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Performance Evaluation of a Solar Thermal Combined Domestic Hot Water and Adsorption Cooling System

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Abstract— To promote sustainable heating and cooling application, a silica-gel and water pair based combined domestic hot water and adsorption cooling system incorporating solar radiation condition of Port Harcourt, Nigeria is studied. Numerical and analytical techniques were employed in evaluating the performance of the system.A novel configuration of the combined system is presented and optimization performed using POLYSUN software. Mathematical models governing the operating processes of the system were developed from the fundamental principle of thermodynamics and resolved to obtain the interactions and correlation between the system parameters incorporating heat recovery technique to improve system performance. An average Coefficient of Performance of 0.43 during hot periods of the day and up to 0.75 during sunset and sunrise hours were obtained from the system using 20 evacuated tube type collectors with total absorption area of 2.481m2 per collector operating under heat recovery mode. Financial analysis, energy savings and CO_2 footprint reduction of the proposed system were also evaluated using software inbuilt packages and results obtained. Inference from system model results under heat recovery mode demonstrates that the proposed application, meets domestic hot water and cooling demands for the location under review under the model conditions.

Index Terms—Adsorption Cooling,Coefficient of Performance, Domestic Hot Water, Optimization, Simulation, Solar Thermal Collector

1. Introduction

1.1 Background Information

SOUTHERN Nigeria has a range of annual temperature varying from November to April considered as the dry season and May to October representing the wet season each year. The dual season depiction of Nigeria climate is based on its geographical location which is near the equatorial belt hence the tropical climatic condition.

During the Wet period in the south, energy in form of heat from electricity and gas is required for heating to meet domestic hot water needs and during the months of higher insolation in dry season, similar amount of energy is required for domestic cooling needs. Relative to other energy demands, the residential electricity demand for heating and cooling account for over 55% of the total annual electricity generation capacity as at 2013 [1].

As shown in Fig. 1, the average monthly temperature for a typical southern city in Nigeria lies between 20^{O}C and 34^{O}C throughout the yearwith the month of February recording the most significant value with average daily annual radiation of about 4kWh/m^2-day as obtained from World Meteorological Agency-WMOweather forecastdatabase [2] and validated using weather stationdataincorporated in simulation software.

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One of the objectives of this work is to evaluate the viability of providing for Domestic Hot water-DHW and cooling needs using solar thermal application considering the assessed climatic condition for the choice location Port Harcourt (4.0N and 7.2E) in southern Nigeria.

Reflecting on the recent increase in the electricity tariff per kWh as published by the focal electricity regulatory body in Nigeria, National Electric Regulatory Commission-NERC [3], and with over 55% of this tariff utilizedto offset residential demand of Domestic Hot Water-DHW and cooling needs as argued earlier in the opening paragraph, sourced from non-renewable resources to a large extent,it can be inferred that there is the necessity for a more sustainable form of application to complement residential electricity demand and this is where Solar thermal heating and cooling application as presented in this work strategically fits in. An excerpt of the tariff projection to 2019 for the selected location is demonstrated in Fig. 2 with electricity tariff significantly increasing to over 75% of the value in 201, though with a projected reduction to about 3% by 2024 probably due to; expected improvement in technology, energy mix and efficiency in the mode of generation, transmission, and distribution. Since the average heating and cooling electricity daily need for a typical residentialhome operating continuouslyin this location is estimated to about 207.07KWh as shown in Table 1 which results in a daily tariff of about USD55.5 (at USD0.26775/KW-h) by 2019 though this cost can be reduced by operating the Hot water appliances less during the dry season and vise versa, the cost implication remains.

This projected increase in electricity tariff infers that over 70% of the population in this location and other parts of the country may not meettheir DHW and cooling demands considering that thisproportion of the population live below USD1.00 per day [4]

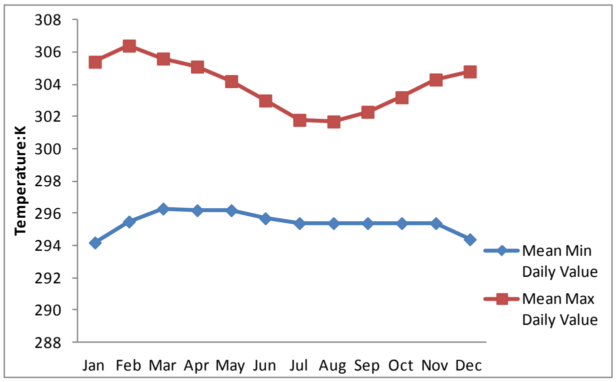

Fig.1. Average Temperature distribution for choice location, Adapted from [1].

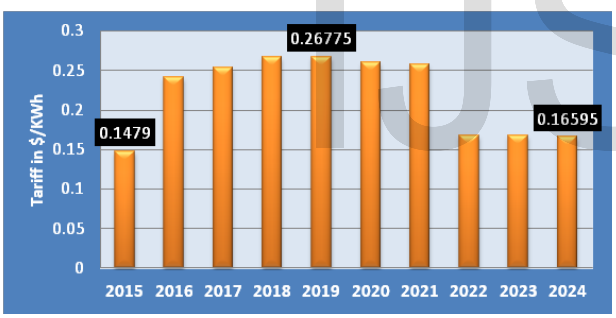

Fig. 2. Electricity tariff increase projection for Port Harcourt., Adapted from [2].

Solar thermal Heating entails the utilization of solar insolation and low-grade heat generated from process plants and other sources for Space and hot water heating while solar thermal Cooling involves the use of solar energy and low-grade heat for; ice making, space cooling in the form of air conditioning and refrigeration [5]. Combined Solar thermal heating and adsorption cooling application in the context of this work, is a renewable energy and energy technology system developed to enhance the performance of basic solar thermal system through the concept of heat and recovery and utilization of excess heat for DHW stored during off-peak cooling periods using solar radiation as driving force.
In a collector system solar thermal technology, solar radiation is captured by the solar collectors and converted to heat energy utilized for domestic hot water, heating, and other process hot water needs [6]. The demonstrated efficiency of the collector systems is about 27.3% [7] possibly due to surface reflections and other associated losses. It is expected that research and design of innovative solar thermal collectors would boost collector conversion efficiency,a development that would reduce the overall cost of the technology through reduction inpayback period. But to maximize the heat energy output use, a combined system of heating and cooling has been researched and developed [7] that produces hot water as well as cooling and one of such systems is the combined DHW and adsorption cooling system.

Adsorption cooling system requires a solid adsorbent to be heated to generate refrigerant vapor in a physisorption (physical adsorption and desorption)process that when condensed and passed through an evaporator absorbs heat from the cooling space at evaporation pressure to produce a cooling effect. The cooling effect and the generated hot water for residential applicationsignificantly improves the Coefficient of Performance-COP of the technology [8] hence the drive to carry out performance evaluation of combined solar thermal DHW and Adsorption Cooling Application-ACA incorporating solar radiation condition ofthe selected location since the climatic condition is assumed to be favorable [1], [6] for the needed insolation required to drive the technology.

1.2 Literature Review

The concept of solar thermal systems for heating and cooling application could be assumed to have reached the stage of commercialization and diffusion in some parts of the world [8] due to streams of research, development and demonstration established in recent times and still ongoing. However, there seem to be no published work that relates performance of a solar thermal combined heating and cooling systems to the climatic condition of the choice location used in the present work considering the scope of search conducted using the internet system for published articles. Within the limit of search conducted, it also appears that norelated work has been published using POLYSUN solar thermal simulation software as recommended by [9] to perform simulation and optimization of a combined solar thermal DHW and Adsorption cooling-STDAC systemintegrated with location climatic conditions.
Evaluation of the suitability of adsorption pairs (adsorbate and adsorbent) for solar thermal cooling applications have been performed as investigated by[10] and the report recommends Silica-Gel and Water pair for space cooling. The extent to which different composite pairs impact solar thermal-ST driven ACA performance is evaluated by [11]. The use of composite pairs in evaluating the performance of a ST cooling system has also been demonstrated [12]. Though these works; identify and isolate suitable pairs for a given application, the evaluated system performance indicators were not tied to the weather condition of a typical location considering solar energy as heat source.
While single bed configuration irrespective of the type of pair applied is known for intermittent cooling [13], it is demonstrated that a 3-bed system [14] has positive impact on system performance a claim also validated by [15] and [16] using 4-

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bed configuration systems. The present study is basically for residential application and as such, a 2-bed system is adopted since it is the minimum requirement for bed configuration to achieve continuous cooling [6], [17]. This is also preferred to 3-4 bed systems due to system compactness and size reduction [18] thereby reducing maintenance costs.

In terms of heat sources, solar radiation and waste heat in form of low grade heat from process plants have been demonstrated [19] as driving forces for solar thermal cooling. When compared with solar radiation [20], lowgrade heat sourceshave the demerit of early system performancedegradation due to inadequate heat supply though this negative impact can be reduced using heat and mass recovery approach [6], [21]. In the present study, heat and mass recovery approach is applied using solar energy as a heat source with the expected foredeal of improving system performance since excess heat would be used for DHW.This approach is also demonstrated by [12], [21] though with no consideration to DHW.

Solar thermal systems are applied in space cooling, ice making and also for DHW as demonstrated by Rouf et al. [22] using concentrated solar application. Apart from the work of Suleiman [6], none of the reviewed works evaluated the performance of the system with respect to combined cooling and DHW production under similar condition as established in the present study. The perspective considered in the present study differs with that of [6] in terms of choice of location, methodology and adsorbent pair.

Hybrid systems have also been developed in a bid to enhance system performance. The use of bed rotation to establish continuous cooling is suggested in [17] while the use of natural and synthetic blend refrigerants have also been demonstrated by [23], [24]. The use of hybrid cycles (Absorption and Adsorption) as suggested by Ullah et al. [25] should improve the COP of single cycle applications, but it can be reasoned that rotation would require increase in the amount of input energy while the use of synthetic adsorbents like HFCs as suggested in [25] would indicate significant increase in the level of Global Warming Potentials-GWP as opposed to natural refrigerants like water. Also, the use of absorption cycle has the concern of safety and excess heat due to the use of ammonia [26] which may not be suitable for domestic use especially for a developing country like the intended location. Solar thermal collectors are required for solar thermal heating and cooling applications if solar radiation is considered as the driving force. Various works [6], [22] have been conducted on the suitability of a particular collector type and impact on system performance [27]. But each of these works,were conducted using locations different from that intended for the present research.

One of the novelties of this study is on the choice of location. Though location integrated solar thermal cooling applications combined with DHW has been evaluated by Suleiman [6], Alam et al. [28], Baiju and Muraleedharan [29], Khairul et al [30], Salvatore et al.[31] and Rifat et al. [32] none integrated the choice of location used for this present study. Suleiman in his work [6] did integrate a location in northern Nigeria into a combined solar thermal heating and cooling system using bed configuration arranged in tubes, but the choice of location for the present study is in southern Nigeria using 2-bed silica-gel and water pair system configuration as opposed to activated carbon and methanol pair which pose the challenge of safety and degradation due tochemical reaction between the pairs at elevated temperatures [10].

In the present research work, thermodynamic relationships of system parameters are developed and determined with respect to their interactions and impacts on the performance of the system withrespect toCOP, Cooling Power-CP and Solar-Fraction-SF using a novel 2-bed Silica-Gel water pair combined solar thermal DHW and cooling configuration as shown in Fig.5. Also evaluated, is heat recovery impact on the performance indicators of the proposed system. POLYSUN software is employed in performing the validation of simulated resultsand optimizationof the system integrating meteorological data for the choice location. Performance of a typical evacuated tube solar thermal collector-ETSC is validated using direct measurement from installation site within the choice location.

1.3 Adsorption Cooling Process Description

Adsorption cooling process is described using a profile of pressure against inverse temperature typical of Clapeyron-Clausius thermodynamic cycle diagram [33] as depicted in Fig. 3. This cycle scheme is widely applied in the literature for adsorption [34], [35] cooling process in describing the thermodynamics processes involved in adsorption cooling.

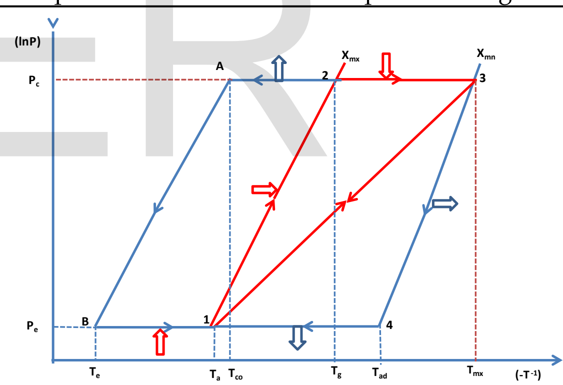

Fig.3.Clausius-Clapeyron depiction of Adsorption Cooling Cycle, Adapted from [35]

From Fig. 3, Stage 1-2 represents Isosteric Heating of the bed since the concentration of the adsorbateremains uniform at Xmx. The heating is initiated at ambient Temperature-Ta until the generation temperature-Tg, just before desorption. Stage 2-3, represents Isobaric Heating of the bed from Tg to the maximum Temperature-Tmx at constant Pressure-Pc. This stage results in desorption of the bed and subsequently condensation of desorbed vapor. Stage 3-4 represents Isosteric Cooling after the bed has attained its maximum Temperature-Tmx for maximum desorption of the bed such that X=Xmn and temperature is reduced to the adsorption Temperature-Tad. Stage 4-1 demonstrates Isobaric Cooling during which the TC-bed further cools from Tad to the initial point of ambient Temperature-Ta. During this stage, the refrigerant is adsorbed back into the bed at constant Pressure Pe. In terms of heat transfer, heat input is required for Stage 1-3 which is a resultant of Stages 1-2 and 2-3 while heat is rejected from Stage 3-1 due to cooling representing the resultant cooling process of the cycle stages 3-4 and 4-1 as shown.In area 1-2-A-B-1 of Fig. 3,the interaction of the refrigerant with the processequipment is demonstrated.Stage 2-A is refrigerant vapor condensation due to heat release to cold water stream, Stage A-Bdemonstrates refrigerantexpansion due to throttling valve effect resulting in pressure and temperature drop and Stage B-1 depicts refrigerant vapor evaporation due to heat transfer from chill water fluid to the refrigerant. The red arrow highlights indicate stages of heat input into the system while the blue arrow highlights demonstrate stages of heat release from the system.

1.4 Solar Thermal Collector Principle

As demonstrated in Fig. 4, the ETSC is composed of a heat pipe system responsible for heat transfer into the manifold collector box. The heat pipe is enclosed by a layer of black coating with significant heat transfer coefficient value aided by fins. The tubes are arranged such that the bulb head terminate into a header pipe enclosed by the collector box. During exposure to the sun, solar radiation strikes the surface of the ETSC collector and the fluid enclosed in the tubes is heated up thereby forcing its vapor up the manifold where cold water absorbs the heat from the vapor while the vapor condenses using latent heat. Condensed fluid returns to the down end of the ETSC and the cycle continues. ETSCcan capture both beam and diffuse radiation as opposed to concentrated solar type collectors making it suitable for the intended location known for characteristic diffuse radiation.

1.5 Aims and Objectives

This present study is structured to achieve the following aims and objectives;

- Conduct onsite measurement of solar thermal collector temperature profile,
- Evaluate the performance of the proposed DHW production and integrated ACA using solar radiation from Port Harcourt Nigeria as the driving force to heat up a packed silica-gel and water pair TC,
- Propose a combined cooling and heating system configuration,
- Conduct thermodynamic analysis of the system operating parameters
- Critically investigate previous works on combined solar thermal cooling and DHW provision,
- Resolve developed mathematical models using numerical and analytical technique.
- Use proprietor software suitable for STDAC to optimize the system for improved performance by incorporating the location solar data,
- Make resource data available for further studies

To ensure the actualization of the research objectives, this work has been structured into 5 different sections. Section 1 covers the background information as regards the research question, review of previous works on related subject, research justification and novelty as well as the aims and objectives. Section 2 presents the apparatus engaged for measurement, schematized description of proposed application and the methodology employed in meteorological analysis of choice location while mathematical modeling and thermodynamics analyses of system parameters and equipment are treated in section 3. Results and findings of the evaluated system is presented in section 4 and discussed in section 5 while section 6 draws conclusion on the research question and answers.

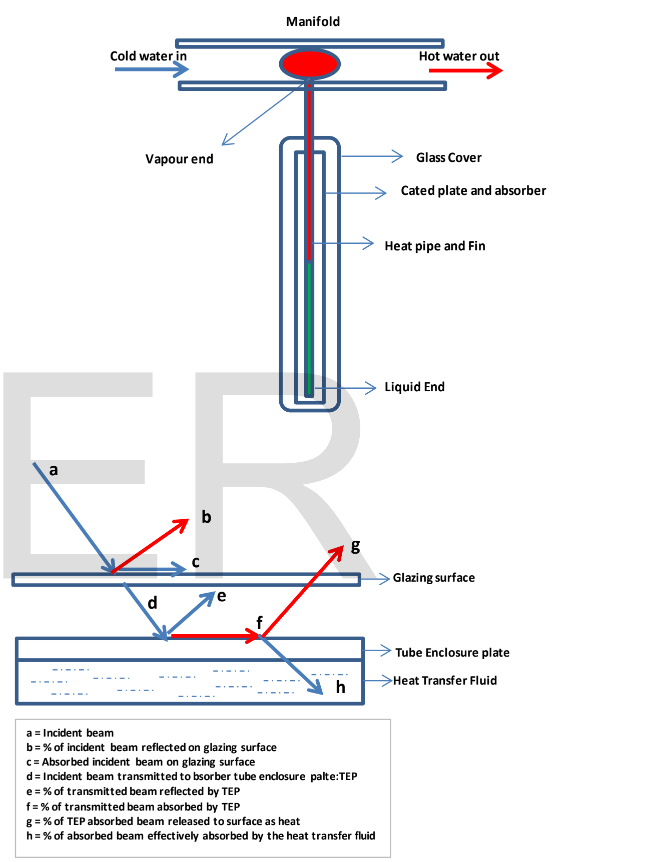

Fig.4.ETSC showing single tube and beam distribution, Adapted from [9], [40]

2. Instruments and Mathematical modeling

2.1 Instruments Analysis

Detail description of the solar collector type, measured parameters and measuring instruments used during the onsite measurement exercise is presented in Appendix A while Fig. 5 represents the solar thermal collector-STC and measurement apparatus used for data acquisition. While the various temperature profile of the STC is evaluated in Fig. 9 as part of primary result and discussed in section 3.0.

2.1.3 Weather Analysis

Meteorological analysis for the selected location being Port Harcourt (4.9N, 7.0E) was conducted using RETSCREEN resource softwarebased on Ground station and NASA satellite measurements as stated in the software. The obtained data were validated using onsite climatic condition data provided by the NigerianMetrological Agency-NIMETthrough the World Meteorological Organization [2].Outcome of the validation demonstrates acceptable level of correlation between the two data sources and as such the climatic condition for the selected location is reliable for use in the intended purpose of combined heating and cooling application. The solar data was integrated into the modeling tools and software.

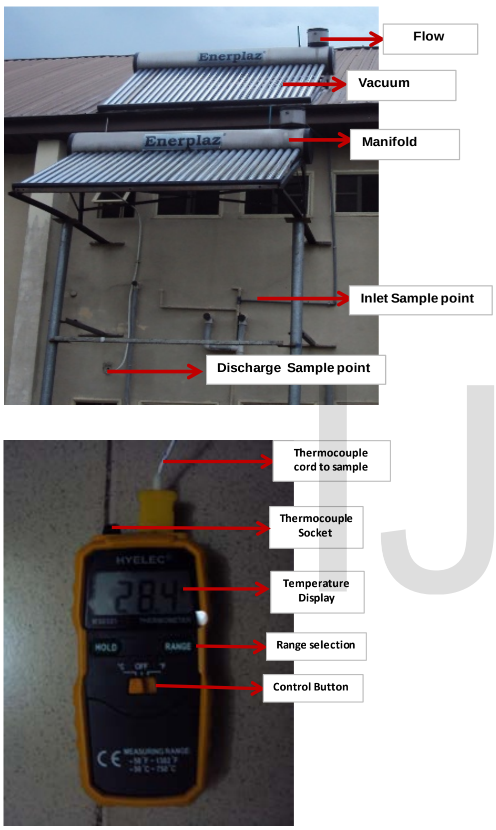

Fig.5.STC and Digital Multi-meter used during data sampling and measurement

2.1.4 Description of proposed system

With reference to Fig.6, the proposed system is made up of; non-tracking and non-concentrating solar collector array, Hot water reserve tank with domestic hot water supply outlet, Two adsorption beds of Silica-gel and Water pair, Condenser, Expander, Refrigerant reserve tank and cold-water supply tank. Detail explanation of the system depicted in Fig.5 is expressed inAppendix B.

2.2 Mathematical and Thermodynamics Modeling

2.2.1 System Mass Relationships

Prior to Isosteric heating, the fundamental mass relationships as a function of mass of adsorbentporous media massbased on fixed bed principles [35] are delineated herein.

$$m_s = (1-\varepsilon)\rho_s V_T \quad (1)$$

$$m_a = V_a \rho_a \quad (2)$$

$$m_g = [(\varepsilon - V_{fa}) \times \rho_g V_T] \quad (3)$$

$$X = {}^{m_a}/_{m_s} \quad (4)$$

$$m_a = m_s X \quad (5)$$

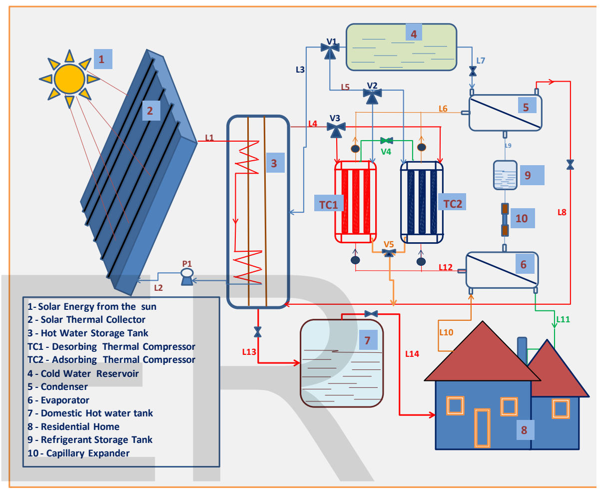

Fig.6.Proposed Solar Thermal DHW and ACA

2.2.2Collector Heat Input

Collector heat input rate is evaluated from the available solar radiation as a function of time using Hottel-Whillier-Bliss equation for efficiency term [9] and the hour angle as widely used in the literature [28]. The energy balance on the ST collector system assuming that the efficiency has accounted for inherent loses is deduced as;

$$\dot{Q}_{uc} = \eta_c N_C A_c G_t = \dot{m}_{cf} C_{pcf} (T_{of} - T_{if}) \quad (6)$$

The solar radiation is correlated tolocal time as;

$$dG_t = G_{mx} \, \mathrm{Cos}\theta_h \, dt \quad (7)$$

Where;

$$\theta_h = \mathrm{HourAngle} = \left({}^{\pi t}/_{12}\right) (8)$$

Combining equations (7) and (8);

$$dG_t = \mathrm{G_{mx}}\,\mathrm{Cos}\left({}^{\pi t}/_{12}\right)\mathrm{dt} \quad (9)$$

In order to evaluate the local solar global radiation, equation (9) is integrated with respect to time using the boundary conditions; t = sunrise time and t = daytime while 12hr timeis expressed as difference between sunrise and sunset hours. Thus;

$$G_t = \mathrm{G_{mx}} \int_{\mathrm{tr}}^{\mathrm{t}} \mathrm{Cos}\left({}^{\pi t}/_{12}\right)\mathrm{dt} = \mathrm{G_{mx}}\,\mathrm{Sin}\left[\frac{\pi(\mathrm{t}-\mathrm{tr})}{(\mathrm{ts}-\mathrm{tr})}\right] (10)$$

Equations(6) and (10) are combined to obtain the exit collector fluid temperature as;

$$T_{of} = \mathrm{T_{if}} + \frac{\eta_c \mathrm{N_c} \mathrm{G_{mx}}\,\mathrm{Sin}\left[\frac{\pi(\mathrm{t}-\mathrm{tr})}{(\mathrm{ts}-\mathrm{tr})}\right]}{(\dot{\mathrm{m}}_{\mathrm{cf}} \mathrm{C_{pcf}})} (11)$$

2.2.3Isosteric Heat of Sorption

This isdeveloped from first principle using Clapeyron-Clausius equation of phase change for correlation of pressure and temperature at equilibrium or saturation condition since the adsorbate vapor and liquid phases exist at equilibrium during sorption. The concept is validated using the works of [36].

$$^{\mathrm{dP}}/_{\mathrm{dT}} = {}^{\mathrm{L_v}}/_{\mathrm{TV_{fg}}} \quad (12)$$

In determining the sorption heat relationship from equation (12), it is assumed that; the TC is operating at low Pressure, the adsorbate liquid phase is at equilibrium with the vapor phase adsorbate at saturation point, the liquid phase volume is insignificant compared to the vapor phase volumeand enthalpy change within the bed system is constant. Based on these assumptions, the Isosteric heat of sorption is deduced from equation (12) as;

$$\Delta H_s = -\mathrm{R_{gs}}\left(\frac{\mathrm{T_{mx}}\,\mathrm{T_{bi}}}{\mathrm{T_{bi}} - \mathrm{T_{mx}}}\right)\ln\left({}^{\mathrm{P_{co}}}/_{\mathrm{P_e}}\right) \quad (13)$$

2.2.4 Equilibrium Uptake

Equilibrium uptake is considered as the mass concentration of the adsorbate at saturation condition dependent on TC temperature and pressure. Though the literature [37]plays host to some set of models that fittingly evaluates the equilibrium concentration, the modified Dubnin-Astakhov (D-A) adsorption Isotherm equation is used in the present work to correlate equilibrium concentration with TC temperature and pressure at saturation condition. The equation in this context is also modified to evaluate mass concentration of the adsorbate phase at any given TC Pressure and Temperature as shown in equation (14). The D-A model is considered in the present work due to its suitability for micro sizes of porous media like the Silica-Geland its suitability in carrying out thermodynamic analyses of system properties as also claimed by [38].

D-A equation, is expressed thus;

$$\ln\left({}^{\mathrm{X}}/_{\mathrm{X_{mx}}}\right) = (-\mathrm{DT^n})\left[\ln\left(\frac{\mathrm{P}_s}{\mathrm{P}}\right)\right]^{\mathrm{n}} (14)$$

For equilibrium uptake as function of TC temperature and saturation temperature, equation (14) is further simplified to;

$$X = \mathrm{X_{mx}} \exp\left[-\mathrm{D_k}\left(\frac{\mathrm{T}}{\mathrm{T_s}} - 1\right)\right]^{\mathrm{n}} (15)$$

Where;

$$D_k = D^{1/n}\frac{\Delta H_s}{R_{gs}} (16)$$

2.2.5 Rate of sorption

This is used in evaluating the rate at which uptake varies with time as the process proceeds from initial to final state. Experimentally determined linear model as widely applied in similar works [37], [38] is also adopted in the present study.

Such that;

$$^{dX}/_{dt} = \mathrm{KD}\left(\mathrm{X_{eq}} - \mathrm{X}\right) \quad (17)$$

Where;

$$K = \left(15/\mathrm{r_{sp}^2}\right) and D = \mathrm{D_o} \exp\left(-\frac{\mathrm{E_a}}{\mathrm{R_{gs}}\,\mathrm{T}}\right) \quad (18)$$

This rate expression in equation (17) is used in evaluating the mass concentration of adsorbate adsorbed or desorbed as a function of process time, temperature, equilibrium concentration and particle size specific to Silica-Gel water pair porous media. It is also applied in section 2.3 in heat energy transfer rate analogy of the TC and sorption processes.

2.3 Thermal Compressor Thermodynamics Model

2.3.1 Isosteric Heating (Stages 1-2)

As depicted in Fig.7, for this stage, it is assumed; that heat loss from the hot water enclosure to the surrounding is insignificant, no mass transfer between the TC and the ambient during heating and overall heat transfer coefficient overTC casing and hot water is constant.

From Fig.7a, the heat energy balance is deduced thus;

$$\dot{Q}_{in1} = \dot{Q}_{b1} \quad (19)$$

The components of the TC are; Casing enclosing the bed, Silica-Gel porous media and adsorbate.

Since heat transferred from the storage tank is supplied by the collector, it is assumed that Hot water inlet stream temperature to TC system is equal to the collector fluid outlet stream temperature of equation (11). Thus;

$$T_{hwi} = T_{of} = T_{if} + \frac{\eta_c N_C A_c G_{mx} \,\mathrm{Sin}\left[\frac{\pi(t-tr)}{(ts-tr)}\right)\Big]}{\left(\dot{m}_{cf} C_{pcf}\right)} \quad (20)$$

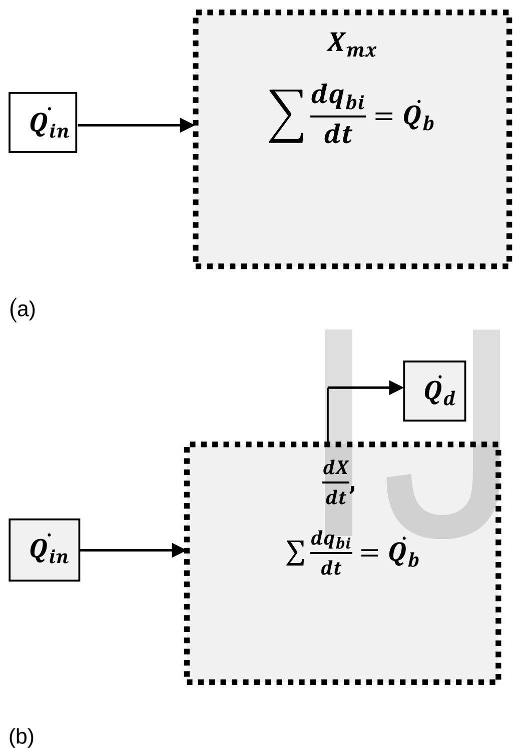

Fig.7.Heat flow during (a) Isosteric Heating and (b) Isobaric Heating

Assuming uniform heat capacity for collector fluid and hot water and also considering that the exit hot water temperature is the same as the inlet collector fluid temperature and including the sunrise-06:30hrs and sunset-18:48hrs times for the location, then equations (19 and (20) are combined as;

$$\dot{Q}_{in1} = \frac{\dot{m}_{hw}}{\dot{m}_{cf}} \times \eta_c N_C A_c G_{mx} \,\mathrm{Sin}\left[\frac{\pi(t-6.5)}{(12.3)}\right] \quad (21)$$

Also in terms of the hot water supply to the bed,

$$\dot{Q}_{in1} = -\left[\dot{m}_{hw} C_{phw} (T_{hwo} - T_{hwi})\right] \quad (22)$$

To evaluate the rate at which temperature varies within the TC bed system during Isosteric Heating, equation (23) is used;

$$dT_b/dt = \left\{G_m \,\mathrm{Sin}\left[\frac{\pi(t-6.5)}{(12.3)}\right]\right\} \Big/ (C_{m1}) \quad (23)$$

Where;

$$C_{m1} = m_s\left(\frac{m_c}{m_s} C_{pc} + C_{ps+} + X_{mx} C_{pl}\right. \text{ and}$$

$$G_m = \left(\dot{m}_{hw} \times \eta A_c G_{mx}\right) \Big/ \dot{m}_{cf}$$

2.3.2 Isobaric Heating and Desorption (Stages 2-3)

In developing the governing model equations for this stage using Fig. 7b, it is assumed that; the system pressure is constant at condenser pressure, mass of adsorbate is continually changing with respect to its concentration from maximum to a minimum value due to desorption while the mass of the solid components of the bed remain fixed, equilibrium has been attained by the system and the overall heat transfer coefficient remainsconstantwith respect to condenser and adsorbate vapor phase.

From Fig.7b,

$$\dot{Q}_{in1} - \dot{Q}_d = \dot{Q}_{b2} \quad (24)$$

From equation (24), the rate of heat required during Isobaric Heating is deduced thus;

$$\dot{Q}_{b2} = \frac{\dot{m}_{hw}}{\dot{m}_{cf}} \times \eta_c A_c G_{mx} \,\mathrm{Sin}\left[\frac{\pi(t-6.5)}{(12.3)}\right] + \left(m_s \times KD(X_{mx} - X_{mn})\Delta H_s\right) \quad (25)$$

To evaluate the rate at which temperature varies within the TC bed system during Isobaric Heating, equation (26) is used;

$$dT_b/dt = \left\{G_m \,\mathrm{Sin}\left[\frac{\pi(t-6.5)}{(12.3)}\right]\right\} \Big/ (C_{m2}) + \left[m_s KD(X_{mx} - X_{mn})\Delta H_s / C_{m2}\right] \quad (26)$$

$$C_{m2} = m_s\left[\frac{m_c}{m_s} C_{pc} + C_{ps} + \left(X_{mn} - X_{mx}\right) C_{pl}\right)\Big] \quad (27)$$

During this stage, condensation of the desorbed vapor phase also occursusing cold water cycle as coolant. Therefore;

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$$-\dot{Q}_{co} = \dot{Q}_{cw} \quad (29)$$

$$\dot{Q}_{co} = -\dot{m}_{cw} C_{pw} (T_{cwo} - T_{cwi}) (30)$$

In addition to previously made assumptions that are also valid for this stage, from Fig. 8a, it is also assumed that; mass concentration of the adsorbate is constant at minimum uptake value, bed cooling water is introduced at a constant flow rate through V2 of Fig. 6, sensible heat released by the bed components is recovered through V5 and temperature gradient is from bed maximum temperature to the adsorption temperature. From Fig. 8a, the heat energy balance is deduced thus;

$$-\dot{Q}_{b3} = \dot{Q}_w (31)$$

$$\dot{Q}_{b3} = -\dot{m}_{cw} C_{pw} (T_{cwo} - T_{cwi}) \quad (32)$$

2.3.3 Isosteric Cooling (Stages 3-4)

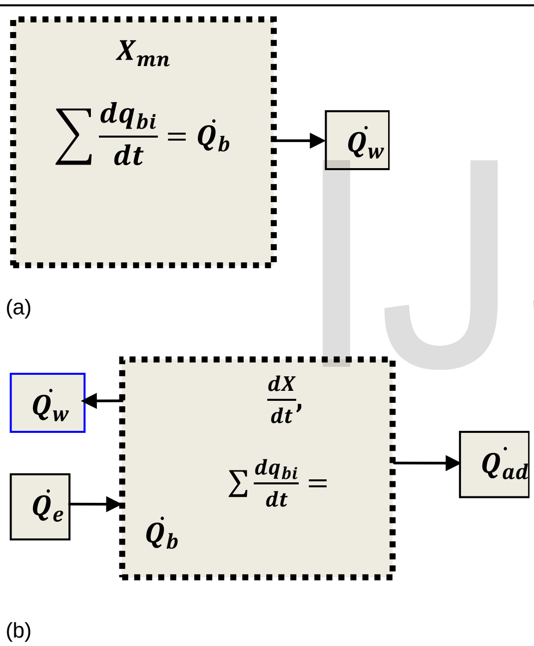

Fig.8.Heat flow during (a) Isosteric Heating and (b) Isobaric Heating

2.3.4 Isobaric Cooling (Stages 4-1)

Assumptions; the coefficient of heat transfer over refrigerant andchilled water heat exchanger in the evaporator is constant, rate of uptake is varying from minimum value at maximum bed temperature to maximum value at initial bed condition, evaporator pressure is constant, adsorption occurs during this stage and desorbed heat is released as adsorbed heat as depicted in Fig.8b.

From Fig.8b, it is deduced that;

$$-(\dot{Q}_{b4} + \dot{Q}_{ad}) = \dot{Q}_w (33)$$

Rearranging and substituting for cold water terms, equation (33) is developed to;

$$\dot{Q}_{b4} = m_s KD (X_{mx} - X_{mn}) \Delta H_s - \dot{m}_{cw} C_{pw} (T_{cwo} - T_{cwi}) \quad (34)$$

The second term of the RHS of equation (34) correlates the heat of adsorption tothe uptake rate.

During this stage, evaporation of the condensedrefrigerant due to heat from chill water also occurs, such that;

$$\dot{Q}_e = (-\dot{Q}_{ch}) \quad (35)$$

$$\dot{Q}_e = -\dot{m}_{ch} C_{pch} (T_{cho} - T_{chi}) \quad (36)$$

Considering the cycle time, equation (36) is expressed thus;

$$\dot{Q}_e = -\dot{m}_{ch} C_{pch} \int_0^{tcycle} (T_{cho} - T_{chi}) dt \quad (37)$$

In terms of refrigerant cooling capacity, the cooling effect is determined thus;

$$Q_e = m_s (X_{mx} - X_{mn}) [L_v @ T_e - C_{pl} (T_c - T_e)] \quad (38)$$

$$\dot{Q}_e = Q_e / t_{cycle} \quad (39)$$

2.4 Heat Recovery Effect

This is evaluated by considering that heat released during cooling and adsorption are transferred to the cooling water system and recovered through V5 and L8 of the proposed application as shown in Fig. 6.

Therefore;

$$\dot{Q}_r = \dot{Q}_{ad} + \dot{Q}_{cw} + \dot{Q}_w \quad (40)$$

But overall heat input rate from the hot water system is from stages 1-3 expressed as;

$$\dot{Q}_{intotal} = \dot{Q}_{in} @ 1-2 + \dot{Q}_{in} @ 2-3 \quad (41)$$

Effect of heat recovery rate is therefore expressed as efficient heat input using equation (42);

$$\dot{Q}_{effin} = \dot{Q}_{intotal} - \dot{Q}_r \quad (42)$$

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2.5 Domestic Hot water and Cooling demand load

DHW demand for the average family size of four in the evaluated location is modelled as a rule of thumb widely applied in the literature domestic hot water chapter [9] which implies that a family size of requires about 265l/day equivalent to a rate of 0.003kg/s of hot water demand.

In the analysis of the DHW demand load, the is further assumed that; the highest demand occurs during the wet season with average air temperature of about 25.6°C for the months of July -September, Cold water supply for heating is at ambient temperature of 25.6°C and DHW storage temperature is maintained at about 70°C.Based on the assumptions, the quantity of heat required per second is evaluated as;

$$\dot{Q_{dhw}} = \dot{m_{hw}} C_{phw} (T_{hwi} - T_a) \quad (43)$$

The residential cooling demand is evaluated using the analysis presented in Table 1 and validated using the works of [39]. This analysis considers that a standard residential home would utilize a minimum of 4 units of cooling devices with standard ratings operating for 24hrs.

TABLE 1:
DHW AND COOLING DEMAND ANALYSIS FOR A STANDARD HOME IN PORT HARCOURT

Appliance	Quantity	Unit Rating (W)	Total rating (W)	KW-h
Fan	4	100	400	9.6
Refrigerator	4	850	3400	81.6
Air Conditioner	4	1000	4000	96.0
DHW	-	-	828	19.87
TOTAL			8628	207.07

2.6 Performance Evaluation

Using MATLAB simulation package, the application performance is evaluated by resolving the model equations describing the proposed system to determine the heat input and the solar thermal collector system size required and configuration to meet the heating and cooling demandincorporating the solar meteorology of the focal location. Based on this, the coefficient of performance is evaluated at normal and recovery operating modes to capture both heating and cooling conditions while the specific cooling power is evaluated for various cooling effects as a function of adsorbent mass. Solar fraction for the collector is calculated using equation (46) as the ratio of solar energy through the STC contributed to the total system energy demand. It is the percentage of solar energy used in offsetting the total energy demand from other energy sources [31]. System optimizationfor the selected location is performed using POLYSUN 8.2 simulation software with focus on economic viability, energy savings and carbon footprint reduction. Total heat supplied to the system by the hot water system is determined using equation (22) considering that hot water inlet temperature maintains a range of 90- 95OC while it exits temperatures between ambient and 40OC.Base conditions used in simulation and evaluation of system parameters are presented in the software script.

The system performance indicators are determined thus;

$$COP_n = \dot{Q_e} / \dot{Q_{in}} \; \boldsymbol{and} COP_r = \dot{Q_e} / \dot{Q_{effin}}$$

$$= \frac{\boldsymbol{m_s}(\boldsymbol{X_{mx}} - \boldsymbol{X_{mn}})[\boldsymbol{L_v}@\boldsymbol{T_e} - \boldsymbol{C_{pl}}(\boldsymbol{T_c} - \boldsymbol{T_e})]}{\dot{Q_{intotal}} - \dot{Q_R}} \quad (44$$

$$SCP = \boldsymbol{Q_e} / \boldsymbol{t_{cycle}} \times m_s \quad (45)$$

$$S_f = \eta_c N_C G_{mx} \operatorname{Sin}\left[\frac{\pi(t-tr)}{(ts-tr)}\right) \Big] \Big/ (\dot{Q_{dhw}} + Q_{in}) \quad (46)$$

3. Results

3.1 Collector Result Analysis

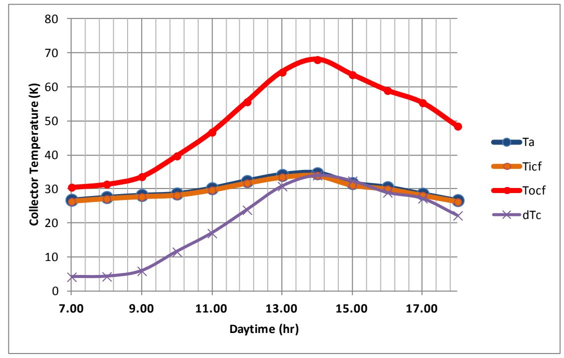
Measurement results on the collector temperature, useful energy produced and effect of collector fluid flow rate all with respect to daytime span of selected location is presented in Fig.9 for a single collector. This demonstrates how the output temperature, inlet temperature to the collector and the temperature gradient on the collector system vary from sunrise to sunset. The STC exit stream temperatureis significant to the evaluation of the system performance as this determines the temperature of the inlet hot water required for heating up the TC system as well as DHW production. The collector temperature gradient has been included in the result analyses since it is the driving force for the evaluation of the ST collector useful energy produced which is required for evaluation of performance indicators. Ambient temperature profile as shown in Fig. 9a forms the basis for collector inlet temperature setting and the cooling set point. In Fig. 9b, the behavior of the ST collector outlet temperature relative to the collector fluid flow rate is shown evaluated at three different flow rates chosen between minimum and maximum values obtained from previous works.

3.2 System Performance Result Analysis

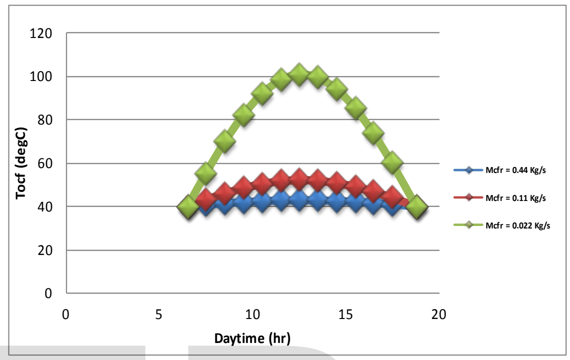
Results of the identified performance indicators as discussed in section 2.6 are presented in Fig. 10 - 13. In Fig. 10a, the behavior of the rate of heat input to the TC system required for process 1-3 within the daytime span of the location is demonstrated for different adsorbent mass while the effect of heat recovery on the heat input within the daytime span is depicted in Fig. 10b. The heat input parameter is invaluable in the determination of other performance indicators of the system and as such, its correlation with daytime span is evaluated using different adsorbent mass.

Considering normal COP and COP due to heat recovery effect, Fig. 11a depicts how these two parameters correlate with the daytime span. These are the minimum fundamental indicators required to evaluate the performance of the system determined from equation (44) as a ratio of the cooling and DHW production of the STC to the total heat input with the inclusion of heat recovery effect. The average values of the indicators over the selected daytime hours are also determined depicted with black lines across the profiles

In Fig. 11b, the solar fraction to the system and produced cooling effect are profiled at normal and recovery conditions of the TC with respect to the daytime span. This enables the assessment of the viability of the use of solar energy as a driving force to the evaluated application considering the maximum average insolation available within the choice location. The effect of cycle time on the cooling effect produced is depicted in Fig. 12b while the specific cooling power, which depicts the adsorbent mass required to produce a desired cooling effect at a given period of the day in line with ambient temperature is shown in Fig. 12a. It is a function of the evaporator temperature and cycle time determined from equation (45). The cycle time considered represent the process time required to complete stage 1-4 of the system.



(a)



(b)

Fig.9.STC temperature profiles with respect to Daytime (a) All Temperature (b) Output Temperature.

The system COP at recovery and normal conditions is compared with rate of heat input as shown in Fig. 13 and appropriate regression equations with the fitting R-squared values also determined for the correlated parameters. This result has been obtained using the heating temperature range of 299.85K to 359.85K representing the temperature operating limit of the system.

3.4 Simulation and Optimization results

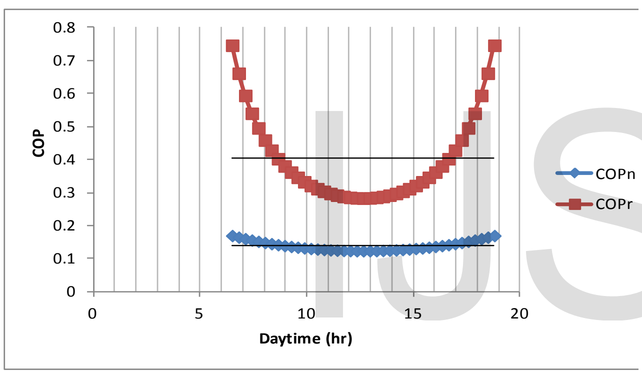
The result of the system performance simulated and optimized for the selected location using POLYSUN software package is shown in the professional report. The result demonstrates system performance in terms of; energy savings, CO_2 emission footprint, payback time and financial analysis.

3.5 Result validation and uncertainty

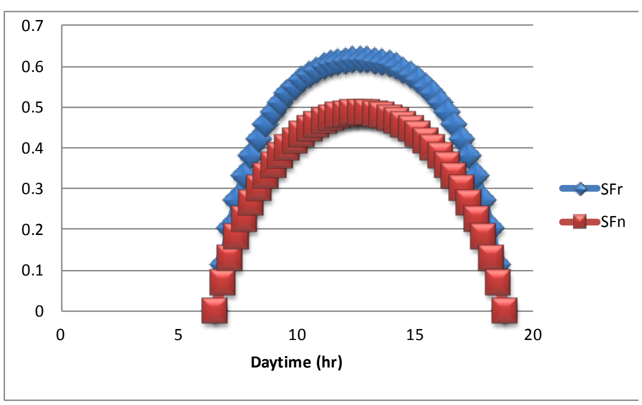
Measurement results obtained in course of this study for collector temperature profiling were subjected to error analysis as earlier mentioned in section 2.1.2 and validated using similar experimental work [28] conducted in similar climate to the choice location. However, the collector flow ratewas not considered during measurement of STC temperature profile for the specified period which is expected to have an impact on the output temperature of the collector as shown in Fig. 9. The measurement results were not subjected to wind interference analysis especially at the sampling points and variation of solar radiation during the sampling time due to limited access to other measuring devices though readings were taken with the assumption that the solar radiation is constant over an averaged maximum value for the selected month of February.

In estimating the cooling and DHW need daily measurements were not taken and results are only based on mathematical and analytical assumptions validated by published works, as in [39].

Other performance results obtained from the evaluation of model equations were validated using POLYSUN simulation and optimization software and trend compared to previous experimental works related to the present work and the results obtained compare within acceptable error margin of less than 5% for system COP.



(a)

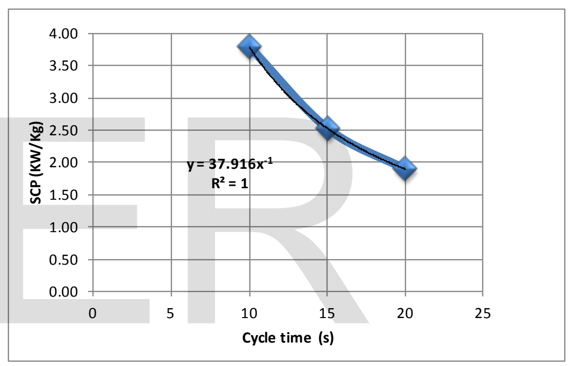


(b)
Fig.11.Comparison of COP and SF at different modes with respect to daytime (a) COP at normal and recovery modes (b) SF at normal and recovery modes

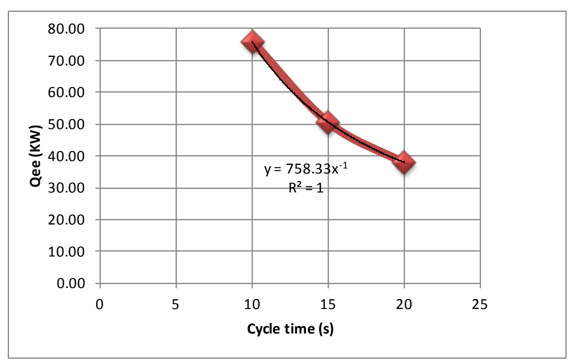
4.0 Discussion

4.1 Collector performance

As depicted in Fig. 9, the collector temperature profile demonstrates an incremental trend with respect to variation in the trajectory of the sun from sunrise to sunset with maximum peakvalues recorded between 12noon and 14:00hrs which is understandable as the sun is assumed to be overhead within this daytime period. This trend implies that the maximum heat energy converted by the ST collector from local solar radiation is attained at this peak period with air temperature also reaching itsmaximum peak average value of about 35^{O}C for hot days considered as worst case for cooling need. The output temperature of the collector fluid within this period is about 68^{O}C but increases to about 100^{O}C when the system is optimized for 20 collectors at a flow rate of about 0.022kg/s as shown in Fig. 9.It is established that the ST collector fluid at reduced flow rate improves the system performance due to increase in the output fluid temperature with corresponding increase in residence time of the fluid within the hot water storage tank at reduced



(a)



(b)

Fig.12.Effect of Cooling with respect to Cycle time (a) Specific Cooling power (b) Cooling Effect produced

collector fluid flow rate. Since the DHW storage tank Temperature is maintained at 70^{O}C and the maximum bed tempera-

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ture determined as T_{mx} = 86.7ᴼC, it can be inferred that the ST collector should serve the purpose of operating the system for adsorption cooling and DHW provision since the hot water storage tank could be heated up to 100ᴼC.

4.2 System performance

It has also been proven within the scope of the present work as shown in Fig.10a, that the total heat input rate required by the system follows similar trend of increase with daytime up to 14:00 daytime hours at mean values of about 158kW and 187kW for an adsorbent mass of 15kg and 20kg respectively assuming the collector fluid flow rate and collector numbers are kept at the base values. Which further implies that the system requires more heat input as the adsorbent bed is increased. Considering the effect of heat recovery as represented in Fig.10(b), total heat input or effective heat input rate reduces by 35% of the initial values for any adsorbent mass considered provided all other initial conditions remain constant.

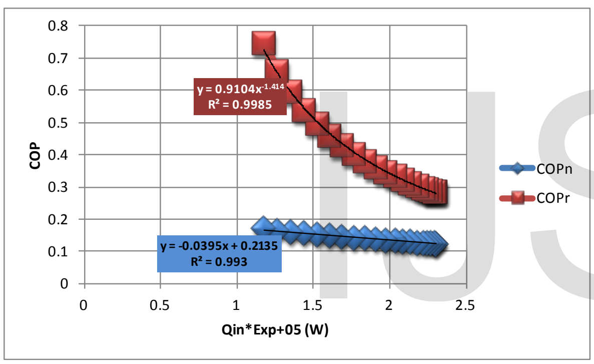

Fig.13.Effect of Total heat input rate on COP

Operating the system under heat recovery mode yields an average COP of about 0.43 (75% increase) while under normal mode without heat recovery, a mean COP of <0.25 is actualized as demonstrated in Fig. 11a, further emphasizing the positive effect of heat recovery to the system. This trend also applies to solar fraction indicator in Fig. 12b where an increase of 35% from normal operating mode is actualized. Analytically, it could be deduced that the COP value in both cases tend to reduce to an average value in both cases as daytime tends to noon due to increase in ambient temperature which affects the system cooling set point and gradually picks to a maximum value after noon periods.

The cooling effect produced as well as specific cooling power, decrease with increase in cycle time for stage 4-1 as shown in Fig. 12a. This condition requires the system to be operated at reduced cycle time for efficient performance. The cooling need for the average home is estimated as 163.2kWh per day which is about 6.8kW while the average cooling effect produced for a period of 15s within the evaporator is about 54.77kW as depicted in Fig. 13a. Also, the system is able to produce required heating temperature of up to 90OC which is greater than the temperature requirement for DHW storage (70ᴼC) and Adsorption cooling process (86.7ᴼC). Based on this analogy under the given base operating conditions it could be established that the Cooling and DHW demand are met.

In Fig. 13, reduction in total heat input amounts to significant increase insystem COP irrespective of the operation mode weather recovery or normal. Total heat input of about 120KW results in a COP of almost 0.7 and 0.2 for heat recovery and normal modes respectively as shown in Fig. 14 making a case for efficient heat supply to the system through heat recovery and other input heat reduction techniques.

Simulation and optimization results obtained using software inbuilt analysis tools and the base parameters specified in Fig. 14 demonstrate an average COP of 0.21 though with no heat recovery effect. The system reports; an energy saving of 20MW-h, a CO_2 footprint reduction of 4616.3kg and a fuel saving of 1898.4m³ of natural gas assuming the entire system were operated using heat produced from the combustion of natural gas. An average solar fraction of 0.62 was obtained, a value that corresponds with Fig. 11b that returns a Solar fraction value >0.60 for heat recovery mode.

5.0 Conclusions

Performance of solar thermal DHW integrated with ACA is numerically and analytically evaluated in this work incorporating the meteorological condition of Port Harcourt in Nigeria a tropical region. Within the scope of the study, it can be established that;

- COP of 0.43 and above up to 0.75 is feasible when appropriate heat recovery technique is applied,
- heat recovery improves the performance of the system by 75%,
- the available maximum solar radiation of Gmx =180/m2 is adequate to provide the needed heat input for the system at a temperature range of 95 – 100ᴼC using optimal flow rates and 20 collectors tilted and oriented based on the position of the location,
- Cooling demand based on a standard family home is met since the system could generate an average cooling power of about 55kW within a 15s cycle time for evaporation process
- DHW daily demand of 828W combined with daily cooling demand is also attained using the simulation base conditions.

Though the system could be theoretically operated in a combined mode of DHW and adsorption cooling, it is recommended that a demonstration be built within the location with more detailed experimental set up for collector temperature profile analyses. The use of different collector fluid and application of process intensification techniques to the system is also recommended for further studies.

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7.0 Appendices

Appendix A: Nomenclature.

Symbols
A area [m2]
Cp specific heat capacity [J/kg-K]
D constant in Dubnin-Astakhov equation
E_a sorption activation energy [J/kg]
G global radiation [W]
ΔH_s heat of sorption [J/kg-K]
L_v latent heat of vaporization [J/kg]
m mass [kg]
N number
P pressure [N/m2]
Ps saturation pressure [N/m2]
Q quantity of heat [J]
R_{gs} specific gas constant [J/kg-K]
r_{sp} particle size (m)
Sf solar fraction
t time [s]
T temperature [K]
Ta ambient temperature [K]
V volume [m3]
X mass concentration/uptake

Subscripts
a adsorbent
ad adsorption
b bed
c bed casing/collector
cf collector fluid
co condenser
cw cold water
d desorption
dhw domestic hot water
e evaporator
eq equilibrium
f fluid
g gaseous phase
hw hot water
Sf solar fraction
i inlet
l liquid phase
mn minimum
mx maximum
o outlet
r, s recovery, silica-gel
Numeric
1 process stage 1-2
2 process stage 2-3
3 process stage 3-4
4 process stage 4-1

Appendix B: Description of proposed system.

With reference to Fig.6, the proposed system is made up of non-tracking and non-concentrating solar collector array, Hot water reserve tank with domestic hot water supply outlet, Two adsorption beds of Silica-gel and Water pair, Condenser, Expander, Refrigerant reserve tank and cold-water supply tank. Solar radiation is converted to heat energy by the array of collectors used in heating up water content in the hot water storage tank provided through L11 from the collector. Hot water stream is connected to both beds using L4 for Desorption which requires heat (Endothermic process) with V3 serving as the hot water flow control valve to the beds while the beds are also connected to cold water stream through L4 to cool the beds during Adsorption which releases heat to the cold-water system (Exothermic process) with V2 serving as the cold-water control valve to the beds. Desorbed vapor mass in the heated TC migrates through a check valve installed on L6 to the condenser which only opens when the pressure in the Desorbing or heated bed is at equilibrium with the condenser pressure. Water from the cold-water tank supplied through L7 condenses the vapor and the condensate in form of refrigerant liquid is stored in the refrigerant tank for storage for peak and off-peak usage. With the aid of the Expander, the pressure and temperature of the refrigerant is further reduced to allow for cooling. From the Expander, the refrigerant is channelled to the evaporator and gains heat from the Chilled water supplied through L10 thereby creating a cooling effect on the chilled water system due to heat extraction. The Chilled water exit stream L11 provides the cooling effect to the required space air conditioning or refrigeration depending on the cooling need. The proposed system is such that the vapor leaving the evaporator at evaporation pressure is adsorbed in TC2 while TC1 is being desorbed - a configuration that provides for continuous cooling. To allow for mass recovery, valve-V4 connects the beds and it operates on differential pressure between the two beds (TC1 and TC2), stimulating transfer of trapped vapor mass from the desorbed bed undergoing cooling to the adsorbing bed as the valve opens to this direction due to higher system pressure in the desorbed bed. System heat recovery is further achieved through the recovery of; discharge water used for bed cooling, Adsorption heat release from the beds and condenser effluent fluid which are all channelled back to the hot water storage through L8 for upgrade to desired temperature. Provision for Domestic Hot Water-DHW is achieved by supplying hot water from the storage tank through L13 using a separate DHW tank to prevent interference with required heating loads for the bed.

Appendix C: Instruments description and measurement methods.

The operation principle of the ETSC type for the conversion of solar energy to heat has been evaluated in section 1.4. Under special arrangement, the ETSC used for the study is an installed roof type, already put to domestic use. In Fig. 5a, the ETSC used during data acquisition from the site location is depicted. Apart from the hot water temperature measurement, other measurements collected are; the STC orientation in terms of angle to the horizontal plane and hemispheric face. These data were obtained under special arrangement with the installer and subjected to scrutiny.Measurement analyses of ST collector hot water output temperature for the intended use relative to average daily solar radiation was implemented in February 2016 for 1 week using HYELEC Digital Multi-meter with a K-type thermocouple probe as shown in Fig. 5. To uphold the safe use of the apparatus, the material safety data sheet (MSDS) of the measuring instrument is consulted and recommendations were implemented before each use. The set of data obtained were subjected to error analysis using the design error margin specification provided by the manufacturer. Though the instrument could measure between -50°C to 750°C (-50≤oC≤750) range but the focus is on the 0°C to 200°C range which the manufacturer specified ± (0.01 read 2°C) as the measure of accuracy. Based on this specification, obtained data during the measurement were put through error analysis to ascertain a measure of accuracy. To stay within an acceptable degree of accuracy, the lower limit of the error margin is adopted and this implies that - 2°C is subtracted from all values obtained during the measurement. The choice of this instrument is based on its affordability and accuracy in terms of error analysis and 1year calibration validity as provided by the Original Equipment Manufacturer (OEM). Measurement results are presented in Fig. 9a showing collector input temperature and output temperature at observed ambient temperature between 07:00hrs and 18:00hrs when the solar thermal collector is tilted to the south and oriented to about 15°C to the roof horizontal plane. Collector Temperature gradient is also evaluated. To ensure control, samples were collected for two different collectors within the same location.

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