Journal of the Nigerian Association of Mathematical Physics
Volume26, (March, 2014), pp 235 – 245
© J. of NAMP

Design and Construction of Multi Stage Solar Still

Maryam Idris and Fatima. S. Koki

Department of Physics,
Bayero University Kano

Abstract

Multi-stage solar still was designed and constructed. Basically, it is an indirectly heated solar still incorporating a distillation unit. Experimental studies were carried out in order to evaluate the system performance. It was observed that the collector efficiency was higher than the maximum working temperature. An average value of 56% was achieved against the theoretical value of 50%. The average value of useful energy, $\dot{Q}_U$ evaluated was 324.03Watts which was supplied from the solar collector to the distillation chamber. Moreover a minimum value of distillate yield recorded 22ml per day and maximum of 634ml per day out of the entire test conducted..

1.0 Introduction

Water is essential to life; man has been dependent on rivers, lakes and underground water reservoir for fresh water requirement in domestic life, agriculture and industry. However, the use of water from such sources is not always hygienic because of the presence of impurities. The impact of many diseases afflicting mankind can be drastically reduced if fresh hygienic water is provided for drinking [1]

1.1 Review of Previous Work

Solar distillation is certainly not a new phenomenon. It has been in practice for a long time. The earliest literature in this field is that of the Arab alchemists Mouchot. In their historical review on desalination of water, Nebbia and Menazzi mentioned the work of Delta Porta (Published in [2] and [3].

Folayan and Ajayi also carried out experiments to compare the performance of single slope and double slope solar stills under varying thermal inertia and addition of charcoal dust to aid solar absorption [4].

Olalekan carried out study on improving the performance of single slope solar still with sun tracking mechanism. For this experiment he constructed two single slopes solar still, were he kept one fixed and the other one tracked. It was observed that the still with the tracking mechanism gave an increase in overall estimated efficiency of 3.8% [5].

Madhlopa carried out an experiment on the Development of an advanced passive solar still with separate condenser. He constructed two stills, were by one was conventional solar still(CSS) and the other advanced solar still(ASS).It was observed that the ASS yield more distilled water than the CSS[6].

Despite the fact that a number of works have been carried out on the mathematical modeling and simulations of solar still, there is still a need for further studies in this area, especially in Nigeria.

2.0 Theoretical Aspects

In order to design and evaluate the economics as well as the performance of a solar energy system, it is necessary to consider various theories and relations on which the design of the system must be based.

2.1 The Flat Plate Collector

Heat transfer analysis of a solar collector is a simple heat balance. The performance of a solar collector is described by an energy balance that indicates the distribution of incident solar energy into:

Corresponding author: ***Maryam Idris***, E-mail: maryamjahun84@gmail.com, Tel.: +2347035853691

1. Useful energy gain $(\dot{Q}_U)$
2. Stored energy $(\dot{Q}_S)$
3. Various losses of energy $(\dot{Q}_L)$

The energy balance on the whole of the collector can be written as

$$A_c G_b(\tau\alpha) + A_c G_d(\tau\alpha) = (\dot{Q}_U) + (\dot{Q}_S) + (\dot{Q}_L) \; [7] \tag{2.1a}$$

Where;

$A_c =$ Collector area, $G_b =$ Beam solar radiation, $G_d =$ Diffuse solar radiation

$\tau\alpha =$ transmissivity and absorptivity of the collector , $\dot{Q}_U =$ Rate of useful heat transfer to a working fluid in the solar absorber / exchanger , $\dot{Q}_S =$ rate of energy storage in the collector and

$\dot{Q}_L =$ rate of energy losses from the collector to the surroundings.

Equation (2.1) can be simplified as:

$$A_c \; G_T(\tau\alpha) = \dot{Q}_U + \dot{Q}_S + \dot{Q}_L \tag{2.1b}$$

Where;

G_T is the sum of beam and diffuse radiations

The total useful energy gain for the entire collector $\dot{Q}_U$ may be obtained from the following relationship:

$$\dot{Q}_U = A_C F_R\left[G_T(\tau\alpha) - U_L\left(T_{fi} - T_a\right)\right] \tag{2.2}$$

where F_R is the collector's heat removal factor.

In equation (2.2), the total useful energy gain is expressed as a function of inlet fluid temperature.

Neglecting any temperature drops of the fluid between the collector and distillation chamber (First tray), $\dot{Q}_U$ can also be written as:

$$\dot{Q}_U = \dot{m}C_p(T_{fo} - T_i) \tag{2.3}$$

Where;

T_i= fluid temperature in the first tray , T_{fo}= outlet fluid temperature from the collector

The rate of energy storage capacity $\dot{Q}_S$ of the fluid (water) in the still, is given as the usual heat capacity equation [8]

$$\dot{Q}_S = \dot{m}C_p(T_i - T_o) \tag{2.4}$$

T_i and T_o are the operating temperature limits.

The rate of heat loss $\dot{Q}_L$ from the system is that loss from the collector absorber plate at temperature T_p to the surroundings at the ambient temperature T_a . The heat loss occurs from three ways:

(i) Conduction loss through the bottom insulation
(ii) Conduction loss through the sides (four) of the collector
(iii) Convection and radiation losses through the top cover.

The rate at which the absorber plate losses heat to the surroundings is given as in[9]:

$$\dot{Q}_L = U_L \, A_C(T_B - T_a) \tag{2.5}$$

where

U_L Is the overall heat transfer coefficient

T_a Is the ambient temperature

Equation (2.5) can be re-written as:

$$\dot{Q}_L = A_c(U_f)(T_p - T_a) + A_c(U_b)(T_B - T_a) + A_s U_s (T_B - T_a) \tag{2.6}$$

2.1.1 Determination of the solar collectors overall heat transfer coefficient

2.1.1.1 Bottom Insulation Heat Transfer Coefficient (U_b)

The insulation at the bottom of the collector is in two stages. The sawdust insulation is having a thermal conductivity K_{11} and thickness e_{11} at the first stage. At the second stage, the bottom of the collector casing has a thermal conductivity K_{12} and thickness e_{12}. The air behind the box frame has a convective heat transfer coefficient U_b at a temperature T_a.

The rate of energy loss through the bottom of the collector is given as

$$\dot{Q}_b = U_b A_b (T_P - T_a) \tag{2.7}$$

The bottom heat transfer coefficient can be calculated as

$$U_b = \frac{1}{R_{11}} + \frac{1}{R_{12}} + \frac{1}{R_{13}} = \frac{K_{11}}{e_{11}} + \frac{k_{12}}{e_{12}} + \frac{h_{13}}{1} \tag{2.8}$$

R_{13} is negligible as compared to R_{11} and R_{12}.

Hence,

$$U_b = \frac{K_{11}}{e_{11}} + \frac{K_{12}}{e_{12}}$$

2.1.1.2 Side Heat Transfer Coefficient (U_S)

The insulation by the collector sides is in three stages. The collector support having thermal conductivity K_{21} and thickness e_{21}. Secondly is the saw dust insulation having thermal conductivity K_{22} and thickness e_{22}. In the third stage, the collector casing has the thermal conductivity K_{23} and thickness e_{23}.

The rate of energy loss through the sides of the collector is given as

$$\dot{Q}_S = U_S A_S(T_B - T_a) \quad (2.9)$$

The side heat transfer coefficient can be calculated as:

$$U_S = \frac{1}{R_{21}} + \frac{1}{R_{22}} + \frac{1}{R_{23}} + \frac{1}{R_{24}} = \frac{K_{21}}{e_{21}} + \frac{K_{22}}{e_{22}} + \frac{h_{24}}{1}$$

R_{24} is negligible as compared to $R_{21,} R_{22,} R_{23.}$

Also $R_{21} = R_{23}$.

Therefore,

$$U_S = \left(2\left(\frac{e_{21}}{k_{21}}\right) + \frac{e_{22}}{k_{22}}\right)^{-1} \quad (2.10)$$

2.1.1.3 Front Heat Transfer Coefficient (U_f)

The rate of energy loss through the top cover is given as:

$$\dot{Q_f} = U_f A_f(T_p - T_a) \quad (2.11)$$

The front heat transfer coefficient can be calculated as:

$$U_f = \frac{1}{R_{31}} + \frac{1}{R_{32}} + \frac{1}{R_{33}} \quad (2.12)$$

where

$$R_{31} = \frac{1}{R_{31.c}} + \frac{1}{R_{31.r}} = \frac{R_{31.r} \times R_{31.c}}{R_{31.r} + R_{31.c}} \qquad R_{32} = \frac{e_{32}}{K_{32}}$$

$$R_{33} = \frac{1}{R_{33.C}} + \frac{1}{R_{33.r}} = \frac{R_{33.r} \times R_{33.C}}{R_{33.r} + R_{33.C}}$$

$R_{31,c}$ Is the convection resistance of air gap between the absorber and the glass cover plate.

$R_{31,r}$ Is the thermal radiation resistance of the air gap between the absorber and the glass cover plate.

$R_{33,c}$ Is the thermal resistance of the wind driven convection

$R_{33,r}$ Is the radiation heat transfer for resistance between the cover surface and the sky.

2.2 Energy Loss from the Absorber Plate to the cover

(i) Radiation heat transfer between parallel plates

The heat transfer by radiation between two surfaces (Parallel) is determined from [10]:

$$\dot{Q}_{rad} = A\varepsilon_e\sigma(T_p^4 - T_g^4) \quad (2.13)$$

where A = cross sectional area of the plate, σ=Stefan Boltzmann constant, ε=Effective emittance for the two plates, T_p=Temperature of emitting surface (Absorber plate), T_g=Temperature of the receiving surface (glass cover) and;

$$\varepsilon_e = \left(\left(\frac{1}{\epsilon_p} + \frac{1}{\epsilon_g}\right) - 1\right)^{-1} \quad (2.14)$$

Therefore,

$$\dot{Q}_{rad} = A_c\sigma(T_p^4 - \dot{T}_g^4)\left[\left(\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_g}\right) - 1\right]^{-1} \quad (2.15)$$

To linearize equations for simplicity, equation (2.15) can be re-written as follows:

$$\dot{Q_{rad}} = h_r A(T_P - T_g) \quad (2.16)$$

Where;

$$h_r = \sigma(T_p + T_g)^2(T_p - T_g)^2\left[\left(\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_g}\right) - 1\right]^{-1} \quad (2.17)$$

(ii) Convention Heat Transfer between Parallel Plates

The Nusselt number N_u, and Rayleigh number R_a, are customary based on the gap width L between the hot and cold walls. The heat transfer coefficient may be obtained from the reliable correlation of [11].

$$Nu = 1 + 1.44\,(1-1708/R_{ae})\,(1-1708\,(\sin 1.8\theta)^{1.6}/R_{ae}) + \{(R_{ae}/5830)^{1/3}-1\} \quad (2.18)$$

where $R_{ea} = R_{aL}\cos\theta$

Also $Nu = h_c L/k \Rightarrow h_c = N_u k/L \quad (2.19)$

$$R_a = G_r P_r \quad (2.20)$$

$$G_r = g\beta\Delta T L^3/v^2 \tag{2.21}$$

where G_r=Grashof number,P_r=Prendtl number=heat transfer coefficient of air in the gap, W/m^2 ^{0}C,L=air gap,m,K=thermal conductivity of air, W/m ^{0}C,G=acceleration due to gravity, m/s^2,Θ=angle of tilt, in ^{0}(degree)

2.3 Energy loss through the glass cover $(\dot{Q}_g)$

The rate of energy loss through the glass cover is purely by conduction. This is obtained in as[12]

$$\dot{Q}_g = \frac{K_g A(\Delta T)}{e_g} \tag{2.22}$$

Where; ΔT is the temperature difference between the two surfaces of the glass (top and bottom of the glass)

The thermal resistance, R_{32} is

$$R_{32} = \frac{e_{32}}{k_{32}} = \frac{e_g}{k_g} \tag{2.23}$$

2.4 Energy Loss from the Glass cover to the surrounding ($Q_{g\text{-}s}$)

The energy loss from the glass cover to the surrounding occurs by convection and radiation. The rate of energy loss is given as[13]:

$$\dot{Q}_{g-s} = h_w(T_g - T_{sky}) + h_{33.r}(T_g - T_{sky}) \tag{2.24}$$

where

$$h_w = 5.7 + 3.8V \tag{2.25}$$

h_w=heat transfer coefficient due to wind recommended by Duffie and Beckman[14],

V=wind velocity in m/s

The radiative heat transfer coefficient $h_{33.r}$ is given as

$$H_s = h_{33.r} = \varepsilon_g\sigma (T_g + T_{sky})(T^2_g + T^2_{sky}) \tag{2.26}$$

$$T_{sky} \approx 0.0552\, T_a^{1.5}$$

Therefore, from the equations (2.22) and (2.26) the convective and the radiative component of the thermal resistance R_{33} can be obtained. For a single glass cover and collector tilt of 45^0 loss coefficient, U_f is given as:

$$U_f(45°) = \left(\frac{1}{h_c+h_r} + \frac{1}{h_w+h_s}\right)^{-1} \tag{2.27}$$

For tilt other than 45^0, Adhikari at el [15] proposed a relation for the determination of the front loss coefficient as:

$$\frac{U_f(\theta)}{U_f(45°)} = 1 - (\theta - 45)\left(0\cdot 00259 - 0\cdot 00144\varepsilon_p\right) \tag{2.28}$$

2.5 The Transmittance Absorbance Product

The $\tau\alpha$ is theoretically calculated from the following relation

$$(\tau\alpha) = \frac{\tau\alpha}{1-(1-\alpha)0{\cdot}016} \tag{2.29}$$

τ=transmittance of the cover material at a given incidence angle, α=absorptivity of the absorber material.

2.6 Collector Heat Removal Factor

It is mathematically given by:

$$F_R = \frac{GC_p(T_{fo}-T_{fi})}{[s-U_L(T_{fi}-T_a)]} \tag{2.30}$$

where G is the flow rate per unit collector area ,S is the absorber solar energy per unit area.

$$G = \frac{\dot{m}}{A_C} \tag{2.31}$$

$$s = G_T(\tau\alpha)/A \tag{2.32}$$

2.7 Collector efficiency

This is the ratio of the useful heat gain per unit **area** of the collector to the solar energy flux on the collector.

$$\eta = \frac{\dot{Q_U}}{A_C G_T} \tag{2.33}$$

Substituting equation (2.2) into (2.33) yields

$$\eta = F_R(\tau\alpha) - F_R U_L \frac{T_f - T_a}{G_T} \tag{2.34}$$

2.8 The Distillation Still

The tray temperatures can be obtained from the following expressions[16].

$$T_{p.i} = \frac{H_1T_1 + (h_bA_{b.i} + f_1)\,T_a}{h_b\,A_{b.i} + f_i + H_i} \tag{2.35}$$

$$T_{p.i} = \frac{h_{i-1}\,A_{b.i-1}T_{1-1} + H_1T_1 + f_1T_a}{h_{i-1}\,A_{b.i-1} + f_i + H_i} \quad i = 2, N+1$$

Where

$f_i = h_{s.i}\,A_{s.i}$,fi = $h_{s,i}\,(A_{s.i}\,h_{\cdot p} + A_{b.1}h_p)$ i = 1, N + 1 (2.36)

2.9 Performance Parameters

Various parameters that are useful for determining performance of the system will be evaluated by the equations below:

Distillate yield (kg/sec) for i^{th} Stage:

$\dot{m}_{e.i} = h_{e.i}\,(T_e - T_{p.i}\,A_{b.i}/\ell_i)$ (2.37)

Hourly distillate yield (kg/hr):

$M_{e.i} = \dot{m}_{e.i} \times 24$ (2.38)

Daily distillate yield (kg/day)

$M_{d.i} = M_{e.1} \times 9$ (2.39)

The performance ratio: This is the sum product of distillate yield per second and latent heat of condensation at each stage to the useful heat from the solar collector. It is mathematically written as:

$PR= \sum_i^N \frac{\dot{m}_{e.i}l_i}{\dot{Q}}$ (2.40)

2.10 The Thermo-physical properties

Following[17] thermo-physical properties have been evaluated using the expression below:

$K = 0.0244 + 0.763 \times 10^{-4}\,T_{av}$ (2.41)

$$\mu = 1.718 \times 10^{-5} + 4.620 \times 10^{-8}T_{av} \quad (2.42)$$

$$P = 353.44(T_{av} - 273.35) \quad (2.43)$$

$$l = 2324.6(1.0727 \times 10^3 - 1.0167T_{av} + 1.4087 \times 10^{-4}T_{av}^2 - 5.146210 \times 10^{-6}T_{av}^3) \quad (2.44)$$

where T_{av} represents the arithmetic mean of the temperatures of evaporation and condensation surfaces and can be expressed as follows [18]:

$T_{av} = (T_e + T_c)/2$ (2.45)

The value of saturation vapour pressure is calculated by the following expression:

where

$T' = 1.8T + 491.69$ (2.46)

Convective heat transfer coefficient [19]:

$$h_{c,i} = C(G_{ri}P_{ri})^n\,K_i/X_i \quad (2.47)$$

where

$G_{ri} = (X_i^3\rho_i^3\beta g\Delta T')/\,\mu_i^2$ (2.48)

$$\Delta T' = \left[(T_1 - T_{p,i+1}) + \frac{(P_{e,i}-P_{c,i})(T_i+273.15)}{268.9\times10^3 - P_{e,i}}\right] \quad (2.49)$$

$$c = 0.21, n = {}^1/_4 \; for\; 10^4 < Gr_i < 2.51 \times 10^5 \quad (2.50)$$

$$C = 0.1255, n = {}^1/_3 \; for\; 2.51 \times 10^5 < Gr_i < 10^7 \quad (2.51)$$

The evaporative heat transfer coefficient is determined by

$$h_{r,i} = 16.273 \times 10^{-3}h_{c,i} \quad (2.52)$$

The radiative heat transfer coefficient is given by

$$h_{r,i} = \frac{\varepsilon_g\sigma(T_i + 273.15)^4 - (T_{p,i+1} + 273.25)^4}{(T_i - T_{p,i+1})} \quad (2.53)$$

3.0 Methodology/ Procedure

3.1 Design Considerations For Typical Solar Collector

A flat plate collector of liquid type should be heat transfer effective, pressure drop, free flow (i.e. minimum fouling and corrosion) accessibility and simple maintenance.

3.1.1 Collector Design Calculations

Suppose that 35 litres of water at 25℃ is to be heated to a temperature of 87℃ per day by the collector, then the main daily energy requirement will be

$$\dot{Q}_R = MC_P\Delta T \qquad (2.54)$$

$m = \rho V$

But $\rho\ of\ water = 1000Kgm^{-3}$

Therefore, $m = 1000 \times 35 \times 10^{-3}kg = 35kg$

$$\dot{Q}_R = 35 \times 4.18 \times 10^3 \times (87 - 25) = 9.070MJ$$

This energy required must be equal to the mean daily total useful energy gain for the collector from the equation below

$$\dot{Q}_U = \eta A_C G_T \qquad (2.55)$$

The average daily horizontal radiation for the month of June is $23 \times 10^6 J/m^2 day$[20]

$$\dot{Q}_U = 0 \cdot 50 \times 23 \times A_C = 11.5A_c$$

$$A_C(11.5)MJm^{-2} = 9.070MJ$$

$$\therefore\ A_C = \frac{9.070MJ}{11.5MJm^{-2}} = 0 \cdot 788m^2$$

Now if the width b of the collector is taken to be $0 \cdot 65m, then$ the length will be

$$l = \frac{0.788m^2}{0 \cdot 65m} = 1.21m$$

The dimension of the collector area will be 1 $1 \cdot 21m\ by\ 0 \cdot 65m$. Let us consider the depth δ; of the absorber to be 0.03m, then the volume of the flat absorber is;

$V_A = 1 \cdot 21 \times 0.65 \times 0.03$

$= 0 \cdot 024m^3$

Now the volume of water in the first tray is:

V_T = (35-24) litres = 11 litres

Assume 0.43 litres in the piping network

Actual V_T = 10.57 litres

Consider a water level h, in the tray to be 0.05m and the width b to be 0.34m, then the length l will be,

$$l = \frac{0.01057}{0.34 \times 0.05} = 0.622m$$

3.2 Construction Detail

In order to understand the operation of the system (stacked tray solar still), it is quite desirable to depict the system constructional details as much as possible.

3.2.1 Construction of the Solar Collector

An iron sheet metal of 2mm thick was used. Pieces A and B were cut from the sheet. The thin lines were folded or bent so that the portions d, e, f are perpendicular to g, and portion h is perpendicular to j.

The pieces A and B are welded together to form C i.e. the collector absorber. The shorter sides are drilled to 25mm diameter to give an in and out for fluid flow.

Before the welding was effected, the interior of the collector was painted black in order to prevent corrosion and effect absorptance.

3.2.2 Construction of the Trays

A galvanized iron sheet of 0.5mm thick is used. A rectangular size was cut out from the sheet. The dotted lines A…A were folded so that portions k and l are 90^0 to portion 0.Three pieces of the same size were also cut from sheet metal, the thin lines was folded a little to angle of about 167^0.The portion m and n were folded vertically parallel to each other.

Another two pieces were also marked and cut, to form the tray ends (two ends per tray). The portions R were gas weld drilled. A V-trough in which the condensate trickles from the V-shape cover was also gas welded to the rectangular size sheet to form the bottom tray.

The castings were made out of 17mm thick plywood. The sides and bottoms of the four portions of S were gas welded to the three pieces sheet to form the second and third trays.

Finally, the two portions T were also gas welded to the fourth component in S (after folding) to form the fourth or top tray.

Holes for water overflow, supply and condensate were drilled along the length of the bottom, second and third trays.

3.3 The Construction of the Castings/Collector Support

The solar collector and the stacked tray are enclosed both casing components were cut accordingly and nailed to form the desired configuration.

3.4 The Cover Glazing

The required size of glass 900x500x3(mm) was cut and bought from the market.

3.5 The Solar Collector and Distillation Chamber Stands

These were constructed from lengths of 20mm square pipes

4.2 Discussion

From figure 3.1, it can be seen that the distillate yield increases as the various temperatures increases. This shows that the more the sun radiation the more the production yield.

From figure 3.2, it can be noticed that the distillate yield was very low due to little solar intensity and also the various temperatures of the absorber plate, glass, ambient and outlet fluid were also low.

From figure 3.3, it can be found that the distillate began to yield at 9:00hr from the first stage. But from figure 3.4, the distillate started to yield at 12:00hr from the first stage, due to little sun radiation. The difference of three to four hours before starting to yield was the period for water to get heated enough for evaporation to take place. This difference depends on the visibility of the solar intensity.

Readings were also taken in the month of December and it was noticed that the various temperature and the production yield were very low compared to the temperatures and yields of the month of June/July. This was due to the harmattan period that affects the solar radiation.

In this report, the minimum yield from the still was 22ml per day on the 23rd June, 2012 and the maximum yield of about 634ml per day was collected on 29th June, 2012.The yield however was quite less than expected. This was due to weather fluctuations (cloud, rain and wind) and loss of heat from the distillation unit. But the performance would definitely improve during summer (i.e. months of February through May)

5.0 Conclusion

A multistage solar still has been designed and constructed which is a two in one system. One part is the flat plate solar collector that supplies heat energy and the other part is the distillation unit. Tests were carried out in order to evaluate the performance of the still for one month.

It was observed that the collector efficiency used to be higher before it reached the maximum working temperature. From were it declined. An average value of 56% was achieved against the theoretical value of 50%.

The average value of useful energy, $\dot{Q}_U$ evaluated was 324.03 Watts which was supplied from the solar collector to the distillation chamber.

However a minimum value of distillate yield recorded 22ml per day and maximum of 634ml per day out of the entire test conducted. This is quite lower than expected. It is evidently connected with weather fluctuations during the experiment.

Finally, it is concluded that the still will perform better during summer, careful design of a very efficient solar energy collector and also casing to prevent excessive heat loss. At least the model has performed and is quite applicable to Nigerian climate.

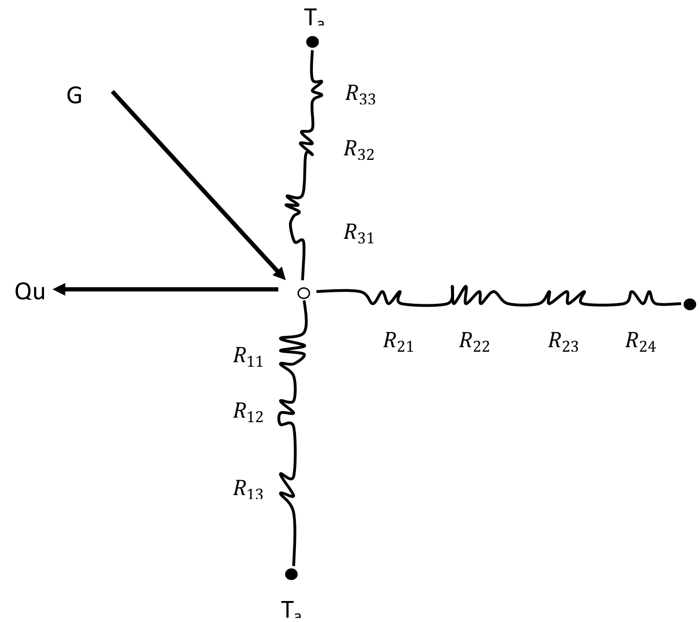

Fig. 2.1 Collector Thermal Network

Table 2.1: Results of the performance analysis of the collector

U_L	4.16 W/m^2 ^{0}C
Q_i	83.48W
F_R	0.85
$\dot{Q}_U$	324.03W
η_c	0.56
T_e^1	583.49k
T_c^1	565.49k
P_1	5123.048 Pa
ΔT^1	16.5K
T_{av}	46 ^{0}C
K	0.0279 W/m ^{0}C
M	1.93x10^{-5}N$_s$ /m^2
ρ	1.107 Kg / m^3
l	2378.4x10^3 J/Kg
G_r	2.0855 x 10^5
	4.51 x 10^7
h	72.35 W/m^2 ^{0}C
$m_{e,i}$	$2.52 \times 10^{-8}\ kg/s$
$M_{e,1}$	9.07 x 10^{-5} Kg / hr
$M_{d,1}$	8.1648 x 10^{-4} Kg / day

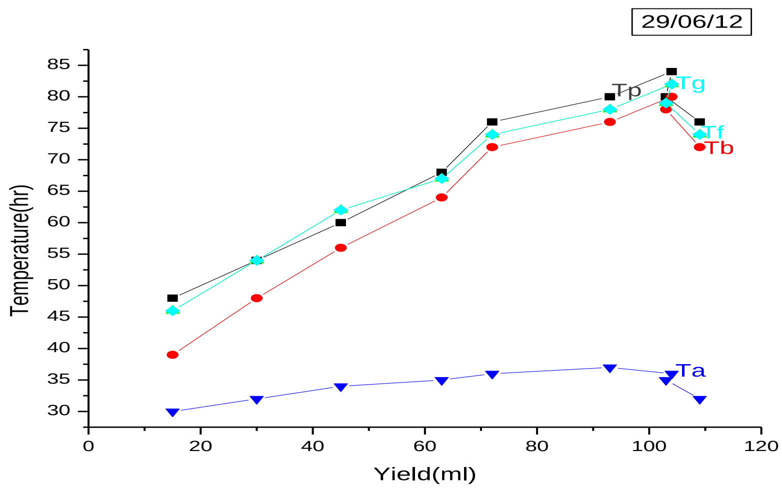

Fig. 3.1: Temperatures against Yield (29/06/12)

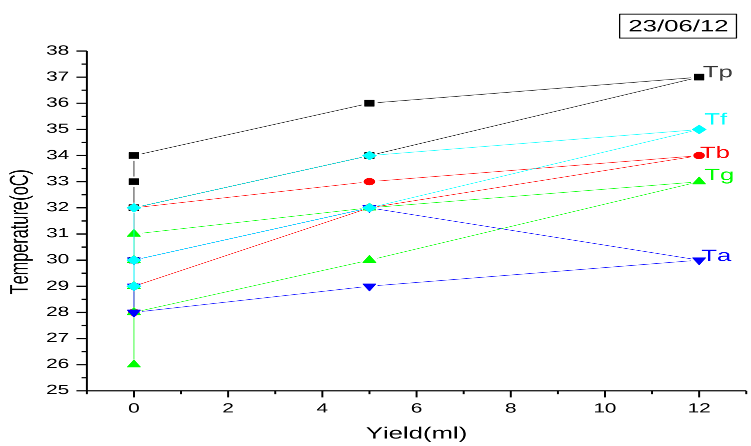

Fig. 3.2: Temperatures against yield (23/06/12)

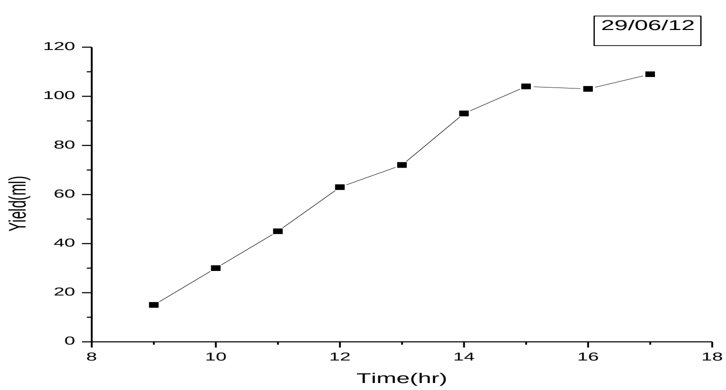

Fig3.3: Yield against Time (29/06/12)

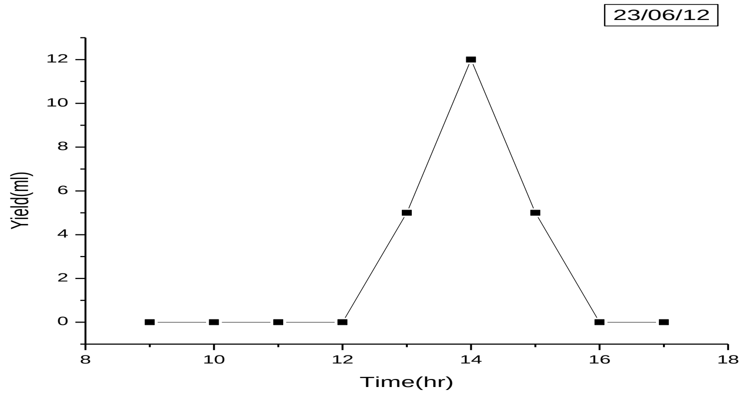

Fig. 3.4: Yield against Time (23/06/12)

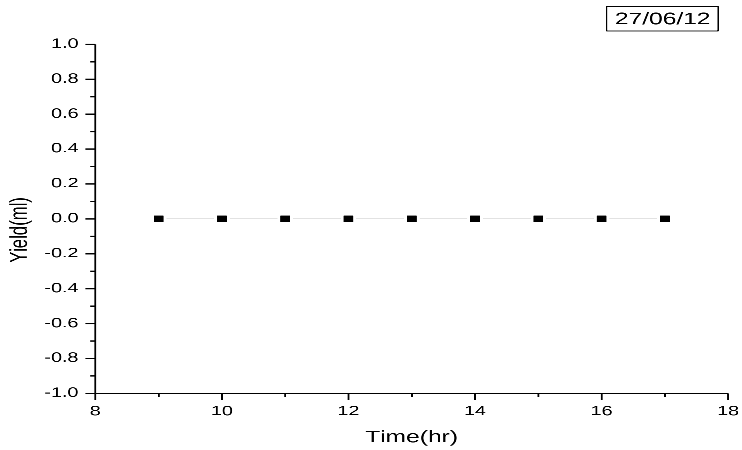

Fig. 3.5: Yield against Time (27/06/12)

References

[1] http://hdr.undp.org/en/report/global/hdr 2006. World Health Organization (WHO),Accessed: June 29,2011.

[2] http//WWW.Solaqua.com/solstilbas .html, Solar Still,Accessed: 22nd September, 2006

[3] G.Nebbia and G.Mcnozzi,(1966),'A Short History of Water Desalination;Acqua Dolec Dal Mare, ‖ inchlests international Milano,p.129.

[4] C.O.Folayan and K.T.Ajayi, (1986) A Compassion of Single Slope and Double Slope Stills", Nigerian Journal of Solar energy, Vol.5, p.169.

[5] WWW.esru.strath.ac.UK/--/Taiwo.pdf. Development of an advance passive solar still with separate condenser. .Accessed:28 November, 2011

[6] WWW.esru.strath.ac.UK/--/Madhlopa.pdf. Improving the performance of solar still using sun tracking mechanism.Accessed:28 November, 2011.

[7] G.O.G.Lof, J.A.Eibling, and Bloemer, Energy Balances in Solar Distillers", American Institute of Chemical Engineers Journal, Vol.7, No.4, 1961, p.641.

[8] M.M Kashimbila,©2009, Fundamental of Physics, First Edition,A.B.U Press Ltd.,Zaria,kaduna State,Nigeria,p.101.

[9] A.Beiser, (1995), Applied Physics, Third Edition, McGraw-Hill, p.261.

[10] A.Beiser, (1995), Applied Physics, Third Edition, McGraw-Hill, p.262.

[11] D.P.Pitts and L.E.Sissom, (1977), Heat Transfer.Schaum Outline series.

[12] M.M Kashimbila,©2009, Fundamental of Physics, First Edition, A.B.U Press Ltd.,Zaria,kaduna State,Nigeria.p.109.

[13] M.M Kashimbila,©2009, Fundamental of Physics, First Edition, A.B.U Press Ltd.,Zaria,kaduna State,Nigeria, p.112-114.

[14] J.A.Duffie and W.A.Beckman, (1980), Solar Engineering of Thermal processes. Wiley New York.

[15] R.S.Adhikari, A. Kumar and G.D.Sootha,(1990), Simulation studies on a Multi-stage stacked tray solar still, Solar Energy Vol.54, NO.5, Pergamum Press Plc.PP.317-325.
[16] T.S Ahmed and M.F Mohammad,(1995), New Development in the Theory of Heat and Mass Transfer in solar stills, Solar Energy Vol.55, No.6, pergamon press plc.
[17] R.S.Adhikari, A. Kumar and G.D.Sootha,(1990), Simulation studies on a Multi-stage stacked tray solar still, Solar Energy Vol.54, NO.5, Pergamum Press Plc.PP.317-325.
[18] J.A.Cusido,J.L.Gonzalez,J.Batlle and J.E.Pelay,(1986),Physics Foundations of Natural Heat Distribution Mechanisms in Passive Solar buildings, An international journal ,Vol.3,No.2,p.112-113.
[19] H.Y Wong, (1977), Handbook of Essential Formulae and Data on heat transfer for Engineers. Longman Group Limited, London
[20] Metrological Station at Malam Aminu Kano International Airport, June 2012.