Numerical Modelling of Solar Parabolic Trough Receiver Employed for Thermal Energy Storage System

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***Abstract*—Solar energy is noted for its high reliability than other systems and allows more energy generation than other renewable resources. Solar Parabolic Trough Collector (PTC) is the most proven industry-scale solar generation technology available today. The thermal performance of such devices is of major interest for optimizing the solar field output and increase the efficiency of solar thermal systems. The present work investigates the performance of parabolic trough collector employed for thermal energy storage system. The PTC is integrated with Thermal Energy Storage (TES) system and consists of concentrator made of aluminium sheets of 7.5 m^2 and receiver tube of 3m length made of chrome black plated stainless steel and is enclosed by a glass envelope. One dimensional numerical model has been developed and examined to study heat transfer and thermal losses in the receiver. The receiver and glass envelope are divided into small segments. Mass and energy balance are applied to each segment to obtain Partial Differential Equations (PDE). These equations are discretized for transient state and solved numerically for real time solar intensity values. The numerical model is validated by comparing the results with experimental data and the agreement is observed to be good. Experimentation has been carried out in PTC for different mass flow rates. Variations of thermal efficiency, thermal losses and HTF outlet temperatures in PTC and the temperature distribution and cumulative heat energy stored in TES system with respect to charging time for various solar irradiance and mass flow rates have been studied.**

***Index Terms*—Solar energy, parabolic trough collector, numerical model, thermal energy storage.**

I. Introduction

Global warming or climate change has become pressing issue for years, due to the rapidly increasing energy demand and growing environmental degradation of burning fossil fuel. The renewable energy is the promising technology for the enhancement of thermal system performance. Solar energy is the oldest of all technologies. Solar energy is highly reliable and allows more energy generation rate than other renewable resources. Solar collectors are special heat exchangers that transform solar radiation into internal energy of the transport medium. HTF absorbs the concentrated solar thermal energy and raise its enthalpy while circulating through the heat collecting element (HTC). The energy collected by the HTF is used as source for power plants, space conditioning equipment and others. This heat can be stored in TES unit for further usage of energy during night times. Flat plate collectors are commercially available for collecting solar energy and mostly used for water heating applications (30-80 ℃). Parabolic trough collector (PTC) is preferred mostly for industrial applications (60-400 ℃) [1]. High performance solar collector is required for delivering high temperature fluid with high efficiency. PTC's are employed in a variety of application including industrial steam production for electricity, industrial steam production for process heat application, and large scale community level solar cooking. PTC plays a lead role in Integrated Solar Combined Cycle (ISCC) power plant [2].

Velraj *et al.* [3] suggested that the systems with light structures and low cost technology for process heat applications up to 400 ℃ could be obtained with parabolic trough collectors (PTCs). Mirrored surfaces linear parabolic reflector is used to focus direct solar radiation onto a tubular solar receiver. Dudley *et al.* [4] proposed the receiver outer surface coated with black chrome and cermet, which is exposed to solar radiations, shows better improvement in performance of PTC when compared to the bare tube. Giorgio Cau *et al.* [5] compared the performance of medium-size Concentrating Solar Power (CSP) plants based on an Organic Rankine Cycle (ORC) power generation unit integrated with parabolic trough and linear Fresnel collectors and suggested that the use of parabolic troughs gives better values of energy production per unit area of solar collector. Mazloumi *et al.* [6] reported that the parabolic trough collector mass flow rate has negligible effect on minimum collector area and has significant effect on optimal capacity of storage tank.

The performance of PTC under different operating conditions and parametric changes can be easily analyzed by numerical heat transfer techniques. This heat transfer analysis is used for sizing of the solar power plant, to evaluate the degradation of collector, to analyze the performance of collector for various mass flow rates and to determine the overall plant performance. Heat loss is the major factor which influences the system efficiency and occurs due to the receiver tube temperature, ambient temperature, wind velocity and emissivity of the coating. Heat loss due to convection, occurs between the glass envelope and the receiver, is fully responsible for the reduction in plant performance. Padilla *et al.* [7] carried out a non-transient mathematical model to calculate the outlet temperature of the fluid and also noted the improvement in performance of PTC considering the radiative heat losses. In the above proposed model, the partial differential equations were obtained by application of energy balance equations to the receiver model and were discretized to obtain the outlet temperature of the fluid. Odeh *et al.* [8] developed a mathematical model to determine the heat loss occurring in the direct steam generation (DSG) system. They also

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developed a transient numerical model to examine the performance of industrial water heating system using PTC [8]. Forristall [9] developed a transient, one dimensional heat transfer model for short and long receivers to evaluate the performance of PTC. In the present work, one dimensional transient state numerical model is developed for the receiver section in this article to analyze the performance of solar PTC. A simple algorithm is developed for solving the energy conservation equations to investigate the performance parameters.

II. Materials and Methods

The parabolic trough receiver consists of 3m long receiver tube covered with glass envelope with supports at the ends connected to the concentrator and tracker device. Polished aluminium sheets are used as concentrators with an aperture area of 7.5 m^2, since the temperature requirement in this case is about 60-90 ℃ [10]. The receiver tube is made of chrome black coated stainless steel. The tracking device consists of single phase induction motor, gear box and micro controller operated control box. The timer control having on/off at a calculated rate specified with a fixed interval, helps to ensure that the collector remains pointed towards sun and moves at a speed of rotation of 5°/20 min The parabola orientation is North-South direction and tracks the sun from East-West. Water is used as the heat transfer fluid (HTF), which also acts as a sensible heat storage material, to transfer heat from the PTC to the storage tank. Water is a good heat transfer fluid for low temperature applications (< 90 ℃) such as domestic heating, hospital sterilization purposes etc. The technical specification of PTC is shown in the Table I.

The cylindrical TES tank is made of stainless steel and has a capacity of 60 litres with a diameter of 370 mm and height of 530 mm. There are two plenum chambers on the top and the bottom of the tank and a flow distributor is provided on the top of the tank to make the HTF flow uniform. The storage tank and the whole circulating pathways are insulated with asbestos material of 8 mm thick and provided with aluminium cladding. Spherical capsules made of high-density polyethylene (HDPE) are used as PCM containers as spherical shape gives the best performance among the various types of containers. The outer diameter of spherical capsule is 60 mm and its wall thickness is 0.8 mm. The total number of capsules in the TES tank is 216. The spherical capsules are uniformly packed in eight rows and a wire mesh supports each row. In the analysis two rows of spherical capsules are considered as one segment. The PCM capsules occupy 67% of the total volume (test section) of storage tank and the remaining volume is occupied by SHS material i.e. water. The present work involves the use of industrial grade granulated paraffin with a melting point range of 60-62 ℃.

Experimentation was carried out in PTC for different fluid flow rates (0.033, 0.066 and 0.1 kg s^{-1}). The flow rates are measured using rota-meter, and varied using mechanical valves. The input solar radiation in this location (12.7525 °N, 80.1961 °E) varies from 500 – 1000 W m^{-2} a day during the month of April and May. These solar radiations are measured using pyranometer.

TABLE I: Technical Specification of PTC

Parameters	Specifications
Aperture area	7.5 m^2
Rim angle	70^0
Focal length of the PTC	0.9 m
Aperture width	2.5 m
Aperture Length	3 m
Concentration ratio	13
Receiver tube	D_o = 0.06 m, D_i = 0.052 m
Glass envelope	D_o = 0.085 m, D_i = 0.080 m
Absorber tube material	Stainless steel coated with chrome black material
HTF	Water

The inlet and outlet temperatures of fluid are measured using resistance temperature detectors (RTD). The experimental duration are 3-4 hours in a day (which is sufficient for storing heat in TES unit) with a particular fluid flow rate. The temperature and solar radiation are measured for every 20 minutes. The PTC experimental setup with TES is shown in the Fig. 1.

Fig. 1. Experimental setup.

III. Numerical Modelling and Heat Transfer Analysis of PTC

In this work one dimensional transient/unsteady state heat transfer analysis is carried out to evaluate the performance of solar PTC numerically for various mass flow rates and real time solar intensity values. The mathematical model of the HCE constitutes of receiver tube and glass envelope and the HTF flows through the receiver tube continuously.

A. Numerical Model of Solar Receiver

The numerical model of receiver is based on the energy balance between the HTF, receiver, glass envelope and surrounding. The cross sectional view of the solar receiver model is shown in the Fig. 2. The energy balance considers the normal solar radiation falling on the collector, thermal losses and the heat gained by the HTF. The partial differential equations are obtained by applying the energy balance for each section of the receiver and each section is considered as the control volume. Proper tracking, specular reflection of concentrator, constant mass flow rates and negligible conduction losses at the receiver ends are the assumption made in this analysis. The incoming solar radiation is absorbed by the glass envelope ($Q_{g\text{-}abs}$) and the chrome black coated receiver tube ($Q_{r\text{-}abs}$) and optical losses have been considered in the heat flux terms. The heat from the receiver tube is carried by the HTF by forced convection ($Q_{r\text{-}f,\ conv}$). The radiation ($Q_{r\text{-}g,\ rad}$) and convection ($Q_{r\text{-}g,\ conv}$) losses from the receiver tube to the glass envelope are accounted in this numerical model. Similarly the convection ($Q_{g\text{-}s,\ conv}$) and radiation ($Q_{g\text{-}s,\ rad}$) losses for the glass envelope to the atmosphere have also been considered. The conduction heat losses occurring in the brackets are neglected since; the brackets are made of small wood holdings. The temperature, heat flux, thermal properties are uniform throughout the circumference of the receiver model at that instant. The thermal resistance network is given in Fig. 3.

B. Receiver Tube to HTF

Energy balance is applied to the control volume shown in Fig. 3 and the following partial differential equation is obtained considering HTF as incompressible fluid.

$$A_{r,i}\,\rho_f C_{p,f}\frac{\partial T_f}{\partial t} = -\dot{m}_f\frac{\partial}{\partial x}\left(C_{p,f}T_f\right) + \dot{Q}'_{r-f,conv} \qquad (1)$$

where $A_{r,i}, \rho_f, C_{p,f}, \dot{m}_f$ and T_f represent the inner surface area of the receiver section, density, specific heat, mass flow rate and temperature of HTF respectively. The convection heat gained by the HTF is given by

$$\dot{Q}'_{r-f,conv} = \pi Nu_f k_f\left(T_r - T_f\right) \qquad (2)$$

where Nu_f, k_f and T_r represent the Nusselt number, thermal conductivity of the HTF and receiver tube temperature respectively. For laminar flow inside the pipe, Reynolds number (Re_D) will be less than 2300 and the Nusselt number is assumed as constant (Nu = 4.36) [11]. For turbulent flow, Reynolds number is greater than 2300 and the following correlation is used [12].

$$Nu_f = \frac{\left(\frac{C_f}{2}\right)(Re_D - 1000)Pr}{1+12.7\left(\frac{C_f}{2}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)}\left(\frac{Pr}{Pr_w}\right)^{0.11} \qquad (3)$$

$$C_f = (1.58 ln Re_D - 3.28)^{-2} \qquad (4)$$

$$Re_D = \frac{\rho_f v_f D_{r,i}}{\mu} \qquad (5)$$

where C_f, v_f and μ are the friction coefficient of inner surface of the receiver tube of diameter $D_{r,i}$, velocity and dynamic viscosity of the HTF respectively.

C. Receiver to Glass Envelope

This control volume considers the heat transfer by convection and radiation. Convective heat transfer mainly depends on the annular pressure between the receiver tube and glass envelope. If pressure is greater than 1 Torr i.e. loss of vaccum in between glass envelope and receiver tube, free convection is considered, while for pressure less than 1 Torr, forced convection is applicable. In this work, free convection is considered (Pressure > 1 Torr). The radiative heat transfer occurs due to the temperature difference between the outer receiver tube temperature and inner glass envelope temperature.

$$A_r\rho_r C_{p,r}\frac{\partial T_r}{\partial t} = A_r\frac{\partial}{\partial z}\left(k_r\frac{\partial T_r}{\partial x}\right) + \dot{Q}'_{r-abs} - \dot{Q}'_{r-f,conv} - \dot{Q}'_{r-g,conv} - \dot{Q}'_{r-g,rad} \qquad (5)$$

The subscript r and g correspond to receiver tube and glass envelope respectively. o and i are the outer and inner geometrical representations in this numerical model. Convective heat loss from the receiver tube to the glass envelope is given below. Raithby and Holland's correlation for natural convection in an annular space between horizontal cylinders is used for this case [9].

$$\dot{Q}'_{r-g,conv} = \frac{2.425k_{air}(T_r - T_g)\left(Pr_r Ra_{D_{r,o}}/(0.861+Pr_{air})\right)^{0.25}}{\left(1+\left(\frac{D_{r,o}}{D_{g,i}}\right)^{3/5}\right)^{5/4}} \qquad (6)$$

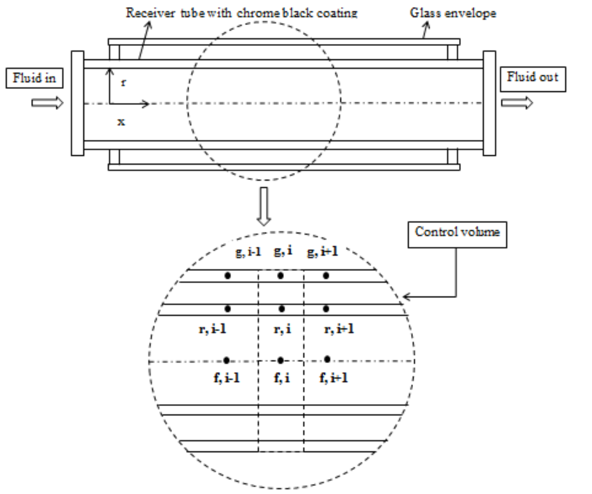

Fig. 2. Cross section view and control volume of HCE.

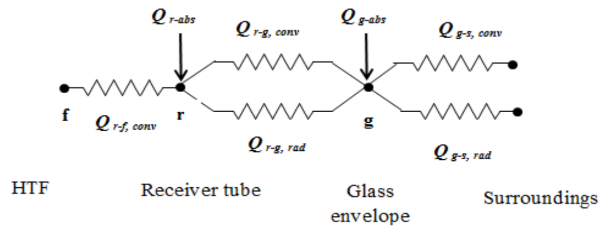

Fig. 3. Network of thermal resistance.

$$Ra_{D_{r,o}} = \frac{g\beta(T_r - T_g)D_{r,o}{}^3}{\alpha\gamma} \qquad (7)$$

$$\beta = \frac{2}{(T_r - T_g)} \qquad (8)$$

$Ra_{D_{r,o}}$, g, β , α and γ denote the Rayleigh number corresponds to the exterior of receiver tube, acceleration due to gravity, volumetric thermal expansion coefficient, thermal diffusivity of solid and kinematic viscosity of the fluid present inside the annulus and these properties are all correspond to the receiver and glass envelope temperature (T_g). The radiative heat transfer between the receiver tube and glass envelope is given by Forristall *et al.* [9]. The radiation correlations are based on assumptions of long concentric cylinders, diffuse reflection and irradiation, grey surfaces, nonparticipating gas inside the annulus. This contributes minor errors but they are less significant or relatively very small.

$$\dot{Q}'_{r-g,rad} = \frac{\sigma\pi D_{r,o}\left(T_r{}^4 - T_g{}^4\right)}{\left(\frac{1}{\varepsilon_r} + \frac{(1-\varepsilon_r)D_{r,o}}{\varepsilon_r D_{g,i}}\right)} \quad (9)$$

where σ and ε_r represent the Stefan-Boltzmann constant and emissivity of receiver tube surface.

D. Glass Envelope to Environment

Convection and radiation are the major means of heat transfer between the glass envelope and atmosphere. The convective heat transfer is either forced or natural depends on the presence of wind in the surroundings. The radiation depends on the temperature difference between glass envelope and surrounding. The energy balance equation of this control volume is given below.

$$A_g\rho_g C_{p,g}\frac{\partial T_g}{\partial t} = A_g\frac{\partial}{\partial x}\left(k_g\frac{\partial T_g}{\partial x}\right) + \dot{Q}'_{g-abs} + \dot{Q}'_{r-g,conv} + \dot{Q}'_{r-g,rad} - \dot{Q}'_{g-sa,rad} - \dot{Q}'_{g-sa,conv} \quad (10)$$

where A_g, ρ_g, $C_{p,g}$ and T_g represent the cross section area, density, specific heat and temperature of glass envelope respectively. The convective heat transfer between glass envelope and atmosphere is given below.

$$\dot{Q}'_{g-sa,conv} = Nu_g k_g \pi\left(T_g - T_a\right) \quad (11)$$

In this case, wind flow is considered, so forced convection mode is involved. Zhukauska's correlation for forced convection normal to an isothermal cylinder is used in the calculation of Nusselt number [9]. The values of constants *C*, *m* and *n* are arrived from the Tables II and III. This correlation is valid only for 0.7 < Pr_{air} < 500, and 1 < $Re_{D_{g,o}}$ < 10^6.

$$Nu_g = C Re_{D_{g,o}} Pr_{air}{}^n\left(\frac{Pr_{air}}{Pr_{air,glass\ envelope}}\right)^{0.25} \quad (12)$$

TABLE II: VALUES OF *C* AND *M* FOR VARIOUS REYNOLDS NUMBER

$Re_{D_{g,o}}$	C	m
1 – 40	0.75	0.4
40-1000	0.51	0.5
1000-200000	0.26	0.6
200000-500000	0.076	0.7

TABLE III: VALUES OF *N* FOR VARIOUS PRANDTL NUMBERS

$Pr \leq 10$	n = 0.37
$Pr > 10$	n = 0.36

The radiative heat transfer of glass envelope to atmosphere is given below, here the glass envelope is considered as grey body placed in a large black body cavity [7].

$$\dot{Q}'_{g-sa,rad} = \sigma D_{g,o}\pi\varepsilon_e\left(T_g{}^4 - T_a{}^4\right) \quad (13)$$

E. Solar Energy Absorption

Energy absorbed by the solar receiver is affected mainly by the optical and geometrical parameters. Concentrator focal errors due to the imperfections in polished aluminium sheets, dust and other miscellaneous errors affect the proper absorption of solar irradiation on the glass envelope. The fraction of solar radiation reflected by the reflector but not received by the glass envelope is given by the intercept factor (υ) [9]. The shadows of HCE, bellows, supports, tracking errors, geometrical errors, mirror reflectivity, dust on the concentrator, and other miscellaneous errors affects this intercept factor which is given below in Table IV. For polished aluminium the mirror reflectivity is less when compared to mirror (ρ_{cl} = 0.732).

$$\upsilon = \prod_{i=1}^{6}\upsilon_i \quad (14)$$

In this work, the incident angle modifier ($K(i)$) and geometrical end losses ($\psi(i)$) are taken as unity and are mainly based on the geometry and optical characteristics of solar collector. The glass envelope transmittance, absorptance and receiver tube absorptance are not affected by the temperature. The optical efficiency (η_o) of the solar PTC is given below.

$$\eta_o = \upsilon\rho_{cl}\tau_e\alpha_r \quad (15)$$

The solar irradiation absorption in the glass envelope is;

$$\dot{Q}'_{g-abs} = \rho_{cl}\upsilon\alpha_g K(i)\psi(i)I_{solar} \quad (16)$$

The solar irradiation absorption in the receiver tube is;

$$\dot{Q}'_{r-abs} = \eta_o K(i)\psi(i)I_{solar} \quad (17)$$

TABLE IV: FACTORS AND OPTICAL PROPERTIES

Factors and optical properties	Value
Errors due to shadows of HCE, supports and bellows υ_1 (Luz chrome black)	0.974
Tracking errors (υ_2)	0.971
Geometrical errors (υ_3)	0.994
Clearness of the reflector material (υ_4)	0.950
Dust on HCE (υ_5)	0.980
Miscellaneous errors (υ_6)	0.960
Glass envelope transmittance (τ_e) – chrome black coated receiver tube.	0.935
Lux black chrome coated receiver absorptance (α_r)	0.940
Glass envelope absorptance (α_g)	0.023
Mirror reflectivity for Al (ρ_{cl})	0.732

IV. NUMERICAL SOLUTION

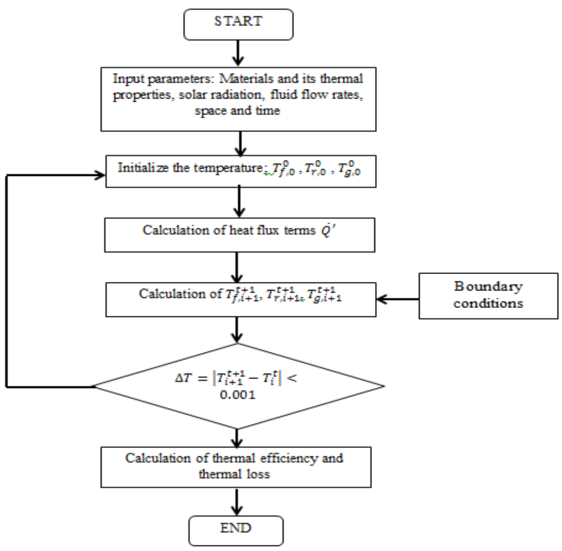

Fig. 4. Flow diagram for numerical simulation.

The partial differential equation obtained by the application of energy balance on the control volume is discretized for transient/un-steady state and solved using MATLAB. The procedures for solving the PDE's are listed below (Fig. 4).

The initial and boundary conditions are applied to the numerical model to solve the energy balance equations and are listed below; and i correspond to the iteration and t represents the time step for solving the energy balance equations.

For receiver to HTF

$$i = 0, t = 0\ ;\ T_{f,0}^{0} = T_{g,0}^{0} \tag{18}$$

For Receiver to glass envelope

$$i = 1, t = 0\ ;\ T_{r,1}^{0} = T_{r,0}^{0} \tag{19}$$

$$i = N, t = 0\ ;\ T_{r,N+1}^{0} = T_{r,N}^{0} \tag{20}$$

For glass envelope to environment

$$i = 1, t = 0\ ;\ T_{g,1}^{0} = T_{g,0}^{0} \tag{21}$$

$$i = N, t = 0\ ;\ T_{g,N+1}^{0} = T_{g,N}^{0} \tag{22}$$

The thermal efficiency (η_{thermal}) of the PTC and the thermal loss are given below;

$$\eta_{\text{thermal}} = \frac{m \cdot C_{pf}\left(T_{f,i+1}^{t+1} - T_{f,i}^{t}\right)}{I_{\text{solar}} A_c} \tag{23}$$

Thermal heat loss=

$$\left(\textstyle\sum_{i}^{N}\left(\dot{Q}'_{r-g,conv} + \dot{Q}'_{r-g,rad}\right)\right)\frac{\Delta x}{I_{solar} A_c} \tag{24}$$

V. Results and Discussion

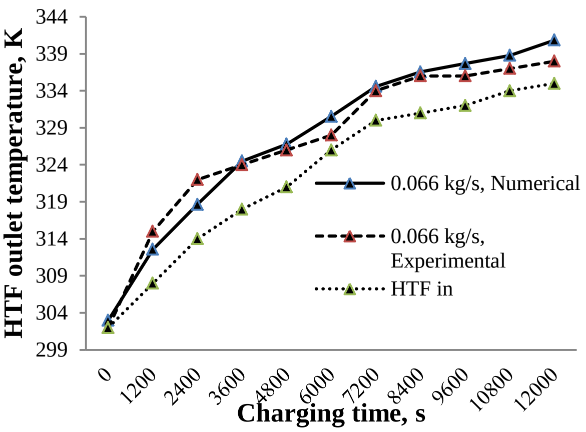

Fig. 5. Variation of HTF temperature with respect to charging time.

Experimental and numerical investigations were carried out for different mass flow rates (0.033, 0.066 and 0.1 kg s^{-1}) for different days in the months of April and May. Solar intensity values and TES unit outlet temperatures were measured and given as input to the numerical model to estimate the HTF outlet temperature, thermal efficiency, thermal loss and instantaneous heat rate and the model is validated by comparing with experimental findings. Fig. 5 shows the variation of HTF outlet temperature with respect to the charging time. Experimental values are compared with that of numerical model and agreement holds good. Plot of HTF inlet temperature in Fig. 5 helps in finding the temperature increased. Fig. 6 represents the variation of instantaneous heat rate for 0.066 kg s^{-1} with respect to charging time. Experimental values are compared with the numerical model and the agreement is good. The deviations are due to varying solar irradiance and the temperature increment/decrement in the collector.

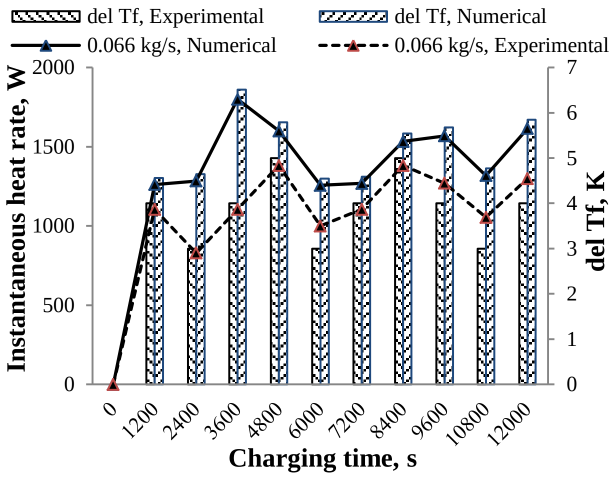

Fig. 6. Variation of instantaneous heat rate with respect to charging time and temperature difference.

Variation of thermal efficiency with respect to charging time for 0.066 kg s^{-1} is shown in Fig. 7. Experimental values are compared with those of numerical model and the deviation could be due to intermittent solar irradiance, recirculation of eddies and geometrical alignment, aberration in concentrator sheet as well as tracking errors in the PTC, but the agreement is acceptable. Variation of thermal efficiency is legitimized by comparing it with the temperature difference in Fig. 8. HTF temperature increment/decrement plays a vital role and this is due to the intermittent solar irradiance.

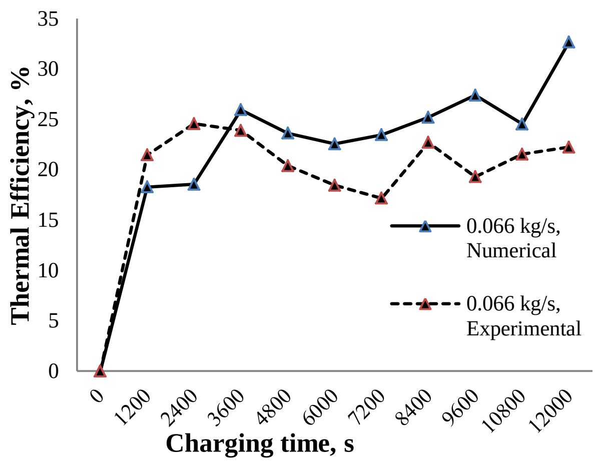

Fig. 7. Variation of thermal efficiencies with respect to charging time.

Fig. 9 depicts variation of thermal loss with respect to HTF outlet temperature. The losses due to convective and radiative heat transfer play a dominant role in the performance of PTC. The collector's thermal efficiency degrades with increase in heat loss and is fully depend on the temperature distribution in the receiver tube, glass envelope and also the heat carried away by the HTF. The presence of air in the annulus between glass envelope and receiver tube also increases the heat loss.

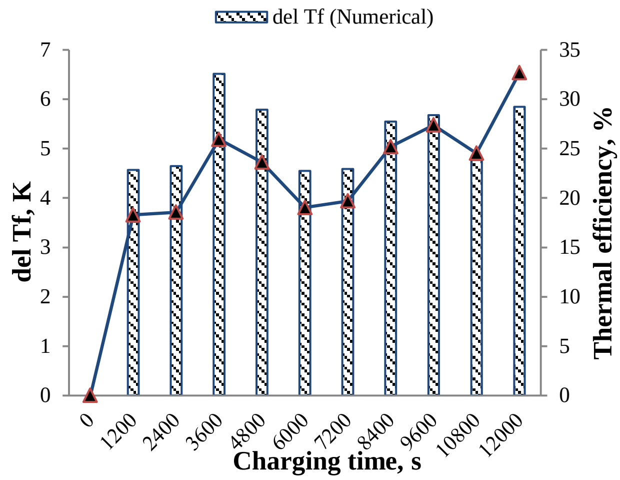

Fig. 8. Comparison of thermal efficiency and temperature difference for 0.066 kg s^{-1} with respect to charging time.

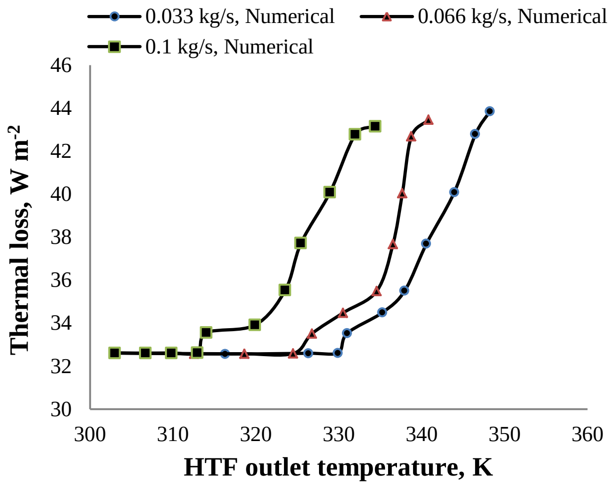

Fig. 9. Comparison of thermal loss for different mass flow rates with respect to charging time.

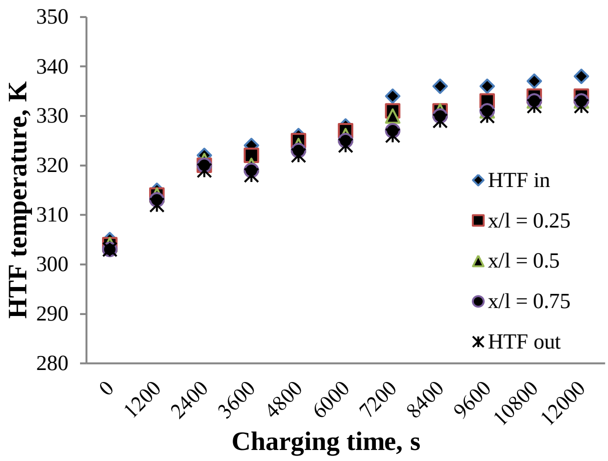

Fig. 10. Axial Temperature distribution in the TES unit with respect to charging time.

Axial temperature distribution of HTF in the TES unit with respect to charging time is shown in the Fig. 10. The axial temperatures inside the tank are measured using resistance temperature detectors spaced in equal intervals. HTF from the receiver transfers its heat to the TES unit at a faster rate initially and as the charging time increases the heat transfer rate decreases due to the decrease in temperature difference between the entry and exit of the TES unit. The cumulative energy stored varies depending on the temperature difference of HTF as well as the heat received by the PCM inside the TES system (Fig. 11). The energy stored increases with increase in time and reaches the maximum of 11500 kJ for the flow rate 0.066 kJ s^{-1} against the designed value of 12000 kJ.

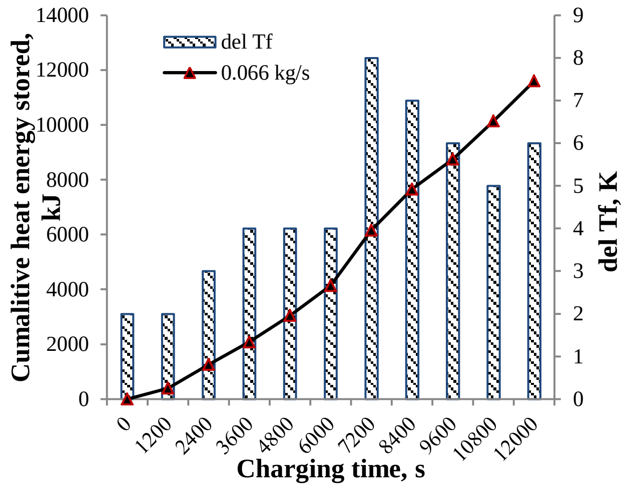

Fig. 11. Variation of cumulative energy stored and HTF temperature difference in TES unit with respect to charging time.

VI. Conclusion

A numerical model based on energy balance across the HCE for the heat transfer analysis of PTC has been developed. The PDE's are solved for one dimensional transient/unsteady state and FDM technique is used to solve the model and the performance parameters of PTC such as heat fluxes, thermal efficiencies and thermal losses are studied for different mass flow rates and real time solar intensity values. The convective and radiative heat transfer occurring in the HCE is also accounted in this work. The numerical model is validated by comparing it with the experimental data and the agreement is good. These close correlations illustrate the relevance in the good nature of the model. The deviations could be due to varying solar intensity values and geometrical and tracking errors in the PTC. Based on these studies, this numerical model is found to be suitable for predicting the thermal performances of HCE under varying operating conditions.

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