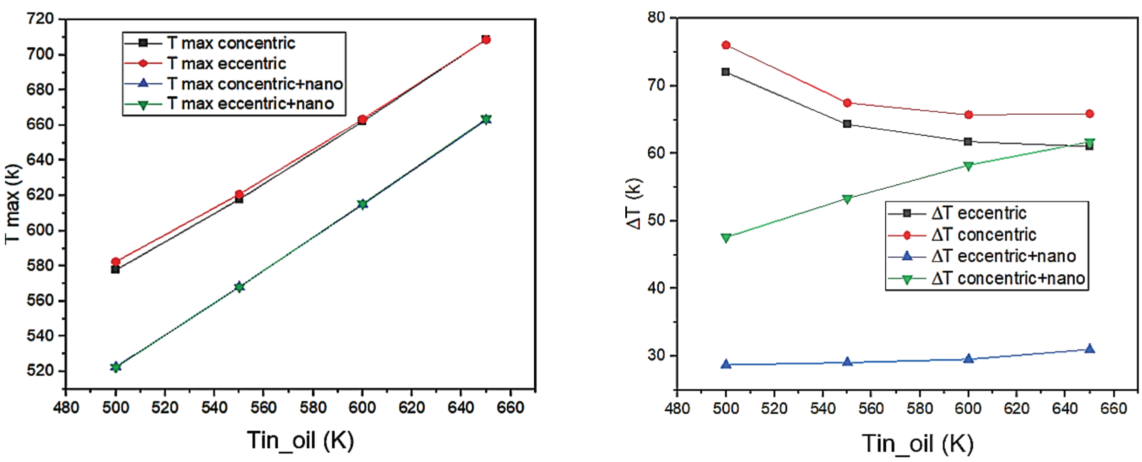


Figure 9. T_{max} and ΔT vs LT - HTF Temperature.



(a) (b)

Figure 8. T_{max} and ΔT vs HT - HTF Temperature.

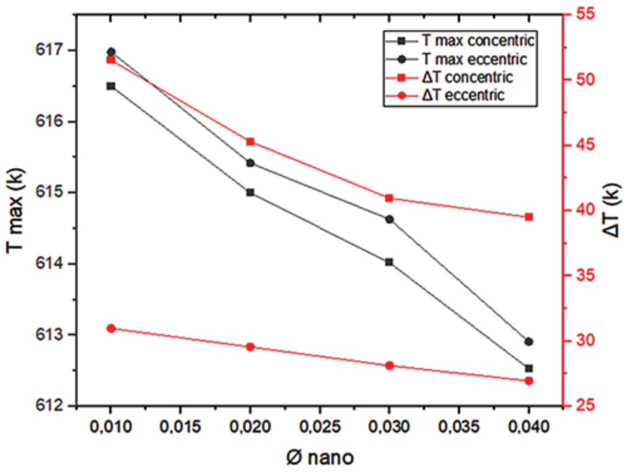


Figure 10. T_{max} and ΔT vs nanofluid concentration.

tube inner surface for the concentric eccentric receivers' tubes respectively.

Effects of the masse flow rate

Figure 11.(a) shows that the *LT-HTF* mass flow rate has a slight impact on the Nusselt number ratio for the base fluid. When using nanofluid, the Nusselt number ratio decreases slightly with increasing flow rate. This indicates that the heat transfer performance for the eccentric receiver configuration improves as the mass flow rate increases, given that the ratio is less than 1.0. The friction factor ratio remains close to 1.0 across all mass flow rates, indicating nearly identical flow resistance for both concentric and eccentric configurations for the base fluid. The friction factor ratio for the nanofluid stays around 1.05, indicating a slight increase in flow resistance about 5% for the concentric receiver compared to the eccentric receiver. This trade-off is considering the significant heat transfer benefits provided by the nanofluid.

Figure 11(b) shows the mass flow variations influence the heat transfer coefficient ratio as well as the friction factor ratio for the cases of the concentric and the eccentric receivers. The use of nanofluid elevates the heat transfer

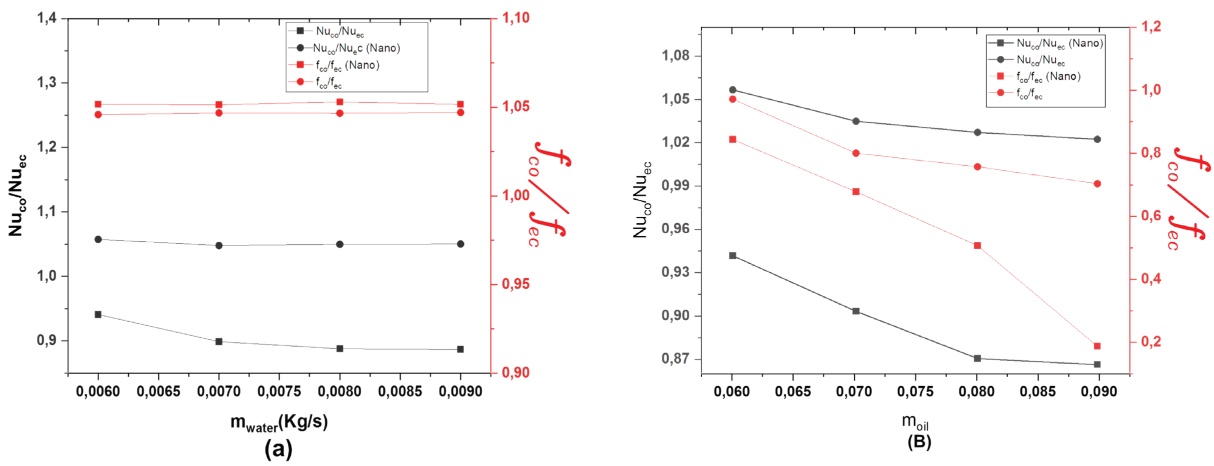


Figure 11. Nu_{co}/Nu_{ec} and f_{co}/f_{ec} vs *LT - HTF* temperature.

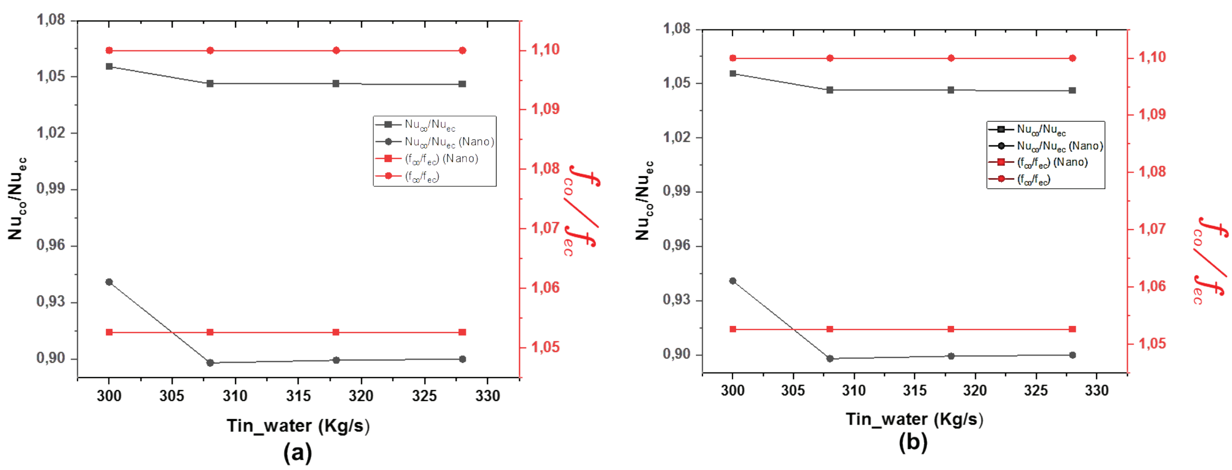


Figure 12. Nu_{co}/Nu_{ec} and f_{co}/f_{ec} vs inlet *LT - HTF* and *HT - HTF* temperatures.

coefficient on the inside of the absorber tube by 16.5% and raises the flow resistance to approximately 1.18 times that of the system without nanofluid.

Effects of the inlet LT-HTF and HT-HTH temperatures

In Figure 10.(a), alterations in the inlet temperature of water (*LT-HTF*) within the inner tube exhibit minimal impact on Nusselt number ratio and no effect on friction factor ratio for both receivers.

However, employing nanofluid enhances the heat transfer coefficient of the inner surface of the absorber tube by 14.5% and increases the flow resistance to approximately 1.04 *times* compare to without nanofluid.

As shown in Figure 12(b), as the inlet oil temperature *(HT-HTF)* increases, the values of the Nusselt number ratio and friction factor ratio for both receiver designs are higher. The eccentric receiver gives a better performance compared to the concentric design across the range of temperatures, with nominally larger ratios. This is because the decrease in oil viscosity with a rise in temperature, which increases convective heat transfer and hence overall thermal efficiency. Consequently, the heat transfer coefficient of the inner surface of the absorber tube improves by 18% and the flow resistance increases to about 1.22 *times* compare to without nanofluid.

Eccentric receiver operates more efficiently at increased *HT-HTF* flow rates, experiencing smaller relative increases in flow resistance and improving heat transfer with less drop-off at higher flow rates. Such findings help to adjust the system to the desired performance and operating parameters, where both heat transfer effectiveness and flow resistance are taken into account.

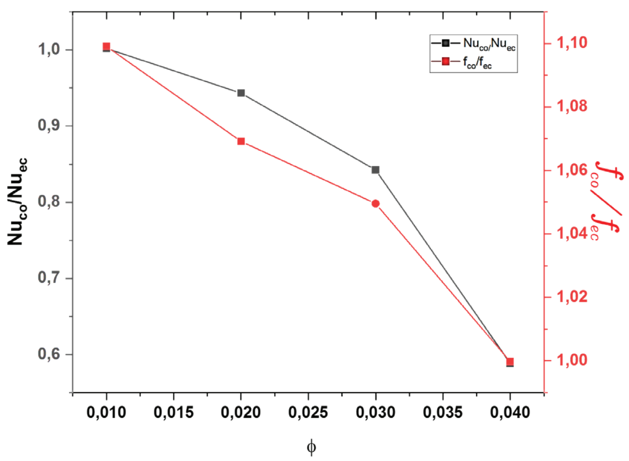


Figure 13. Nu_{co}/Nu_{ec} and f_{co}/f_{ec} vs nanofluid concentration.

Effect of nanofluid concentration

Figure 13 illustrates how the nanofluid concentration (f) affects the heat transfer enhancement ratio (Nu_{co}/Nu_{ec}) of the inner surface of the absorber tube and the ratio of increase in resistance to flow (f_{co}/f_{ec}) of nanofluid. It is observed that both (Nu_{co}/Nu_{ec}) and (f_{co}/f_{ec}) decrease with increasing concentration of nanofluid. This indicates that the heat transfer improvement ratio and the ratio of flow resistance in the eccentric receiver are larger than those in the concentric receiver, and they further increase with increasing nanofluid concentration.

Thermal performance

Figure 14 compares the effects of varying mass flow rates and inlet temperatures of the *HT - HTF*, both with and without nanofluid, on the thermal efficiency of different receiver configurations: conventional, concentric, and eccentric tubes. Figures 14. (a) and Figure 14. (b) depict the changes in heat collection efficiency of the concentric and

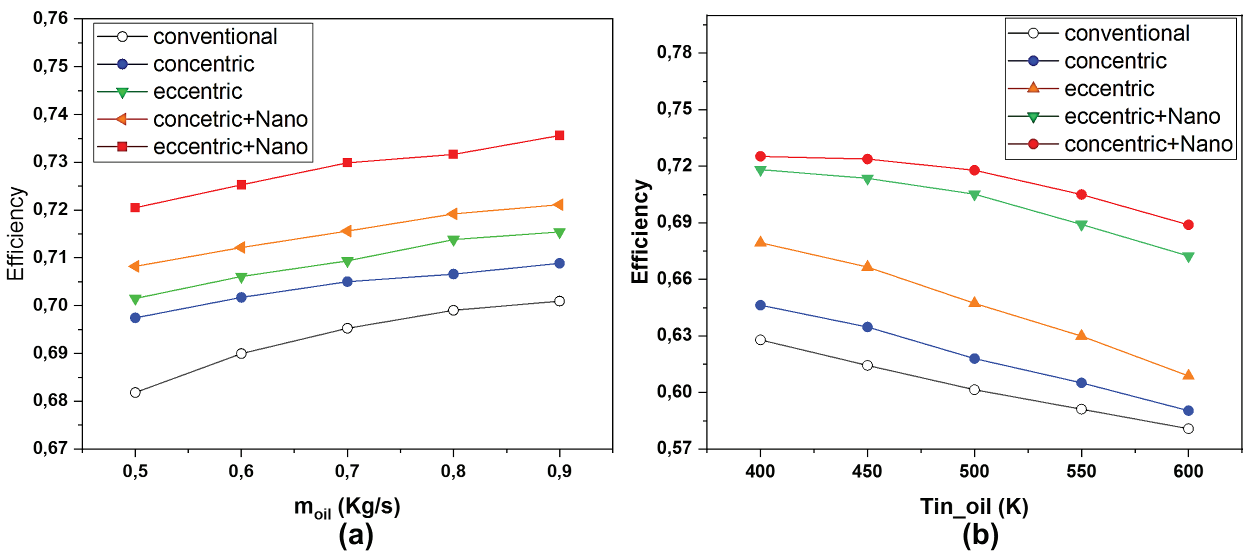


Figure 14. Thermal efficiency vs *HT - HTF* temperature and mass flow rate.

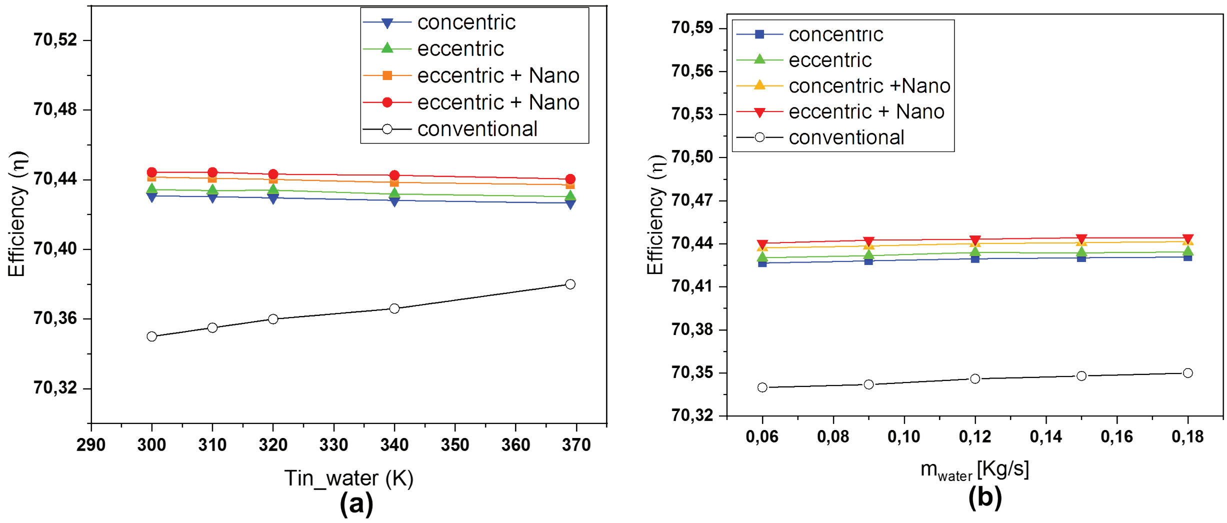


Figure 15. Thermal efficiency, vs *LT - HTF* temperature and mass flow rate.

eccentric receiver tubes over the conventional PTC design. These figures illustrate that the two types of receiver tubes are capable of achieving efficiency improvements up to 2.62% and 3.26% respectively. But this improvement of efficiency is dependent on the increase of the mass flow rate of the *HT-HTF*. Moreover, the presence of nanofluid leads to a very significant increase in efficiency, with an improvement of 6.12% for the concentric tube and 8.23% for the eccentric tube, thus making it possible for the two new designs to outperform the conventional receiver tube.

Figure 13, illustrates the impact of changes in the *LT-HTF* inlet temperature and the mass flow rates of different receiver configurations: conventional, concentric and eccentric receivers' tubes. It is noticed that the efficiency increases considerably with the declining the inlet temperature for *LT-HTF* and the growing of the masse flow rate. But the effects of the *LT-HTF* inlet temperature and the mass flow on the efficiency are limited.

As shown in Figure 14, the thermal efficiency increases by increasing the concentration of nanofluid, the PTC with eccentric receiver tube shows the best efficiency with a difference of 34.78% compared with the conventional receiver tube.

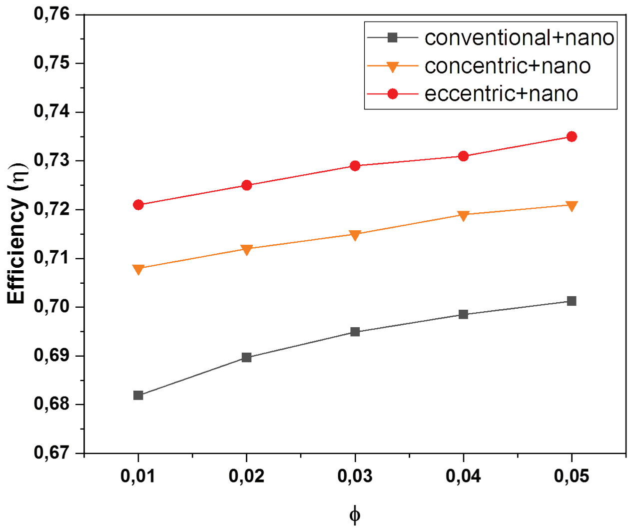


Figure 16. Thermal efficiency vs nanofluid concentration (ϕ).

Graphical Analysis of Results

This section presents findings on the heat performance of parabolic trough solar concentrators with different receiver tube designs. The changes in heat performance and efficiency of these configurations explored in the study are depicted by figures 17, 18, 19, and 20.

Using a concentric and eccentric receiver tube configuration reduced the receiver tube temperature gradients compared to a conventional absorber tube. Which is reduced to 42*K* and 34*K* for the concentric which translates to a 60.6% improvement compared to the conventional single-tube design. The concentric tube configuration also showed substantial benefits, with a ΔT reduction of 13*K*, or 14.82%, highlighting its effectiveness in minimizing thermal stresses and improving system reliability see Figure 15.

Increasing the mass flow rate of the HT-HTF fluid led to the reduction of the temperature gradient (ΔT) for both the concentric and eccentric receivers. Lowering the ΔT is attributed to the more effective convective heat transfer at the higher HT-HTF flow rates. Nonetheless, the inlet temperature and mass flow rate of the water in the inner tube only have a limited effect on this. From the Figure 18, it is also evident that the employment of nanofluid further brought down ΔT by 5K in the concentric tube and 30K in the eccentric tube, thus implying that the enhancements of 14.82% and 50.57%, respectively, as compared to the system

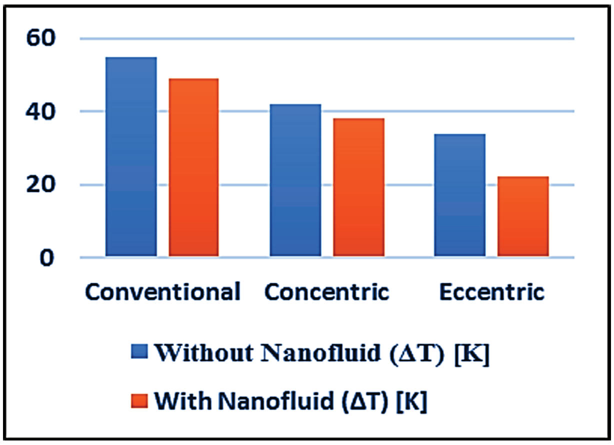


Figure 17. Temperature gradient (ΔT) reduction for different configurations.

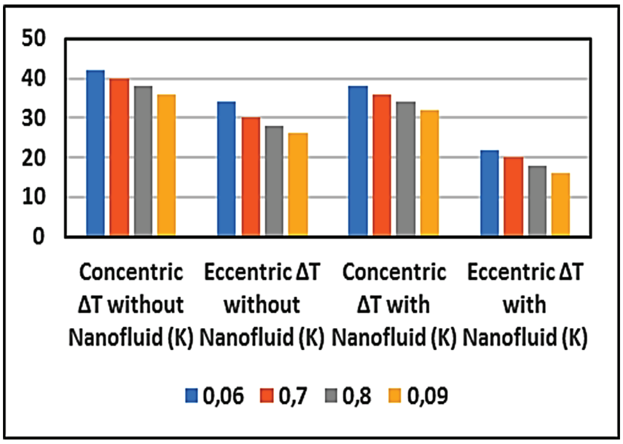


Figure 18. Temperature gradient (ΔT) vs *HT - HTF* mass flow Rate.

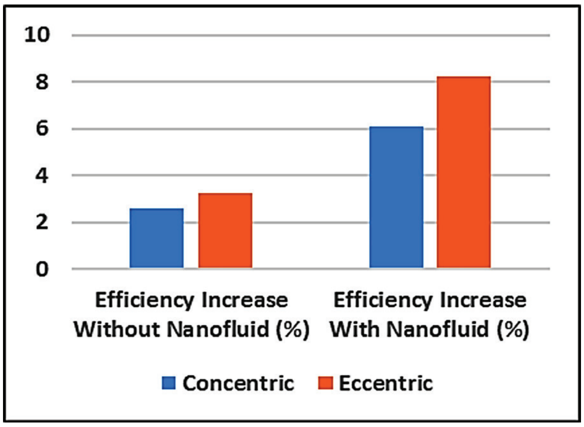


Figure 19. Temperature gradient (ΔT) reduction for different configurations.

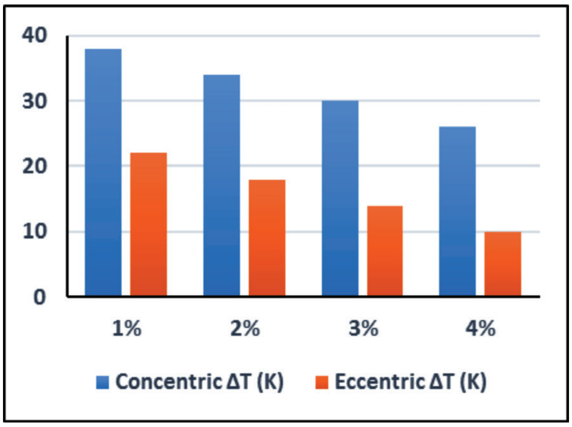


Figure 20. Temperature gradient (ΔT) vs *HT - HTF* mass flow Rate.

without nanofluid were attained, hence the application of nanofluid being notably appropriate for the solar receiver tubes to achieve the thermal performance increment.

CONCLUSION

This study analyzes the thermal performance improvement of a parabolic trough solar concentrator (PTC) using concentric and eccentric receiver tube geometries and nanofluids has been elaborated in the current research. The main aim was to address the issue of non-uniform heat flux (NUHF), which may result in temperature gradient and thermal stress and reduce system efficiency.

Both concentric and eccentric configurations are found to be effective in minimizing heat temperature gradients in the absorber tube. The percentage of reduction in the concentric receiver tube was 60.6% whereas it was only 45% for the eccentric design when both were compared to the smooth tube. Such decreases are significant as they alleviate the thermal stress on the tube, thus extending the reliability and the service life of the system.

Besides, an enhancement of the mass flow rate of the HT-HTF enhances convective heat transfer. Consequently, it reduces peak temperatures and temperature gradients in both configurations. Notably, employment of a 4% nanofluid concentration revealed much enhancement of performance. When the concentration was at this level, the temperature gradient was reduced by another 50.57% in the eccentric tube, while thermal efficiency in the PTC system was 34.78% greater than in conventional systems without nanofluids.

In terms of efficiency, both the eccentric and concentric designs performed better than the traditional PTC. The concentric tube recorded an efficiency gain of up to 2.62%, while the eccentric configuration reached 3.26%. Efficiency improuvement increased to 6.12% and 8.23% for the new PTC designs proposed combined with nanofluids for the eccentric and concentric tubes respectively. The above results support the idea that combining innovative receiver tube geometries with and optimized nanofluids can significantly increase thermal efficiency.

To summarize, the research highlights the influence of a receiver configuration and the selection of a heat transfer fluid in the performance enhancement of the parabolic trough solar collectors (PTCs). The use of fluid compositions and geometrically optimized flow will lead to a better distribution of the temperature even, higher heat transfer, and lower thermal stress that will basically result in system lifetimes being extended and their efficiency getting better. These kinds of improvements are particularly relevant to the clean energy sectors, which are now facing the challenge of bringing down the total costs of the plants while at the same time, making a less significant contribution to the atmospheric load of greenhouse gases. In the future, experimental validation and prototype fabrication will be necessary steps to turn these design concepts into feasible technologies that can be easily integrated into the sustainable energy systems.

NOMENCLATURE

PTC	Parabolic Trough Collector
PTR	Parabolic Trough Receiver
NUHF	Non-Uniform Heat Flux
HTF	Heat Transfer Fluid
CFD	Computational Fluid Dynamics
LT-HTF	Low-Temperature Heat Transfer Fluid
HT - HTF	High-Temperature Heat Transfer Fluid
AL_2O_3	Aluminium Oxide (nanoparticule)
DNI	Direct Normal Irradiance
MCRT	Monte Carlo Ray Tracing
UDF	User Defined Functions

Subscripts

co	Refers to concentric
ec	Refers to eccentric
f	Refers to fluid
s	Refers to solid
p	Refers to particle

Nomenclature

C_p	Specific heat, $kJ / kg°C$
d	Inner diameter, m
k	Thermal conductivity, $W/m°C$
L	Length of the receiver tube, m
$\dot{m}$	Mass flow rate, Kg/s
ρ	Density, kg/m^3
μ	Dynamic viscosity, ($kg/m{\cdot}s$)
Q_c	Collected heat, W/m^2
T	Température, $°C$
T_m	Main temperature, $°C$
T_w	Wall temperature, $°C$
q_w	Heat flux on the wall, W/m^2
ΔP	Pressure drop, Pa
u	Velocity, m/s

Greek symbols

ϕ	Nano fluid concentration, %.

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article

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