2E (Energy and Exergy) Analysis of ET-CPC Solar Collector Integrated with Different Configuration of Thermal Storage System

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Research Article

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2E (energy and exergy) analysis of ET-CPC solar collector integrated with different configuration of thermal storage system

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Abstract

The intermittency of solar thermal energy warrants the integration/utilization of thermal energy storage system for efficient operation. Effective utilization of solar water heating (SWH) system can reduce nearly 70 - 90 % of the energy cost incurred for water heating applications. In this study, a compound parabolic concentrator (CPC) solar collector is paired with thermal energy storage (TES) system for the improvement of thermal performance of the collector through enhanced heat transfer rate and minimizing the heat losses. Effects of varying mass flow rate and different arrangement of phase change materials (PCMs) on the performance of the CPC solar collector are investigated. A study of the influence of PCMs configurations in TES systems viz three PCMs (Case 1) and five PCMs (Case 2) on the energy efficiency, exergy efficiency and overall loss coefficient of the solar collector and TES system is made and compared with sensible TES system. The results show the attainment of maximum thermal efficiency of 70 % for 'Case 2'. Comparison with 'Case 1', 'Case 2' exhibited a reduction heat loss of 4 % from the TES system. Results of exergy study reveal a superior performance in Case 2 over other configurations.

Keyword: compound parabolic concentrator, thermal energy storage, phase change materials, exergy efficiency and overall loss coefficient of the solar collector

1. Introduction

Availability of enormous solar energy is perceived as the most effective source of clean energy than other resources (Panwar et al., 2011). Mitigation of global warming in a sustainable manner can be achieved through recent developments seen in solar thermal technology. Dependence of solar harnessing mainly on a kernel solar collector that transfers the solar radiation into heat energy is well known (Son et al., 2014). Among the available solar collectors, a flat plate collector and the evacuated tube are being used for several applications due to ease in installation in buildings along with relatively low operating and maintenance costs. Nevertheless, a large collector area and low optical efficiency of the glass covers, abundant loss of useful heat gain through top, side and bottom of the collector are the major drawbacks. A non-tracking concentrated solar collector consisting of evacuated tubes and parabolic reflectors referred to as compound parabolic concentrator (CPC) collector has been proposed for overcoming the above said major drawbacks and for effective utilization of solar energy. This CPC collector renders the benefits of a selective surface coating with low emissivity, enhanced heat transfer rate with the presence of vacuum in the evacuated tube and additional heat energy gained by the parabolic reflectors (Sobhansarbandi and Atikol, 2015). In most of the domestic applications, water is used as heat transfer fluid (HTF) owing to its desirable thermal transport properties, compatibility with the collector material and availability. However, considerable reduction is seen in thermal efficiency of the collector with respect to rise in temperature of the HTF at the inlet resulting in temperature lift at all sections of the collector (Kürklü et al., 2002). Attempts have been made to reduce temperature of the circulating HTF through storage of the required quantity of thermal energy in a thermal storage system thereby simultaneously reduces the mismatch between demand and supply and also the temperature of HTF is brought to the possible low temperature condition.

Several researchers have undertaken studies for addressing the aforesaid issues associated with solar collectors through structural modifications and other methods like integration of TES system. A brief literature review on the latest development/innovative methods used in the

amelioration of the thermal performance of the CPC solar collector is summarized as follows. Gang et al (Gang et al., 2012) performed experimental investigation of thermal performance of an evacuated CPC collector with U-type absorber pipes. The experimental results showed the achievement of a better thermal performance by the collector through increase in HTF temperatures and this collector could be suitable for applications such as desalination, water heating, space heating, etc. Xu et al (Xu et al., 2020) made a numerical study on a novel multi section CPC solar collector and, according to the results, the uniform distribution of solar radiation in the absorber tube was appreciably higher than the conventional CPC solar collector, under the similar conditions. Xia and Chen (Xia and Chen, 2020) made experimental study of an evacuated solar collector, coupled with a mini CPC reflector and compared the collector's performance, without the mini CPC reflector. The results showed an enhancement in the range of 24.3 % - 29.2 % in the thermal efficiency of the collector integrated with mini CPC reflector compared to a conventional evacuated collector.

Studies involving solar collector integrated with TES system and the subsequent influence of the integration on thermal performance of the collector are discussed as under. Yang et al (Yang et al., 2014) analyzed numerically the performance of the solar flat plate collector combined with single and three PCMs configurations. The average energy and exergy collection efficiency of solar collector with three PCMs configuration was high, as against to the single PCM configuration. An experimental study on solar flat plate collector paired with a nanocomposite PCM designed for SWH application was conducted by Saw et al (Saw et al., 2013). The composite prepared used paraffin loaded with copper nanoparticle (20 nm) to compensate for the inherent low thermal transport properties of the base PCM. The efficiency of collector reportedly, manifested a 6.9 % and 8.4 % increase using the paraffin and nano-composite PCM respectively. Kilickap et al (Kılıçkap et al., 2018) examined the thermal performance of solar collector integrated with latent heat thermal energy storage (LHTES) system and sensible thermal energy

storage system, under different seasonal conditions. Results showed a 1.5 % improvement in thermal efficiency of the solar collector with the use LHTES system compared to sensible heat storage system. Anbazoglu et al (Canbazoglu et al., 2005) carried out a similar study on the performance of the solar collector integrated with LHTES system using sodium thiosulfate pentahydrate as the PCM. The outlet HTF temperature in the solar collector was seen as 3.5 times higher using LHTES system than the sensible TES system.

Exergy analysis is considered as an established tool for evaluating the overall thermal performance of the system (Lior et al., 2006). Kalogirou (Kalogirou, 2004) used an exergy analysis for isothermal and non-isothermal solar collectors for determination of the optimum temperature of the absorber tube for a reduced entropy generation. The optimized absorber tube temperature was the geometric mean of the stagnation temperature of the HTF and the ambient temperature. Farahat et al. (Farahat et al., 2009) used exergy analysis on double glass flat plate solar collector and noticed, the collector reaching a maximum exergy efficiency of 3.9 %, under optimized conditions. Reportedly, the exergy efficiency increased, with increase in the temperature difference between the collector inlet and the ambient. A similar behavior of exergy efficiency, was presented by Ge et al. (Ge et al., 2014) and additionally, a study on the influence of the mass flow rate on the exergy was also carried out. The study revealed an increase in flow rate resulting in decrease in exergy rate. Jafarkazemi and Ahmadifard (Jafarkazemi and Ahmadifard, 2013) used a similar approach described by Farahat et al. (Farahat et al., 2009) for quantifying the solar radiation exergy. A comparison study related to the performance of the flat plate solar collector with three different HTFs (propylene glycol- water, ethylene glycol and water) was made. The exergy studies showed water as the best suited HTF under nonfreezing conditions. Nevertheless, under freezing conditions, the mixture of water and propylene glycol was the best choice of circulating fluid.

In the view of the recent technological advancements made in the CPC solar collectors and the merits accrued with the integration of collector with TES system, the present study aims to

investigate the thermal performance of CPC solar collector, integrated with different configurations of TES system. A comprehensive experimental investigation at system level, including the heat losses from various components and exergy analysis under real time conditions as not been reported to the best of the authors' knowledge.

2. Experimental test facility

An experimental setup was designed and fabricated for studying the charging characteristics of the TES tank with a cascaded arrangement of PCMs. It consisted of a CPC solar collector, TES system, makeup HTF tank, pump and necessary measuring devices and schematic arrangements of the test facility is illustrated in Fig 1. Eight evacuated glass tubes having a diameter and length of 0.05 m and 1.8 m, were connected in series using a serpentine arrangement. A copper tube of 6 mm outer diameter with a wall thickness of 1mm runs through the entire length of each evacuated glass tube. The above arrangement was placed on an aluminum reflector with 0.15 m² aperture area and 55 ° half acceptance angle. The entire CPC arrangement was tightly fastened to the ground for providing the necessary support and minimizing vibrations. A TES system having as cylindrical shape of 0.5 m diameter and 1 m height was fabricated with galvanized iron. Sufficient space was provided at the top of the tank to accommodate valves and sensors. Including this space, the effective height of the TES was 0.92 m. A perforated mesh plate was placed at the bottom of each zone for ensuring a uniform flow of HTF and for supporting the weight of the PCM balls. Selection of the PCMs and arrangement was based on the degree of stratification, which is detailed in the forthcoming section. A cylindrical encapsulation made of aluminum with 4.5 cm inner diameter and 14.5 cm height, was used for the encapsulation of the different PCMs. The capsule was filled with PCMs up to 90 % of the volume in order to accommodate the volume change occurring during the phase change. A total of three hundred and sixty cylindrical capsules in all were placed inside the TES tank, with equal distribution in each zone.

The experimental facility was placed at the roof top in Thermal Sciences Block, Anna University, Chennai, India. Water, the selected HTF, was circulated from the makeup water tank through the copper tube in the receiver using a centrifugal pump. The mass flow rates of HTF were continuously monitored by a Coriolis mass flow meter. The solar radiation was effectively collected using a reflector assembly. The presence of vacuum in the glass tubes minimized the heat losses through convection and radiation. Hot water from the collector entered the top of the TES tank, where the sprinklers distributed the incoming water uniformly. During the charging process, heat was transferred to PCMs that were arranged in a cascaded manner with decreasing melting temperature from the top to the bottom. The above arrangement of PCMs is advantageous for achieving a high temperature driving potential, along the flow direction of the HTF, which, in turn, was more beneficial for effective charging of hot thermal energy in the PCMs. The HTF leaving the TES tank was recirculated back to CPC collector and the charging cycle was continued from 8.00 h to 17.00 h at the intended mass flow rate. The charging experiments were performed at three different flow rates (60kg/h, 120 kg/h and 180 kg/h), with a sensible variation in ambient temperature (± 1.3 °C) and solar radiation (± 30 W/m²). The TES tank, connecting pipes and fittings were all wrapped by glass wool insulation, to reduce heat infiltration and a photographic view of the experimental setup was presented in Fig 2. The temperature of HTF at the inlet, the center and the outlet of each zone was continuously measured every 1 min using a data logger (Agilent 34970). A digital pyranometer (Hukseflux LP02) was used for recording the solar radiation on the surface of the solar collector.

3 Data analysis

Various parameters of ET-CPC and cascaded LHTES system are evaluated from the experimental data as given below;

Rate of heat transfer to the HTF across the ET-CPC solar collector is calculated from Eqn (3.1).

149
$$Q_u = \dot{m} \times c \times \Delta T \times dt \quad (3.1)$$

150 where $\dot{m}$ is the HTF mass flow rate ($kg\ s^{-1}$), c is the specific heat of HTF ($kJ\ kg^{-1}\ K^{-1}$), ΔT is the
 151 difference in temperature of the HTF (K) and dt is the time interval (s).

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Fig 1 Schematic of experimental test facility

Fig 2 Site Photographic view of experimental facility

The total energy stored in the TES system is evaluated by summing of energy stored in each zone.

$$Q_s = Q_1 + Q_2 + Q_3 \quad (3.2)$$

where, ' Q_1 ', ' Q_2 ' and ' Q_3 ' represent the energy stored in each PCM zone.

The heat losses from the LHTES system to the surrounding is calculated from

$$Q_{loss} = (Q_{losszone1} + Q_{losszone2} + Q_{losszone3}) \times dt \quad (3.3)$$

$$Q_{losszones} = \frac{T_{avg} - T_a}{\frac{1}{2\pi r_1 L h_{in}} + \frac{\ln(r_2/r_1)}{2\pi k_{Gl} L} + \frac{\ln(r_3/r_2)}{2\pi k_{ins} L} + \frac{1}{2\pi r_3 L h_{out}}} \quad (3.4)$$

where T_{avg} - average temperature of the zone (K), T_a - surrounding temperature (K), r_1 - inner radius of the LHTES (m), r_2 - outer radius of the LHTES system without insulation (m) and r_3 - outer radius of the LHTES system (m) with insulation ($0.023 \text{ W m}^{-1} \text{ K}^{-1}$).

The convective heat transfer coefficient (h_o) of air and TES system is predicted using the correlation (VDI-Gesellschaft, 2010) as given in the following Eqn (3.8).

$$h_o = \frac{[0.825 + 0.387 \times [Ra \times f \times (pr)]^{1/6}]^2 \times k_{air}}{H} \quad (3.5)$$

where,

$$f(Pr) = \left[1 + \left(\frac{0.492}{Pr} \right)^{9/16} \right]^{-16/9}$$

Grashof number is given by

$$Gr = \frac{g \times \beta \times \Delta T \times H^3}{\nu^2} \quad (3.6)$$

The convective heat transfer coefficient of HTF in inside area of TES evaluated from correlation given in (VDI-Gesellschaft, 2010) (3.10)

$$h_i = \frac{k_f}{d_p} \times (2.58 \times Re_p^{1/3} + 0.094 \times Re_p^{0.8} \times Pr^{0.4}) \quad (3.7)$$

where,

$$Re_p = \frac{\rho_f \times d_p \times \varepsilon \times u_f}{\mu_f}$$

$$Pr = \frac{c_{p,f} \times \mu_f}{k_f}$$

The utilization ratio characterizes the amount of energy retrieved (Q_d) versus the stored energy (Q_s) during the discharging process:

$$U_T = \frac{Q_d}{Q_s} \quad (3.8)$$

CPC collector thermal efficiency (η_c) is expressed,

$$\eta_c = \frac{Q_u}{A_a \times G_t} \quad (3.9)$$

where, A_a is the aperture area of solar collector (m^2) and G_t is the incident solar radiation on the surface of solar collector ($W m^{-2}$).

The useful energy gained with 'F_R' and 'U_L' terms is expressed as

$$Q_u = F_R \times [G_t \times A_a - A_r \times U_L \times (T_p - T_{amb})] \quad (3.10)$$

where, 'F_R' is the heat removal factor, A_r is the receiver-absorber area (m^2) and T_p is the mean plate temperature (K)

The absorbed radiation 'S' is obtained from Eqn

$$S = G_t \times \frac{A_{ap}}{A_{ab}} \times \tau_1 \times \tau_2 \times \alpha_{ab} \times \rho_r^n \times (1 - \frac{r_{p1} - r_{ab}}{2\pi p r_{ab}}) \times (1 + \rho_{ab} \rho_1 \frac{A_{ap}}{A_{p1}}) \quad (3.11)$$

where, τ_1 and τ_2 are transmissivity of inner and outer glass; α_{ab} , ρ_r , n , ρ_{ab} , ρ_1 , A_{p1} , r_{p1} and r_{ab} are absorption coefficient, receiver reflectivity, reflection number, absorber reflectivity, glass reflectivity, area of glass evacuated glass (m^2), radius of glass (m) and radius of an observer (m) respectively.

The heat losses 'Q_L' associated with the solar collector is determined from Eqn (3.17)

$$Q_L = S - Q_u \quad (3.12)$$

Radiative thermal resistance (R_{abs-g2}) between absorber and glass tube is calculated from

$$R_{r,abs-g2} = \frac{1}{h_{abs-g2} \times A_{abs}} \quad (3.13)$$

where,

$$h_{r,abs-g2} = \frac{\varepsilon_{abs} \times \sigma \times (T_{ab}^2 + T_g^2) \times (T_p + T_g)}{1 + \frac{\varepsilon_{abs}}{\varepsilon_g} \times \frac{D_{abs}}{D_{g1}} \times (1 - \varepsilon_{abs})}$$

where ε_{abs} - emissivity of the selective absorbing coating. ε_g - emissivity of the inner surface of an outer glass tube, σ - Stefan Boltzmann constant ($5.67 \times 10^{-8} W m^{-2} K^{-4}$) and D_g - inner diameter of an outer glass tube.

Convective thermal resistance between outer surface of the glass tube and ambient is calculated from

$$R_{c,g2-a} = \frac{1}{h_{c,g2-a} \times A_{g2}} \quad (3.14)$$

where,

$$h_{c,g2-a} = \frac{Nu_{air} \times k_{air}}{d_{g2}}$$

The value of Nusselt number in the above expression is predicted using the following correlation (Kalogirou, 2014)

$$Nu_{air} = 0.4 + 0.54 \times (Re)^{0.52} \text{ for } 0.1 < Re < 1000$$

Radiative heat resistance between the glass tube's outer surface and the surrounding environment is calculated.

$$R_{r,abs-g2} = \frac{1}{h_{r,g2-a} \times A_{g2}} \quad (3.15)$$

219 where,

$$220 \quad h_{r,g2-a} = \varepsilon_{g2} \sigma (T_a^2 + T_{g2}^2) \times (T_a + T_{g2})$$

221 Equation for exergy balance in the solar thermal collector is given by
222 Suzuki (1988)

$$223 \quad E_{ex,in} - E_{ex,s} - E_{ex,out} - E_{ex,d} = 0 \quad (3.16)$$

224 where, ' $E_{ex,in}$ ', ' $E_{ex,s}$ ', ' $E_{ex,out}$ ' and ' $E_{ex,d}$ ' are the inlet exergy rate, stored exergy rate, outlet exergy
225 rate, and exergy loss rate

226 Exergy rate of fluid flow at the inlet of CPC solar collector is calculated from Eqn
227 (Bejan, 2016)

$$228 \quad E_{ex,in} = mc_p \left[T_{ex,i} - T_{ex,a} - T_{ex,a} \ln \left[\frac{T_{ex,i}}{T_{ex,a}} \right] \right] \quad (3.17)$$

229 Exergy rate of fluid at the outlet of CPC solar collector is calculated from Eqn
230 (Bejan 2016)

$$231 \quad E_{ex,out} = mc_p \left[T_{ex,o} - T_{ex,a} - T_{ex,a} \ln \left[\frac{T_{ex,o}}{T_{ex,a}} \right] \right] \quad (3.18)$$

232 The radiation exergy rate from the sun to the surface of the CPC solar collector ($E_{ex,S}$)
233 is evaluated from the following Eqn as given by (Gang *et al.* 2012).

$$234 \quad E_{ex,S} = I \times A_a \left[1 - \frac{T_{ex,a}}{T_{ex,s}} \right] \quad (3.19)$$

235 where, the apparent solar temperature ($T_{ex,S}$) is considered as 6000 K in exergetic of SWH system
236 (Petela 2003).

237 Exergy efficiency of the CPC solar collector (η_{ex}) is given below (Gang *et al.* 2012).

$$\eta_{ex} = \frac{E_{ex,out} - E_{ex,in}}{E_{ex,S}} \quad (3.20)$$

where, ' $T_{ex,in}$ ', ' $T_{ex,out}$ ' and ' $T_{ex,a}$ ' are temperature of inlet, outlet and ambient temperature HTF fluid respectively.

4. Results and discussion

4.1 Thermal efficiency

Thermal efficiency of the ET-CPC, defined with regard to the heat loss factor is presented in Fig 3 from 11.00 h - 17.00 h. As can be observed, the slope of the line ' $F_R U_L / C_R$ ' is the lowest for 'Case 2, portraying the minimum heat losses. Thermal efficiency tends to decrease with respect to increase in heat loss parameter as a result of drop in thermal resistance between the outside surface and ET. Upon pairing the ET-CPC with sensible heat storage, the value of y- intercept representing ' $F_R \eta_o$ ' is 0.52, while it is 0.69, and 0.7 for 'Case 1' and 'Case 2', respectively. As the mass flow rate of HTF increases, the intercept value is 0.61 at 120 kg h⁻¹ and 0.67 at 180 kg h⁻¹, which are lower than 'Case 1' and 'Case 2' at 60 kg h⁻¹. This can be due to occurrence of destratification in sensible heat storage system due to increase in HTF flow rate. The synergetic benefits of the cascaded arrangement of PCMs and compound parabolic concentrator will be advantageous towards the development of an effective SWH system.

Fig 3 Variation in thermal efficiency of ET-CPC

4.2 Heat losses from the SWH system

A detailed analysis of heat losses from the ET-CPC and different configurations of TES system has been made and discussed in the following section.

4.2.1 Heat Losses from the ET-CPC Collector

The solar rays reaching the center of the ET-CPC is accepted by the absorber tube directly, but those reflecting from the edge of ET-CPC reach the receiver after numerous reflections. This reduces the intensity of solar radiation and causes a heat loss. The energy is lost in the collector due to radiation, convection and conduction from the top, bottom and edge of the collector. Fig 4 shows the thermal network for ET-CPC solar collector and the corresponding thermal resistances are determined from the Eqs (3.12 - 3.15).

Fig 4 (a) Thermal network (b) Cross-section of ET

The variations in thermal resistance from absorber surface to ambient, at three different mass flow rates (60 kg h^{-1} , 120 kg h^{-1} and 180 kg h^{-1}) under sensible heat storage configuration are shown in Fig 5. The temperature of the absorber plate should be close to the ambient temperature for maintaining higher thermal resistance. This can be attained at a higher flow rate, for instance, thermal resistance is 2.85 K W^{-1} at 180 kg h^{-1} . While its reduces to 2.45 K W^{-1} at 60 kg h^{-1} . This can be attributed to an increase in total heat transfer from the plate to the HTF, which lowers the plate temperature.

Fig 5 Variations in (a) Radiative (b) Convective (c) Conductive resistance of ET-CPC paired with sensible heat storage

For ‘Case 1’ and ‘Case 2’, all the thermal resistances at $60 kg h^{-1}$ are relatively higher than sensible heat storage mode, as observed from Fig 6. The maximum radiative thermal resistance of $0.50 K W^{-1}$ and $0.52 K W^{-1}$ for ‘Case 1’ and ‘Case 2’ respectively, indicating a minimum heat loss resulting from their higher thermal performance as discussed in section 4.5. This loss depends on the inlet HTF temperature, as an increase in HTF temperature at the inlet of collector decreases the rate of useful heat energy gained. There is the possibility of higher plate temperature for the given incident solar radiation causing a reduction in radiative thermal resistance. Through the proposed cascaded arrangement in ‘Case 1’ and ‘Case 2’, the effective charging helps in the reducing of HTF temperature at the inlet of solar collector that increases thermal resistance considerably. The convective and conductive resistances are attained to a maximum of $1.4 K W^{-1}$ and $0.77 K W^{-1}$ in ‘Case 2’ configuration.

Fig 6 Variations in (a) Radiative (b) Convective (c) Conductive resistance of ET-CPC collector paired with ‘Case 1’ and ‘Case 2’

Fig 7 shows the details of the hourly energy loss of ET-CPC integrated with sensible heat storage system at three different HTF flow rates, evaluated during the charging cycle. At the flow rate of $60 kg h^{-1}$, the hourly energy loss increases from the start of experimentation till 15 h, followed by a declining trend till the end of the experimentation. As observed, the maximum energy loss is nearly 1911.5 kJ for sensible heat storage. It can be ascribed to a higher temperature difference between the absorber tube and ambient temperature as illustrated in Fig 8. This starts declining, reducing between 14.45 h to 17.00 h. Similar trends were reported by (Hobbi & Siddiqui 2009) in the collector.

For cascaded LHTES system in ‘Case 1’ and ‘Case 2’, there is a considerable reduction in energy losses due to the existence of lower temperature difference between plate and

315 ambient as depicted in Fig 9. Due to this, there is a reduction in energy losses, varying between
 316 660 kJ - 1773 kJ and 656 kJ - 1619 kJ for 'Case 1' and 'Case 2' respectively as shown in Fig 10.

317

318 **Fig 7 Heat losses on hourly basis from ET-CPC collector paired with sensible heat storage**

319

320 **Fig 8 Temperature variation between plate and ambient under sensible heat storage**

Fig 9 Heat losses on hourly basis from ET-CPC paired with ‘Case 1’ and ‘Case 2’

Fig 10 Temperature variation between plate and ambient under ‘Case 1’ and ‘Case 2’

4.3.2 Heat losses from TES System

Initially, the TES system is supplied with HTF at $85\text{ }^{\circ}\text{C}$ at 240 kg h^{-1} and the HTF circulation has been taking place until the steady-state condition is achieved. Fig 11 shows a variation of HTF temperature inside the TES system lasting for 44 h . As noticed, the temperature drop is by $37.2\text{ }^{\circ}\text{C}$, $39\text{ }^{\circ}\text{C}$, $40\text{ }^{\circ}\text{C}$ and $41\text{ }^{\circ}\text{C}$ at $x/L=0.95, 0.65, 0.35$ and 0.05 respectively. During the initial time, cool down begin with a steeper slope at $1.01\text{ }^{\circ}\text{C h}^{-1}$ at top and $0.93\text{ }^{\circ}\text{C h}^{-1}$ at the bottom. After ensuring the proper insulation from the above results, the experiments were conducted.

Fig 11 Transient temperature variation of HTF

The results of overall heat losses evaluated from Eqn (3.6-3.10) for sensible heat storage at three different flow rates are plotted in Fig 12. As can be seen from the Fig, there is a maximum heat loss of 585 kJ at 60 kg h^{-1} during $14.00\text{ h} - 15.00\text{ h}$. Obviously, the HTF temperature is higher at a low flow rate, leading to a considerable reduction in heat loss at a higher flow rate throughout the experimentation. As an

example, the heat losses vary between 140 *kJ* - 520 *kJ* at 180 kg h^{-1} against the range of 192 *kJ* - 585 *kJ* at 60 kg h^{-1} .

Fig 12 Energy loss from sensible heat storage

The heat losses of TES system in ‘Case 1’ and ‘Case 2’ are compared with the results of sensible heat storage at 60 kg h^{-1} in Fig 13. It is noticed that the heat losses are more less same during between 9.00 - 11.00 *h* for all configurations. However, the losses tend to decrease considerably between 11.00 *h* - 14.00 *h* as a result of melting of PCMs in a nearby isothermal manner. In addition, the temperature driving potential between the HTF and ambient temperature is so small, resulting in the reduction of heat losses. As the charging proceeds, the heat losses in ‘Case 1’ and ‘Case 2’ approach to that of sensible heat storage due to an increase in convective heat losses. During the experimentation, the cumulative heat loss of 4138 *kJ*, 3991 *kJ* and 3574 *kJ* was perceived for sensible heat storage, ‘Case 1’ and ‘Case 2’ respectively. In summary, the mean temperature of HTF is found to be lower in ‘Case 1’ and ‘Case 2’ because of higher thermal mass of the PCMs.

Fig 13 Energy losses from different configurations of TES system.

4.4 Exergy analysis of solar collector

Fig 14 describes the variations in the exergy efficiency with the independent parameter for ET-CPC with sensible heat storage at three different flow rates. It is observed that the exergy efficiency increases with respect to increase in independent parameter $(T_{mean}-T_{amb})/G_t$, reaching maximum exergy efficiency of 7.3 % at 180 kg h^{-1} . This is because of enhanced useful energy gained in the ET-CPC, lowering the plate temperature thereby increasing the useful exergy rate as shown in Fig 15. Similar trends of results are seen in other flow rates, but evidently with lower exergy efficiency at 120 kg h^{-1} and 60 kg h^{-1} . The maximum efficiency drops to 5.4 % at 120 kg h^{-1} and 3.6 % at 60 kg h^{-1} . These results indicate an increase in the energy destruction with respect to increase in both ambient and mean HTF temperature. After 11 h, there is no significant variation in the exergy efficiency. This can be seen from the existence of a minimal temperature variation between the mean HTF and ambient. Gang *et al.* (2012) reported, exergy efficiency of 4.62 % and 5.89 % at inlet water temperature of 55 °C to 75 °C for

370 similar kind of ET-CPC collector. The present values are slightly higher than the above results,
 371 resulting from the difference in collector area, the intensity of solar radiation and surrounding
 372 temperature.

373
 374 **Fig 14 Exergy efficiency of ET-CPC as a function of heat loss coefficient (sensible heat**
 375 **storage)**

376
 377 **Fig 15 Instantaneous exergy rate of ET-CPC (sensible heat storage)**

Fig 16 shows the variations in exergy efficiency of ET-CPC paired with ‘Case 1’ and ‘Case 2’ at 60 kg h^{-1} and also compared with the results obtained through paring it with the sensible heat storage. As noticed, exergy efficiency for all three configurations is almost the same till 10.00 h. The sensible heating of PCM in ‘Case 1’ and ‘Case 2’ in the solid state does not cause any appreciable increase in exergy efficiency over the sensible heat storage. Following the initiatiion of the melting of PCMs at 12.00 h, the exergy efficiency shows a sharp rising trend, with the impication of its superior performance in attaining higher energy charging with higher exergy rate as illustrated in Fig 17 The continuous charging of thermal energy in a nearby isothermal manner maintains the required temperature driving potential with minimum irreversibility till the end of melting PCMs up to 17.00 h.

Fig 16 Exergy efficiency of ET-CPC as a function of heat loss coefficient (‘Case 1’ and ‘Case 2’)

Interestingly, for the cascading arrangement of PCMs in 'Case 2' exergy efficiency increases over the ‘Case 1’. For example, the Figs of maximum exergy efficiency are 6.2 % and

8.6 % for 'Case 1' and 'Case 2' in comparison with the sensible heat storage case of 3.6 %. This can be explained by an increase in useful heat energy gained in ET-CPC due to the supply of HTF at a lower temperature which reduces the exergy destruction considerably. The proposed CPC solar collector exhibited a higher exergy efficiency, compared to a flat plate solar collector (Gunerhan & Hepbasli 2007) and ET solar collector (Ataee & Ameri 2015). In conclusion, the superior performance of ET-CPC in attaining a higher exergy efficiency through integration of cascaded arrangement of PCMs has a huge potential for solar water heating applications.

Fig 17 Instantaneous exergy rate of ET-CPC ('Case 1' and 'Case 2')

4.6 Conclusions

Experiments were performed for the exploration the effect of PCM addition in augmenting the useful heat energy gained in ET-CPC collector, decides the phenomenal increasing energy stored compared to sensible heat storage Thermal performance of ET-CPC collector is analyzed in terms of thermal resistance and exergy analysis. The main conclusions drawn from the experimental studies are summarized as below;

- Inlet temperature of HTF to the collector influences predominantly thermal performance of collector and TES system. Lower inlet temperature causes an increase in useful heat energy gained and reduction in thermal losses in the collector at particular available solar radiation. Radiative thermal resistance is much higher than convective thermal resistance in ‘Case 2’, indicating a better heat transfer between the HTF and inside wall of the absorber tube.
- It is ascertaining that the heat losses from TES system is more or less same between 9.00 – 11.00 h for all three TES configurations. After wards, there is a drastic reduction in heat losses from ‘Case 1’ and ‘Case 2’, as a result of melting of the PCMs there by lowering the temperature driving potential between the HTF and ambient.
- Maximum exergy efficiency of 8.6 % for ‘Case 2’ while it is 6.2 % and 3.2 % for ‘Case 1’ and sensible heat storage. This can be ascribed to an appreciable reduction in exergy destruction in ET-CPC when pairing with ‘Case 2’, resulting from a supply of HTF at relatively lower temperature to the collector. It is possible to reduce the exergy destruction further in the collector through the effective charging of PCMs by employing an appropriate cascaded arrangement in the TES system.
- The aforesaid promising results with the integration of CPC and cascaded PCM configuration will lay the framework for the development of efficient solar water heating systems for medium temperature applications. The synchronized set of strategies from the present study will be applicable for effective utilization of abundant available solar energy and in realizing the potential benefits of solar water heating system in terms of energy savings and mitigation of greenhouse gas emission.

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