Performance Simulation of an Active Solar Water Heating System under the Weather of Zaria Using TRNSYS

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Abstract— Solar radiation and weather data of a location which are important driving function for solar system design vary randomly according to location and time of the year. This random variation implies that the performance of a solar system could vary according to the time of the year. This research predicts the long term performance of an active solar water heating system for domestic hot water application under the weather condition of Zaria Nigeria latitude 11.2 ^{o}N and longitude 7.8^{o}N using TRNSYS 16 software. The simulated system consists of a flat plate solar collector, having a total surface area of 2.2 m^2 tilted at 12^0 from horizontal, a thermally insulated vertical storage tank of 100L capacity, interconnecting piping and a solar pump. The monthly average hourly performance of the systems for the recommended average days for the months was numerically simulated based on Typical Meteorological Year (TMY) weather data for the said location. TRNSYS components (TYPES) which represent each physical components of the real system are selected from the components library in the simulation studio. These components are then linked in such a manner that represents how the system works. The parameters describing each component (TYPE) of the system is then modified according the system characteristics. The result of the simulation reveals that the system is capable of meeting a daily domestic hot water load of 100 litres at a minimum temperature 61 ^{o}C at the end of the day (5; 00 pm) for most part of the year except the month of July where the tank temperature dropped to temperature below 34 ^{0}C at the end of the day.

Index Terms— Simulation, Tank outlet temperature, Solar fraction, Collector efficiency TRNSYS, Solar radiation.

1. Introduction

Domestic hot water (water for washing dishes and clothes, for bathing and cooking) use represents a large proportion of energy need in most homes. This energy need accounts for approximately one third of the total annual energy consumption for domestic purposes and therefore a better portion of the family income is spent on domestic hot water.(Retscreen International, 2004). In most homes, domestic hot water usually is provided by an electric or gas-fired water heater, or boiler or furnace that also heats the home. Heating water with electricity is expensive and will become even more costly as the price of fossil fuels used to generate it continues to rise.

The way out of these problems is to explore options which capture and utilizes the abundant solar energy through efficient system design to increase system efficiency. The overall importance is the reduction of the dependence on fossil fuel for our domestic hot water energy need which in turn has the advantage of decreasing the amount of carbon dioxide in the atmosphere and increasing family saving.

The heat converted from solar radiation, is well suited to provide domestic hot water and space heating. The Sun emits energy at a rate of 3.8 x 10^{23} kW, of which, approximately 1.8 x 10^{14} kW is intercepted by the earth, which is located about 150 million km from the sun. About 60% of this amount reaches the surface of the earth. The rest is reflected back into space and absorbed by the atmosphere. About 0.1% of this energy, when converted at an efficiency of 10% would generate four times the world's total generating capacity of about 3000 GW (Mirunalini, *et al.*, 2010). However this energy is known to be highly random and varies with time. This makes solar system performance to also vary through the year. For the system to be able to meet all domestic loads, solar water heating system are operated in concert with an electric or gas-fired backup unit, (Greg, 2003).

In this research work, TRNSYS 16 software is used to evaluate and predict the annual performance of a previously designed active solar water heating system (not presented here) operated under the weather condition of Jos, latitude 10.2^{o}N and longitude 7.8^{o}E.

Several research have been conducted which validated this software. Sebaii *et al.,* (2010) developed a transient mathematical model based on an analytical solution of the energy balance equations for a single pass flat plate solar air heater. The model was found to be able to predict the heater performance with good accuracy. The thermal performance of the heater was investigated by computer simulation for various black painted and selectively coated absorbers. It was found that the best performance was achieved using Ni-Sn as a selective coating material.

Shariah and Shalabi, (1997) have studied optimisation of design parameters for a thermosyphon solar water heater for two regions in Jordan represented by two cities, namely Amman and Aqaba through the use of TRNSYS simulation program. Their results indicate that the solar fraction of the system can be improved by 10±25% when each studied parameter is chosen properly. It was also found that the solar fraction of a system installed in Aqaba (hot climate) is less sensitive to some parameters than the solar fraction of a similar system installed in Amman (mild climate).

2. Theoretical Background.

2.1 Collector heat removal factor

The collector heat removal factor, F_R, is the ratio of actual useful energy gain of a collector to the useful gain if the whole collector surface were at the fluid inlet temperature, F_R is analogous to the heat exchanger effectiveness. For a header-riser flat-plate collector, the collector heat removal factor can be expressed as (Duffie and Beckman, 1991).

$$F_R = \frac{\dot{m}C_p}{A_c U_L}\left[1 - exp\left(-\frac{A_c U_L F'}{\dot{m}C_p}\right)\right] \quad (1)$$

Where F' is the collector efficiency factor expressed as.

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L[D_i + (W - D_i)F]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}}\right]} \quad (2)$$

h_{fi} is the internal fluid heat transfer coefficient and F is the standard fin efficiency for straight fins with rectangular profile, given as:

$$F = \frac{tanh\left[\frac{m(W-D_i)}{2}\right]}{\frac{m(W-D_i)}{2}} \quad (3)$$

Where :

$$m = \sqrt{\frac{U_L}{K\delta}} \quad (4)$$

U_L is the overall heat transfer coefficient
$k\delta$ is the plate thermal conductivity andthickness product. D_i is the internal diameter of tube.

2.2 Collector Top Loss

An approximate relation for collector top loss coefficient (U_{top}) given by (Duffie & Beckman, 1991):

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

Where:

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

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

$$e = 0.430\left(1 - \frac{100}{T_{pm}}\right) \quad (8)$$

2.3 Collector useful energy

In steady state, the performance of a flat-plate solar collector can be described by the useful gain from the collector, Q_u, which is defined as the difference between the absorbed solar radiation and the thermal loss. Duffie and Beckman, (1991) expresses the useful energy gain of a solar collector for glazed or evacuated collectors as:

$$Q_u = F_R[S - U_L(T_i - T_a)]^+ \quad (9)$$

The + superscript indicates that only positive values of the terms in the square brackets are to be used. Thus, to produce useful gain greater than zero the absorbed radiation must be greater than the thermal losses.

2.4 Sensible heat requirement.

The sensible heat requirement is the energy needed to raise the temperature of water to desire temperature. If water flowing at the rate of $\dot{m}_L$ is to be heated from a mains supplied temperature T_i to a desire temperatureT_L, the energy requirement over a specified time horizon may be expressed as follows (Alireza, *et al.*, 2009)

$$Q_L = \dot{m}_L C_p (T_L - T_i) \quad (10)$$

2.5 Storage tank losses.

Losses from the storage tanks may be significant. The rate of tank losses is estimated from the tank loss coefficient-area product $(UA)_{st}$ and the ambient temperature T_a surrounding the tank, calculated as (Govind, 2006),

$$Q_{st} = U_{st}A_{st}(T_{st} - T_a) \quad (11)$$

For a cylindrical tank, the surface area of the tank is related to the storage volume of the tank by the following equation (Kulkarni, *et al.*, 2006).

$$A_{st} = 1.845\left(2 + \frac{h}{d}\right)V_{st}^{\frac{2}{3}} \quad (12)$$

2.6 Collector and Storage tank temperature

On the basis of Pierson and Javelas (1983) and Duffie and Beckman (1991) analysis, the collector hourly outlet hot fluid temperature T_{co} is determined by the absorber thermal equilibrium equation as:

$$T_{co} = T_{ci} + \left[T_a - T_{ci} + \frac{I_T(\tau\alpha)}{U_L}\right]\left[1 - exp\left(-\frac{U_L A_c}{\dot{m}C_p}\right)\right] \quad (13)$$

Temperature at the inlet of the storage tank could be predicted from the expression (Koffi, 2008):

$$T_{sti} = T_{co} + \left(T_{fco} - T_a\right)\left[1 - exp\left(-\frac{U_o \pi d_o L_o}{\dot{m} C_p}\right)\right] \qquad (14)$$

By neglecting the thermal wall resistance, the heat exchange coefficient between the connecting pipes and the ambient conditions can be determined by the equation (Qin, 1997).

$$U_o = \frac{1}{\frac{1}{2\pi k_{ins}} Ln\left(1+\frac{2t_{ins}}{d_o}\right) + \frac{1}{\pi(d_o+2t_{ins})h_w}} \qquad (15)$$

2.7 Storage tank temperature

Assuming solar radiation intensity on tilted surface, ambient temperature , and load demand to be constant over the specified time step, Solution of the energy balance equation of the solar water heating system of figure 1 can be obtained using simple Euler's integration, to solve for the tank storage temperature T_{St}^{+} (Duffie and Beckman, 1991) as:

$$T_{St}^{+} = T_{st} + \frac{\Delta t}{\left(\rho V_{st} C_p\right)_s}\left[Q_u - Q_{Ls} - U_{st} A_{st}(T_{st} - T_a)\right] \qquad (16)$$

Where Δt is the time horizon which may be a day, a month or a year depending on the period of interest. For satisfying the entire thermal demand, (solar fraction S.F=1) storage tank temperature during the time of the demand must be greater than the desired load temperature.

2.8 System solar fraction.

Solar fraction, is the fraction of the total hot water energy that is supplied by solar system, is calculated using the equation from Buckles and Klein (1980),

$$SF = \frac{Q_L - Q_{Aux}}{Q_L} \qquad (17)$$

Where, Q_L is the total energy removed from the system to support the water heating requirement.
Q_{Aux} is the total auxiliary energy supplied to the system to support the portion of the total load that is not provide by the solar energy. The solar fraction is a better indicator of the system performance compared to the other parameters such as collector efficiency or heat removal factor, since it manifests the overall performance of the entire system not a component.

2.9 Collector efficiency.

The Hottel-Whillier equation defines the efficiency for a solar collector in terms of the collector heat removal factor F_R, given in equation form as (Govind *et al.*, (2008) :

$$\eta_C = \frac{Q_u}{I_T A_c} = \frac{F_R\left[S - U_L(T_i - T_a)\right]^{+}}{I_T} \qquad (18)$$

3.0 Materials and Methods

3.1 System description

The system consists of a flat plate solar collector, having a total surface area of 2.2 m^2 tilted at 12^0 from horizontal, a thermally insulated vertical storage tank of 100L capacity and interconnecting piping. The external casing of the collector is made of wood due to its low conductivity to prevent heat lost in the collector. The casing is covered with a 4 mm thick low iron double glass and sealed with a rubber gasket. The absorber plate is made of aluminum. High radiation absorption is achieved by the use of black fine matt paint finish on the aluminum surface, which has a high absorption coefficient. The undersides of the absorber plate and the side casing are well insulated to reduce conduction losses with 50 mm and 30 mm fiberglass insulation, respectively. The flat plate collector has 12 evenly spaced parallel copper pipes 15 mm in diameter embossed by semi-circular grooves formed in the flat plate absorber. A solar pump driven by a photovoltaic panel is installed to drive the fluid round the system.Figure 1 is the photograph of the constructed system used for the simulation.

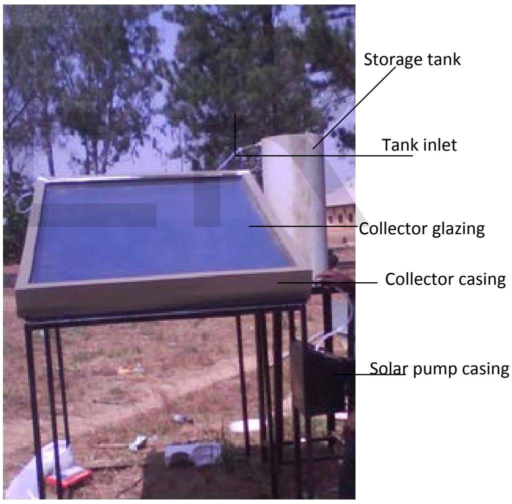

Figure1: Photograph of the simulated system

3.2 Working Principle.

Solar radiation from the atmosphere falling on the collector glazing (glass) passes through the glazing and it is trapped in the collector. This trapped radiation heats up the absorber surface within the solar collector. A heat-transfer fluid (water) flowing through the tubes attached to the absorber plate by means of a solar pump driven by a photovoltaic panel picks up the heat from the absorber plate and move to the top of the storage tank, and a denser water flows from the bottom of the storage tank to the collector to again pick the heat trapped within the collector. The process is self-controlled because it continues until there is no heat from the sun. The heated water is reheated by other forms of conventional heaters when the solar system fails to meet up the water temperature requirement.

3.3 Simulation Procedure.

The monthly average hourly performance of the systems for the recommended average days for the months (table 1) was numerically simulated using TRNSYS 16 software (Klein S. A. *et al.*, 1996) based on Typical Meteorological Year (TMY) weather data for the location Zaria. TRNSYS components (TYPES) which represent each physical components of the real system are selected from the component library and dropped in the simulation studio. These components are then connected in such a manner that represents the flow of information from one component of the system to another. The parameters describing each component (TYPE) of the system is then modified according the system characteristics of figure 1 as indicated in table 2.

Table1 Recommended average days for the months

Month	Average day	Value of n (day) for simulation
January	17	17
February	16	47
March	16	75
April	15	105
May	15	135
June	11	162
July	17	198
August	16	228
September	15	258
October	15	288
November	14	318
December	10	344

Source: klein(1977); Duffie and Beckman, (1991)

Table 2: System characteristics

Description	Value/Type
Total collector area	2.2 m^2
Collector tilt	12.0°
Collector no. of glazing	2
Pump flow rate	100kg/hr
Storage tank capacity	100L
Riser tubes material	Copper
Number of riser tubes	12
Absorber surface	Painted matt black
Glass type	4 mm low iron glass
Collector insulation	Saw dust 30 mm thick for both sides and back insulation.

The system schematic model used for simulation is as shown in figure1. The results of the simulation as predicted by the model are plotted by the plotters and are shown in the next section.

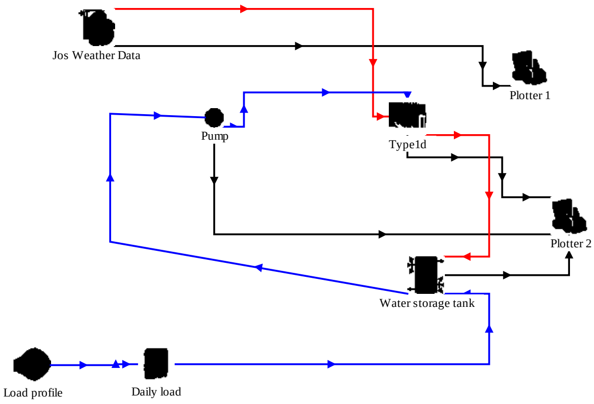

Figure 2: System simulation model in TRNSYS studio.

4.0 Results and Discussion.

Figure 3 to figure 5 shows the monthly average hourly temperature of water in the storage tank simulated for the recommended average days of the months as in table1. Figure 3 shows an increase in the storage tank water temperature of the system increases from temperature slightly above 22 °C in the morning (8.00 am) to a maximum temperature of slightly above 72°C in the evening (5.00pm) for the months of January to April. This implies that the system would be capable of supplying 100L of domestic hot water at a minimum temperature of 70 degrees at the end of each day for the months of January to April.

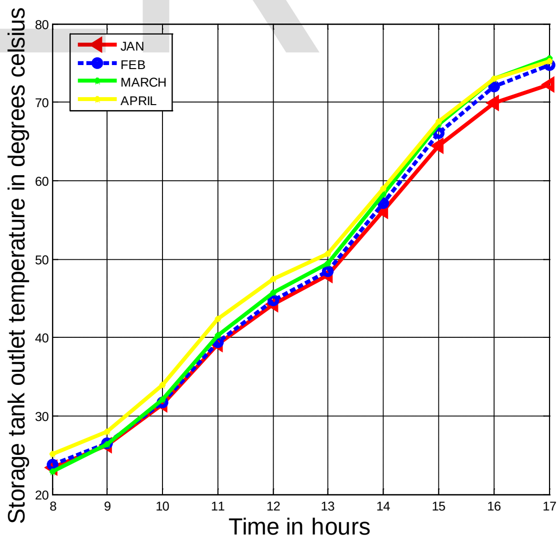

Figure 3: Hourly variation of system storage tank outlet temperature for the months of January to April.

Figure 4 is the hourly variation of the storage tank water temperature for the recommended average day of the month for the months of May to August. The graph shows that a maximum temperature of 64 degrees at the end of the day is attainable in August. The storage tank temperature for the months of May, June and July is less than 64 °C. The month of July has the least storage tank water temperature of 34°C at the end of the day. This indicates a drop in performance of the system as compare to the performance in the months of January to April. The poor performance of the system is attributed the poorer amount of solar radiation received on the surface of the collector for these months as seen from the simulation (not shown here). To meet up with all domestic hot water need, higher amount of auxiliary energy must be provided for the month of July.

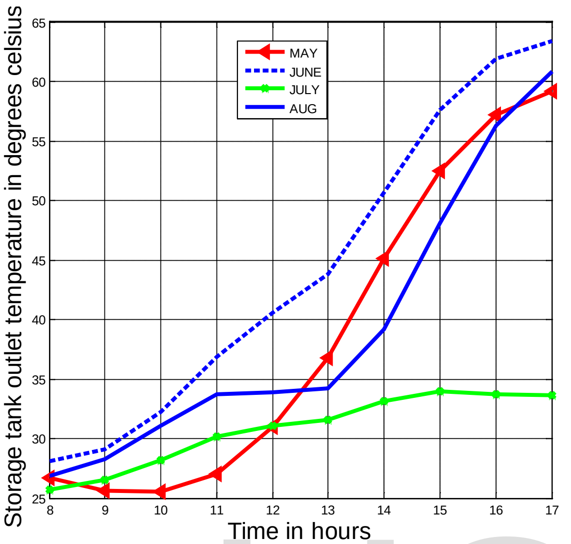

Figure 4: Hourly variation of system storage tank outlet temperature for the months of May to August.

Figure 5 shows the hourly performance of the system for the months of September to December. The figure reveals again an increase in performance of the system. The storage tank temperature increased from the initial water temperature in the morning to temperature slightly above 72 ^{o}C at the end of the day for these months. This again implies that the solar system is capable meeting a very large percentage of the domestic hot water energy need for these months.

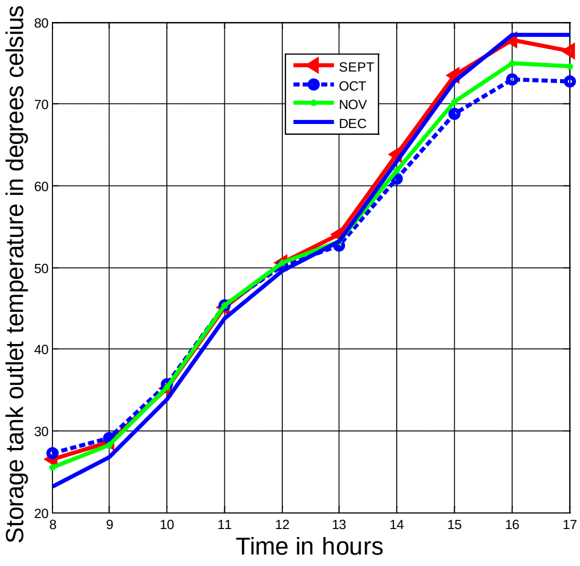

Figure 5: Hourly variation of system storage tank outlet temperature for the months of September to December.

Figure 6 to figure 8 shows the hourly fraction of load temperature (90^{o}C) met by the solar system from 8.00am to 5.00 pm local time for the months of January to December. The graphs show that a minimum of 83% of the domestic hot water load is met by the solar system in January to April. This implies that a greater part of the domestics hot water load would be made by the solar system for these months and only 17 % would supply by other conventional means. The solar fraction for the months of May to August as indicated in figure 7 shows that only 70% portion of the domestic hot water load is met by the solar system in June, 64% in August and only 15% is met in July. The lowest performance of the system as indicated in the month of July implies higher amount of auxiliary would be required to meet load demand compare to its performance in January to April. The portion of domestic hot water load met by solar system in the months of September to December increases again to a minimum of 90% as shown in figure 9. This indicates a good save in energy required to rise water temperature to load temperature by conventional means.

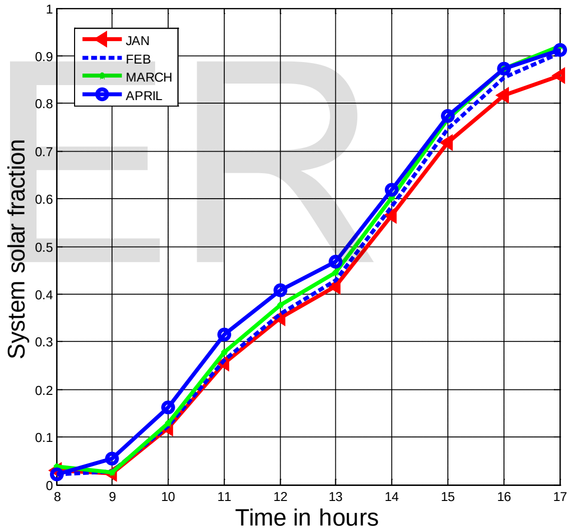

Figure 6: Hourly fraction of domestic hot water load met by solar system for the month of January to April.

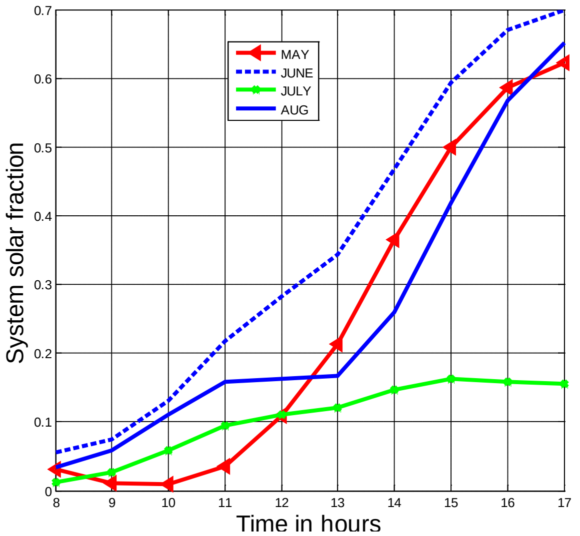

Figure 7: Hourly fraction of domestic hot water load met by solar system for the month of May to August

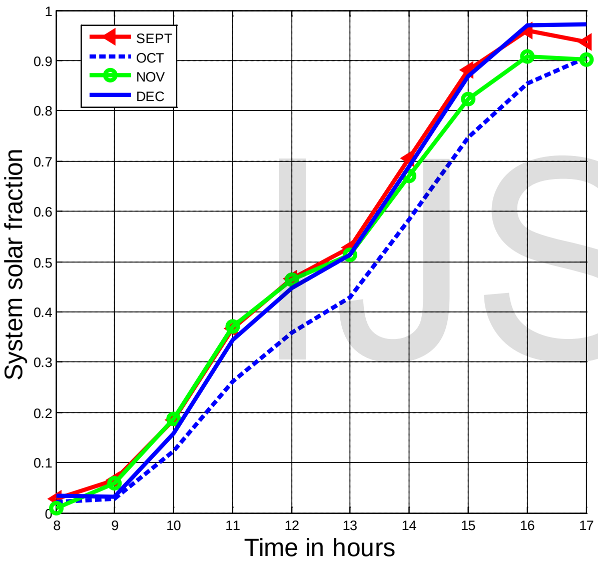

Figure 8: Hourly fraction of domestic hot water load met by solar system for the month of Sept to December..

Conclusions

The annual performance of an active solar water heating system designed, for the city of Zaria latitude 11.2°N and longitude 7.8 °E was successfully predicted through simulation using TRNSYS 16. The aim is to predict the hourly temperature of the storage tank for each month and to determine the extent by which the system would be capable of meeting the domestic hot water need of the location. The following conclusions were drawn from the results of the simulation:

1. A solar collector with area of 2.2 m² and of system characteristics as described in table 2 is capable of meeting 60% to 80% of domestic hot water load of 100L set at 90°C in Zaria.
2. The best performance of the system for this location is in the months of January to April and September to December.
3. The performance of the system needs to be complimented with other sources of energy to meet load for most day in the months of May to August and maximum auxiliary to meet hot water demand is in the month of July.

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Nomenclature

F_R heat removal factor
$\dot{m}$ collector fluid mass flow rate (kg/hr.m²)
C_p fluid specific heat (KJ/kgK)
A_c collector area (m²)
U_L overall collector Loss Coefficient (kJ/h.k)
N_G number of glass covers
β collector tilt (degrees)
ε_g emittance of glass (0.88)
ε_P emittance of plate
Ta ambient temperature (°C)
T_{pm} mean plate temperature (K)
h_w wind heat transfer coefficient (W/m².C)
Q_u useful energy gain per unit time (W/m²)
F_R collector heat removal factor,
U_L overall loss coefficient of the collector (W/m².K)
A_c collector area (m²)
S absorbed solar radiation. (W/m²)
T_i ,Ta collector fluid inlet and ambient temperatures respectively. (°C)
W represents tube spacing, (m)
Cb contact resistance, (W/m k)
C_p Specific heat (J/Kg)
Q_L quantity of heat required to met the load
T_L desired load (hot water) temperature, °C
$\dot{m}_L$ desired load mass flow rate, kg/s
U_{st} tank loss coefficient (W/m².)
A_{st} tank surface area (m²)
T_{st} storage temperature (°C)
T_{co} collector outlet temperature (°C)
T_{ci} collector inlet temperature (°C)
I_T hourly solar radiation on tilted surface.
U_o Loss coefficient of collector outlet pipe plus insulation (W/m².K)
T_{sti} storage tank inlet temperature (°C)
L_o length collector outlet pipe to the tank inlet
d_o outer diameter of collector outlet pipe
t_{ins} thickness of insulation material (m)
T_{ci} fluid inlet temperature (°C)
$\dot{m}$ mass flow rate of fluid the collector (kg/s)
k_{ins} thermal conductivity of insulation material
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