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Techno – economic Simulation and Optimization of a Solar assisted Absorption Air Conditioning System for an Office Block in Zaria, Nigeria

***Ahmadu, T. O., Folayan, C. O., Anafi, F. O. and Yawas, D. S.**
Department of Mechanical Engineering, Faculty of Engineering, Ahmadu Bello University, Zaria, Nigeria

Abstract - *In this study, a techno – economic simulation and optimization of a solar assisted absorption air conditioning system has been done for an office block with an average space cooling demand of 151.8kWh/m^2 – annum. The TRNSYS 16 software was used to model and simulate the solar assisted absorption air conditioning system. Meteorological data over a period of a typical year for Zaria in Nigeria where the office block is located was used. The economic evaluation was done using the Life Cycle cost Savings (LCS) method. The system technical simulation and optimization process was done by employing a monthly average data approach, in which the TRNSYS software was combined with Microsoft excel. The simulation was done on an hourly time step, technical optimization was done by studying effect of varying system component sizes on performance indices: Coefficient of Performance (COP) and Solar Fraction (SF). The Life cycle savings was expressed in a generalized form in terms of two economic parameters P_1 and P_2 which relate all Life cycle cost considerations to the first year fuel cost or the initial solar system investment cost. Results from the techno – economic optimized system indicate that with 40m^2 collector area significant primary energy savings was achieved.*

Keywords: *Solar absorption, air conditioning, TRNSYS, techno-economic, optimization*

1.0 INTRODUCTION

The energy demand for air conditioning has increased continuously in the last few decades, especially in developing countries (Song et al., 2009). This increase is caused among other reasons by a rise in thermal load of air conditioned spaces to ensure occupants comfort, optimum performance of human and equipment in the space.

Conventional vapour compression air conditioning units are usually employed to meet these cooling energy demands. However, these units consume a considerable amount of power. In addition, these units contribute to environmental degradation because of the refrigerants which they use (chloro fluoro carbon (CFCs) and hydro flouoro carbon (HFCs)) (Estiot et al., 2007).

A very attractive and promising alternative is solar cooling, the reason being that the cooling load usually reaches its peak value when solar energy is mostly available (Li and Sumathy, 2000). To this end, absorption air conditioning has received attention in the last few decades. One of the major advantages of this technology is in its ability to use low grade energy, such as waste heat from industrial processes, solar energy etc. which come at little or no cost at all.

Over the years, solar absorption air conditioning has received a lot of attention, the ultimate aim being to achieve high solar fractions to save primary energy. Solar fractions need to be higher than about 50% to start saving primary energy (Mendes et al., 1998).

In spite of favourable energy savings prospect, solar cooling technology is still somewhat a scarcity, due to their high initial cost and to the lack of knowledge on system implementation and expected performance (Blackman et al., 2015).

An economic analysis of a solar cooling facility is necessary in order to determine the least cost of meeting the energy needs, considering both solar and non-solar alternatives.

Zaria is a city geographically located in the northern Guinea Savanah region of Nigeria, on latitude 11.08^{o}N and longitude 7.71^{o}E at 661m above sea level, with ambient temperatures rising to between 35^{o}C and 38^{o}C in the hot months (Eke, 2011). In Zaria, cooling of office buildings is done using conventional vapour compression units. Owing to the high cost of electricity, this leads to financial burden on organizations and institutions.

There have been extensive theoretical and experimental studies in solar absorption air conditioning and its economic analysis. Modelling of a solar powered absorption cooling system for Abu Dhabi has been done by Islam et al., (2012). Simulation using TRNSYS software was done by Balghouthi et al., (2007) to select and size different components of a Solar Absorption Cooling (SAC) system for Tunisia condition. Simulation and optimization of a LiBr/H_20 solar absorption cooling system for Malaysia using evacuated tube collectors in TRNSYS was done by Assilzadeh et al., (2005). (Yeung et al., 1992) carried out the design and installation of a 4.7kW solar absorption chiller using flat plate collectors at the University of Hong Kong. An experimental

*Corresponding author Tel: +234-8023824226
Email: talibahmadu@gmail.com

investigation of part load solar energized cooling system for typical Spanish houses in Madrid was carried out by Seyd et al., (2005). (Mendoza et al., 2013) carried out a TRNSYS modelling and simulation of a solar absorption cooling system under the weather condition of Guayaquil, Ecuador resulting in an annual solar fraction of 0.6. An economic viability and optimization of a solar cooling system was done by Abdulateef et al., (2009) for the Malaysian climate. Techno – economic analysis of a solar cooling system for Thailand has been carried out by Yougprayun et al., (2007). Techno – economic evaluation of solar heating and cooling systems for Spanish climate was done by Blackman et al., (2015).

However, despite the vast solar energy potential in Nigeria, especially the northern part, there is limited information on the performance prediction and economic viability of a solar absorption air conditioning system. Also most solar absorption cooling simulation works using TRNSYS software have used data for single day to represent performance for a month. This is done using a single recommended day for each month as reported by Klein (1977).

This work was aimed at carrying out a technical and economic viability study of a Solar assisted Absorption Cooling (SAC) system for an office block in the city of Zaria in northern Nigeria.

2. Building Cooling Energy Demand Simulation

The building to be cooled is an office block in the department of mechanical engineering, Ahmadu Bello University, Zaria, Nigeria. The building consists of 5 office rooms and a passage covering a total floor area of 90m^2 and 3m in height. Fig. 1 is the plan view of the office block showing the orientation and dimensions. The building has been divided into thermal energy zones as seen in figure 1.

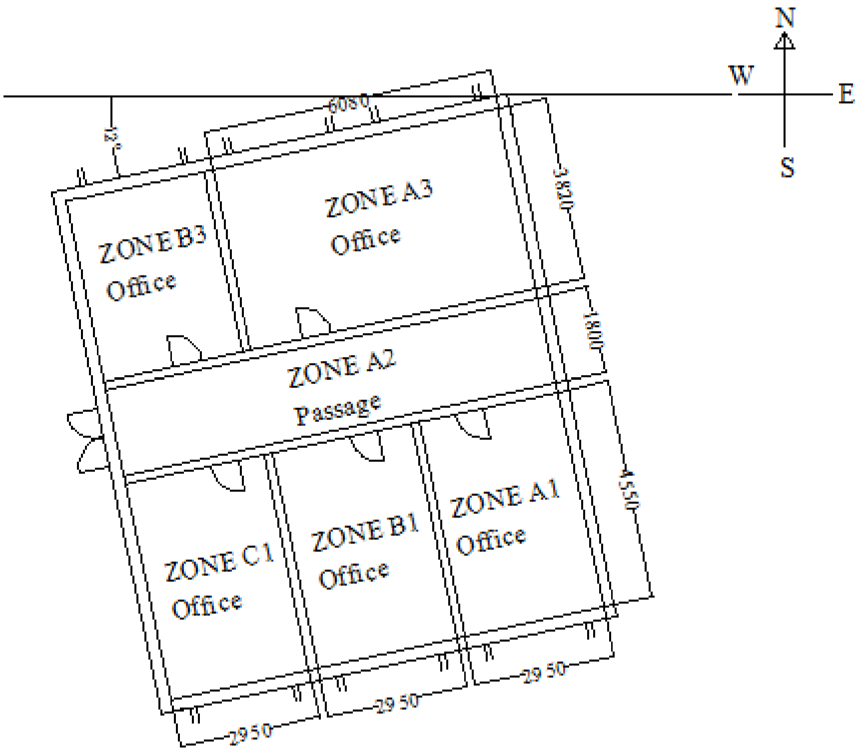

Fig. 1: Plan view of the building (dimensions in mm)

2.1 The TRNSYS software

TRNSYS is a well-known software diffusely adopted in the academia and industry for the design of dynamic thermal energy systems. It is a very reliable tool for the simulation of thermal systems, especially based on solar technology (TRNSYS manual, 2006). Its library consists of a large number of components, these components are referred to as 'types'. Each type is defined by a mathematical model in the TRNSYS simulation engine and has a set of matching profomas in the simulation studio. The simulation engine is programmed in FORTRAN.

2.2. Building cooling energy demand simulation using TRNbuild

The building cooling energy demand simulation was done using a TRNSYS sub program, TRNbuild. A maximum cooling energy demand of 11.05 kW was obtained for the building block and this occurred in the month of April, this formed the basis for sizing the absorption chiller. Also, the month of August had the lowest solar radiation of 795.5 W/m^2, this formed the basis for sizing the solar collector area. A cooling set point of 24^{o}C was chosen for the building block. Details of the building cooling load simulation have already been presented in (Ahmadu et al, 2016).

Fig. 2 shows the variation in average monthly solar radiation on a flat surface for Zaria.

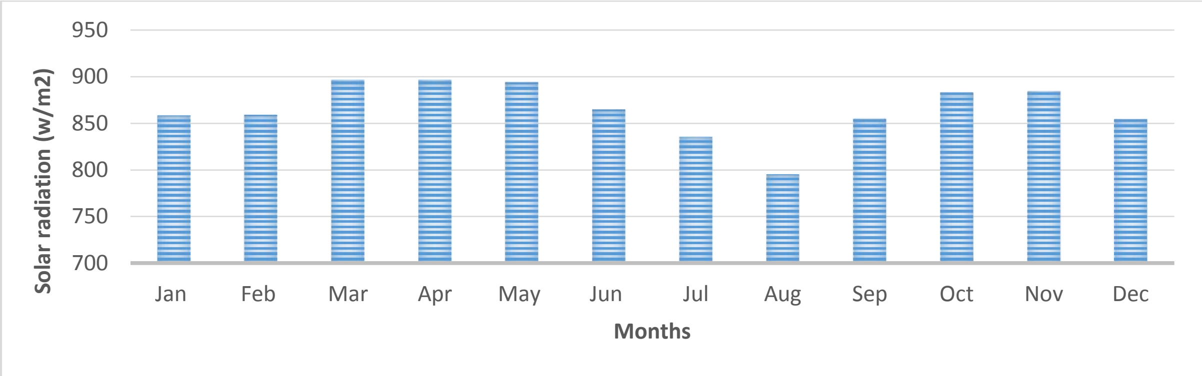

Fig. 2: Average monthly solar radiation on flat surface for Zaria

3.0 SOLAR ABSORPTION COOLING SYSTEM MODEL AND SIMULATION

A SAC system was modelled in TRNSYS to cover the cooling energy demand. Simulation and optimization of the SAC system was done using the TRNSYS software. The TRNSYS software was combined with Microsoft excel, using a monthly average data approach, where hourly data was averaged to obtain daily data and daily data was averaged to obtain monthly data. Life cycle Cost Savings (LCS) method was used in the economic evaluation.

3.1. System description

The SAC system consists of three flow loops, the hot water loop, the cooling water loop and the chilled water loop, as shown in fig. 3. Each of these loops is connected to the absorption chiller.

In the hot water loop, solar radiation heats water in the flat plate collector array, the water is circulated via a pump and it's stored in a hot storage tank, from the tank, the hot water is pumped into the generator (G) of the absorption chiller where desorption of Lithium bromide – water solution takes place. The hot water is then pumped back into the storage tank. From the storage tank, it is pumped back to the collector array where heating again takes place and the cycle continues. An auxiliary heater is attached to heat the water in the hot storage tank during periods of low solar radiation, the auxiliary heater is fuelled by liquefied natural gas.

In the cooling water loop, cooling water from a cooling tower is pumped first to the condenser (C) and then to the absorber (A), (both in the absorption chiller) to take away the heat of condensation and absorption respectively before returning to the cooling tower. In the cooling tower, the water is cooled and the process continues.

In the chilled water loop, chilled water from the evaporator (E) of the absorption chiller is pumped to the building, where it circulates round the building through cooling coils and takes away the cooling load. The water rises in temperature and is then returned to the chiller where it is again chilled at the evaporator.

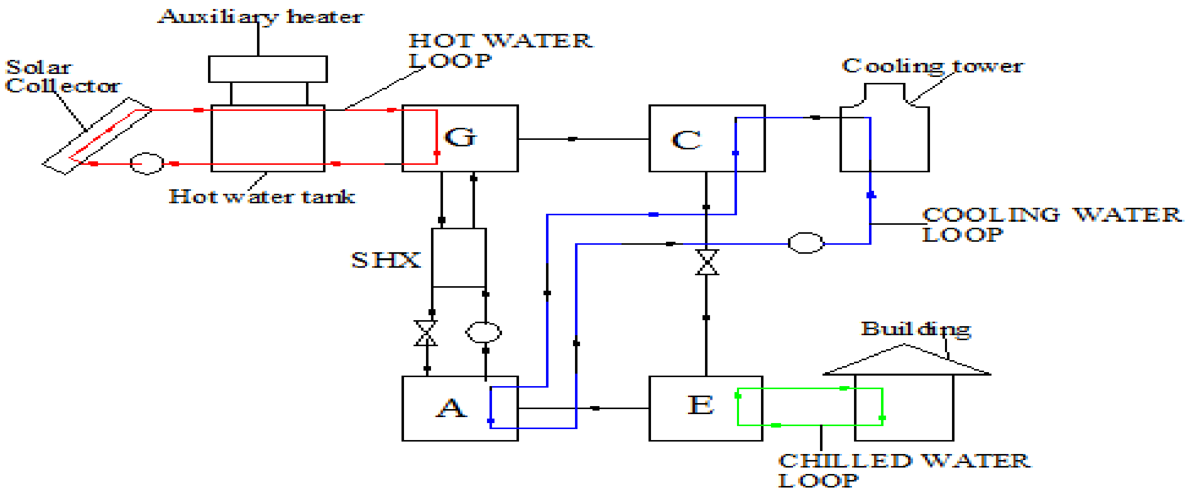

Fig. 3: Schematic of SAC system (G: generator, C: condenser, E: evaporator, A: absorber, SHX: solution heat exchanger)

3.2. *Initial components sizing*

Initial sizing of system components was done to obtain a base line over which simulation was carried out. Table 1 shows initial system component sizes over which simulation was done. Details of system components sizing has already been presented in (Ahmadu et al, 2016).

Table 1: Initial component sizes

Component	Size
Absorption chiller	12 kW
Solar collector array	26.93m^2
Cooling tower	29.14 kW
Hot water storage tank	1.2 m^3
Mass flow rate of hot water loop	0.416 kg/s

3.3. *TRNSYS types used in the model*

The initial components sizes as obtained in table 1 formed the base line over which simulation was done in the TRNSYS studio. The TRNSYS model of the SAC system, as seen in fig. 4, is composed of types that mathematically describe the operation of each of the individual components that make up the SAC system.

Each of these types have inbuilt models programmed in FORTRAN. The types are connected by links, with the output from one being the input to the next. The model is integrated with the building cooling load simulation model from where it reads the cooling load data file.

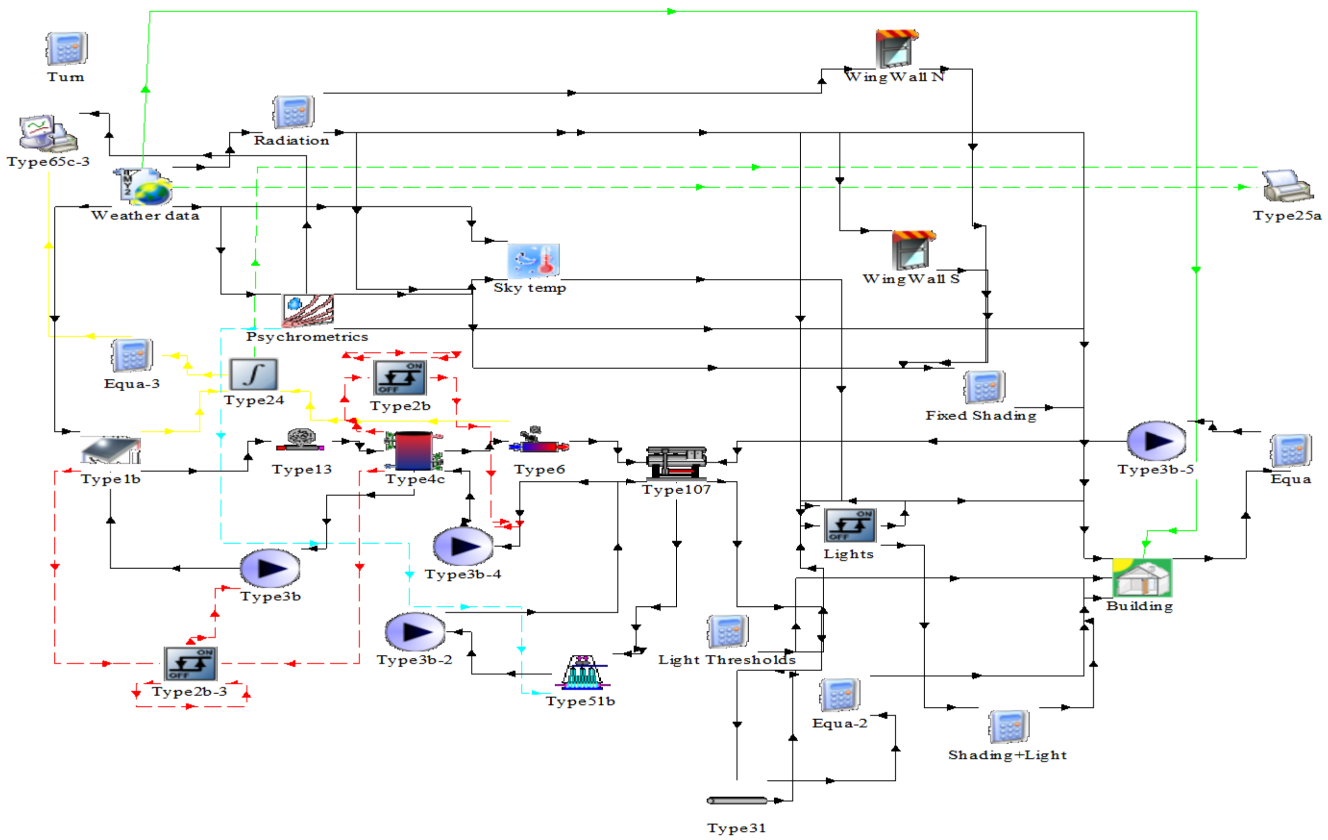

Fig. 4: Model of the SAC system in TRNSYS studio

The types that model the SAC system components are as follows:

3.3.1. *Type 107*

Type 107 models the single effect absorption chiller, operating with hot water at the generator. This it does by interactive calls with an external performance data and using equations 1 – 6 as reported in the TRNSYS manual (2006).

The energy delivered to the hot water stream at the generator is calculated as:

$$\dot{Q}_{hw} = \frac{C_{rated}}{COP_{rated}} f_{dei} \quad (1)$$

Where f_{dei} is the faction of the design energy at which the chiller will operate at the present conditions of chilled water set point, entering hot water and entering cooling water temperatures. This it does by interaction via a dynamic data subroutine with the external performance data.

C_{rated} is the rated capacity of the chiller (kW)

COP_{rated} is the rated Coefficient of performance of the chiller

$\dot{Q}_{hw}$ is the thermal energy transfer at the hot water stream (kJ/s)

The hot water stream outlet temperature is calculated as:

$$T_{hw,o} = T_{hw,i} - \frac{\dot{Q}_{hw}}{\dot{m}_{hw}C_{p,hw}} \quad (2)$$

Where:

$T_{hw,o}$ is the temperature of the hot water at exit (°C)

$T_{hw,i}$ is the temperature of the hot water at inlet (°C)

$\dot{m}_{hw}$ is mass flow rate of the hot water stream (kg/s)

$C_{p,hw}$ is the specific heat capacity of the hot water stream (kJ/kgk).

The chilled water outlet temperature which should be equal to the set point temperature but may be greater if the chiller is not operating at the rated capacity is given as:

$$T_{chw,o} = T_{chw,i} - \frac{\dot{Q}_{chw}}{\dot{m}_{chw}C_{p,chw}} \quad (3)$$

Where:

$T_{chw,o}$ is the temperature of the chilled water at exit (°C)

$T_{chw,i}$ is the temperature of the chilled water at inlet (°C)

$\dot{m}_{chw}$ is mass flow rate of the chilled water stream (kg/s)

$C_{p,chw}$ is the specific heat capacity of the chilled water stream (kJ/kgk).

$\dot{Q}_{chw}$ is the energy removed from chilled water Stream (kJ/s)

The cooling water stream outlet temperature is calculated as:

$$T_{cw,o} = T_{cw,i} + \frac{\dot{Q}_{cw}}{\dot{m}_{cw}C_{p,cw}}$$

(4)

Where:

$T_{cw,o}$ is the temperature of the cooling water at exit (°C)

$T_{cw,i}$ is the temperature of the cooling water at inlet (°C)

$\dot{m}_{cw}$ is mass flow rate of the cooling water stream (kg/s)

$C_{p,cw}$ is the specific heat capacity of the cooling water stream (kJ/kgk).

$\dot{Q}_{cw}$ is the thermal energy transfer to the cooling water stream (kJ/s).

The performance indices are calculated as follows:

The machine coefficient of performance (COP) is calculated as:

$$COP = \frac{\dot{Q}_{chw}}{\dot{Q}_{hw}+\dot{Q}_{aux}} \quad (5)$$

$\dot{Q}_{aux}$ is the energy supplied to the auxiliary Components (kJ/s).

The solar fraction (SF) denotes the fraction of the cooling load that is met by solar energy, it's calculated according to Duffie and Beckman (2013) as:

$$SF = \frac{\dot{Q}_{coll}}{\dot{Q}_{coll}+\dot{Q}_{aux\ heat}} \quad (6)$$

Where:

$\dot{Q}_{coll}$ is the energy delivered to hot water stream by the collector array (kJ/s)

$\dot{Q}_{aux\ heat}$ is the energy delivered to hot water stream by the Auxiliary heater (kJ/s)

3.3.2. *Other types used*

Type 1 models the flat plate solar collector, type 4 models the hot water storage tank, type 51b models the wet cooling tower, type 6 models the auxiliary heater, type 3b models the pumps, type 2b together with the equation editors in which user defined equations are entered, are used to implement the system controls, type 65 is a plotter that displays simulation results.

3.4 Technical simulation and optimization procedure

The simulation was carried out in the TRNSYS studio using the TRNSYS model of the SAC system. Based on the values obtained from the initial sizing of components, system parameters were varied to achieve a SAC system that will meet the cooling energy demand of the building block for the whole year, as well as achieve a maximum COP and solar fraction.

The absorption chiller capacity was kept constant at 12kW. The collector area was varied from 5m^2 to 60m^2. For maximum annual energy availability, according to Duffie and Beckman (2013), a surface slope equal to latitude is best. Therefore, the collector was sloped at an angle of 11.08°. The hot water storage tank volume was varied from 0.3m^3 to 1.8m^3. Cooling tower capacity was kept constant at 29.14kW. The flow rate of the hot water loop was varied from 0.2kg/s to 0.6kg/s.

Simulation was done using an hourly time step for the year round. For every simulation, a graphical display of temperature variation within the zones for the whole year is displayed. In addition, TRNSYS generates an output file where hourly values of COP, and solar fraction are stored. This data file was exported to Microsoft excel. For each parameter, hourly data for each day were averaged to obtain daily average data. The daily average data for each month were then averaged to obtain monthly average data. These monthly averaged data were used in the optimizing process by studying the effect of varying collector area, storage tank volume and mass flow rate on COP and Solar Fraction on a monthly basis. The cooling energy demand was considered to be met when the temperatures within the zones fell below the cooling set point of 24°C for the year round.

3.5 Economic viability simulation and optimization procedure

Several methods of economic analysis of solar cooling systems are available in literature. For the purpose of this work, the Life cycle Cost Saving Method (LCS) was used for the analysis. The life cycle saving of a solar absorption cooling system over a conventional system can be expressed as the difference between a reduction in the fuel cost and an increase in expenses incurred as a result of the additional investment for the solar system (Duffie and Beckman, 2013).

$$LCS = P_1 \times C_F \times L \times F - P_2 \times C_S \qquad (7)$$

Where:

C_F is the unit cost of delivered conventional Energy for the first year of analysis.

L is the annual load (kWh)

F is the annual fraction of load supplied by Solar energy

C_s is the total cost of installed solar energy Equipment (Naira), given by:

$$C_S = C_A \times A_C + C_E \qquad (8)$$

Where:

C_A is the collector area dependent cost (Naira/m^2)

A_c is the collector area (m^2)

C_E is the collector area independent cost (Naira)

P_1 and P_2 are given by:

$$P_1 = PWF(N_e, i, d) \qquad (9)$$

PWF = Present Worth Factor

N_e = Number of years of economic analysis

i = Inflation rate on cost of conventional fuel

d = Discount rate

$$PWF(N_e, i, d) = \frac{1}{(d-i)}\left[1 - \left(\frac{1+i}{1+d}\right)^{N_e}\right] \qquad (10)$$

The factor P_2 accounts for investment related expenses which in the present work consists only of the investment, since the investment is made at once and is only expected in current naira. Therefore for the present work, P_2 is unity.

The building cooling energy demand simulation showed an annual total cooling energy demand of 13,663.14kWh for the building block. This corresponds to an average space cooling demand of 151.8kWh/m^2 – annum.

The present electricity tariff under the Multi Year Tariff Order (MYTO) for Zaria in Nigeria is N 26.37/kWh (MYTO 2.1 report, 2015). The inflation rate on electricity tariff under the MYTO is 3.1% (MYTO 2.1 report, 2015). The Central Bank of Nigeria (CBN) discount rate is 4.25% (World Fact Book, 2016). According to Duffie and Beckman (2013), solar absorption cooling systems have a lifespan of about 20 years. Therefore the number of years of the economic analysis for this work was set at 20 years. These figures were used in evaluation of equation 10.

Table 2 below shows the prices of components used in evaluation of equations 7 and 8. The prices were obtained from a current market survey of components prices from manufacturers in China and Germany. The prices in Dollars were converted to Naira using the current exchange rate of naira to the dollar.

Table 2: Component prices

Component	Price (Naira)
Solar Collector	28,170/m^2
Absorption chiller (12kW)	638,000
Cooling tower (30kW)	550,000
Hot water tank (1.0m^3)	95,700
Installation cost	20% of collector area
Miscellaneous cost	120,000

(Source: online market survey of current manufacturer prices.)

The auxiliary heater is powered by liquefied natural gas. The present cost of liquefied natural gas in Zaria, Nigeria is ₦333.33/kg. This was used in evaluating the cost of auxiliary energy, which contributes to C_E in equation 8.

4.0 RESULTS AND DISCUSSION

Fig. 5 shows the temperature variation within the zones for the whole year with no cooling from the SAC system. The temperatures within the zones can be seen to be high (above the set point temperature of 24°C). These are the simulated temperatures within the zones with no cooling.

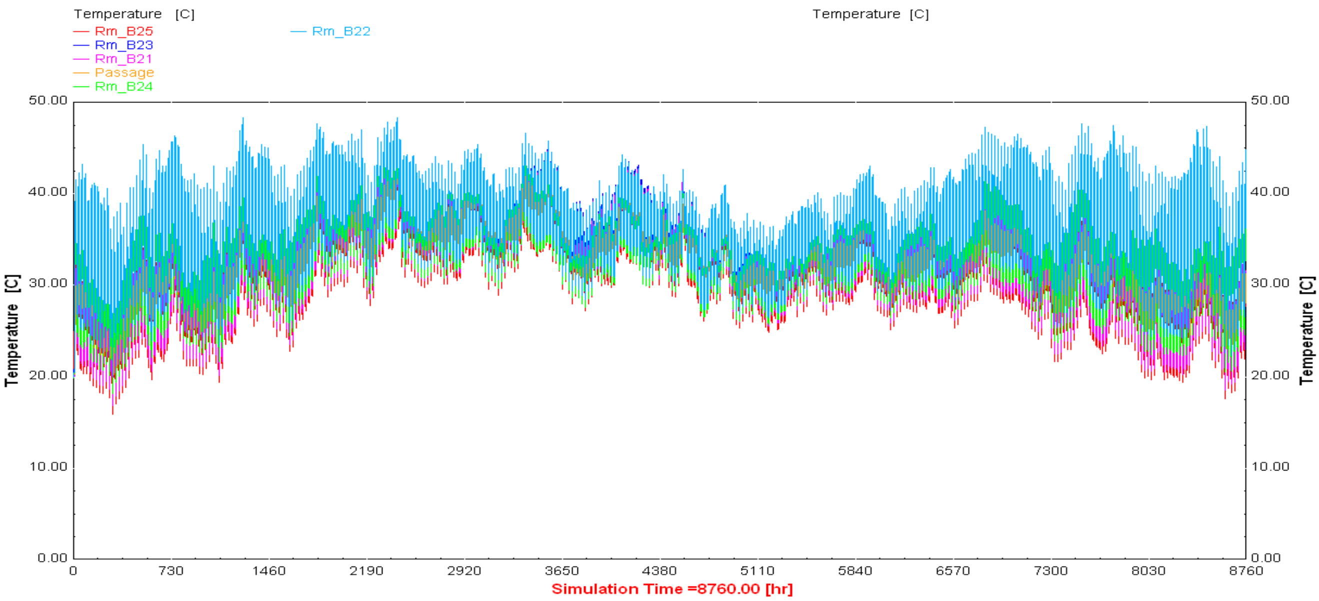

Fig. 5: Year round temperature variation within the zones without cooling

Figs. 6 and 7 show the variation of COP with collector area and storage tank volume for the two extreme months: April with the highest cooling energy demand (also with the highest solar radiation) and August with the lowest solar radiation. Performance in all the other months fall between these extremes.

As seen from both figures, the coefficient of performance increases with increasing collector area; this is because a larger collector area harnesses more thermal energy, increasing the temperature of the hot water supply to the generator, thus resulting to better performance. The COP however gets to a peak where increase in collector area does not translate to significant increase in COP, this starts at collector area between 35m^2 to 40m^2. This is because at this point, the chiller has almost attained its rated COP, increasing the collector area would not necessarily increase the COP beyond the rated limit. Similar trends in COP against collector area have been reported in past works carried out for different climates (Assilzadeh et al, 2005; Mendoza et al, 2013).

Also, the COP decreases with increasing storage tank volume, this is because more thermal energy is required to heat the fluid in the storage tank as the size increases, therefore it takes a longer time interval to get the hot water temperature to the operating temperature range of the absorption chiller. However, as the collector area increases, the larger volume tanks begin to record increase in COP. This is because a larger collector area results in harnessing more thermal energy. The larger tanks now have sufficient thermal energy to raise the temperature of the water in the tank to the required set point to keep the system running even when solar radiation has dropped.

There is a collector area that optimizes the storage tank volume. As seen from both figures, starting from a collector area of 40m^2, the 0.9m^3 tank gives a higher COP than both the 0.3m^3 and 0.6m^3 tanks. For both months, storage tank volume of 0.9m^3 performs best. Maximum COP of 0.62 was attained in the month of April with 40m^2 collector area, while maximum COP of 0.47 was attained with 40m^2 collector area for the month of August. From the system COP therefore, optimum collector area is 40m^2, optimum storage tank volume is 0.9m^3.

Figs. 8 to 10 show the variation of solar fraction with collector area. The solar fraction is seen to increase with increasing collector area for all months. This trend of increasing solar fraction with collector area has also been reported in past works done for different climates (Assilzadeh et al, 2005; Mendoza et al, 2013; Abdulateef et al, 2009). The months of March and April each recorded the maximum solar fraction of 0.9 with a collector area of 40m^2. For all the months, solar fraction of 0.6 can be attained with 30m^2 collector area. However, apart from the months of July and August, which recorded solar fractions of 0.67 and 0.65 respectively with a collector area of 40m^2, a solar fraction of at least 0.7 can be achieved for all the other months with 40m^2 collector area. This is as a result of the lower solar radiation in the cloudy months of July and August. With lower solar radiation, the system gets less efficient, the system then has to rely on the use of more auxiliary heat to meet the cooling energy demand, thus lowering the solar fraction. From the system solar fraction therefore, the optimized collector area is 40m^2.

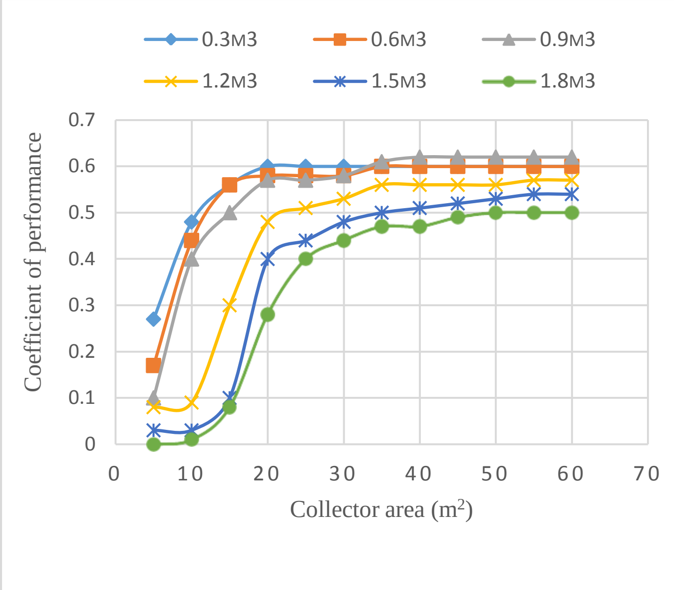

Fig. 6: Variation of coefficient of performance with collector area and hot water tank storage volume for the month of April.

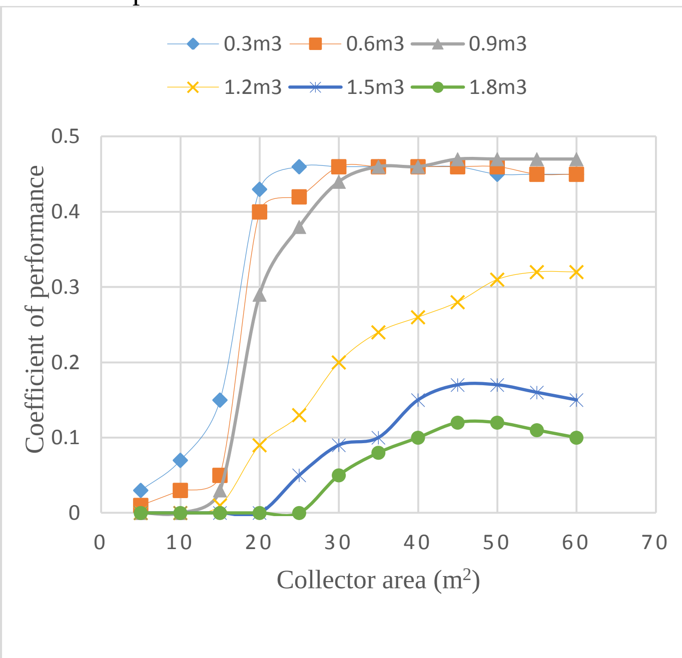

Fig. 7: Variation of coefficient of performance with collector area and hot water tank storage volume for the month of August.

It will be noted here that from the hourly solar fraction data file generated by TRNSYS and exported to excel, solar fraction of 1 was recorded for some hours, especially in the months of March and April. However because of the monthly average data approach adopted in this work, there is no month with an average solar fraction of 1. This has ensured that the system is not over predicted.

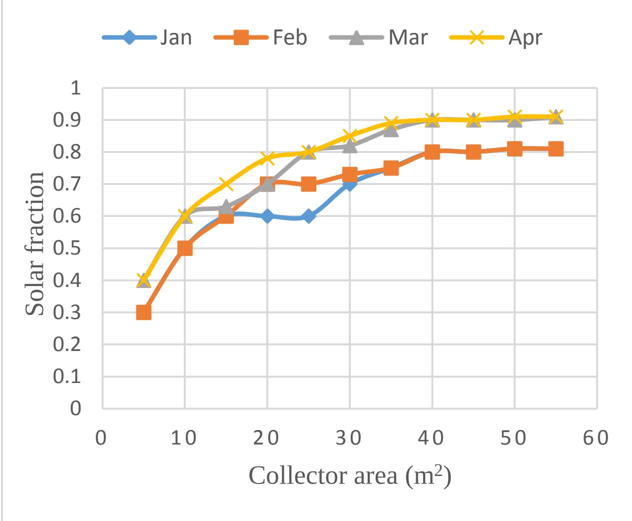

Fig. 8: Variation of solar fraction with collector area for the months of January to April.

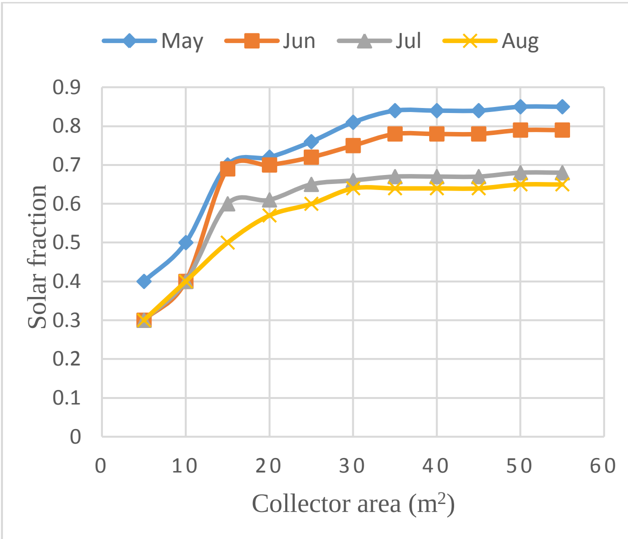

Fig. 9: Variation of solar fraction with collector area for the months of May to August.

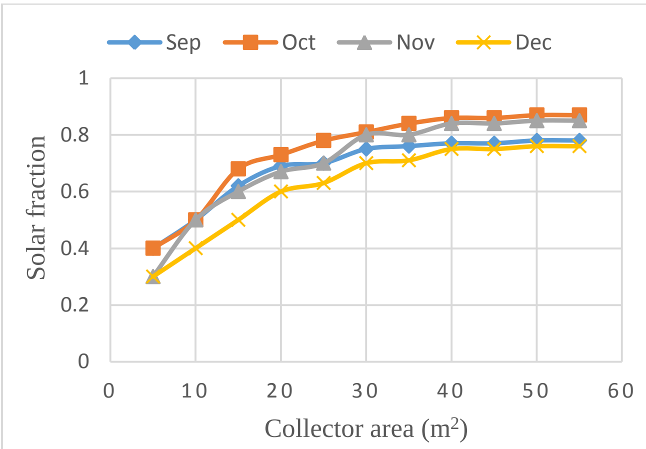

Fig. 10: Variation of solar fraction with collector area for the months of September to December.

Fig. 11 shows the average annual solar fraction as a function of collector area. As seen from the figure, on an average, a maximum solar fraction of 0.8 is achieved as from 40m^2 collector area and this remains so even with further increase in collector area.

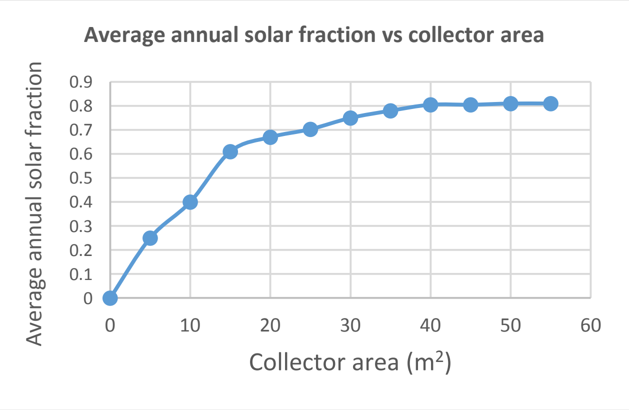

Fig. 11: Variation of average annual solar fraction with collector area.

Considering the COP and SF therefore, the optimized collector area is 40m^2. Fig. 12 shows the variation of total cost of cooling system, cost of electricity covered by the solar cooling system and cost of auxiliary energy used by cooling system as a function of collector area. The total cost of the cooling system comprises the total components cost, installation cost, maintenance cost and running cost. The cost of electricity covered by the solar cooling system is the fraction of electricity cost for cooling the building over the life cycle of the system (20 years) that is met by solar energy. The cost of auxiliary energy is the cost spent on auxiliary energy during the life time of the system.

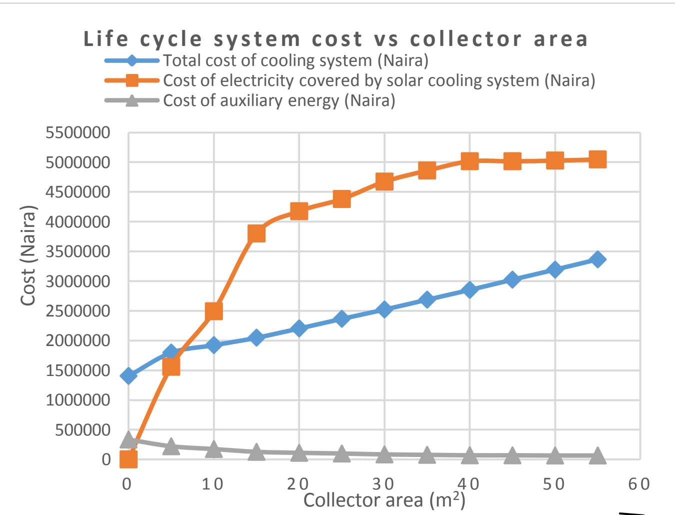

Fig. 12: Variation of total cost of cooling system, cost of electricity covered by solar cooling system and cost of auxiliary energy with collector area.

As seen from the graph, the total cost of the cooling system increases linearly with collector area. This is so as other system components remain constant and increasing collector area also results in an increase in installation cost. The cost of electricity covered by the solar cooling system is seen to increase with increasing collector area. This is so as increasing collector area increases solar fraction, whereby the system is able to meet increasing percentage of the cooling energy demand running on solar energy. However, as from 40m^2 collector area, the cost of electricity covered is seen to remain almost constant with increasing collector area. This is attributed to the fact that the solar fraction also remains fairly constant as from 40m^2 collector area as seen from fig. 11.

Comparing the curves of total cost of cooling system with cost of electricity covered by solar cooling system, it is seen that the curve of cost of electricity covered by solar cooling system begins to peak above that of total cost of cooling system as from 10m^2 collector area. This indicates the savings achievable over the life cycle of the system.

Also, the cost of auxiliary energy is seen to decrease as the collector area increases. This again is because the solar fraction increases with increasing collector area, thus the system depends more on solar energy and less on the auxiliary energy.

Fig. 13 shows the graph of life cycle savings as a function of collector area.

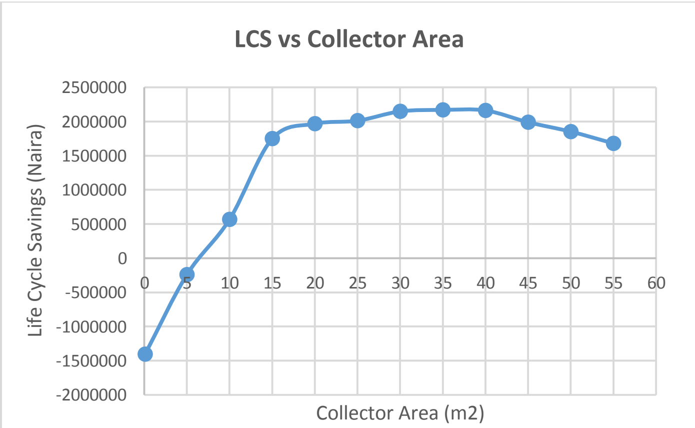

Fig. 13: Variation of life cycle savings with collector area.

As seen from the graph, negative savings are incurred from 0 to 7m^2 collector area, from where the savings start to become positive. As from 10m^2 collector area, savings start to become significant and continues to increase up to 30m^2 collector area. Maximum life cycle savings are achieved between 30m^2 and 40m^2 collector area, with the peak life cycle savings of ₦2, 172,558.00 achieved with 35m^2 collector area. With 40m^2 collector area, the life cycle savings is reduced very marginally to ₦2, 164,401.00. On increasing the collector area above

$40m^2$, then the life cycle savings start to decline rapidly with increasing collector area. This is so because as seen from figure 11, the solar fraction reaches the peak and becomes constant as from $40m^2$ collector area. Increasing the collector area above $40m^2$ results in additional cost of the system without a corresponding offset in the cost of electricity. Similar trend in life cycle savings against collector area have been reported in previous works (Abdulateef et al, 2009; Yougprayun et al, 2007).

Therefore for maximum savings, collector area of $35m^2$ can be adopted. However, in order to maintain maximum possible COP and ensure that the cooling set point temperature in the building is continuously sustained, then a collector area of $40m^2$ should be adopted. This is justified as the maximum COP and cooling capability obtainable with the $40m^2$ collector area far outweighs the marginal decrease in savings on increasing the collector area from $35m^2$ to $40m^2$. From the forgoing, the techno – economic optimized system parameters are: Absorption chiller capacity: 12kW, Cooling tower capacity: 29.14kW, Solar Collector area: $40m^2$, Storage tank volume: $0.9m^3$.

Fig. 14 shows the temperature variation within the zones for the whole year with cooling from the optimized SAC system. From the figure, temperature variation within the zones for the whole year are seen to be below 24°C, an indication that the optimized SAC system has been able to meet the cooling energy demand of the building.

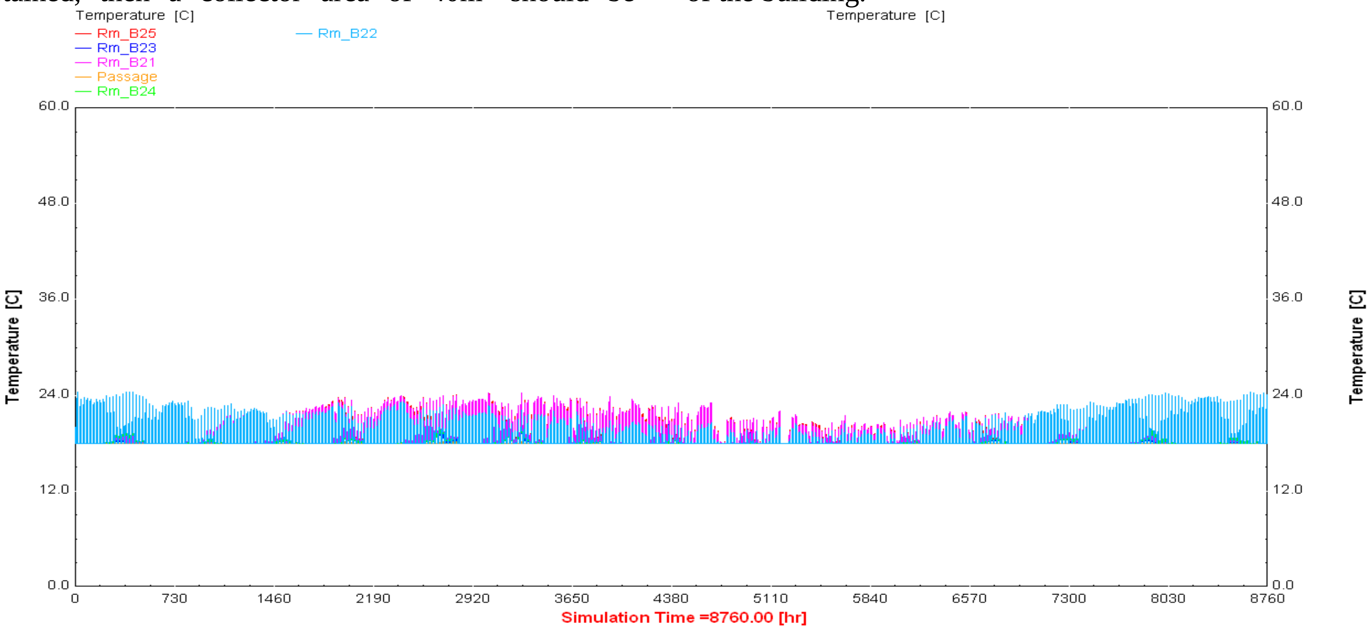

Fig. 14: Temperature variation within the zones for the whole year with cooling from the optimized SAC system.

5.0 CONCLUSION

Techno – economic simulation and optimization of a solar assisted absorption cooling system for an office block in Zaria, Nigeria has been carried out. The office block covers a total floor area of $90m^2$, with an average space cooling demand of $151.8kWh/m^2$ – annum. Results from the technical simulation indicate that with an absorption chiller of 12kW capacity, $40m^2$ area of flat plate collector, $0.9m^3$ hot water storage tank volume, the system attains a maximum COP of 0.62 and average annual Solar fraction of 0.8. Results from the economic analysis indicate that a life cycle savings of ₦2, 164,401.00 is achievable with $40m^2$ collector area.

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