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Experimental Investigation of Heat Losses from a Heat Pipe Based Parabolic Trough Collector used for Direct Steam Generation

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Authors' contributions

This work was carried out in collaboration among all authors. Author ALT designed the study, performed the statistical analysis, and wrote the first draft of the manuscript. Author ACO managed the analyses of the study and the literature searches. All authors read and approved the final manuscript.

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ABSTRACT

The performance of a thermosiphon based parabolic trough collector (PTC) used for direct steam generation depends largely on the heat losses of the solar thermal system. This paper presents an experimental investigation of the heat losses in a thermosiphon based solar thermal system that used a linear receiver with a PTC for the generation of low temperature steam. A locally constructed PTC was used to concentrate sun rays to a linear copper pipe enclosed in an evacuated glass tube and held at the focal line of the PTC to heat water and generate steam. Circulation of the water in the closed-loop solar thermal system was through natural convection. A solar meter was used to measure the incident radiation flux at the experimental site and PT100 temperature sensors were installed at different points of the system to measure the temperature distribution within the system. The thermal efficiency and overall heat losses of the system were investigated by fitting the experimental data to standard equations. The results showed that the instantaneous thermal efficiency of the system was 46.48%, 43.1% and 45.32% respectively for three days examined. The

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overall heat losses in the system were 1211.95, 974.32 and 911.26 kwh per day respectively for the three days investigated. Heat losses from the tank accounted for over 83% of the losses for all the days examined. The evacuated glass tube reduced heat losses from the receiver to very low values of 2.31, 1.63 and 1.43 KWh per day respectively for the three days tested. The use of a better insulating material on the tank was recommended to reduce convective and conductive heat losses, thereby enhancing the performance of the system.

Keywords: Direct steam generation; parabolic trough; heat losses; solar thermal system; thermosiphon.

1. INTRODUCTION

Parabolic trough collectors (PTC) are the most used solar concentrators in the world. This is because they can attain high temperatures of up to 400°C with very little or no noticeable degradation in their performance over time. They are largely used for solar electricity generation systems (SEGS) in the USA and direct solar steam (DISS) in Europe. Mills and Thirugnanasambandam reported the advantages and study progress of PTC in solar thermal energy systems [1][2]. The performance of a PTC as a steam generation system depends largely on the quantity of heat transfer to the receiver and the heat losses of the system. In the report on the operation and maintenance for SEGS plants and the DISS systems, Eck and Steinmann ascertained that heat losses of receivers are very important to the efficiency of steam generating systems, with thermal emittance of selective coating (radiation losses) and convection being the main causes [3]. Several procedures have been used by researchers to estimate the thermal loss of receiver tubes and other parts of the solar thermal system depending on the operating temperature.

Dreyer, et al. carried out experiments to determine the heat losses on three Schott’s parabolic trough receiver components under steady state conditions at German Aerospace Center (DLR) and U.S. National Renewable Energy Laboratory (NREL). The results showed that heat losses of the state-of-the-art parabolic trough receivers are well below 300 W/m and depend directly on the thermal emittance of the absorber tubes [4]. Lüpfert, et al. used different approaches (field measurements and laboratory setups both based on energy balances from the hot inside of receiver tube to the ambient) to measure receiver heat losses. They also measured and analyzed the temperature of a glass envelope for evacuated receivers and modeled the overall heat losses and emissivity coefficients of the receiver. Their work showed a good agreement between the different approaches and independent installations. For solar parabolic trough plants operating in the usual 390°C temperature range, they reported heat loss of about 300W/m (per receiver length) [5]. Zhang, et al. experimentally studied the heat losses of a double-glazing vacuum U-type solar receiver mounted in a PTC natural circulation system used for generating medium-temperature steam. They performed field experiments to determine the overall heat losses of the receiver. The also studied the effects of wind, vacuum glass tube, radiation, and structural characteristics on the heat losses. Their results showed that the heat losses increased from 183 to 255 W per receiver and concluded that neither convection nor radiation heat losses may be negligible in the analysis of U-type solar receivers [6]. Yaghoubi, Ahmadi and Bandehee evaluated heat losses of absorber tubes of PTCs for the collector field of 250 kW Shiraz (Iran) solar thermal power plant for different conditions. Their findings show that any absorber tube with vacuum reduces heat losses in comparison with a broken tube or those without vacuum. This led to the conclusion that poor isolation or glass tube failure should be avoided for any solar thermal power plants when in operation [7].

As an alternative to pump-forced circulation in PTC solar thermal systems, the natural circulation (thermosiphon) technique can be used to stimulate flow. This is because of the several advantages (simple to install and flexible, easy to control, and high heat transfer ability even at low temperature differences) it has over mechanical or electrical pumped circulation. Despite this, information on the application of the natural circulation (heat pipe based) solar receivers to PTC systems for generating low-mid temperature steam is scanty. A solar adsorptive refrigerating system, where the reactor was heated by a PTC system coupled with thermosiphon technique was proposed [8]. However, no experimental validation was provided. A U-type receiver coupled with a natural circulation heat pipe system and applied to a PTC system to generate medium-temperature (120 – 200ºC) steam was developed. Experimental investigation to determine the performance of the system carried out reported that the system generated mid-temperature steam at a pressure up to 0.75 MPa. The maximum thermal efficiency obtained was 38.52% at a discharging pressure of 0.5 MPa during summer [9]. Suggestion were made that optimization on thermal insulation and collector exit design could raise the efficiency and reliability of system. Abiem and Akoshile investigated experimentally the thermal efficiency of a solar thermal steam generating system using thermosiphon technique with PTC and a linear receiver. They reported that the system generated low temperature steam of up to 105ºC at a pressure of approximately 120 kPa. A thermosiphon mass flow rate with a maximum value of 0.042 kg/s was observed. The instantaneous thermal efficiency of the system reached 46.48 % [10]. The aforementioned works show that little or no attention is paid to heat losses of the entire systems (tank, connecting pipes and receiver) by researchers. The objective of this paper is to address this (experimentally investigate the overall heat losses of a heat pipe based parabolic trough collector direct steam generating system).

2. EXPERIMENTAL DETAILS

Experimental test was carried out on a thermosiphon-based PTC solar thermal system. Fig. 1 shows the schematic diagram of the experimental setup. It was made up of a PTC, an insulated water tank, copper pipe (receiver), flexible ascending pipe (to allow for movement of the PTC) and a descending pipe. At the start of the experiment, a dual axis solar tracker moved the PTC to follow the sun and concentrate solar irradiation to the receiver, held horizontally at its focal line. Water from the tank, positioned 40 cm above the height of the trough flowed under gravity to the receiver, passing through the descending pipe, where it absorbed heat, became less dense and had a higher temperature. This less dense and more buoyant hot water in the receiver was moved by natural circulation (thermosiphon) through the flexible ascending pipe back into the tank where it transferred heat to the cold water. The water in the tank then flowed back to the receiver. The cycle continued as long as there was heat absorption, resulting into a potential gradient between the descending and ascending pipes to provide a driving force of natural circulation until the water attained boiling temperature due to continuous thermal exchange. At the boiling stage, the high temperature and pressurized water was flashed into the tank that also served as a steam and liquid water separator [10].

A pressure release valve was installed at the top of the tank to ensure efficient steam discharge. A water level meter was also installed by the side of the tank to help in monitoring the water level in the tank. And as the need arose, water was slowly added to the tank to prevent drastic changes in the temperature of the thermal system. In order to minimize heat losses and enhance the efficiency of the system, the ascending and descending pipes were also insulated with fiberglass. Temperatures in the solar thermal system were monitored with seven (7) temperature sensors [($T_{rec.}$), (T_{in}), (T_{out}), (T_{water}), (T_{vapor}), (T_{tank}) and ($T_{amb.}$)] with an accuracy of ±0.15ºC, as specified by the manufacturers. T_{rec} monitored variations in the receiver temperature. T_{in} and T_{out} at the end of the descending pipe and beginning of ascending pipe respectively, monitored the temperature of the water just before it entered the receiver and when it exited. T_{water} and T_{vapor} were installed inside the tank, T_{water} at the bottom (but not in contact with the wall of the tank) to measure the temperature of water in the tank and T_{vapor} at the top of the tank to measure the temperature of vapor (steam). T_{tank} was placed on the surface of the tank to measure the external temperature. T_{amb} placed close to the PTC (but not in direct contact with any object, other than air) measured the ambient temperature. The specifications of the solar thermal system (PTC and Tank) are presented in Table 1. A digital flow meter with an inbuilt hydraulic electromagnetic flow sensor was installed at the end of the ascending pipe to measure the natural circulating mass flow rate of water in the system. A Universal 500PSI pressure sensor (P_{vapor}) was also installed inside the tank to measure the pressure in the tank. Apart from the temperature sensors (T_{water} and T_{vapor}) inside the tank, all the others were placed on the surface of the pipes or tank and covered with an insulating material. Incident solar irradiation was measured with a Model TM – 206 Solar Power Meter (a pyranometer) with an accuracy of ± 10 W/m^2 (specified by the manufacturer) during the experiment. The temperatures and pressure signals were gathered by an Arduino data acquisition board to a personal Computer at time intervals of 1 minute each [10].

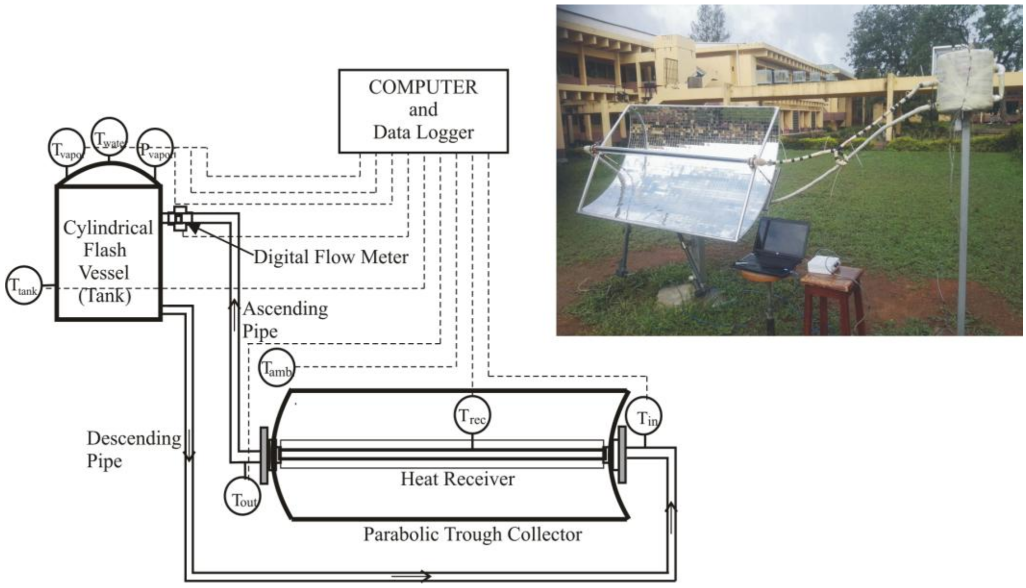


Fig. 1. Schematic diagram of the experimental set up

Table 1. Specifications of the PTC steam generating system

Item	Value/type
Collector aperture area (A_c)	1.50 m^2
Collector aperture width (a)	1.00 m
Concentration ratio (CR)	35.2
Rim angle (ψ_{rim})	78.68^o
Working fluid	Water
Tracking mechanism type	Electronic
Mode of tracking collector axis	N–S horizontal, E–W tracking
Tank (Outer) Height	32.0 cm
Tank (Outer) Diameter	30.0 cm
Tank (Inner) Height	28.0 cm
Tank (Inner) Diameter	23.4 cm
Tank Volume	~ 12.0 L
Insulator	Fiberglass
Thermal conductivity of fiberglass	0.045 W/m.k [11]

The study was carried out on 12th, 15th and 17th of May, 2019. The beginning of the wet (rainy) season, which starts in April and ends in the month of October in the middle-belt region of Nigeria [12]. The average temperatures of the receiver, water, tank and connecting pipes were fitted into standard equations to estimate the thermal efficiency and heat losses in the system.

3. RESULTS AND DISCUSSION

3.1 Natural Circulation (Thermosiphon) Mass Flow Rate

The results of the natural circulation (thermosiphon flow) of water showed that flow started at about 10:20 hours when the temperature of water in the ascending pipe was 50.42^{o}C and ended at about 18:00 hours at a temperature of 72.58^{o}C, with a maximum mass flow rate of 0.042 kg/s recorded at 13:20 hours on 12th May, 2019. On the 15th May, 2019, flow equally started at about 10:20 hours when the temperature of the ascending pipe was 48.59^{o}C and ended at about 17:10 hours at a temperature of 64.5^{o}C, with a maximum mass flow rate of up to 0.03 kg/s attained at 14:00 hours. Thermosiphon flow started at about 11:30 hours when the temperature of the ascending pipe was 47.13^{o}C and ended at about 16:40 hours at a temperature of 66.04^{o}C, with a maximum mass flow rate of 0.042 kg/s recorded at 14:10 hours on 17th May, 2019. These results compared favorably with those of Zhang et al, [9].

3.2 Instantaneous Thermal Efficiency of the Heat Pipe Based PTC System

The instantaneous thermal efficiency of the system was calculated as a function of time of the day using equation 1 [13].

$$\eta = \frac{\dot{m}c_w(T_{out} - T_{in})}{I_b A_c} \quad 1$$

where η is the instantaneous thermal efficiency, $\dot{m}$ is the thermosiphon mass flow rate, c_w is the specific heat capacity of water, T_{out} is the outlet temperature, T_{in} is the inlet temperature, I_b is the global solar irradiation and A_c is the collector's aperture area. The results showed that the efficiency of the system attained a maximum value of 46.48 %, 43.1% and 45.32% respectively for 12th, 15th and 17th May, 2019 experiment was carried out. The thermal efficiency curves are plotted as a function of time of the day and presented in Fig. 2. This result is an improvement compared with Zhang et al. [9] who used a U-tube receiver and obtained a maximum thermal efficiency of 38.52%.

3.3 Heat Losses of the Heat Pipe Based PTC System

3.3.1 Convectional heat loss of the receiver

A linear (traditional) receiver was used in this work. A vacuum was maintained by tightly sealing the both ends of the evacuated glass tube using high thermal resistant and impermeable stopper corks. Air molecules between the surface of the heat receiver and the evacuated glass tube were locally extracted through the use of a syringe and needle. Convective heat losses are negligible in this kind of cases [6]. Hence, this category of heat loss was neglected in this study.

3.3.2 Radiative heat loss of the receiver

Radiative heat loss is the major mode of heat losses in linear receivers and cannot be neglected in this work, even with low test temperature of approximately 100^{o}C. Again, because the outer surface of the receiver was only covered with a black coating, it is necessary to calculate the effects of radiation during the experiment. Because of low experimental temperatures, the glass temperature was assumed to be equal to ambient temperature. Equation 2 was used to calculate the radiative heat loss of the receiver [6].

$$q_{rad} = \frac{\sigma A_r (T_r^4 - T_{gi}^4)}{\frac{1}{\varepsilon_r A_r} + \frac{1-\varepsilon_{gi}}{\varepsilon_{gi} A_{gi}}} \approx \frac{\sigma A_r (T_r^4 - T_a^4)}{\frac{1}{\varepsilon_r A_r} + \frac{1-\varepsilon_{gi}}{\varepsilon_{gi} A_{gi}}} \quad 2$$

where q_{rad}, σ, A, T, and ε are the overall heat losses due to radiation, Stefan Boltzmann constant, surface area, average temperature and emissivity, respectively. Subscripts rad, r, gi and *a* stand for the radiation, receiver, inner glass surface and ambient respectively.

Radiative heat losses per day for the receiver were determined to be 2.31 kWh, 1.63 kWh and 1.43 kWh respectively for the three days examined. These results compare with that of Zhang et al. who reported a heat loss of 255 W per receiver [6].

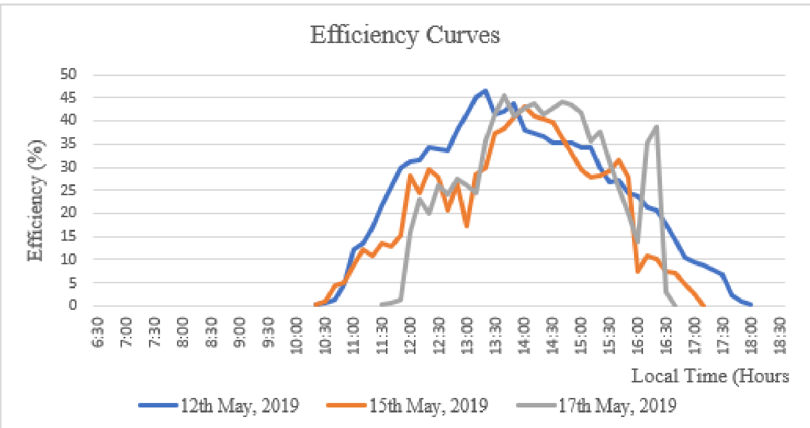


Fig. 2. Instantaneous thermal efficiency curves as a function of time of the Day for 12th, 15th and 17th May 2019

3.3.3 Heat losses from the tank

Conductive and convective heat losses were expected from the tank. Equation 3 was used to calculate the heat losses from the tank [14].

$$Q_{Tank} = \frac{T_w - T_a}{\frac{\ln\left(\frac{d_2}{d_1}\right)}{2\pi k_g L} + \frac{\ln\left(\frac{d_3}{d_2}\right)}{2\pi k_f L} + \frac{\ln\left(\frac{d_4}{d_3}\right)}{2\pi k_s L} + \frac{1}{1.42\pi d_4 L\left(\frac{T_w - T_a}{L}\right)^{1/4}}} \qquad 3$$

where Q_{Tank} is the heat loss from the tank, T_w is temperature of water, T_a is ambient temperature, d_1 is inner diameter of the tank, d_2 is the outer diameter of the tank, d_3 is the inner diameter of the lagging sheet, d_4 is the outer diameter of the lagging sheet, k_g is the thermal conductivity of galvanized steel, k_f is the thermal conductivity of fiberglass, k_s is the thermal conductivity of mild-steel and L is the length of the tank.

The first, second and third terms in the denominator of equation 3 estimates the thermal resistance through an inner galvanized steel tank, fiberglass (insulator) and external wall (used for lagging) respectively, while the fourth term estimates the free convection from the external vessel walls to the environment.

The heat loss per day from the tank were estimated to be 1008.56 kWh, 820.39 kWh and 757.11 kWh respectively for the three days examined. These formed 83.22%, 84.2% and 83.08% of the overall heat losses of the solar thermal system.

3.3.4 Heat losses from the pipes

Conductive and convective heat losses were also expected from the pipes. Equation 4 was used to calculate the heat loss from the pipes [13].

$$q_{pipe} = UA(T - T_a) \qquad 4$$

where q_{pipe} is the heat loss from the pipe, A is a suitable area for the heat flow, T is the temperature of the water, T_a is the ambient temperature and U is the overall heat-transfer coefficient expressed as

$$U = \frac{1}{1/h_1 + \Delta x/k + 1/h_2} \qquad 5$$

where h_1 and h_2 are the coefficient of convective heat transfer for the pipe and insulating material respectively, x is the thickness of the pipe and insulating material and k is thermal conductivity.

For a horizontal receiver, the coefficient of convection heat transfer (h) can be estimated by the use of natural convection heat transfer correlation for horizontal cylinders shown in equation 6 [15].

$$\bar{h} = \frac{\overline{Nu_D} k}{D} \qquad 6$$

where k is the thermal conductivity of the fluid, D is the diameter of the cylinder and $\overline{Nu_D}$ is the Nusselt number expressed in equation (7) [16].

$$\overline{Nu_D} = \left\{0.60 + \frac{0.387\mathrm{Ra}_D^{1/6}}{\left[1 + (0.0559/\mathrm{Pr})^{9/16}\right]^{8/27}}\right\}^2 \qquad 7$$

where Ra_D is the Rayleigh number given by (Gr_D .Pr = $g\rho^2\beta\Delta T D^3 c_p/\mu k$), Gr_D is the Grashof number, Pr is the Prandtl number, g is acceleration due to gravity, ρ is the fluid density, β is the fluid thermal expansion coefficient, ΔT is the change in temperature, c_p is the fluid specific heat capacity and μ is the dynamic viscosity of the fluid.

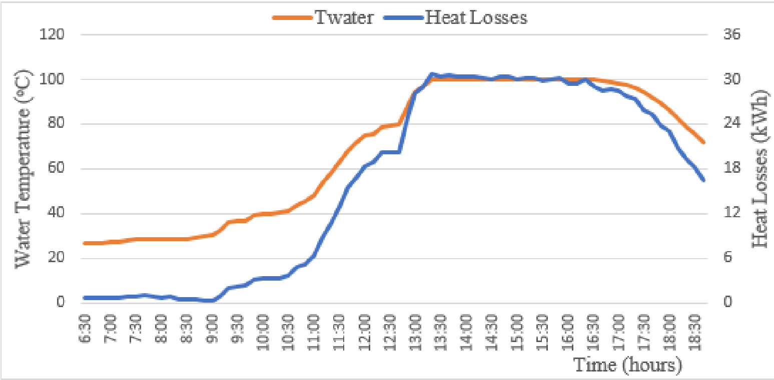
The average convection heat transfer coefficient and Nusselt number were calculated by the use of equations (6) and (7) respectively.

Table 2 gives the Nusselt number and average natural convective coefficient calculated using the experiment data and equations (6) and (7). As shown, the experimental natural convective coefficients of the; tank ranged between 9.00 and 9.53 W/m^2.k, receiver ranged between 56.58 and 57.49 W/m^2.k, descending pipe ranged from 56.39 to 58.15 W/m^2.k and ascending pipe ranged from 74.23 to 75.83 W/m^2.k for the three days examined.

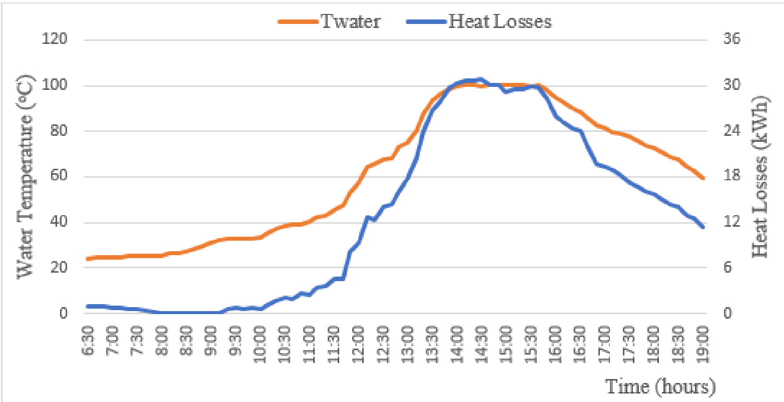
The average natural convective heat transfer coefficient for the connecting pipes shown in Table 2 were used with equations (4), (5) and experimental data to estimate the total heat losses from the pipe. The results showed that the total heat loss from the pipes per day were 201.08 KWh, 152.30 KWh and 152.72 KWh respectively for the three days examined.

Table 2. Nusselt number and natural convective coefficient (h) obtained from experimental data

Items	12th May 2019		15th May 2019		17th May 2019	
	$\overline{Nu}_D$	$\bar{h}$ (W/m²$\cdot$k)	$\overline{Nu}_D$	$\bar{h}$ (W/m²$\cdot$k)	$\overline{Nu}_D$	$\bar{h}$ (W/m²$\cdot$k)
Tank	4.62	9.53	4.36	9.00	4.47	9.23
Receiver	2.59	57.49	2.56	56.88	2.55	56.58
Descending Pipe	2.50	58.15	2.42	56.39	2.44	56.80
Ascending Pipe	2.31	75.83	2.27	74.57	2.26	74.23



(a)



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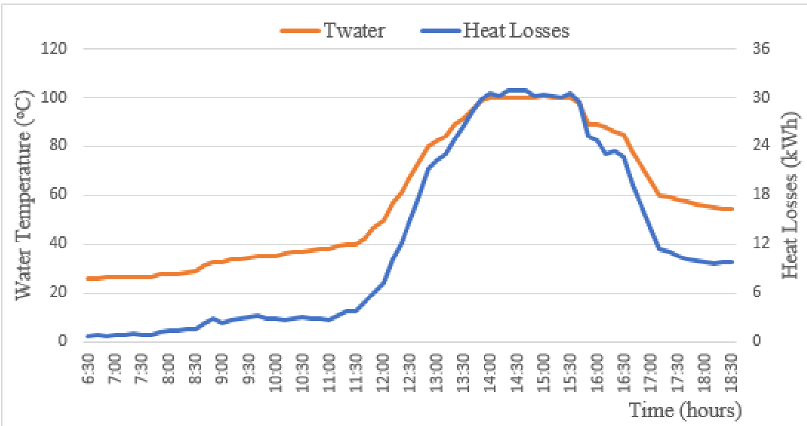

(c)

Fig. 3. Heat loss rate vs water temperature as a function of time of the day for (a) 12/05/2019 (b) 15/05/2019 and (c) 17/05/2019

Table 3. Polynomial curve fit equations for heat losses and temperature of water

Date	Equation	r^2 Value
12th May, 2019	$y = -0.0219x^2 + 3.064x + 27.626$	0.9973
15th May, 2019	$y = -0.0325x^2 + 3.3622x + 27.922$	0.9907
17th May, 2019	$y = -0.0301x^2 + 3.3466x + 25.26$	0.9965

The overall heat losses in the system were 1211.95 KWh, 974.32 KWh and 911.26 KWh respectively for 12th, 15th and 17th May, 2019. Lüpfert, et al. reported heat loss of around 300W⁄m (per receiver length) [5].

3.4 Heat Losses with Temperature of Water

The heat losses of the thermosiphon based PTC solar thermal system were compared with the temperature of the heat transfer fluid (HTF) for the 12th, 15th and 17th of May, 2019 investigation was carried out. The plots were a function of time of the day and the results are presented in Fig. 3 (a-c) respectively.

A strong positive correlation existed between the heat losses and the temperature of the heat transfer fluid (HTF), with coefficient of correlation (r) of 0.9969, 0.9901 and 0.9945 respectively for 12th, 15th and 17th May, 2019. The results show that the rate of heat loss increased as the temperature of the HTF increased until the water (HTF) attained the boiling point of 100°C for all the days examined. Maximum rate of heat loss were recorded at the steam discharging stage of the experiment. Also little changes in the heat loss rate were observed at the steam discharging stage. The rate of heat losses also decreased with decrease in the temperature of the water in the system.

Linear regression analysis was equally carried out on the heat losses and the temperature of water to specify the models that provide the best fit to the curves in their datasets. The models obtained are presented in Table 3. All the model for the heat losses and water temperature gave high coefficient of regression (r^2) of 0.9973, 0.9907 and 0.9965 respectively for the 12th, 15th and 17th May, 2019.

4. CONCLUSION

An experimental investigation of heat losses from a PTC natural circulation solar thermal system used for generating low temperature steam was carried out in this work. The results showed that heat losses from the tank were very high and constituted over 83% of the losses for all the days examined. To obtain higher thermal efficiency a better insulating material should be used or the annular space between the tank and the lagging sheet should be large to accommodate more fiberglass. It was concluded that for a PTC natural circulation solar thermal system used for direct steam generation both conductive and convective heat losses cannot be neglected in the analysis.

PUBLISHING DATA SHARING POLICY

The data that support the findings of this study (including heat balance sheet for the overall system) are available from the corresponding author upon reasonable request.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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