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Research Paper

Analysis to Determine the Thermodynamic Performance of a Flat Plate Solar Collector

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Abstract: This paper presents the thermal performance of solar flat plate collector considering various ambient, weather and design conditions of the site under review (Benin City, Edo State) which are 27 °C, rainy and dry season, 6.5438° as the latitude respectively. Thermodynamic analysis was carried out considering the ambient and boundary conditions. The result shows that the outlet water temperatures were dependent on the weather condition (solar radiation intensity, cloud cover) with outlet water temperature of 61^oC and 328.36W as Energy gained of the collector obtained. This shows that the use of this flat plate solar collector will be viable for domestic heating application.

Keywords: *Thermodynamic performance, flat plate, collector, heat transfer,*

Introduction

The environmental issues and the restrictions on the usage of fossil fuels around the world have made the use of renewable energy sources necessary. Some of the more popular renewable energy sources are solar, wind, hydro, biofuels, and geothermal. This study focuses on solar renewable energy.

An attractive aspect of solar energy is that it can be used to power small applications. It is cheaper to use solar on a large scale, but it is still very easy to implement on a small scale. There are many advantages of solar energy utilization, thereby leading to a substantial development in this energy sector (Gustafson, 2013), some of the advantages are the supply of clean energy without polluting the environment and an ability to replenish itself after use. On the earth, surface solar radiation is only effectively available in hours of clear sky weather. Nielsen (2005) gave an expression for determination of the global horizontal solar radiation intensity in nth hours of the day. Bena and Fuller (2002), Sharma et al (2009) and Iloeje (1993) indicated that solar energy trapped by solar dryers is now commonly used globally in drying and preservation of agricultural products.

Solar radiation passes through a transparent cover and impinges on the blackened absorber surface of high absorptivity, a large portion of this energy is absorbed by the plate and then transferred to the transport medium in the fluid tubes to be carried away for storage or use. The underside of the absorber plate and the side of casing are well insulated to reduce conduction losses. The liquid tubes can be welded to the absorbing plate, or they can be an integral part of the plate. The liquid tubes are connected at both ends by large diameter header tubes (Kalogirou, 2004).

The transparent cover (glazing) is used to reduce convection losses from the absorber plate through the restraint of the stagnant air layer between the absorber plate and the glass. It also reduces radiation losses from the collector as the glass is transparent to the short-wave radiation received by the sun but it is nearly opaque to long-wave thermal radiation emitted by the absorber plate (greenhouse effect). The glazing with low iron content has a relatively high transmittance for solar radiation 11 (approximately 0.85–0.90 at normal incidence) but its transmittance is essentially zero for the long wave thermal radiation (5.0–50 mm) emitted by sun-heated surfaces (Kalogirou, 2003).

A theoretical and computational method was adopted in this study to investigate the thermodynamic performance of a solar flat plate collector, considering Benin City, Edo State as the site location.

Materials and Method

The following initial conditions were obtained as considerations for the collector system.

i. Steady state system.

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ii. 26.1 ^{O}C daily average water inlet temperature (Relates to ambient temperature gotten from National Centre for Energy and Environment (NCEE), University of Benin, Edo State Nigeria).
iii. Active heating time of the water is estimated to be about 6 hours, within which the sun is actively present. Though daily peak temperature may vary with time.
iv. $100^{O}C$ absorber plate temperature.
v. Average daily solar radiation I=$1017.695 W/m^2$ from NCEE, University of Benin.
vi. Specific heating capacity of water C_p=$4190 J/kgK$.
vii. 10^{O} angle of tilt.
viii. 300 litres capacity water tank is used to supply water to the solar collector. The tank is placed at 0.5m height from the collector inlet thus creating necessary pressure head for circulation of water in the system.
ix. The surface azimuth angle is 0° (the collector is facing the south).
x. The solar parameters are calculated with the sun one hour after noon $i.e.$ 15^{O}.

To design a solar flat plate collector, the area has to be determined considering the temperature of fluid (water) expected and the expected solar radiation of the location. The heat requirement is a function of the mass flow rate and the time of heating. Figure 1 shows an isometric view of the flat plate solar collector tilted at 10 ^{O}C and has a water supply tank and a water discharge tank. Table 1 shows the design dimensions for the solar thermal system.

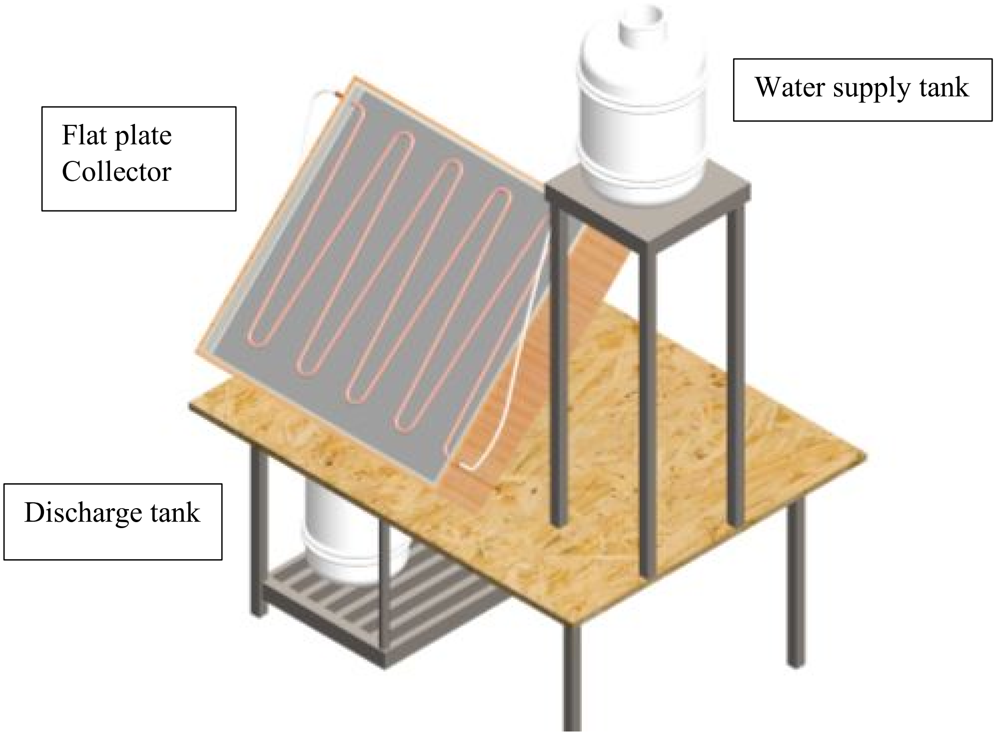

Figure 1. Isometric view of the flat plate solar collector

Table 1. Materials and dimensions for design

S/N	Components	Materials	Dimensions
1	Casing	Wood	0.84 m × 0.79 m × 0.08 m
2	Insulation	Fiber (Polyurethane)	0.80 m × 0.75 m × 0.004 m
3	Absorber plate	Aluminum painted black	0.80m × 0.75m
4	Flow tube	Copper	0.012 m in diameter
5	Glazing cover	Transparent glass	0.80 m × 0.75 m × 0.005 m
6	Water supply tank	Plastic	300 liters
7	Hot water collecting tank	Plastic	20 liters
8	Pipe	Plastic hose	h =1 m , d=0.008 mm

A volume of 300 litres is to be heated in six hours, therefore the volume flow rate is

$$\nabla = \frac{V}{t} \quad (1)$$

Where V = volume, t = heating time.

$$\nabla = \frac{0.3}{6\times3600} = 1.389\times10^{-5} m^3/s$$

Mass flow rate is given by:

$$m = \nabla \times \rho \quad (2)$$

Where ∇ = volume flow rate, ρ = density of water.

Mass flow rate $= 1000\times1.389\times10^{-5} = 0.01389 \mathrm{kg/s}$, which is approximately 0.014kg/s.

Area of collector is given by:

$$C_A = n\times W\times l \quad (3)$$

Where: $n = number\ of\ turns.$

$W = distance\ between\ segments.$

$l = length\ of\ segments.$

$$C_A = 9\times0.1\times0.7 = 0.63m^2$$

The thermodynamic analysis of the collector system consists of the accurate estimation of the heat gained and the heat loss within the system. The overall heat losses in the collector is represented by U_L, in order to properly estimate heat losses in the solar flat plate collector heat gained by the collector ought to be known. "Referring to equation 9, it can be seen that the heat gained cannot be properly estimated without first getting the overall heat losses of the solar flat plate collector".

$$Q_u = A_c(G_{b,t} - U_L(T_{pm} - T_a) \quad (9)$$

The overall heat losses in the collector is given by

$$U_L = U_T + U_B + U_E \quad (4)$$

Where:

U_L = Overall Heat Transfer Coefficient

U_T = The top loss Heat Coefficient

U_B = The bottom loss Heat Coefficient

U_E = The edge loss Heat Coefficient

A. Top loss coefficient (U_T)

The top loss coefficient is the losses that take place at the top of the flat plate solar collector and it is always between the glass cover of the flat plate solar collector and the environment. It is necessary to account for this type of losses so as to effectively know the amount of heat energy left.

The Top loss can be calculated using the heat transfer coefficients, but this method cannot be calculated numerically on a computer, a numerical method was devised by Goswami, (2015). For inclined or sloped collectors, the top loss heat coefficient according to (3) is given by

$$U_T = \left(\frac{N}{\frac{C}{T_{pm}}\left[\frac{(T_{pm}-T_a)}{(N+f)}\right]^e} + \frac{1}{h_w}\right)^{-1} + \frac{\sigma(T_{pm}+T_a)(T_{pm}{}^2+T_a{}^2)}{\frac{1}{\varepsilon_p+0.00591Nh_w}+\frac{2N+f-1+0.133\varepsilon_p}{\varepsilon_g}-N} \quad (5)$$

Where:

$$f = (1 + 0.089h_w - 0.1166h_w\varepsilon_p)(1 + 0.07866N) \quad (6)$$

f = focal distance.

h_w= wind heat transfer (W/m^2 ^{0}C)

ε_p= (0.95) emittance of plate.

N = (1) number of glass covers.

$$C = 520(1 - 0.000051\beta^2) \quad (7)$$

Where:

C = collector factor.

$\beta = (10^O)\ tilt\ angle\ of\ collector.$

$$h_w = 2.8 + 3.0V \quad (8)$$

Where:

V = wind speed.

e= $0.430(1 - 100/T_{pm})$ (9)

e = emissive power.

ε_g= emittance of glass (0.88)

T_a= (299.1K) ambient temperature (K)

T_{pm}= (373K) mean plate temperature (K)

The wind heat transfer is calculated from maximum wind Speed at Benin City which is 5miles/hr. (Relates to wind heat transfer coefficient gotten from National Centre for Energy and Environment (NCEE), University of Benin, Edo State, Nigeria, 2017).

$h_w = 2.8 + 3.0 \times 2.235 = 9.505 W/m^2C$

B. Bottom loss coefficient (U_B)

The bottom loss coefficient is the losses that occurs at the bottom of the flat plate solar collector. It is usually between the bottom of the absorber and the bottom cover of the collector. This form of estimation is necessary to be able to account for heat loss and it helps to improve on the design of the flat plate solar collector.

Bottom loss coefficient is given by the following relation:

$$U_B = \frac{k}{L} \tag{10}$$

Where k is thermal Conductivity (Urethane) =0.028

And L = thickness of urethane layer =30mm =0.03

C. Edge loss coefficient (U_e)

This form of heat loss occurs at the edges and corners of the flat plate solar collector, although this type of heat loss is minimal. It helps us to known how effective the collector system is and how best to reduce this form of losses.

Edge loss coefficient can be estimated using the following relations:

$$U_e = \frac{\frac{k}{L_e} \times P \times C_t}{C_A} \tag{11}$$

Where:

k = thermal conductivity of insulation (polyurethane) =0.028

L_e=Thickness of edge insulation =0.02

P = Perimeter of collector = 2(0.82 +0.77) =3.18m

C_t = Thickness of Collector = 0.08m

C_A = Collector Area = 0.82 x0.77 =0.6314m^2 (Gross area).

Therefore

$U_L = U_T + U_B + U_e$

The useful energy is given by:

$$Q_u = A_c(G_{b,t} - U_L(T_{pm} - T_a)) \tag{12}$$

where, $A_c = Area\ of\ collector.$

$G_{b,t} = radiation\ on\ a\ tiled\ surface.$

$U_L = heat\ transfer\ coefficient.$

$T_{pm} = absorber\ plate\ temperature.$

$T_a = ambient\ temperature.$

Tube Spacing from design is 50mm

Tube Diameter (Inside) is 10mm

Plate Thickness = 1mm

Plate Thermal Conductivity (Aluminium) is 235 W/m^{2o}C (1)

Heat transfer coefficient inside Tubes = 300W/m^2 ^{0}C

D. Bond Conductance

The bond conductance consists of bond conductivity, bond width and bond thickness. It is one of the major parameters used for accurately estimating the flat plate solar collector performance.

The bond conductance C_b is given by

$$C_b = \frac{K_b b}{\gamma} \tag{13}$$

Where:

K_b is the bond conductivity =386W/m^2 ^{o}C

b= bond Width = 12mm =0.012m
γ = Bond Thickness =100m =0.1
The standard Fin efficiency F is given by

$$F = \frac{\tanh[m(W-D)/2]}{m(W-D)/2} \quad (14)$$

Where:
W = Distance between two concentric tubes =100mm = 0.1m
D = Tube Diameter (External) = 12mm=0.012m

$$m = \sqrt{\frac{U_L}{k\delta}} \quad (15)$$

Where:
k = plate thermal conductivity =235W/m^{2o}C
δ = Plate thickness = 1mm= 0.001m

$$F_R = F_1 F_3 F_5 \left[\frac{2F_4}{F_6 \exp\left[-\frac{\sqrt{1-F_2{}^2}}{F_3}\right] + F_5} - 1 \right] \quad (16)$$

Where, F_R= heat removal factor.
For A serpentine Tube conductor, first we check if $\frac{\dot{m}C_p}{F_1 U_L A_c} > 1.0$
The collector has a mass flow rate of 0.014kg/s.
And F_1 is given by

$$F_1 = \frac{\kappa}{U_L W} \frac{\kappa R(1+\gamma)^2 - 1 - \gamma - \kappa R}{[\kappa R(1+\gamma) - 1]^2 - (\kappa R)^2} \quad (17)$$

$$F_2 = \frac{1}{\kappa R(1+\gamma)^2 - 1 - \gamma - \kappa R} \quad (18)$$

$$F_3 = \frac{\dot{m}C_p}{F_1 U_L A_c} \quad (19)$$

$$F_4 = \left(\frac{1-F_2{}^2}{F_2{}^2}\right)^{1/2} \quad (20)$$

$$F_5 = \frac{1}{F_2} + F_4 - 1 \quad (21)$$

$$F_6 = 1 - \frac{1}{F_2} + F_4 \quad (22)$$

$$\kappa = \frac{(k\delta U_L)^{1/2}}{\sinh[(W-D)\left({}^{U_L}/_{k\delta}\right)^{1/2}]} \quad (23)$$

$$\gamma = -2\cosh\left[(W-D)\left(\frac{U_L}{k\delta}\right)^{1/2}\right] - \frac{DU_L}{\kappa} \quad (24)$$

$$R = \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}} \quad (25)$$

Length of one serpentine segment (L)	0.7 m
Distance between tubes (W)	0.1 m
Number of segments (N)	5
Plate thickness (δ)	1 mm
Tube outside diameter (D)	12 mm
Tube inside diameter (D_i)	10 mm
Plate thermal conductivity (k)	235 W/m ◦C
Overall loss coefficient (U_L)	7.82 W/m2 ◦C
Fluid mass flow rate (m)	0.014 kg/s
Fluid specific heat (C_p)	4190 J/kg ◦C
Fluid-to-tube heat transfer coefficient (h_{fi})	1000 W/m2 ◦C
Bond conductance (C_b)	46.32W/m^{2o}C

Therefore, the collector flow factor

$$F_f = \frac{F_R}{F_1} \tag{26}$$

Therefore $= Q_u \times F_R$ (27)

Heat content of the fluid and efficiency of the collector system

The heat content of the fluid comprises the temperature of the water, temperature of the absorber plate and the quantity of heat generated by the flat plate solar collector.

A. The Outlet Fluid Temperature

This is the temperature of water that leaves the flat plate solar collector after absorbing heat energy from the sun, it is necessary to estimate it properly for collector performance purpose.

The outlet fluid Temperature is given by

$$\frac{T_{fo} - T_a - S/U_L}{T_{fi} - T_a - S/U_L} = \exp\left(-\frac{U_L A_c F_1}{\dot{m} C_p}\right) \tag{28}$$

B. Efficiency

This is the efficiency of the solar flat plate collector considering the area of the collector, the solar radiation and the heat energy gained.

$$\eta_i = \frac{Q_u}{A_c G_T} \tag{29}$$

C. The Average Absorptance-Emittance Product

The average absorptance-emittance product is the amount of energy that is gotten considering the fouling factor.

$$Q_u = \dot{m} C_p (T_o - T_i) \tag{30}$$

$$Q_u = A_c F_R [G_T (\tau\alpha)_{av} - U_L (T_i - T_a)] \tag{31}$$

D. Maximum Plate and Fluid Temperature

Equilibrium temperatures (sometimes called stagnation temperatures), encountered under conditions of high radiation with no fluid flowing through the collector, are substantially higher than ordinary operating temperatures, and collectors must be designed to withstand these temperatures. It is inevitable that at some point control problems, servicing, summer shutdown, or other causes will lead to no-flow conditions. The fluid and plate temperatures are the same for the no-flow condition. The equilibrium temperatures of other parts of the collector can be estimated from the ratios of thermal resistances between those parts and ambient to that of the plate to ambient. These maximum equilibrium temperatures place constraints on the materials, which must retain their important properties during and after exposure to these temperatures, and on mechanical design to accommodate thermal expansion. The maximum equilibrium temperature, is a function of the ambient temperature and the incident solar radiation.

$$T_{max} = T_a + \frac{S}{U_L} \tag{32}$$

Where, $T_a = (26.1^{O}C)$ *Ambient temperature.*

s = (1099.11W/m^2) solar radiation.

$U_L = 7.82 W/m^2 {}^{\circ}C$ Overall heat loss.

Result and Discussion

The thermodynamic analysis was carried out considering the heat gained in the flat plate solar collector, heat losses in the collector, bond conductance, flow removal factor, collector flow factor, average emittance product and the maximum fluid and plate temperature respectively. As shown in table 2.

Table 2 presents the results that were obtained after the thermodynamic analysis of the flat plate collector system was carried out. The top loss was found to be 6.33W/m^2 ^{O}C, this occurred between the glass cover and environment. Heat gained of 328.36W was estimated, this amount of power generated is considerably high considering the area that was used for the research. The outlet temperature gotten was about 61 ^{O}C, this can be used for domestic and some industrial use.

The maximum plate and fluid temperature was estimated to be 166.65 °C, this is the temperature the absorber plate and the water within the collector tube will get to (this is also known as the limit). An efficiency of 47.4 % was achieved, this is reasonably high considering the size of the flat plate solar collector.

Table 2. Results from thermodynamic analysis

Top loss coefficient	6.33W/m^2 ^{O}C
Bottom loss coefficient	0.933W/m^2 ^{O}C
Edge loss coefficient	0.560W/m^2 ^{O}C
Heat gained	328.36W
Overall heat loss	7.82W/m^2 ^{O}C
Outlet fluid temperature	61 ^{O}C
Maximum plate and fluid temperature	166.65℃
Collector flow factor	0.271
Efficiency	0.474

Conclusion

Thermodynamic analysis was carried out on the flat plate solar collector in order to determine the behavior and performance of the collector system. The outlet temperature obtained was high considering the collector area that was used, the efficiency (47.4%) and energy gained (328.36W) help to show that the solar flat plate collector can be used for domestic and some industrial applications respectively.

Reference

Bena B, Fuller RJ, (2002) Natural convection solar dryer with biomass back up heater. *J. Solar Energy*, **72** (1), 75-83. DOI: 10.1016/S0038-092X(01)

Duffie J, Beckman WA, (2013) *Solar Engineering of Thermal Processes*. John Wiley and Sons, inc., New York. ISBN: 978-0-470-87366-3

Gustafson M, (2013) A C*omputational Approach to Simulating the Performance of a 24-Hour Solar Hydrogen Fuel Cell Electric Power Plant*, Wright State University, Dayton, https://corescholar.libraries.wright.edu/cgi/viewcontent.cgi?article=1927&context=etd_all

Goswami YD, (2015): Principles of solar engineering. Third edition, New York. https://www.advan-kt.com/principlesofsolarengi.pdf

Iloeje OC, (1993) Design, operation and performance of a solar rice dryer. In proc. 2nd Workshop on ''SDTAI'', Nsukka, Nigeria, 24-27 November. https://core.ac.uk/download/pdf/234071349.pdf

Kalogirou SA, (2003) The potential of solar industrial process heat applications. *Appl. Energy*; **76**, 337–61. http://www.sciencedirect.com/science/article/pii/S0306-2619(02)00176-9

Kalogirou, S.A. (2004): Solar thermal collectors and applications. *Prog. Energy & Combust. Sci.* 30, 231–295. http://www.elsevier.com/locate/pecs

National Centre for Energy and Environment (NCEE), University of Benin, Edo State Nigeria (2017)

Nielsen, R. 2005. *Solar Radiation*. http://home.iprimus.com.au/nielsens/.

Sharma A, Chen CR, Nguyen V, (2009) Solar energy drying systems: a review. *Renewable and Sustainable Energy Review*, **13**(6-7), 1185-1210. https://ideas.repec.org/a/eee/rensus/v13y2009i6-7p1185-1210.html#correct