Experimental validation of energy parameters in parabolic trough collector with plain absorber and analysis of heat transfer enhancement techniques

F R Bilal a U C Arunachala b* H M Sandeep b
aSustainable & Renewable Energy Engineering Department, University of Sharjah, United Arab Emirates.
bDepartment of Mechanical & Manufacturing Engineering, Manipal Institute of Technology, Manipal University, Karnataka State, India

E-mail: *arun.chandavar@manipal.edu

Abstract. The quantum of heat loss from the receiver of the Parabolic Trough Collector is considerable which results in lower thermal efficiency of the system. Hence heat transfer augmentation is essential which can be attained by various techniques. An analytical model to evaluate the system with bare receiver performance was developed using MATLAB. The experimental validation of the model resulted in less than 5.5% error in exit temperature using both water and thermic oil as heat transfer fluid. Further, heat transfer enhancement techniques were incorporated in the model which included the use of twisted tape inserts, nanofluid, and a combination of both for further enhancement. It was observed that the use of evacuated glass cover in the existing setup would increase the useful heat gain up to 5.3%. Fe3O4/H2O nanofluid showed a maximum enhancement of 56% in the Nusselt number for the volume concentration of 0.6% at highest Reynolds number. Similarly, twisted tape turbulators (with twist ratio of 2) taken alone with water exhibited 59% improvement in Nusselt number. Combining both the heat transfer augmentation techniques at their best values revealed the Nusselt number enhancement up to 87%. It is concluded that, use of twisted tape with water is the best method for heat transfer augmentation since it gives the maximum effective thermal efficiency amongst all for the range of Re considered. The first section in your paper

1. Introduction
The increasing trend of energy demand and adverse effect on atmosphere due to greater use of conventional energy sources has mainly influenced the urge to investigate alternative sources of energy. Renewable energy avoids/minimizes pollution and environmental imbalance that leads to climate change and global warming. These resources include, geothermal, hydro, solar, wind, biomass, tidal and wave energy. Among them, solar energy is a high radiant energy source with tremendous advantages over other alternative energy sources. Based on the solar energy conversion principle, the output could be electric or thermal. The medium (100°C-250°C) and elevated (250°C-400°C) thermal applications are centered on the concentration level of the thermal system. A Parabolic Trough Collector (PTC) is basically a line focus concentrator available over a wide range of aperture areas from about 1 to 60 m2 and with widths ranging from 1 to 6 m. Concentration ratio range from 10 to 80, and rim angle from 70° to 120° to work in the temperature range of 150° to 400°C [1]. Numerous studies have been conducted to evaluate the energy efficiency parameters of PTC. A numerical model to evaluate the heat loss from an absorber to surroundings was developed by Tijani and Roslan [2]. It highlighted that 64% of total heat loss across the glass envelope is due to convection and the heat loss due to radiation is 36% at a wind speed of 2 m/s. The effect of geometry of glass cover on heat flux distribution were analysed using
Monte Carlo Ray Tracing (MCRT) by Fuqiang et al. [3]. To minimize the gradient of heat flux distribution on the absorber surface that can reduce the thermal stress of the absorber, a glass cover with elliptic–circular cross section was proposed. The one-dimensional mathematical model for PTC (considering all the heat transfer mechanisms in the energy transformation process) was developed by Liang et al. [4] and validated with the test data from Sandia National Laboratories. This model is precise enough when compared with the 3D models from other research and the experimental data. The average simulated outlet temperature from the model was 0.65°C higher than the test data. Whereas, the result of 3D model gave 2.69°C than the experiment results. Manglik et al. [5] have studied the thermo-hydraulic characteristics of water and ethylene glycol in a tube with twisted tape inserts with twist ratios of 3.0, 4.5 and 6.0 under laminar flow and uniform wall temperature conditions. The authors have proposed a means to predict the mode of heat transfer in terms of Grashoff’s number (Gr) and swirl parameter (Sw). When Gr is much greater than Sw², heat transfer occurs mainly due to free convection and when Gr is much smaller than Sw², the swirl flow intensifies and heat transfer is mainly due to forced convection. A generalized correlation for estimating the Nusselt number and friction factor incorporating thermal entrance effects a fully developed swirl flow as well as the combined effect of forced and free convection have been developed, which is found to be in good agreement with the experimental data. Similar studies were carried out by Manglik et al. [6] even in the turbulent flow regime. Based on the experimental results, correlations for Nusselt number and friction factor have been developed. Further, the friction factor in case of laminar flow [5] has been combined with the correlation developed in the present work and presented as a single expression which is found to agree with the experimental data within ±10%. Lu et al. [7] have investigated the non-uniform heat transfer characteristics of PTC. The heat loss and system efficiency evaluated from both uniform and non-uniform models fit with the experimental results. But, heat loss in case of non-uniform model is slightly higher than the uniform model. Under off-sun condition, the deviation between heat transfer as predicted by both uniform and non-uniform models is not significant. Whereas, during on-sun condition, it has been suggested to use non-uniform model only. Montes et al. [8] have developed a thermo-fluiddynamic model for PTC, which is well suited for different design options. It has been applied to evaluate collector thermal performance with different working fluids: oil, molten salt, and water/steam. The heat transfer analysis involved four parameters viz. PTC length, absorber diameter, working temperature and pressure. The influence of these parameters on heat loss, pressure drop, energy, and exergy efficiencies were also studied. Exergy was considered in optimization process as it accounts for all relevant energy gains and losses, characterized by their corresponding temperature and pressure. Based on the analysis, direct steam generation was found to be more efficient than oil and molten salt systems. Yaghoubi et al. [9] have performed numerical and experimental analysis to find the impact of failure of absorber envelope on performance of PTC based 250 kW plant. The amount of heat loss is compared numerically for three different types of tubes viz. vacuum, lost vacuum and broken glass tube. The experimental data were used to validate the numerical result. The heat loss in vacuum lost receiver was 46% higher compared to vacuum tube, which leads to 3-5% reduction in the system performance. The amount of heat loss was 58.5% (19% drop in system performance) in case of broken glass tube as compared to vacuum tube. Numerical model was presented by Ouagued et al. [10] to simulate the Heat Transfer Fluid (HTF) temperature change, HTF heat gain and PTC heat loss during the sunshine hours using Syltherm, Marlotherm and Therminol oils. The results showed that an increase in absorber temperature increases heat loss as well as a decrease of heat gain. Based on the cost, thermal capacity, temperature range and availability, Syltherm was suggested as the best HTF. MCRT coupled to a finite volume solver was used to model 3D heat transfer in a PTC by Wirz et al. [11]. This allowed the incorporation of non-uniform temperature and heat transfer distributions and the identification of critical peak temperatures and heat fluxes. The computed heat losses and thermal efficiencies agree well with the experimental data. This 3D model can predict glass temperatures more accurately than previous gray models and temperature correlations. A detailed model of heat loss in PTC was developed using commercial heat transfer software by Roesle et al. [12] which included radiation as well as conduction and convection heat transfer. The in-detail simulation involved non-uniform irradiation and complex gas behaviour in the annular gap between absorber tube and vacuum jacket. Simulations have shown the validity of the model with respect to both heat loss and temperature distribution. The simulation result indicated that the
present receivers are suffering from excessive radiation loss. In view of this, solar selective coatings for absorber and antireflective coatings for vacuum unit were suggested.

The thermal performance of a PTC was numerically investigated using FEM method by Wang et al. [13]. The solar energy flux distribution in the circumferential direction is non-uniform due to combination of direct and concentrated radiation. The circumferential temperature difference increases with solar irradiation and decreases with increase in temperature and velocity of the flowing fluid. It was observed that for the range of fluid flow velocity of 1-4 m/s, solar irradiation of 500-1250 W/m² and fluid inlet temperature of 373-673 K, the circumferential temperature difference can attain 22-94 K. The stress distribution and deformation of absorber were calculated which is useful for better design of the absorber. A prototype of an innovative 5 kW medium temperature solar receiver/reactor was manufactured and tested by Jin et al. [14]. The system was used for methanol decomposition which gave a conversion efficiency in the range of 50-95% at a feeding rate of 0.035 lpm for a mean solar flux between 300 W/m² and 800 W/m². The efficiency of solar thermal energy conversion to chemical energy reached was 30-60%. Hence it offers an excellent opportunity for the development of an economical solar thermochemical technology. Research by Giostri et al. [15] dealt with the development and testing of an innovative code for the performance prediction of PTC plants in off-design conditions. This model can be used either for single calculation in a specific off-design condition or for a complete year simulation. This was tested with reference values found in literature in view of real applications and showed good agreement. It can be used to optimize plant components and layout in feasibility studies as well as to select the best control strategy during individual operating conditions. Bellos et al. [16] have investigated the thermal performance of PTC using Syltherm-800 as HTF with internally finned receivers. The performance parameters of receiver with different fin thickness (2, 4 and 5 mm) and lengths (5, 10, 15 and 20 mm) were numerically determined and compared with the plain receiver. Increase in fin thickness and length resulted in enhanced thermal performance and increased pressure drop. Based on the thermal performance index, fin with 20 mm length and 4 mm thickness was found to give optimum thermo-hydraulic performance. Xiangtao et al. [17] have numerically investigated the influence of receiver with pin fin arrays on overall thermal performance of PTC using thermal oil D12 as HTF. The Monte Carlo Ray Tracing technique along with Finite Volume Method has been adopted and the numerical results were validated against the experimental data from DISS test facility in Spain which was found to be in well agreement. The outcome of the analysis revealed that the use of pin fin arrays resulted in 9% increase in Nusselt number and 12% increase in overall thermal performance factor compared to plain receiver. Fuqiang et al. [18] have numerically investigated the thermo-hydraulic and structural performance of a symmetric outward convex corrugated receiver for PTC with thermal oil D12 as HTF. The numerical results were validated against the results of experiments conducted in the DISS test facility in Spain. It has been observed that the following receiver enhanced the overall heat transfer coefficient by 8.4% and decreased the maximum thermal strain of the receiver by 13% when Re = 81728, p/D = 4.3. Several heat transfer enhancement techniques such as use of receiver with Internal hinged blades (Kalidasan et al. [19]), internally finned receivers (Bellos et al. [20]), helical screw tapes (Song et al. [21]), porous disc enhanced receiver (Reddy et al. [22], Jamal-Abad et al. [23]), twisted tape inserts (Jaramillo et al. [24]), use of nano fluid in converging-diverging receiver (Bellos et al. [25]), wall detached twisted tapes (Mwesigye et al. [26]), and porous rings (Ghasemi et al. [27]) have been investigated and obtained satisfactory outcome.

The receiver with evacuated glass envelope in PTC yields thermal efficiency in the range of 65-70% which is about 10% higher than PTC with non-evacuated receiver as discussed by Kasaeian et al. [28]. Though the evacuated receiver traps maximum reflected solar flux, the system efficiency remains low as the trapped heat is not properly transferred to the HTF. To make significant quantum of heat available for HTF, the research is also focused on different heat transfer augmentation techniques which can further increase the thermal efficiency of the system.

The presence of nanoparticles in the base fluid increases the effective thermal conductivity of the mixture, thus increasing the overall heat transfer coefficient. An analytical model of PTC consisting of evacuated quartz receiver with spectrally selective coating covered by quartz envelope has been developed by Risi et al. [29] with (CuO+Ni)/N₂ nanofluid as HTF. Further optimization of the developed model resulted in peak effective thermal efficiency of 62.5% with outlet temperature of 650°C. Numerical study by Ghasemi et al. [30] revealed that the addition of nanoparticles to the base liquid
increases the radiation absorption characteristics of the resulting mixture and hence the thermal efficiency of PTC. Sokhansefat et al. [31] performed three-dimensional numerical analysis of fully developed turbulent flow based mixed convective heat transfer in a PTC receiver for Al$_2$O$_3$/synthetic oil nanofluid under non-uniform heat flux boundary condition. The average convection heat transfer coefficient was found to increase by 14%, 8.6% and 6% with addition of Al$_2$O$_3$ nanoparticles ($\phi = 5\%$) to the base fluid at operating temperatures of 300 K, 400 K and 500 K respectively. Wen et al. [32] have presented the experimental findings on convective heat transfer characteristics of $\gamma$-Al$_2$O$_3$/H$_2$O nanofluid ($\phi = 0.1\%, 1.0\%$ and $1.6\%$) in comparison with water under constant heat flux condition and laminar flow ($500 < Re < 2100$) regime. The rise in heat transfer coefficient of nanofluid compared to water was much higher at a given value of $\phi$. The maximum rise in heat transfer coefficient was at the entrance region due to decrease in thickness of thermal boundary layer and subsequently decreases along the length of the tube. Hence it can be concluded that the enhancement in heat transfer coefficient is due to combined effect of increase in effective thermal conductivity and the particle migration resulting in decreased thickness of thermal boundary layer. However, above a certain value of $\phi$, the thermal parameters show downward trend due to agglomeration of nanoparticles.

The heat transfer enhancement with inserts is due to the combination of increased effective heat transfer area, swirl generation and increase in flow turbulence with interruption to the growth of boundary layer. Mwesigye et al. [33] have numerically investigated the heat transfer and pressure drop characteristics of Syltherm-800 in PTC receiver with wall detached twisted tapes inserts of various twist ratios (0.30 to 2.42) and width ratios (0.61, 0.76 and 0.91) under turbulent flow condition (10260 < Re < 320000). It was observed that the circumferential temperature difference in case of plain receiver increases under elevated concentration ratio at constant Re, whereas the drop in circumferential temperature of 4-76% was achieved with inserts. The augmentation in thermal enhancement factor was 0.74 to 1.25 times compared to plain receiver. Numerical analysis of PTC receiver with Syltherm-800 as HTF under turbulent flow condition was studied by Cheng et al. [34] using the unilateral multi-longitudinal vortexes enhanced parabolic trough solar receiver (UMLVE-PTR). The longitudinal vortex generators were located on the lower portion of receiver which is exposed to concentrated solar flux. The heat loss in UMLVE-PTR was reduced by 1.35% to 12.10% compared to smooth PTC receiver within $3.8 \times 10^4 < Re < 7 \times 10^5$. The numerical simulation of fully developed turbulent flow ($1 \times 10^4 < Re < 2 \times 10^5$) of Therminol VP-1 in a receiver with helical fins, protrusions and dimples were studied by Huang et al. [35]. For the tubes with dimples with same surface characteristics, the ratio of Nusselt number for tube with inserts to that of friction factor plain receiver was increased by 4% to 64%. Experimental investigation on the influence of twisted tape insert ($\gamma = 5, 10$ and $15$) on thermal and hydraulic characteristics of TiO$_2$/H$_2$O nanofluid ($\phi = 0.5\%$ to $3.0\%$) have been investigated by Azmi et al. [36] under turbulent flow regime ($8000 < Re < 30000$) and uniform heat flux conditions. The results revealed that nanofluid ($\phi = 1\%$) in plain tube at Re = 23917 gives the best performance. The effect of perforated louvered twisted tape inserts (LTT) on the convection heat transfer co-efficient of PTC receiver with Behran thermal oil as HTF was numerically investigated by Ghadirijafarbeigloo et al. [37]. Using Soltrace code, Non-uniform heat flux boundary condition was assigned to the outer surface of the receiver. The overall rise in Nusselt number and friction factor in case of perforated LTT was 150% and 210% respectively compared to plain receiver subjected to best flow condition. Based on the thermal performance ratio (2.2), use of perforated LTT with twist ratio of 2.67 at Re = 5000 have been suggested.

Till date research focusses mainly on overall performance analysis, stress and temperature distribution in the receiver, validation of mathematical modelling, use of nanofluids and turbulators in PTC. The information pertaining to the combined application of nanofluid and turbulators in PTC is scanty. Hence the present study is concentrated on validation of plain receiver based mathematical model for performance analysis of PTC and its extension to evaluate the heat transfer enhancement with the combined use of nanofluid and turbulator.

2. Mathematical modelling of PTC
The basic elements making up a PTC are (i) the absorber tube located at the focal axis through which the liquid to be heated flows, (ii) the concentric transparent cover over the absorber tube, (iii) the reflector, and (iv) the support structure (with tracking mechanism).
Significant amount of losses occurs during energy transformation (i.e. between direct solar radiation as input and useful heat gain as output). Hence the degree of optical loss is presented as:

\[ \eta_{opt} = \rho r \alpha \]  

(1)

Based on the trough and absorber geometry, the concentration ratio is known as the ratio between effective aperture area and absorber area.

\[ C = \frac{(W-D_o)}{\pi D_o} \]  

(2)

Geometrical losses are due to the incidence angle (\( \theta \)) of direct solar radiation on the aperture plane of the collector and the sun’s vector, both contained on a plane perpendicular to the collector axis. This angle depends on the day of the year and the time of day. It is a very important factor, because the fraction of direct radiation that is useful to the collector is directly proportional to the cosine of this angle. An equation quoted by [1] is dealt with to account for the incident angle effect considering E-W tracking mode. The tilt factor (\( n_b \)) for the aperture plane is defined based on solar azimuth angle.

\[ \cos \theta = \sqrt{(\sin \phi \sin \delta + \cos \phi \cos \delta \cos \omega)^2 + \cos^2 \delta \sin^2 \omega} \]  

(3)

Hence the flux absorbed by receiver (without glass envelope) is:

\[ S = S_p \eta_{opt} + S_p \eta_{opt} \frac{D_o}{W-D_o} \]  

(4)

The thermal resistance network indicating various heat losses for the bare PTC receiver is shown in Figure 1. The overall heat transfer coefficient between receiver and ambient is evaluated by considering total thermal loss (radiative and convective). To consider the convective heat loss, a correlation by Churchill & Bernstein [38] is incorporated. The radiative heat loss from the receiver to the surrounding is based on sky temperature as discussed in Forristall [39]. The thermophysical properties are estimated at bulk mean temperature of HTF.

\[ T_{bulk} = \frac{T_{f1} + T_{fa}}{2} \]  

(5)

\[ Nu_{rc} = 0.3 + \frac{0.62 Re^{1/2} Pr^{5/3}}{[1+(0.4/Pr)^{2/3}]^{1/4}} \left[ 1 + \left( \frac{Re}{282000} \right)^{5/8} \right]^{4/5} \]  

(6)

\[ h_{rr} = \varepsilon \sigma (T_r + T_{sky})(T_r^2 + T_{sky}^2) \]  

(7)

The overall heat loss coefficient is given by,
The Nusselt number and the friction factor for internal flow in receiver are calculated by Gnielinski’s correlation [40] for single-phase fluid.

\[ Nu = \frac{f L (Re - 1000)}{(1 + 12.7 \left( \frac{Pr}{Re^{2/3} - 1} \right))} \]

(For 2300 < Re < 5 × 10^6, 0.5 < Pr < 2000)

\[ f = (1.58 \ln(Re) - 3.82)^{-2} \]

If the flow is viscous and considered to be thermally and hydrodynamically developing, correlation quoted in Kothandaraman & Subramanayan [41] should be considered.

\[ Nu = 3.66 + \frac{0.104(Re Pr \frac{D_t}{T})^{-0.8}}{1 + 0.16(Re Pr \frac{D_t}{T})^{-0.8}} \]

The convective heat transfer co-efficient of the HTF is given by,

\[ h_{hf} = \frac{Nu \times k}{D_t} \]

The overall heat transfer coefficient is given by,

\[ U_o = \left[ \frac{1}{U_1} + \frac{D_o}{D_1} \times \frac{1}{h_{hf}} + \frac{D_o \times \ln(D_o/D_t)}{2k_r} \right]^{-1} \]

The collector efficiency factor and the collector heat removal factor is given by Equation 14 and Equation 15 respectively.

\[ F' = \frac{U_2}{U_1} \]

\[ F_R = \frac{mc_p}{A_r U_1} \left( 1 - e^{\frac{-U_1 F_R A_r}{m c_p}} \right) \]

The useful heat gain and average receiver surface temperature are evaluated based on the overall heat loss coefficient and heat removal factor [42] which are given by Equation 16 and Equation 17 respectively.

\[ Q_u = F_R (W - D_o) L \left[ S - \frac{U_1}{C} (T_i - T_o) \right] \]

\[ T_r = \frac{(W L) \frac{1}{\eta_{opt}} + m c_p T_i F_R A_r U_1 T_o}{m c_p F_R + A_r U_1} \]

The pumping power is given by,

\[ P_p = \frac{\dot{m} \Delta p}{\rho \eta_p} \]

The effective thermal efficiency is a crucial factor, which provides a tool to compare the basic model with different augmentation techniques. The electrical conversion factor is assumed to be 18.4% as suggested by Kumar [43].

\[ \eta_{eth} = \frac{Q_u \frac{\dot{p}}{W L}}{W L} \]

By considering various energy parameters, a MATLAB code (Figure 2) is written to evaluate the performance of PTC.

3. Heat transfer augmentation techniques

As the base model was validated, heat transfer augmentation techniques were applied to predict the enhanced performance of the system.

3.1. Evacuated glass envelope

Since plain tubes are directly exposed to the ambient, large amount of concentrated heat is lost by convection. Hence PTC with plain receiver gives less thermal efficiency. With the advancement of solar thermal technologies, evacuated receivers were developed which improves the thermal efficiency of the system. Here, plain metallic tube is surrounded by a glass envelope which is sealed to the absorber tube at both the ends. The annular space between plain metallic tube and the glass cover is vacuumed, which substantially eliminates the heat loss. Hence, the modified mathematical model is considered.
dimensions used in the calculation is \( D_{g}=45.9 \text{ mm} \) and \( D_{t}=45.4 \text{ mm} \). This size is of proportional scaling to the commercial SCHOTT [44] evacuated glass receivers.

Considering the absorber tube and the glass cover around it to constitute a system of long, concentric tubes, two equations to describe the heat loss from the system is discussed by Sukhatme & Nayak [1], which would help in solving the algorithm.

\[
\frac{Q_{l}}{l} = h_{w}(T_{r} - T_{c}) \pi D_{o} + \left( \frac{\sigma \pi D_{o}}{T_{r}^{4} + T_{g}^{4}} \right) \quad \text{------------------------------------------ (20)}
\]

\[
\frac{Q_{l}}{l} = h_{w}(T_{c} - T_{a}) \pi D_{og} + \sigma \pi D_{og} \varepsilon_{g}(T_{c}^{4} - T_{sky}^{4}) \quad \text{------------------------------------------ (21)}
\]

After the algorithm loop achieves convergence, the overall heat transfer coefficient is calculated as:

\[
U_{l} = \frac{Q_{l}}{A_{r}(T_{r} - T_{a})} \quad \text{------------------------------------------ (22)}
\]

Other calculations are same as mentioned in base model.

3.2. Nanofluids

Nanofluids have novel properties that make them potentially useful in many heat transfer applications. The nanoparticles used are typically made of metals, oxides, carbides or carbon nanotubes and the common base fluids include water and ethylene glycol. In the present study, effect of Fe\(_{3}\)O\(_{4}\)/H\(_{2}\)O nanofluid on the performance of PTC is investigated. Here the nanoparticles can be separated from base fluid due to its magnetic nature which is not possible with other nanoparticles like Al\(_{2}\)O\(_{3}\), Cu, TiO\(_{2}\) etc.

Density and specific heat of nanofluids are calculated by using Pak & Cho [45] correlations, which are defined as follows:

\[
\rho_{nf} = (1 - \varphi) \rho_{w} + \varphi \rho_{np} \quad \text{------------------------------------------ (23)}
\]

\[
C_{p,nf} = (1 - \varphi) C_{p} + \varphi C_{np} \quad \text{------------------------------------------ (24)}
\]

Timofeeva et al. [46] introduced the effective medium theory for computing thermal conductivity of nanofluids, as mentioned below:

\[
k_{nf} = (1 + 3\varphi) k_{w} \quad \text{------------------------------------------ (25)}
\]

Viscosity of the nanofluid is calculated by the correlation [47]:

\[
\mu_{nf} = \mu_{w} \left[ \frac{1}{(1+\varphi)^{1.7}} \right] \quad \text{------------------------------------------ (26)}
\]

The Nusselt number and friction factor correlation [48] are referred as:

\[
Nu = 0.02172 Re^{0.10} Pr^{0.5} (1 + \varphi)^{0.05181} \quad \text{------------------------------------------ (27)}
\]

\[
f = 0.3491 Re^{-0.25} (1 + \varphi)^{0.1517} \quad \text{------------------------------------------ (28)}
\]

which is valid for: \( 3000 < Re < 22000, 0 < \varphi < 0.6\% \), \( 3.72 < Pr < 6.50 \).

3.3. Inserts

Inserts increase the Reynolds number due to swirling, which in return enhances the heat transfer coefficient. High heat transfer rates are achieved due to the longer helical path taken by the fluid and resulting fluid mixing. It prevents scale or fouling build up in the receiver tube as a side benefit because it is swept off by the turbulence created, and extends time needed for tube cleaning which is also easier to clean by extracting the turbulator manually.

One of the best types of twisted tape is considered in the model which was studied by Murugesan et al. [49] and the correlations to predict Nusselt number and friction factor are as follows:

\[
Nu = 0.0207 Re^{0.862} Pr^{0.33} X^{-0.215} \quad \text{------------------------------------------ (29)}
\]

\[
f = 2.642 Re^{-0.474} X^{-0.302} \quad \text{------------------------------------------ (30)}
\]

valid in the range of \( 2000 \leq Re \leq 12000 \) and \( 2 \leq X \leq 6 \).

3.4. Combination of nanofluid and insert

Sundar et al. [50] have investigated the effect of combining both heat transfer augmentation techniques in their work and provide empirical equation for friction factor and Nusselt number which includes different volume concentrations of Fe\(_{3}\)O\(_{4}\) as well as different twist ratios of inserts.

\[
Nu = 0.0223 Re^{0.88} Pr^{0.55} (1 + \varphi)^{0.03} (1 + X)^{0.028} \quad \text{------------------------------------------ (31)}
\]

\[
f = 0.3490 Re^{-0.25} (1 + \varphi)^{0.21} (1 + X)^{0.017} \quad \text{------------------------------------------ (32)}
\]
Figure 2. Flowchart for the base model
The influence of different heat transfer enhancement techniques on the performance of PTC have been studied analytically by suitable modifying the base model accordingly. The value of the overall heat loss coefficient changes due to evacuation of the annular space in the receiver (evacuated receiver), rise in effective thermal conductivity of HTF (nanofluid) and generation of swirl (twisted tape) and, combined effect of thermal conductivity and swirl motion (combination of nanofluid and twisted tape).

Based on cost and effectiveness, Fe3O4/ H2O nanofluid is taken for the study. As the correlations used (Equation 31 & 32) has the limitation on nanofluid volume concentration, the analytical study is restricted to φ=0.6%.

4. Experimental setup

Setup consists of a PTC with single axis tracking mechanism (supplied by ECOSENSE™, New Delhi, India) which is shown in the Figure 3. Tracking device works on timer algorithm and sensor based algorithm which rotates the complete structure from east to west direction. Receiver tube made of stainless steel is used to circulate water and through copper receiver tube thermic oil is circulated as shown in Figure 4.

Two separate turbine type flow meters are incorporated in the circuit to measure the fluid flow rate. The parabolic reflector has a width of 1.68 m and length of 1.25 m, which lead to aperture area of 2.1 m². The reflector is made of stainless steel with mirror film with reflectivity of 0.8. The emissivity of both receiver surface is 0.66, and the absorptivity of the copper tube and stainless-steel tube are 0.5 and 0.9 respectively. Both tubes have internal diameter of 23 mm and outer diameter of 25.4 mm. The fluid is recirculated using separate water and oil pumps. Six thermocouples are used to detect fluid inlet and outlet temperature in the receiver, receiver temperature, ambient, water inlet temperature to the storage tank and water temperature in the storage tank. Additional instruments like digital sunshine indicator, anemometer and protractor are also used in the experiment. The detailed specification of different instruments used is provided in the annexure.

Before starting the experiment, storage tank and the complete fluid flow line is filled with water/thermic oil. One of the receiver is adjusted to focal line and then proper orientation of concentrator is set using tracker actuating switches. After selecting the water/oil mode, fluid pump is switched on and flow rate is adjusted accordingly. In this condition, the concentrator is continuously tracking the sun and various readings viz. temperatures, flow rate, solar radiation, wind speed is noted at regular intervals. Similar procedure is followed in case of other heat transfer fluid (HTF). Experiments were repeated five times.
times to validate the basic mathematical model. Water was used in three trials (in summer and winter). Oil based experiments are done twice during winter.

5. Results and discussion

The experiments are conducted using both water and thermic oil as HTF. The operating conditions for water are: HTF inlet temperature (29°C to 38°C), ambient temperature (28°C to 32°C), solar radiation level (600 to 950 W/m²) and HTF flow rate (2.5 to 20 LPM). The corresponding ranges measured in case of thermic oil as HTF are: 37°C to 48°C, 35°C to 39°C, 850 W/m² to 1050 W/m² and 2 LPM to 4 LPM. By incorporating HTF inlet temperature to receiver, ambient temperature, wind speed and solar radiation values, energy parameters are calculated analytically using MATLAB code and are in good agreement. In case of water, a deviation of 5.5% and 4.5% in exit temperature is noticed in comparison with experimental values for two trials as shown in Figure 5. Further, as depicted in Figure 6, thermic oil on the other hand have shown a deviation of only 3%.

From calculated receiver exit temperature in case of water and thermic oil as HTF, it is concluded that the mathematical model is showing excellent agreement with experimental values. Hence, the model is modified to consider the effect of nanofluid, turbulator and evacuated glass envelope.

5.1. Evacuated glass envelope

The earlier discussed algorithm is implemented and compared with the basic model which results in an increase of 4.4-5.3% useful heat gain over the range of solar radiation as shown in Figure 7. It is understood that more heat is trapped inside due to the vacuum, eliminating the heat loss to surroundings. This enhancement in useful heat gain will increase the efficiency of the PTC.

5.2. Nanofluid

Increasing the volume concentration of Fe₃O₄ in water increases the heat transfer, which is represented by the Nusselt number as shown in Figure 8. At highest Reynolds number, the Nusselt number is 38.4 for the base model and a maximum enhancement of 56% is achieved at 0.6% concentration.
5.3. **Inserts**
The insertion of twisted tape in the flow line enhances heat transfer. Decreasing the twist-pitch to tube diameter ratio lead to increase in heat transfer. As shown in Figure 9, 59\% enhancement in the Nusselt number at twist ratio of 2 compared to the base model is obtained when evaluated at highest Reynolds number.

5.4. **Combination of Nanofluid and Twisted tape**
The best nanofluid volume concentration of φ=0.6\% and twist ratio of X=2 for the typical twisted tape are implemented in the basic model.

Figure 10 shows a comprehensive comparison between all three heat transfer augmentation techniques. When nanofluid and the twisted tape insert are used individually, better performance is observed with an enhancement range of 56-75\% and 59-73\% in Nusselt number respectively. On the other hand, the combination of twisted tape and nanofluid shows an enhancement of Nusselt number in the range of 63-87\% as compared to the base model.

Effective thermal efficiency is an important parameter to consider for comparison in the case of heat augmentation techniques since the addition of twist tape inserts obstructs the flow of the fluid which results in more pumping power. So, it’s not always the heat enhancement to look at but also the effective thermal efficiency. It is concluded that all the three heat augmentation techniques are better than the base model. It is also noticed that the order of the highest effective thermal efficiency enhancement is not the same order as the heat transfer enhancement shown in Figure 9.

The abrupt drop in effective thermal efficiency is observed in all the cases at Re ≅ 4250 as seen in Figure 10. This is due to the domination of pumping power over useful heat gain at defined narrow range of Re. Based on effective thermal efficiency, for the given range of Re, use of twisted tape with water as HTF is justified.

6. **Conclusion**
From both analytical and experimental analysis on performance of PTC, the following conclusions are drawn:

i. The use evacuated glass envelop for the existing bare receiver would increase the useful heat gain up to 5.3\% due to the reduced convection and radiation losses.

ii. Use of nanofluid (φ = 0.6\%) showed a maximum enhancement of 56\% in Nusselt number at highest Reynolds number resulting in better heat transfer characteristics.

iii. An improvement of 59\% in Nusselt number is achieved by using twisted tape inserts (X = 2) with water as HTF also at highest Reynolds number.

iv. The combined use of Fe₃O₄ nanofluid (φ = 0.6\%) and twisted tape insert (X = 2) resulted in 63-87\% enhanced heat transfer rate.

v. Since the effective thermal efficiency considers both useful heat gain and pumping power, twisted tape with water as HTF is found to give the highest rise of 1.6\% compared to water in a plain tube.
Figure 5. Comparison of experimental and analytical exit temperature of water

Figure 6. Comparison of experimental and analytical exit temperature of oil

Figure 7. Comparison of useful heat gain with and without vacuum glass tube

Figure 8. Effect of twist ratio and concentration of nanoparticles on Nusselt number

Figure 9. Variation of Nu with Re for different heat transfer augmentation techniques

Figure 10. Variation of $\eta_{	ext{heat}}$ with Re for different heat transfer augmentation techniques
Nomenclature:

\( A \)  Area, \([m^2]\)
\( C \)  Concentration Ratio, \([-\]
\( C_p \)  Specific heat, \([J/kgK]\)
\( D \)  Diameter, \([m]\)
\( f \)  Friction Factor, \([-\]
\( F_R \)  Heat removal factor, \([-\]
\( F' \)  Collector efficiency factor, \([-\]
\( h \)  Heat transfer coefficient, \([W/m^2K]\)
\( I \)  Irradiance, \([W/m^2]\)
\( k \)  Thermal conductivity, \([W/mK]\)
\( l \)  Length, \([m]\)
\( L \)  Absorber Length, \([m]\)
\( \dot{m} \)  Mass flow rate, \([kg/s]\)
\( Nu \)  Nusselt number, \([-\]
\( P_p \)  Pump power, \([W]\)
\( Pr \)  Prandlt’s number, \([-\]
\( Q \)  Heat transfer rate, \([W]\)
\( Re \)  Reynolds number, \([-\]
\( \tau_b \)  Tilt factor, \([-\]
\( S \)  Absorbed flux, \([W/m^2]\)
\( T \)  Temperature, \([K]\)
\( U \)  Overall coefficient, \([W/m^2 K]\)
\( W \)  Width, \([m]\)
\( X \)  Twist Ratio, \([-\]

Greek Symbols

\( \alpha \)  Absorptivity, \([-\]
\( \gamma \)  Intercept factor, \([-\]
\( \rho \)  Reflectivity, \([-\]/\) Density, \([kg/m^3]\)
\( \sigma \)  Stefan Boltzmann constant, \([W/m^2 K^4]\)
\( \eta \)  Efficiency, \([\%]\)
\( \varepsilon \)  Emissivity, \([-\]
\( \theta \)  Incident angle, \([degree]\)
\( \phi \)  Latitude, \([degree]\)
\( \delta \)  Declination, \([degree]\)
\( \omega \)  Hour angle, \([degree]\)
\( \mu \)  Dynamic viscosity, \([N.s/m^2]\)
\( \nu \)  Kinematic viscosity, \([m^2/s]\)
\( \varphi \)  Volume concentration, \([\%]\)

Subscripts

\( a \)  Ambient
\( bulk \)  Bulk
\( c \)  Cover
Electrical conversion
Effective thermal
Inner, Inlet
Inner glass
Loss
Nano fluid
Nano particle
Outer, overall
Outer glass
Optical
Receiver
Receiver convection
Receiver radiation
Sky
Useful
Vacuum receiver
Wind, Water

7. References
1] Sukhatme, K., and Sukhatme, S. P., 1996. Solar energy: principles of thermal collection and storage. Tata McGraw-Hill Education.
2] Tijani, A. S., and Roslan, A. M. B., 2014. Simulation analysis of thermal losses of parabolic trough solar collector in Malaysia using computational fluid dynamics. Procedia Technology, 15, pp. 842-849.
3] Fuqiang, W., Jianyu, T., Lanxin, M., and Chengchao, W., 2015. Effects of glass cover on heat flux distribution for tube receiver with parabolic trough collector system. Energy Conversion and Management, 90, pp. 47-52.
4] Liang, H., You, S., and Zhang, H., 2015. Comparison of different heat transfer models for parabolic trough solar collectors. Applied Energy, 148, pp. 105-114.
5] Manglik, R. M., and Bergles, A. E., 1993. Heat transfer and pressure drop correlations for twisted-tape inserts in isothermal tubes: part I-laminar flows. Journal of heat transfer, 115(4), 881-889.
6] Manglik, R. M., and Bergles, A. E., 1993. Heat transfer and pressure drop correlations for twisted-tape inserts in isothermal tubes: part II- transition and turbulent flows. Journal of Heat Transfer, 115(4), 890-896.
7] Lu, J., Ding, J., Yang, J., and Yang, X., 2013. Nonuniform heat transfer model and performance of parabolic trough solar receiver. Energy, 59, pp. 666-675.
8] Montes, M. J., Abánades, A., and Martínez-Val, J. M., 2010. Thermofluidodynamic model and comparative analysis of parabolic trough collectors using oil, water/steam, or molten salt as heat transfer fluids. Journal of Solar Energy Engineering, 132(2), 021001.
9] Yaghoubi, M., Ahmadi, F., and Bandehee, M., 2013. Analysis of heat losses of absorber tubes of parabolic trough collector of Shiraz (Iran) solar power plant. Journal of Clean Energy Technologies, 1(1), pp. 33-37.
10] Ouagued, M., Khellaf, A., and Loukarfi, L., 2013. Estimation of the temperature, heat gain and heat loss by solar parabolic trough collector under Algerian climate using different thermal oils. Energy Conversion and Management, 75, pp. 191-201.
11] Wirz, M., Roesle, M., and Steinfeld, A., 2012. Three-dimensional optical and thermal numerical model of solar tubular receivers in parabolic trough concentrators. Journal of Solar Energy Engineering, 134(4), 041012.
12] Roesle, M., Coskun, V., and Steinfeld, A., 2011. Numerical analysis of heat loss from a parabolic trough absorber tube with active vacuum system. Journal of Solar Energy Engineering, 133(3), 031015.
13] Wang, Y., Liu, Q., Lei, J., and Jin, H., 2015. Performance analysis of a parabolic trough solar collector with non-uniform solar flux conditions. International Journal of Heat and Mass Transfer, 82, pp. 236-249.
14] Jin, H., Sui, J., Hong, H., Wang, Z., Zheng, D., and Hou, Z., 2007. Prototype of middle-temperature solar receiver/reactor with parabolic trough concentrator. Journal of Solar Energy Engineering, 129(4), pp. 378-381.
15] Manzolini, G., Giostri, A., Saccilotto, C., Silva, P., and Macchi, E., 2012. A numerical model for off-design performance prediction of parabolic trough based solar power plants. Journal of Solar Energy Engineering, 134(1), 011003.
16] Bellos, E., Tzivanidis, C. and Tsimpoukis, D., 2017. Thermal enhancement of parabolic trough collector with internally finned absorbers. Solar Energy, 157, pp.514-531.
17] Gong, X., Wang, F., Wang, H., Tan, J., Lai, Q. and Han, H., 2017. Heat transfer enhancement analysis of tube receiver for parabolic trough solar collector with pin fin arrays inserting. Solar Energy, 144, pp.185-202.
18] Fuqiang, W., Qingzhi, L., Huaizhi, H. and Jianyu, T., 2016. Parabolic trough receiver with corrugated tube for improving heat transfer and thermal deformation characteristics. Applied Energy, 164, pp.411-424.
19] Kalidasan, B., Shankar, R. and Srinivas, T., 2016. Absorber Tube with Internal Hinged Blades for Solar Parabolic Trough Collector. Energy Procedia, 90, pp.463-469.
20] Bellos, E., Tzivanidis, C. and Tsimpoukis, D., 2017. Multi-criteria evaluation of parabolic trough collector with internally finned absorbers. Applied Energy, 205, pp.540-561.
21] Song, X., Dong, G., Gao, F., Diao, X., Zheng, L. and Zhou, F., 2014. A numerical study of parabolic trough receiver with nonuniform heat flux and helical screw-tape inserts. Energy, 77, pp.771-782.
22] Reddy, K.S., Kumar, K.R. and Ajay, C.S., 2015. Experimental investigation of porous disc enhanced receiver for solar parabolic trough collector. Renewable Energy, 77, pp.308-319.
23] Jamal-Abad, M.T., Saedodin, S. and Aminy, M., 2017. Experimental investigation on a solar parabolic trough collector for absorber tube filled with porous media. Renewable Energy, 107, pp.156-163.
24] Jaramillo, O.A., Borunda, M., Velazquez-Lucho, K.M. and Robles, M., 2016. Parabolic trough solar collector for low enthalpy processes: An analysis of the efficiency enhancement by using twisted tape inserts. Renewable Energy, 93, pp.125-141.
25] Bellos, E., Tzivanidis, C., Antonopoulos, K.A. and Gkinis, G., 2016. Thermal enhancement of solar parabolic trough collectors by using nanofluids and converging-diverging absorber tube. Renewable Energy, 94, pp.213-222.
26] Mwesigye, A., Bello-Ochende, T. and Meyer, J.P., 2016. Heat transfer and entropy generation in a parabolic trough receiver with wall-detached twisted tape inserts. International Journal of Thermal Sciences, 99, pp.238-257.
27] Ghasemi, S.E. and Ranjbar, A.A., 2017. Numerical thermal study on effect of porous rings on performance of solar parabolic trough collector. Applied Thermal Engineering, 118, pp.807-816.
28] Kasaeian, A., Daviran, S., Azarian, R. D., and Rashidi, A., 2015. Performance evaluation and nanofluid using capability study of a solar parabolic trough collector. Energy Conversion and Management, 89, pp. 368-375.
29] De Risi, A., Milanese, M., and Laforgia, D., 2013. Modelling and optimization of transparent parabolic trough collector based on gas-phase nanofluids. Renewable Energy, 58, 134-139.
30] Ghasemi, S. E., and Ahangar, G. R. M., 2014. Numerical analysis of performance of solar parabolic trough collector with Cu-Water nanofluid. International Journal of Nano Dimension, 5(3), pp. 233-240.
31] Sokhansefat, T., Kasaeian, A. B., and Kowsary, F., 2014. Heat transfer enhancement in parabolic trough collector tube using Al 2 O 3/synthetic oil nanofluid. Renewable and Sustainable Energy Reviews, 33, pp. 636-644.
32] Wen, D., and Ding, Y., 2004. Experimental investigation into convective heat transfer of nanofluids at the entrance region under laminar flow conditions. International journal of heat and mass transfer, 47(24), pp. 5181-5188.
33] Mwesigye, A., Bello-Ochende, T., and Meyer, J. P., 2013, November. Heat Transfer Enhancement in a Parabolic Trough Receiver Using Wall Detached Twisted Tape Inserts: In ASME 2013 International Mechanical Engineering Congress and Exposition (pp. V06BT07A031-V06BT07A031). American Society of Mechanical Engineers.
34] Cheng, Z. D., He, Y. L., and Cui, F. Q., 2012. Numerical study of heat transfer enhancement by unilateral longitudinal vortex generators inside parabolic trough solar receivers. International Journal of Heat and Mass Transfer, 55(21), pp. 5631-5641.
35] Huang, Z., Yu, G. L., Li, Z. Y., and Tao, W. Q., 2015. Numerical Study on Heat Transfer Enhancement in a Receiver Tube of Parabolic Trough Solar Collector with Dimples, Protrusions and Helical Fins. Energy Procedia, 69, pp. 373-380.
36] Azmi, W. H., Sharma, K. V., Sarma, P. K., Mamat, R., and Anuar, S., 2014. Comparison of convective heat transfer coefficient and friction factor of TiO2 nanofluid flow in a tube with twisted tape inserts. International Journal of Thermal Sciences, 81, 84-93.
37] Ghadirjafarbeigloo, S., Zamzamian, A. H., and Yaghoubi, M., 2014. 3-d numerical simulation of heat transfer and turbulent flow in a receiver tube of solar parabolic trough concentrator with louvered twisted-tape inserts. Energy Procedia, 49, pp. 373-380.
38] Churchill, S. W., and Bernstein, M., 1977. A correlating equation for forced convection from gases and liquids to a circular cylinder in crossflow. ASME, Transactions, Series C-Journal of Heat Transfer, 99, pp. 300-306.
39] Forristall, R., 2003. Heat transfer analysis and modelling of a parabolic trough solar receiver implemented in engineering equation solver (pp. 1-145). National Renewable Energy Laboratory.

40] Gnielinski, V., 1976. New equations for heat and mass-transfer in turbulent pipe and channel flow. International chemical engineering, 16(2), pp. 359-368.

41] Kothandaraman, C. P., 2004. Heat and mass transfer data book. New Age International.

42] Kibaara, S., Chowdhury, S., and Chowdhury, S. P., 2012, May. A thermal analysis of parabolic trough CSP and biomass hybrid power system: In Transmission and Distribution Conference and Exposition (TandD), 2012 IEEE PES (pp. 1-6). IEEE.

43] Kumar P., 2012. Effect of Differential Mass Flow Rate on the Thermal Performance of Double Duct Packed Bed Solar Air Heaters. International Conference on Renewable Energies, Environment and Power Quality.

44] Dudley, V., Kolb, G., Sloan, M., and Kearney, D., 1994. SEGS LS2 solar collector—Test results. Report of Sandia National Laboratories, Report No. SANDIA94-1884.

45] Pak, B. B., and Cho, Y. I., 1998. Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles. Expt. Heat Transfer, pp. 151-170.

46] Timofeeva, E. V., Gavrilov, A. N., McCloskey, J. M., Tolmachev, Y. V., Sprunt, S., Lopatina, L. M., and Selinger, J. V., 2007. Thermal conductivity and particle agglomeration in alumina nanofluids: experiment and theory. Physical Review E, 76(6), 061203.

47] Brinkman, H. C., 1952. The viscosity of concentrated suspensions and solutions. The Journal of Chemical Physics, 20(4), pp. 571-571.

48] Sundar, L. S., Naik, M. T., Sharma, K. V., Singh, M. K., and Reddy, T. C. S., 2012. Experimental investigation of forced convection heat transfer and friction factor in a tube with Fe₃O₄ magnetic nanofluid. Experimental Thermal and Fluid Science, 37, pp. 65-71.

49] Murugesan, P., Mayilsamy, K., and Suresh, S., 2011. Heat transfer and friction factor in a tube equipped with U-cut twisted tape insert. Jordan J. Mech. and Indust. Engng, 5, pp. 559-565.

50] Sundar, L. S., Kumar, N. R., Naik, M. T., and Sharma, K. V., 2012. Effect of full length twisted tape inserts on heat transfer and friction factor enhancement with Fe₃O₄ magnetic nanofluid inside a plain tube: an experimental study. International Journal of Heat and Mass Transfer, 55(11), pp. 2761-2768.
Annexure
Specification of the instruments used in the experiment

(a) Hall Effect type flow meter

| Make/Model | Nuclus/P812 |
|------------|-------------|
| Display    | 16 × 2 lines LCD |
| Supply voltage | 230 V AC |
| Linearity  | ± 1% of full range |
| Repeatability | ± 0.5% of full range |
| Output     | 230 V AC; 4 to 20 mA/2 to 10 V DC |

(b) Thermocouple

| Type                  | Chromel-Alumel (K-type) |
|-----------------------|--------------------------|
| Operating temperature range | 0 – 200°C |
| Accuracy              | ± 2.2°C of 0.75% |
| Insulation            | Teflon |

(c) Anemometer

| Make/Model             | Work Zone/AVM-03 |
|------------------------|-------------------|
| Measurement range for wind velocity | 0 to 45 m/s (± 3%) |
| Measurement range for temperature | 0°C to 45°C (± 2°C) |

(d) Digital sunshine indicator (Make: Kaizen Imperial)

| Sensor (Pyranometer) |
|-----------------------|
| Radiation range       | 0 – 2000 W/m² |
| Absolute accuracy     | ± 5% |
| Repeatability         | ± 1% |
| Sensor output         | 0.200 mV/Wm² |
| Sensitivity           | Custom calibrated to exactly 5.00 Wm⁻²/mV |
| Operating environment | -40°C to 55°C; 0 to 100% Relative humidity |

| Data logging unit |
|-------------------|
| Display           | 16 characters × 2-line LCD |
| Measured parameter| Date, Time, Solar radiation intensity |
| Number of channels | 8 |
| Power supply      | 12 V SMF |
| Data storage      | 512 K EEPROM |
| Battery charging  | Through solar PV panel |
| Data logging interval | User programmable from 1 minute to 24 hours |
| Operating environment | -40°C to 75°C; 0 to 95% non-condensing (humidity) |
| Data retrieval    | Via Data shuttle to computer |

| Solar PV panel |
|----------------|
| Output voltage | 12 V DC |
| Wattage        | 10 W |