FABRICATION AND CHARACTERISZATION OF A PROTOTYPE PARABOLIC TROUGH SOLAR CONCENTRATORS FOR STEAM PRODUCTION

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Abstract— In this paper, the potential for a solar-thermal system to produce steam has been studied. Three parabolic trough solar concentrators (PTSCs) for roof tops of similar dimension were designed, fabricated, characterized and their efficiencies compared when closed and when open. The PTSCs´ were made of appropriate materials and were manually tracked. They were designed with principal focus at 0.4 m so that the receiver heat loss was minimized by covering the collectors with glass which was 0.0025 m in thickness. The dimensions of the collector were aperture width: 1.2 m, Collector length 5.8 m, aperture area: 6.95 m^2. The absorber pipe was a copper tube which carried water as the heat transfer fluid. The concentration ratio of the solar concentrators was 128. The concentrator testing was carried out for each of the concentrators. The maximum temperature of steam obtained was 248.3^oC while average temperature of steam was produced was 150^oC. When closed their efficiencies were: Aluminium sheet reflector PTSC; 55.52 %, Car solar reflector PTSC; 54.65 % and Aluminium foil reflector PTSC; 51.29 %. The open solar concentrator efficiencies were 32.38 %, 34.45 % and 27.74 % respectively. The results obtained show that production of power using the sun flux is a viable undertaking. The concentrators can be used to provide power to remote areas which are away from the gridlines. This will make power readily available to the marginalized rural poor people. Improvement of the tracking system and optical efficiency can improve the efficiencies of the fabricated concentrator systems.

Keywords—parabolic trough concentrator, solar-thermal, transmittance-absorptance product, thermal and optical efficiency..

I. INTRODUCTION

SOLAR energy is a renewable source of energy. Its use does not contribute to emissions of greenhouse gases and other pollutants to the environment. It is sustainable since it cannot be depleted in a time relevant to the human race. Kenya has very high solar irradiance in most of arid and semi- arid regions. Therefore her potential for deployment of solar collectors for thermal steam production is a viable commercial undertaking. However research efforts to that direction are still wanting. The aim of this study was to use appropriate materials in design, fabrication and characterization of local solar thermal steam producing systems for power generation. Solar collectors transform short wave radiation of the range 0.29 µm - 2.5 µm into long wave radiation and trap this energy in form of heat which is transferred into a heat storage vault by the heat transfer fluid.

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Parabolic trough power plants are the only type of solar thermal power plant technology with existing commercial operating systems (Wikipedia, 2009). A parabolic reflector reflects all the rays that are parallel to its principal axis to a point focus. When this parabola is extrapolated in three dimensions, a parabolic trough is generated, whose focus lies along the axis of the trough. In terms of capacity solar genix's has produced collector modules which are commercially viable and produce 176 Kw of peak energy for dual functions factory roof and for solar heating with an efficiency of 56 % (Cleaveland, 2005). In Africa, a solar thermal plant in Cairo, based on 1,900 m^2 of parabolic trough collector provides steam for pharmaceutical plant, el Nasr project (Ecoworld, 2009). Globally parabolic trough power plants technology with existing commercial operating systems include Nevada solar one which operates on a 250 acre site in Nevada desert and generates 134 MW of power per year. A larger solar based facility already exists in Mojave Desert in USA, generates 354 MW of solar energy power. In Spain Torresol and Arcosol have a 50 MW parabolic trough based plants located in Seville Cadiz. Other Spanish plants using this technology are Andasol 1, 2, 3 in Granada province (Ecoworld, 2009).

II. MATERIALS AND METHODS

A. FABRICATION OF SOLAR CONCETRATOR TROUGH

Angle iron beams of aperture width 1.2 m were bent into parabola with a focus at 0.4 m using the formula the common equation for plotting parabola.

Black sheets were folded and welded into angle iron beams on the outside and the ends closed. The edges of the sheets were folded to provide a rail which was lined with rubber sheets so that the glass covers slid smoothly. A rubber sheet was put between glass and metal to prevent the glass from cracking. The length of the collector was 5.8 m and an aperture width of 1.2 m. A manual tracker system was fabricated using a general gears of gear ratio 1:30 and a class B black pipe of external diameter 0.008 m and internal diameter 0.003 m. This pipe was fitted onto one of the bigger diameter slots of the gears while the other slot a winch of radius 0.04 m was fitted to effect minute turning of the collector on the North – South axis. The turning would be affected when a 0.005 m by

0.003 m pin that was placed at the plane of aperture width would cast a shadow. The collector was laminated with three appropriate materials each in its turn i.e. aluminium sheet, car solar reflector and aluminum foil each in its turn. The receiver was a cylindrical copper pipe painted black with appropriate black paint. The paint coat was kept as thin as possible so that there was minimum resistance of flow of heat through the coat to the pipe and to the heat transfer fluid. The collector was covered with a 0.0025 m thick glass cover.

The fabricated collector parameters were:

Aperture area = 6.95 m^2

Collector area =15.75 m^2

Aperture width = 1.2 m

Focal length = 0.4 m

Collector length = 5.8 m

Outer diameter of absorber pipe = 0.025 m

Inner diameter of absorber pipe = 0.002 m

Concentration ratio = 128

Fig. 1 shows the setup for the fabricated trough solar concentrator used for production of solar thermal steam.

Fig. 1. Fabricated prototype parabolic trough solar concentrator for steam production.

The parabolic trough was laminated with aluminium sheet, car solar reflector and aluminium foil appropriate materials each in its turn.

B. COLLECTOR TEST

To perform steady state collector testing: a heat exchanger and a 6 kW heater were fabricated. The testing was done at 2.0 m surface above the ground since the collector was designed for rooftops. The inlet temperatures used were 90^o C, 140^o C, 190^o C and 240 ^{o}C. The test times were taken systematically about solar noon, to give a total of sixteen data points.

The test period contained a time of fifteen minutes with desired fluid temperature which was followed by a steady state period of fifteen minutes. The solar irradiance and intercept factor were determined by calorimetric method.

The pressure gauges gave the values of pressure drop across the collector and the thermocouples at both ends of collector mixing joints were directly read to obtain the fluid inlet temperature, T_1 and the fluid outlet temperature, T_2.

Direct solar irradiance from the sun was computed from calorimetric method by placing a calorimeter with water and a thermocouple at the plane of collector aperture and continuously recording temperature changes.

C. MEASUREMENT OF INTERCEPT FACTOR

The amount of solar radiation that was intercepted by the receiver was determined by calorimetric method using the equation 2. Four calorimeters with measured amount of water were placed at equidistant points along the receiver and the solar thermal heat they absorbed was determined. The ratio of area of absorbing surface in the direction of the sun to the area of receiver superimposed was also evaluated using

$$\alpha\tau A I_{bc} = mc\frac{\Delta\theta}{\Delta t} \quad \ldots\ldots\ldots 1$$

Where:-

$\alpha\,\tau$ = beam transmittance-absorptance product, m = mass of water, kg and density of water = 1000 kg/m^3, c = specific heat capacity of water J / kg ^{O}C, 4200 J/kg ^{o}C. $\frac{\Delta\theta}{\Delta t}$ = temperature change per unit time (^{o}C/s)and A = aperture area and I_b = beam irradiance.

For a black body α = 1. The absorber was made of copper tube of diameter 0.0025 m whose thermal conductivity was taken as 3.9×10^2 J/kg/K and absorptance = 0.9 (Hanssan, 1972).

D. AVERAGE HEAT ABSORBED BY RECEIVER

To determine the average heat that was absorbed by the receiver per unit time, it was partially filled with water and one end was closed using a control valve and the other end leading to a condenser was left open. The condenser was made of a plastic cylinder in which a coiled copper tube was fixed. The plastic cylinder was then filled with cold water to condense the steam. The setup was operated until a steady state that was shown by water boiling in the receiver was observed. The mass of steam condensed was measured using a beam balance and the time it took to obtain the steam was measured using a stop watch. The temperature at which steam was obtained was also recorded.

E. MEASUREMENT OF SOLAR IRRADIANCE

A copper calorimeter of radius 0.03 m was insulated on the outside using aluminium foil and on the inside it was painted black. 0.05 kg of water was poured inside and after settling the probe of thermocouple was used to read initial temperatures of water in the calorimeter and recorded. This calorimeter was left exposed at collector plane of the collector at noon and the time of exposure recorded. The final temperature reached by water in calorimeter was recorded and Equation 2 was used to determine solar irradiance.

The measurement of solar power intensity was measured throughout the day by recording the temperature changes of the water in the calorimeter every twenty minutes. The calorimeter was placed at the middle of collector plane and secured with a thin cotton thread.

F. THERMAL EFFICIENCY

The efficiency of the concentrators was obtained from equation 2,
which similar to the expression shown in equation 3.

$$\text{Efficiency } \eta = \frac{\text{heat out put}}{\text{heat in put}} = \frac{Q_u}{A_c A_b} \ldots\ldots\ldots 2$$

Hottel–Whillier–Bliss equation was also used to find efficiency of the fabricated concentrators as shown in equation 3

$$F_R \frac{A_a}{A_c} [I_b \rho \gamma < \tau\alpha > - U_L \left(\frac{T_m - T_a}{I_b}\right) \ldots\ldots\ldots 3$$

(Rai, 1987)

The slope of the efficiency graph represents the quantity:

$$-F_R U_L \frac{A_a}{A_c}$$

The optical efficiency of the plotted graphs was given by:

$F_R \rho \gamma < \tau\alpha >$ (John, *et al*, 1991)

G. TRANSMITTANCE - ABSORPTANCE PRODUCT $< \alpha\tau >$

It refers to the ratio of flux absorbed in the receiver to the one incident on cover system, $\alpha\tau$. The transmittance of the cover system was found by use of Spectro- 320 analyzer and the absorptance of copper was taken from 0.9 (Hanson, 1972). The product of transmittance-absorptance was determined from equation 4 as shown.

$$< \tau\alpha > = \frac{\tau\alpha}{1-\rho_d(1-\alpha_g)} \ldots\ldots\ldots 4$$

(Rai, 1987)

ρ_d represents the reflectance of cover system for diffuse radiation incident from bottom side. $\rho_d = 0.15$, $\alpha_g = 0.3$ for single glass cover, τ = transmittance of cover, α = absorptance of receiver. (Rai, 1987)

Collector heat removal factor F_R is calculated from experimental data as shown in equation 5.

$$F_R = \frac{\dot{m} c_p}{A_r U_L} [1 - e^{-\frac{F' A_r A_a}{\dot{m} CP}}] \ldots\ldots\ldots 5$$

Collector efficiency factor F ' was also obtained from equation 6

$$q_u = F' \frac{A_a}{L} [S - \frac{A_r}{A_a} U_l (T_m - T_a) \ldots\ldots\ldots 6$$

(John, et al, 1991)

Collector flow factor was calculated from:

$$F'' = \frac{F_R}{F'} \ldots\ldots\ldots 7$$

H. OPTICAL EFFICIENCY

Optical efficiency η_{opt} was calculated from equation 8 as

$$\eta_{opt} = \rho \Upsilon < \tau\alpha > \ldots\ldots\ldots 8$$

(Kalogirou, 1997)

III. RESULTS AND DISCUSSIONS

A. INTERCEPT FACTOR

Intercept factor obtained at noon was calculated as follows: The surface area of the calorimeter was determined as:- 0.010995 m^2 while the surface area of absorber superimposed by calorimeter was determined as: 5.498×10^{-4} m^2. Surface area ratio was found to be 1: 19. Solar thermal energy collected after exposing calorimeter to the sun was 1.299×10^3 W/m^2. Intercept factor was determined as a ratio of surface area of calorimeter to that of absorber in relation to energy collected and was found to be

$$\frac{1.229 \times 10^3}{19} = 649.84 \ W/m^2$$

The average values of thermal energy collected was obtained as: 649.84 W/m^2, 591 W/m^2, 658.7 W/m^2 and 633.1 W/m^2 for the four calorimeters that were used. These were averaged and the intercept factor obtained was 627.8 W/m^2. This value compared well with the value found by integration of amount of solar energy intercepted by receiver at four equidistant points on the receiver that was obtained as 687.3 W/m^2.

B. AVERAGE HEAT ABSORBED BY ABSORBER

To compute amount of heat that was absorbed by receiver per unit time, data obtained from section 2.7 i.e. mass of steam: - 0.0778 kg in 198 s at a temperature of 150 °C, was used. Average power that was absorbed was found to be 886.9 Wm^{-2}. The insolation that was measured by calorimetric method was found to be 852.7 Wm^{-2}. The collector aperture was 6.96 m^2. Power collected in one hour neglecting external losses would be 3.19×10^6 W. If one acre was occupied by parabolic trough solar concentrator then the power that would be collected in one hour would be 9.27×10^6 MW of steam at 150 °C. The N-S tacking mode ensured that the solar image at the absorber was not very much enlarged in the morning and in the evening.

C. DETERMINATION OF SOLAR IRRADIANCE

The heat gain was calculated from the equation given by:

$$Q = mc_p \frac{\Delta\theta}{\Delta t}$$

Where Q = quantity of heat gained, m = mass of heat transfer fluid (Twidel *et al.*, 1986).

The initial temperature of water that was in the copper calorimeter was 21.51 ^oC and its time of exposure to the sun was 1200 s. The final temperature recorded for the water in the calorimeter was 35.26 ^oC.

Rate of heat gain = $\frac{0.05 \times 4200 \times 4.122}{20 \times 60}$ = 2.4 J/s

Area of calorimeter base is given by:-

$\pi r^2 = 2.82 \times 10^{-3}\ m^2, \pi = 3.142$

Solar power per m^2 was evaluated as:-

$$\frac{2.4045}{2.82 \times 10^{-3}} = 852.7\ W/m^2$$

D. THERMAL EFFICIENCY

Temperature rise across the collector (T_1-T_2) during testing was directly read from thermocouple thermometers and the pressure drop values obtained from pressure gauges. Direct beam from the direction of the solar disc was computed by calorimetric method.

Collector efficiencies for the fabricated solar concentrators were graphically obtained by use of Hottel-Whillier-Bliss equation given as equation 3.

The collectors were tested under steady irradiation; intercept factor and fluid flow rate such that $F_R\rho\gamma\ (\tau\alpha)$ and U_l were considered constant.

Graphs of the collector efficiency as a function of $(T_m - T_a/I_b)\ ^oCm^2$ for the open and closed aluminium sheet PTSC, car solar reflector PTSC and aluminium foil PTSC studied were as shown in Fig.s 2-4.

The suitability of aluminium sheet as a reflecting system for thermal solar steam generating system was studied using Fig. 1.

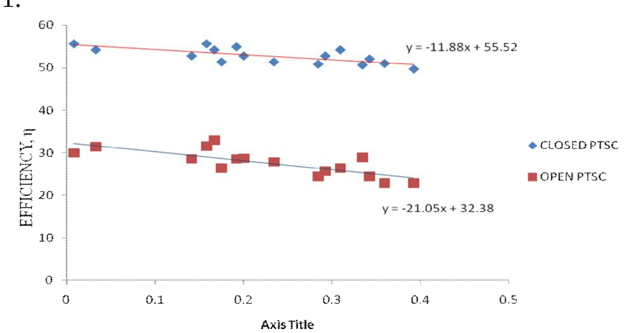

Fig. 2. Efficiency graph for aluminium sheet solar concentrator when closed with glass and when open.

The experimental parameters under which the efficiency testing was carried out were:

$\dot{m}$ = 8.8 kg/s, I_b = 752 W/m^2 and ΔPa = 921000 Pa for the closed PTSC and for the open PTSC the parameters were $\dot{m}$ = 3.98 kg/s, I_b = 749.3 W/m^2 and ΔPa = 372000 Pa. For this analysis average T_a = 21.8, T_m = 152.2 ^oC.

Efficiency was plotted against $(T_m - T_a/I_b)\ ^oCm^2$ where T_m was the average temperature for the fluid outlet temperature and inlet temperatures. Efficiency varied linearly with $\frac{T_m - T_a}{I_b}$.

Most useful energy gain was available for collection when solar power intensity is high.

As shown in the graphs instantaneous efficiencies were obtained and plotted as a function of $(T_m - T_a/I_b)\ ^oCm^2$

It was assumed that U_LF_R and $\alpha\ \tau$ were constants and U_L, is the overall heat loss coefficient. For the open concentrator efficiency by use of experimental variables was 35.73 %, and the experimental efficiency was 32.38 % from graph 1. F_R = 0.9 Where F_R – collector heat removal factor, γ – Intercept factor, ρ – Reflectance and $< \tau\alpha >$ - Transmittance absorptance product.

U_L (overall heat loss) was found to be a weak function of temperature and corresponded to F_R as seen from energy balance equation 3.

The scatter in the data results from variations of relative proportions of beam proportion of solar radiation, temperature dependency, wind effects and variations in angle of incidence. The value of efficiency is based on effective aperture area, A_a.

From the graph efficiency of aluminium sheet concentrator was found as 55.52 % and 56.8% by use of experimental variables. The heat loss coefficient for the closed concentrator was = -11.88 W/m^2 while the heat loss coefficient from graph 1 of the open concentrator was = -21.05 W/m^2. This depicts a serious loss in heat for open solar concentrator. For this reason thermal efficiency of the open concentrator is low since it losses heat by air convections from collector and sky radiation. The efficiency of the closed collector was higher because the collector was covered and heat losses were minimal.

The value of efficiency obtained by experiment was lower than the one obtained with use of experimental variables. This was because during performance of the fabricated aluminium sheet concentrator environmental factors such as wind contributed to more heat losses lowering the capacity of the concentrator to utilize the solar flux.

The transmittance absorptance product for the glass covered concentrator was found to be 0.8 from equation 4. This was because a stagnant air mass could not be established around the receiver for the open concentrator due convection currents established by air flow. These caused heat losses that made the open thermal collector to have low thermal efficiency and hence poor solar thermal energy collection

Emissivity of the cover system (glass) was calculated to be 0.17

Collector heat removal factor F_R was calculated from experimental data and found as 0.9 for the closed concentrator and 0.53 for the open concentrator from equation 5.

Collector flow factor was obtained as 1.5 for the closed concentrator and 0.82 for the open concentrator from equation 7. Collector efficiency factor for the closed concentrator was 0.64 and for the open concentrator 0.48 as obtained from equation 6. The optical efficiency of the aluminium sheet solar concentrator was determined as 0.53 from equation 8.

The study of Car solar reflector material for its suitability for use as a reflecting system for thermal solar steam generation system gave the efficiencies for the closed and open concentrators as shown by Fig. 3.

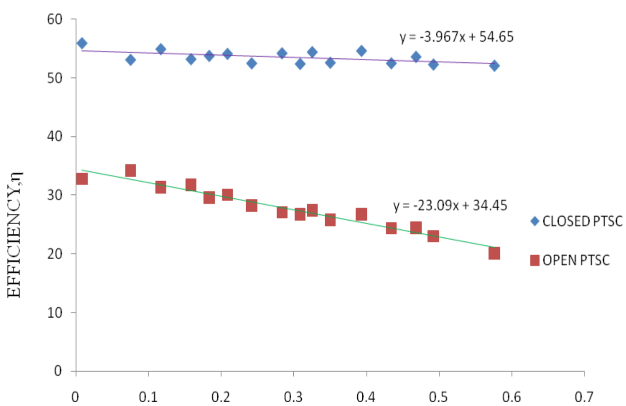

Fig.3. Efficiency graph for Car solar reflector solar concentrator when covered with glass and when open.

The experimental parameters under which concentrator

$T_m - T_a / I_b$ ($^oCm^2/W$)

g/s, I_b = 752.1 W/m^2 and ΔPa = 870000 Pa for closed PTSC and $\dot{m}$ = 4.78 kg/s, I_b = 803.5 W/m^2 and ΔPa =463000 Pa for the open solar concentrator.

The average parameters used in instantaneous efficiency analysis were; T_m = 210 ^oC, T_a = 22.7 ^oC and average solar irradiation is as shown above. Efficiency was calculated as 54.9 % using equation 3 and by use of graph 2, it was obtained as 54.7 % .

The value of efficiency obtained by experiment was lower than the one obtained by of calculation. This was because during the operation of the concentrator, environmental factors increased the heat losses hence lowering the efficiency.

The transmittance absorptance product was obtained from equation 4 as 0.78

The heat removal factor F_R was obtained as 0.87 for closed PTSC and 0.88 for the open PTSC from equation 5.

The collector efficiency factor F' was obtained as 0.64 for closed PTSC and 0.42 for the open PTSC from equation 6.

The collector flow factor for closed car solar reflector PTSC was obtained as 1.42 while for the open PTSC it was 0.84 as evaluated from equation 7.

Collector flow factor was determined as 1.5 from equation 7.

The slope for the closed concentrator was – 3.967 W/m^2 while for open concentrator it was – 23.09 W/m^2. Therefore the open concentrator was losing more than half of the solar thermal energy that the closed concentrator was able to absorb.

Optical efficiency was calculated as 0.56 from equation 8 hence much energy was being lost in form of diffuse radiation.

The car solar reflector collector also had a lower operational efficiency when open due excessive heat losses. The heat loss coefficient obtained from theoretical data was lower because it was not influenced by environmental factors.

The study of aluminium foil material for its suitability for use as a reflecting system for a thermal solar steam generating system was carried out as shown by Fig. 4.

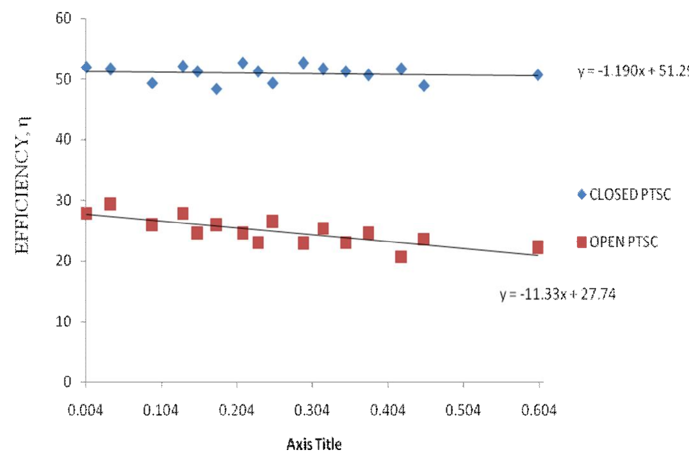

Fig 4. Efficiency graph for Aluminium foil solar concentrator when covered with glass and when open

The experimental parameters during testing for the closed concentrator were: $\dot{m}$ = 7.8 kg/s, I_b = 837.4 W/m^2 and ΔPa =745000 Pa and when open the test parameters were: $\dot{m}$ = 4.08 kg/s, I_b = 850 W/m^2 and ΔPa = 411000 Pa

In this analysis the average parameters that were used to determine efficiency of the fabricated concentrator were; T_m = 198 ^oC, T_a = 22.5 ^oC and solar power intensity was as given above. The instantaneous efficiency obtained from equation 3 was 51.29 % while for the open concentrator the efficiency obtained was 27.74 %. The efficiency that was obtained by experimental variables were 53.5 % and 28.7 % respectively.

The value of efficiency obtained by experiment was lower than the value obtained by use of data variables. During performance of the concentrator energy is lost in form of heat by convection especially when wind speed increases. The losses contribute to reduced tor to efficiency. The heat loss coefficient for the closed PTSC was – 1.190 W/m^2 while the one for open concentrator was – 11.33 W/m^2 again indicating serious heat losses that make the concentrator to deliver limited solar thermal energy.

The transmittance absorptance product, $< \tau\alpha >$ of fabricated car solar reflector was obtained as 0.78 from equation 4.

Collector heat removal factor F_R was obtained equation 5 as 0.85 for the closed concentrator and 0.57 for the open concentrator.

The collector efficiency factor F' was obtained as 0.56 from the equation 6 for the closed

Collector flow factor F" was obtained as 1.52 for the closed concentrator and 0.73 for the open concentrator from equation 7.

Optical efficiency was obtained as 0.47 from equation 8.

The efficiency graphs for the appropriate materials PTSC s' when closed with glass cover were Aluminium sheet PTSC; 55.52 %, Car solar reflector PTSC; 54.65 % and aluminium foil PTSC; 51.296 % and when open 32.38 %, 34.45 % and 27.74 % respectively. It is also evident that when the operating temperatures of collector increase the heat losses also increase. Comparison for the tendencies of efficiency heat loss for closed and open collectors shows that a closed collector minimized heat losses by conduction and convection to a large extent. Some radiation absorbed by the glass cover slightly raised the temperature of the cover hence reducing rate of upward heat loss from the receiver. This reduction in the losses has an effect in producing some increase in transmittance i.e. improves absorptivity - transmittance product $<\propto \tau>$. Reduction of heat losses by introducing covers increased efficiency of each concentrator.

TABLE I
UNITS FOR MAGNETIC PROPERTIES

Reflector	ΔP (Pa)	$\dot{m}$ (kg/s)	I_{bav} (W/m^2)
Aluminium sheet	921000	8.8	752
Car solar reflector	870000	8.0	752.1
Aluminium foil	745000	7.8	837.4

TABLE II
UNITS FOR MAGNETIC PROPERTIES

Reflector	ΔP(Pa)	$\dot{m}$ (kg/s)	I_{bav} (W/m^2)
Aluminium sheet	372000	3.98	749.3
Car solar reflector	463000	4.78	803.5
Aluminium foil	411000	4.08	850

E. COLLECTOR FLOW FACTOR

PTSC	CLOSED CONCENTRATOR	OPEN CONCENTRATOR
Aluminium sheet	1.50	1.12
Car solar reflector	1.42	0.84
Aluminium foil	1.52	0.73

The characteristics of the fabricated prototype parabolic trough solar concentrators were as shown in the tables III-V, which played a role in the design of the collectors.

TABLE III
UNITS FOR MAGNETIC PROPERTIES

TABLE IV
UNITS FOR MAGNETIC PROPERTIES

PTSC	CLOSED CONCENTRATOR	OPEN CONCENTRATOR
Aluminium sheet	0.90	0.88
Car solar reflector	0.53	0.86
Aluminium foil	0.56	0.81

TABLE V
UNITS FOR MAGNETIC PROPERTIES

PTSC	CLOSED CONCENTRATOR	OPEN CONCENTRATOR
Aluminium sheet	0.64	0.48
Car solar reflector	0.60	0.40
Aluminium foil	0.56	0.34

IV. CONCLUSIONS AND RECOMMENDATIONS

In this paper a study was made to enhance Kenyan research in solar thermal steam generation for steam production by use of appropriate materials. The efficiencies for the closed PTSC were as follows: Aluminium sheet PTSC, 55.52 %, Car solar reflector 54.65 % Aluminium foil, 51.29 % which is lower than the efficiencies of Luz collector, 68 % , Euro trough, 65.2 % (Suhas, 1990) and Sky fuel parabolic trough, 73 % (Wikipedia, 2011). This is due to evacuation of absorber tube surroundings and automatic tracking systems that they use. The reflectance of the fabricated PTSC was as follows: aluminum sheet PTSC 0.83; car solar reflector PTSC 0.81; Aluminium foil PTSC 0.78; Aluminium for the Luz collector, 0.94 and Euro trough, 0.96 (John *et al*, 1991). The difference is as a result of the surface treatment of the optical systems, which improves their performance. The glass cover transmitivity of the 0.0025 m glass that was used in fabricated PTSC was 0.8 while the Luz collector has 0.965 and the Euro trough has 0.95. This is because they use selective coatings. The efficiencies observed for closed prototype parabolic trough solar concentrators demonstrates that this technology with appropriate reflector systems can produce solar steam that is hot enough for solar thermal conversion power systems. This can be achieved by use automatic tracking system and smoother reflecting surfaces. In this case higher temperatures and higher efficiencies would be realized. On the other hand use of open parabolic trough systems, where the absorber is exposed, led to high energy losses resulting to low operating efficiencies of the concentrating collectors made from various materials. Enclosing the receiver inside a transparent medium and evacuating it and also surface treatment of receiver as evident with the Euro trough, sky fuel trough, Luz collectors' e.t.c. would significantly improve the efficiencies of the fabricated solar concentrators.

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