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Design and Testing of a solar Parabolic Trough System for Electricity Generation in Sudan

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Abstract: The performance characteristics of a parabolic trough collector system have been characterized using ASHRAE 93 standard. The standard includes collector efficiency test and the incident angle modifier test. The fabricated trough-receiver unit and its down stream system were installed in an open area at the roof of the Mechanical Engineering Department, University of Khartoum. The test result shows that the obtained thermal efficiency is 32%. The output shows that it is possible to generate electricity using parabolic trough system with a considerable efficiency and competitively with the other conventional systems due to high local solar potential through the year and availability of site areas

Keywords: *Solar radiation; parabolic trough; angle modifier; thermal; efficiency trough; angle modifier; thermal; efficiency.*

1. Introduction

Lack of electricity in Sudan has been one of the major problems at many remote areas not connected to the National Electricity Grid (NEG) due to population growth and higher living standard. These areas have abundant solar radiation all over the year. Accordingly, solar thermal electricity generation is available solution to electricity supply there. Out of many solar systems, parabolic trough collectors are the most preferable for solar steam generation which is needed to drive steam turbine/generator for electricity generation since high temperature can obtained without any series degradation in the collector efficiency [1, 2]. The parabolic trough collectors constitute a proven source of thermal energy for power generation and industrial process heat.
In this work, three models of a parabolic trough collector have been designed, constructed and tested for thermal power plant developed for electricity generation. A counter flow heat exchanger is designed and used as steam generator. A MATLAB code has been developed for analyze the proposed design of trough power plant. As the designed power plant is a combination of different components, the code consists of different subprograms to solve each individual component of the power system. At the end all these components have been run together to investigate the performance of the system.

2. System design

From the local market available materials, a parabolic trough collector with the data given in table 1 is fabricated and is shown as a photo in Figure 1. The fabricated trough-receiver unit and its down stream system were installed in an open area at the roof of the Mechanical Engineering Department, Faculty of Engineering-University of Khartoum.

Table (1). Geometrical data collector

Length	5m
Aperture	2.35m
Rim angle	72 ^{o}
Focal length	0.78m
Receiver diameter	4.67cm
Geometrical C.R	50.3
The concentrator height	0.42m

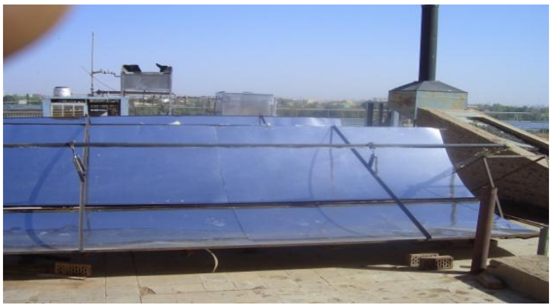


Fig (1). The developed trough model

Three models were connected in series to form solar power source of total aperture area of 35m^2. Each model consists of three fixed base parts and one movable part for rotating the parabolic mirror. The set is oriented in east-west axis to avoid tracking. The tracking takes place each day to overcome daily declination change. This orientation causes collector's end losses at early mornings and late evenings. The reflecting mirror is made of glazed stainless steel sheet and the receiver from steel galvanized pipe coated with black board paint. At the edge of the absorbing pipes, a flexible tube is used for conveyance of the heat transfer fluid. The collectors were connected with other dawn stream units to form a Rankin Power Cycle System with two major cycles', solar cycle and power cycle as illustrated by figure 2.

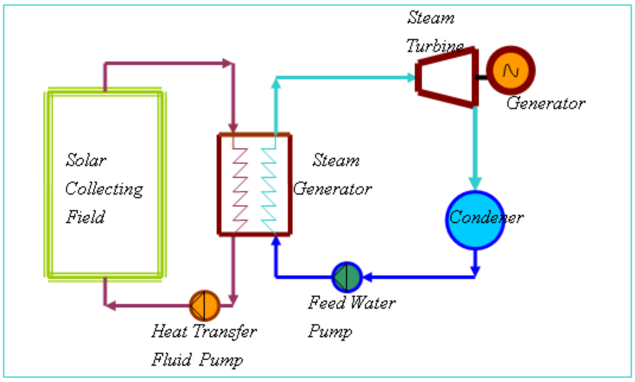
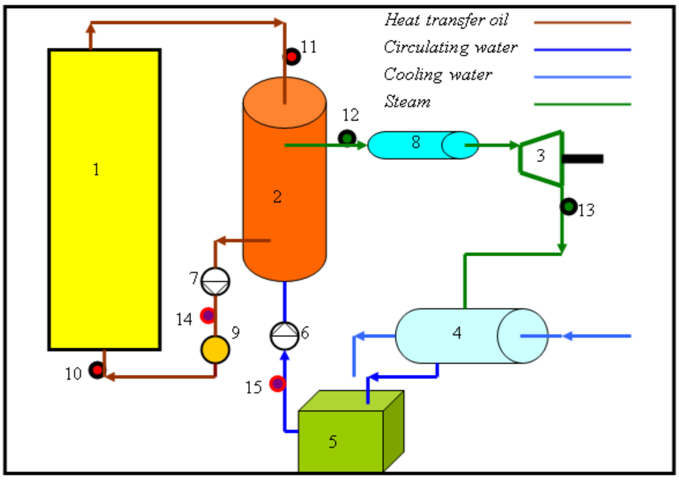


Fig (2). Solar Power Plant Process Diagram

The solar cycle has a set of three solar collectors, a heat exchanger and a pump. In this cycle the collectors collects the solar radiation and convert it into thermal energy by heat transfer fluid (HTF). The heat transfer fluid exchanges heat to the power cycle via heat exchanger. The oil feed pump circulates the fluid in the cycle. The power cycle convert the thermal energy into kinetic energy. In this cycle the pump increases the fluid pressure from the condenser pressure to enter the boiler at the boiling pressure. The steam leaves the heat exchanger at superheated state to drive the steam engine.
Due to some problems associated with the use of known heat transfer fluids and their high cost, and according to previous studies taken about the thermal properties of some vegetable oils [3], sesame seat oil is used as the heat transfer fluid. The thermal properties of the oil are [4, 5]:
Viscosity = 23x10^{-6} m^2/s
Thermal conductivity = 0.17 W/m.K
Specific heat capacity =2.47 kJ/kg.K
Density = 923 kg/m3
Flash point = 295 ^{o}C
Fire point = 340 ^{o}C

3. Experimental setup and methodology

The experimental setup for testing the collectors is shown schematically in Figure 3. As illustrated by Figure 3, with a set of temperature sensors, pressure sensors, flow meters and control valves the temperatures of the HTF and the water at inlets and outlets, the inlets and outlets pressures, the mass flow rates, the ambient temperature and the solar radiation intensity are continuously measured during the experiment



1 Solar collectors
2 Steam generator
3 Steam engine sensor
4 Condenser
5 Water tank
6 Water feed pump
7 Oil feed pump
8 Seam reservoir

9 Oil flow meter
10 Oil inlet temperature sensor
11 Oil outlet temperature sensor
12 Steam inlet pressure sensor
13 Steam outlet pressure sensor
14 Oil control valve
15 Water control valve

Figure (2). Schematic layouts of the test equipments

Baseline performance was established for the trough models. From these measured data, the collector thermal efficiency test and the incident angle modifier test were evaluated. The collector was tested according to a recognized solar standard to allow for comparison of performance with other collectors. ASHRAE 93-standard was chosen [6].

4. The thermal efficiency test

The thermal efficiency η_{th} is obtained by measuring the temperature increase of the working fluid through the receiver Δt together with the fluid properties and mass flow rates. This gives the rate of thermal energy input. Dividing this by solar radiation falling on the collector gives a measured of the collector efficiency.

$$\eta_{th} = \frac{\dot{m} c_p \Delta t}{A_a I_b} + C$$

(1)

By repeating the test for increasing Δt, a linear model of the collector efficiency can be obtained. ASHRAE 93 gives the following equation for efficiency calculation

$$\eta_{th} = \frac{Actual\ useful\ energy\ collected}{Solar\ energy\ int ersepted\ by\ absober\ area} = \frac{\int_{T_1}^{T_2} \dot{m} c_p (t_o - t_i) dT}{A_a \int_{T_1}^{T_2} I_b dT}$$

(2)

Where

T_1, T_2 = the start and finished time for the test period respectively
t_i, t_o=the inlet and outlet heat transfer fluid temperature
I_b = the beam irradiance in the plain of the collector aperture
$\dot{m}$ =heat transfer fluid flow rate
c_p=heat transfer fluid thermal capacity
A_a=collector's aperture area

From ASHRAE 93 standard, the performance of the trough collector is described by straight line as:

$$\eta_{th} = -(F_R U_L \frac{A_r}{A_a})(\frac{\Delta t_a}{I_b}) + F_R \eta_o$$

(3)

The y-axis intercept occurs at efficiency where the inlet to ambient temperature is zero [2]. Experimental work was carried according to this method. The collector efficiency η_{th} is plotted versus the collector heat loss factor $\frac{\Delta t_a}{I_b}$

An overall testing time of 11 days only in part under clear sky condition, was sufficient to extract relevant collector performance parameter set. By curve fitting, the best performance curve of the model is obtained with slope of -7.8576 and intercept of 0.6508.

Therefore the collector equation can be written as;

$$\eta_{th} = -7.8576\left(\frac{t_i - t_a}{I_b}\right) + 0.6508$$

(4)

The coefficient of determination R^2 is 0.97 indicating a good fit with the data.

From equation (3);

$\frac{A_r U_L F_R}{A_a} = 7.8576$, and $F_R \eta_o = 0.6508$.

For geometrical concentration of 7.8 (after fitting the fins), the gradient of the equation gives $U_L F_R$= 61.3 W/m^{2}K. From the optical properties of the collector and the statistical estimation of the intercept factor, the optical efficiency ($\eta_o = K_{tn}\Gamma\tau\alpha\rho$) is estimated to be 0.67 at normal angles of incident from which the heat removal factor F_R is 0.97. This in turn yields an overall heat transfer coefficient U_L of 63.2 W/m^{2}K. A result from the test performance is presented in Figure 3.

The result shows that the performance curve from this study is greatly lower than that of typical collector, (Brooks, South Africa – 60.1%) [7], (Kalogirou, 64.2%) [8], which can be ascribed to the higher thermal losses for the collector's end losses due to east-west orientation and surface imperfections (measured as the standard deviation of the surface errors) which affect the intercept factor and consequently the efficiency of the collector. Other research troughs have value close to this with values about 40% (Coventry, Australia) [9], (Ahmed Hegazy 32.1%-60.4%) [10].

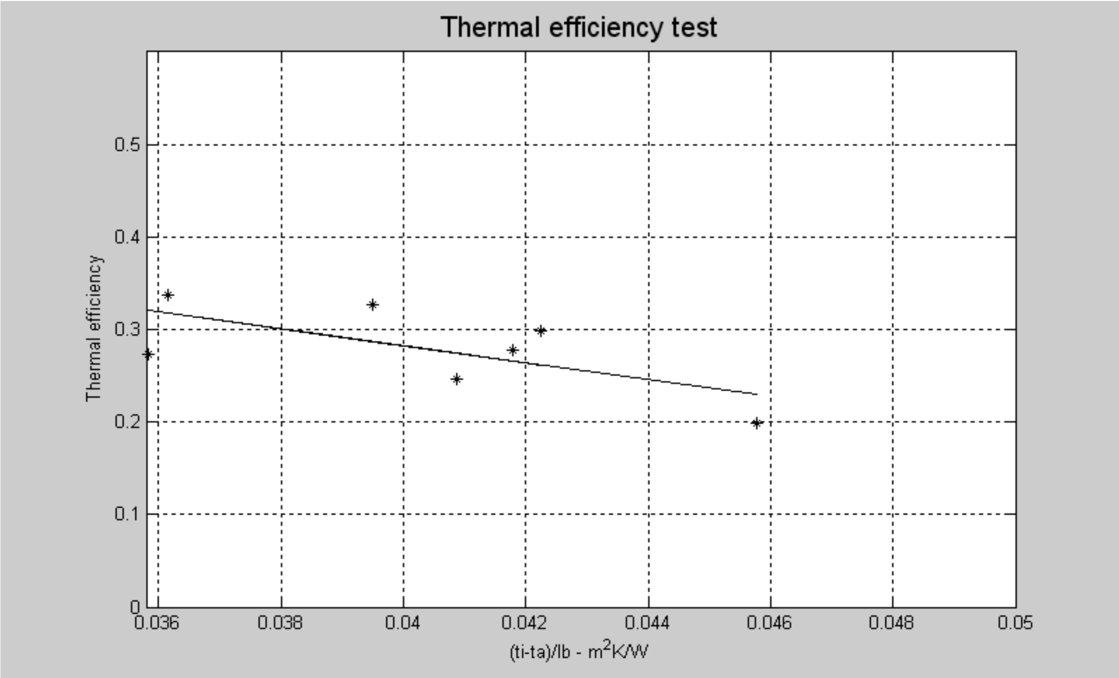


Fig (3). Thermal efficiency test result

5. Incident angle modifier test

The incident angle modifier K_t enables the performance of the collector to be predicted for solar angles of incident other than 0° [2, 6]. It determines the drop in optical efficiency due to the change in incident radiation. The tests are run essentially the same way as normal efficiency test, with PTSC performance measured for a receiver inlet temperature near the ambient air temperature. For a set value of angle of incidence θ_i, and the incident angle modifier K_t is then calculated from the equation [6]:

$$K_t = \frac{\eta_{th}}{F_R \eta_o} = \frac{\eta_{th}}{F_R[((\tau\alpha)\rho\gamma)]_n} \qquad (5)$$

This is the ratio of a thermal efficiency at specific value θi to the peak efficiency of the collector at zero incidence angles. The value of θi is calculated from the solar angles using equations [6]. ASHRAE 93 recommends that the value of θi be increased from zero to a maximum of 60oC for the incident angle modifier test and that of a total of four data points be generated with one each at 0o, 30o, 45o, and 60o [6, 11]. The equation is modeled by a 2nd order polynomial according to ASHRAE 93 standard [6, 11]. Regression analysis provided the following equations for K_t:

(1) For θ_i in degrees;

$$K_t = (3.337x10^{-6})\theta_i^2 - (0.14)\theta_i + 0.8938$$

(6)

(2) For θ_i in radian;

$$K_t = 0.011\theta_i^2 - 0.8012\theta_i + 0.8939$$

(7)

(3) For K_t as a function of (1/cos θ_i);

$$K_t = 0.431\left(\frac{1}{\cos\theta_i}\right)^2 - 2.132\left(\frac{1}{\cos\theta_i}\right) + 2.451$$

(8)

The coefficient of determination R^2 is 0.8637.

The test result performance is presented in Figure 4.

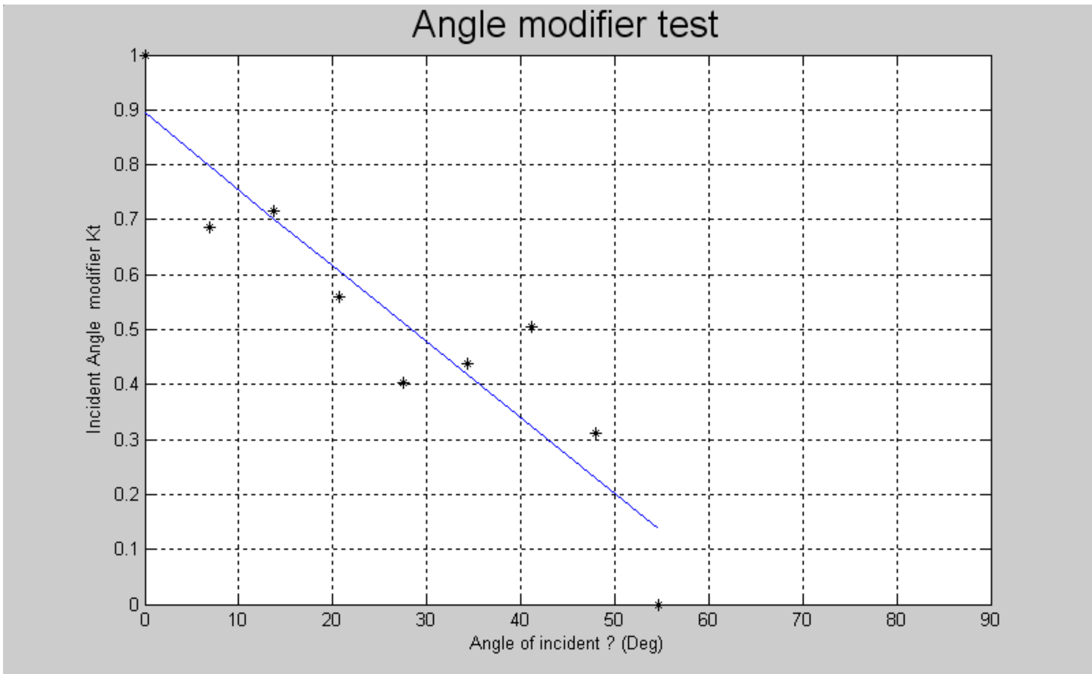


Fig (4). Angle modifier test results – for θ_i in degrees

From figure 4 the calculated value of the maximum K_t is 0.90 at zero incidence angles and 0.14 at maximum incidence angle of 60°. The factors affect the performance of the parabolic trough collector with increasing θ_i the geometrical reduction in irradiance falling on aperture as θ_i increases, called the cosine effect and change in optical efficiency as light interact differently with the reflective surface of the collector [6].

6. Thermal losses tests

These tests are not a part of ASHRAE 93 standard method. The test is carried according to the European Standard EN 1297S-2 [12]. The aim of the tests is to determining steady state heat loss as a function of day time. Only the measured obtained during clear weather were considered. The heat energy gained by the heat transfer fluid was obtained as:

$$\dot{Q} = \dot{m}c_p(T_i - T_o) \qquad (10)$$

Where:-

$\dot{m}$ = HTF mass flow rate kg/s
c_p = Thermal capacity of the HTF J/kg.K
T_i = HTF inlet temperature K
T_o = HTF outlet temperature K

From the measured data, the thermal efficiency of the collector was calculated. The test was repeated from 14 April to 13 May 2007 and from 11 May to 10 July 2010. With MATLAB code, the calculated collector efficiency test result is presented in Figure 5 as efficiency against solar time for oil flow rate of 0.43 kg/s. The Figure shows clearly that the higher values of the efficiency are obtained at the highest intensity of the solar radiation. Table 2 gives the average and maximum efficiencies for 4 most clear days.

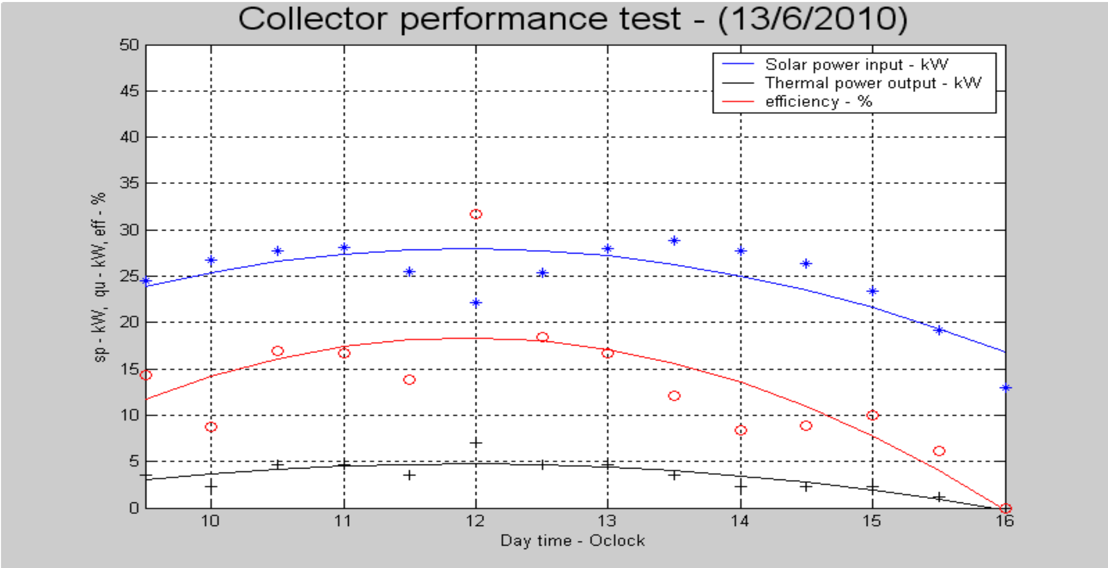


Fig (5). Output thermal power and solar irradiance chart for 13-6-2010

Table (2). Estimated thermal efficiencies

Date	Average thermal efficiency	Maximum efficiency
13-6-2010	0.14	0.32
14-6-2010	0.12	0.27
15-6-2010	0.06	0.09
16-6-2010	0.14	0.31

7. Conclusion

The performance characteristics of a locally manufactured parabolic trough collector are presented. The test was carried according to ASHRAE 93 standard. The testing task includes sequential tests of solar efficiency and incidence angle modifier. The performance obtained by testing the collector is greatly lower than that of typical collectors. This can be improved by:

1. a precise tracking system is needed with north-south orientation
2. increasing construction accuracy by using more advance equipments

The physical output shows that it is possible to generate electricity using parabolic trough collectors. However the average thermal efficiency is 32% which is fairly acceptable, considering that it is the first attempt to manufacture such collector locally**.** Adding evacuated glass envelop around the receiver will improve the performance that make the model competitive with the other power sources.

Nomenclature

A_a the effective aperture area of the collector, m^2
A_r surface area of the receiver or absorber, m^2
c_p specific heat capacity, *J/kg.K*
F_R heat remover factor
I_b the beam irradiance in the plain of the collector aperture, W/m^2
I_d direct solar beam, W/m^2
K_t incident angle modifier

$\dot{m}$ Mass flow rate, *kg/s*
$\dot{Q}$ the thermal energy added to the system, *J*
T_a ambient air temperature, ^oK
T_{in} Temperature of the heat transfer fluid entering the absorber, K
T_{out} Temperature of the heat transfer fluid leaving the absorber, *K*
t_i the tube inlet temperature, ^oC
t_o the tube outlet temperature, ^oC
U overall heat transfer coefficient $W/m^2.K$
α absorbtivity of the surface
Γ fraction of the reflected energy on a receiver
γ Surface azimuth angle, *degree*
ΔT mean temperature difference
ε emittance of the absorber surface.
η_o optical efficiency
η_{th} thermal efficiency
θ_i Angle of incidence, *degree*
ρ collectors' surface reflectivity
τ transmittance of the glass

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