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Design of Solar Powered VARS for Application in a Tropical Region

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Abstract— **This work is on the design and analysis of a Water - Lithium Bromide (H2O - LiBr) absorption refrigeration system for application in the rural communities of South-Eastern geopolitical zone of Nigeria putting into consideration the geographical location and climatic conditions of the country of study. The geographical and climatic position of Nigeria have placed it to receive abundant solar radiated energy, but with its teeming population of rural and some urban dwellers living with poor conventional energy supply. This work, therefore, looks at the need for good preservative system for food, agricultural produce and temperature cooling using this abundant solar energy in the absence of conventional source. Water - lithium bromide absorption refrigerating system was adopted in this design because Nigeria is not a sub-zero zoned country and she has an average room temperature of 300C - 360C. This system was designed with a generator/desorber temperature of 950C, evaporator temperature of 350C, and absorber temperature of 450C. Through the application of thermodynamic models, the analysis of the designed system parameters yielded the pressure limits of 0.87kPa at the evaporator of the system, and 5.29kPa at the condenser point of the system. The analysis of the designed VARS gives a very reasonable coefficient of performance (COP) of 0.807 and 1.259 at maximum. These results show that the designed system with its thermodynamic parameters is efficient and effective to be applied in Nigeria and some sub-saharan African nations.**

Index Terms— **Coefficient of Performance, Refrigerant, Solar Power, Vapour Absorption**

I. INTRODUCTION

Utilizing and managing all available energy resources in a smart and wise manner is unarguable regardless of its renewability, sustainability or lack thereof, many efforts have been made to advantageously use any wasted and un-utilized energy. Solar energy is a very large, inexhaustible source of energy. The power from the sun intercepted by the earth is approximately 1.8 x 10^{11} MW which is much larger than the present consumption rate on the earth, of all commercial energy sources known to man. In principle, solar energy could supply all the present and future energy needs of the world on continuous basis [1]. In addition to solar energy size, it has two other advantages over fossil fuels and nuclear power; (a) it is an environmentally clean energy source (b) it is free and available in adequate quantities in almost all parts of the world.

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Nigeria is blessed with abundant amount of solar radiated energy. It has been found that there is an estimated 3000hrs of annual sunshine. Reference [2] reported that Nigeria has an estimated number of rural communities of over 97,000 whose population is characterized with deprivation from conventional energy, arising from poor supply infrastructure. For instance, about 18% only of the rural population have access to electricity as at 1991/1992, however, where this convectional energy is available, it supply is unreliable. The readily available and widely utilized energy in the rural areas is the renewable energy type such as wood, agricultural and animal wastes, wind energy and solar energy which are mainly used for cooking, cottage industrial applications, winnowing and open-to sun-drying process [3]. Therefore, with abundant availability of solar energy in this West African nation Nigeria, it becomes imperative to utilize this energy for cooling processes such as refrigeration and air-conditioning and other areas of applicability.

Presently, most of the cooling produced is by vapour compression or vapour absorption refrigeration system. The compressor of these vapour-compression systems use a huge amount of electrical energy generated by burning fossil fuel. However, the scarcity of energy and depletion of fossil fuel sources around the world creates the need for the development of a refrigeration system that may run on an alternative source of energy. Therefore, this is where a solar powered vapour absorption refrigerating system (VARS) comes into play with great advantage.

The price of energy has been increasing exponentially world over. It is a fact that industrial and home refrigeration are one of the most energy consuming sectors. The solution to this problem lies in the design of a refrigeration system which uses no energy or minimal amount of energy, and this system is an Absorption Refrigeration System running on solar energy source. By producing an absorption refrigeration system energy costs are not only being cut down but the environment is also preserved as this system does not use any of the CFCs or HCFC gases. It is an established notion that CFC and HCFC gases used as refrigerants in conventional vapour compression refrigeration and air conditioning systems not only have high global warming potential (GWP) but also high Ozone depletion potential (ODP). These effects can be remedied by using environmental friendly systems like H_20-LiBr or NH_3-H_20 vapour absorption refrigeration system.

Also energy consumption for industrial agriculture, commerce and residence in many countries have been increasing, which leads to the increase in demand of alternative energy. The government of the Federal Republic of Nigeria has come up with policies that will help to ensuring adequate supply of power. The cost of having this electric energy monitored through metering system cannot be ideally contended with by the rural dwellers due to cost of services from appropriate authorities in Nigeria [4].

The privatization of the Nigeria electric power sector (generation, transmission and distribution) has ushered in an era of a better and improved power supply which is still not sufficient for teaming population of the most populous black nation in the world. Her rural communities still suffer from shortage of power supply or total unavailability of this energy source.

II. LITERATURE REVIEW

A. Refrigerant/Absorbent Combination for Solar Powered VARS System

The need for two or more substances that should work together as a single solution of working fluid, produced several variants of refrigerant–absorbent pairs in the absorption refrigeration industry. The refrigerant should be more volatile than the absorbent so that the two can be separated easily. Water is usually used as the refrigerant for solid absorbents [5]. Many researchers have reported the use of solar energy for the absorption systems working in various refrigerant–absorbent combinations, namely:

- Ammonia – Water [6], [7], [8], and [9].
- Water –Lithium Bromide [10], [11], and [12].
- Lithium Bromide – Zinc Bromide Methanol [13].
- Lithium Chloride – Water [14].
- Ammonia –Calcium Chloride [15], and [16] etc.

The cycles of operation were either intermittent or continuous – open or closed. A suitable refrigerant–absorbent fluid is probably the single most important item in an absorption refrigeration system. The selection of a suitable combination involves simultaneous consideration of various factors which include qualitative considerations of desirable properties as well as an analysis of its theoretical performance in a refrigeration system. Ammonia –Water and Water-Lithium Bromide as conventional fluids still have desirable properties compared to other working fluid variants, especially for the high number of latent heat, so can minimize the need of refrigerant flow rate. The water-lithium bromide (H_2O – LiBr) is limited to temperature above the freezing point of water (4^0C - 5^0C) while Ammonia – Water (NH_3 – H_20) is favourable for subzero refrigerant temperatures [17].

However, the desirable properties of refrigerant absorbent mixtures or combinations for VARS can be outlined as follows:

(1) The refrigerant should exhibit high solubility with solution in the absorber. This is to say that it should exhibit negative deviation from Raoult's law at absorber.
(2) There should be large difference in the boiling points of refrigerant and absorbent (greater than 200^0C) so that only the refrigerant is boiled off in the generator.
(3) It should exhibit small heat of mixing so that a high COP can be achieved. However, this requirement contradicts the first requirement. Hence, in practice a trade-off is required between solubility and heat of mixing.
(4) The refrigerant-absorbent mixture should have high thermal conductivity and low viscosity for high performance.
(5) It should not undergo crystallization or solidification inside the system.
(6) The mixture should be safe, chemically stable, non-corrosive, inexpensive and readily available.

B. Solar Driven Refrigeration Systems' Review

Recently, many researchers have carried out research on refrigeration systems powered or driven by renewable energy source, achieving the conservation of conventional energy and reduced environmental pollution. Among these renewable energy sources, solar energy has proven to be the best for air cooling and refrigeration because of the achievement of maximum cooling load at the highest solar radiation input [18] and [19]. Solar energy can be converted to electricity using photovoltaic cells and used to operate a conventional vapour compression refrigeration system, hence, a refrigeration cycle driven by a combination of heat and electrical energy addresses difficulties associated with the solar–heating systems' long-term operation at steady state [20].

There are innumerable number of researches that have been carried out on solar powered absorption refrigeration system, thus we try to review some of them. Reference [21] developed and tested a prototype design of a 2KW refrigeration equipment driven by solar radiation with a low COP of 0.5. Reference [22] also did much review on various theoretical and practical applications of solar refrigeration system. [23] developed a single effect H_20–LiBr absorption chiller of 7kw nominal cooling capacity assisted by a 32.2m^2 array of flat plate solar collectors achieving a 6.5KW cooling capacity during summer. However, [24], emphasized that theoretically a high COP of about 1.7 can be achieved using a triple effect cycles.

Reference [25], published their study on dynamic performance of a flat-plate solar solid-adsorption refrigeration for ice making operating with activated carbon/methanol. Also, [26] presented the description and operation of a novel solar powered adsorption refrigeration system operating with activated carbon/methanol. In their work [27] investigated on an absorption refrigerator driven by solar cells, thought it produced a low COP of 0.25 it was able to maintain a refrigeration temperature of 5.8^0C. Again [20] with their experimental research on a new solar pump free lithium bromide absorption refrigeration system with a second generator where the maximum COP was 0.787. During his research [28] examined the potential energy benefits of integrated refrigeration system with microturbine and absorption chiller driven by solar energy. Reference [29] conducted an hourly simulation and performance of solar electric vapour compression refrigeration system. [30] applied a double generator principle in a solar powered system using ammonia–water absorption refrigeration system. Under solar driven system, [31] conducted a stationary analysis of a solar driven H_20-Libr absorption refrigeration system.

Reference [32] performed a thermodynamic analysis of different working fluid pairs of solar driven absorption refrigeration system they observed that with the increase in evaporator temperature; the COP values of each cycle increases [33], ran a thermodynamic analysis of vapour absorption refrigeration cycles using NH_3-H_20 and H_20-Libr solutions their system consists of three heat exchangers (refrigerant, solution and refrigerant – solution heat exchangers). [34] carried out a design and analysis of a solar powered VARS using H_20-LiBr, observing that higher evaporator and generator temperature and low condensing temperature result in higher COP. In their work [35] developed a computer program for simulation of a solar powered ARS. Also [36] simulated the performance of a solar ARS using H_20 with a COP of 0.93. The area of solar energy or renewable energy as a source to power a cooling and refrigeration systems is a very interesting research area with numerous recorded literatures, hence, we can only review but a few in this study.

III. METHODOLOGY

The approach in designing this system is based on the consideration that the system will be put into use in the South–East geo-political zone of Nigeria. This solar powered absorption refrigeration system was designed on actual data obtained through metrological reports on solar radiation over Nigeria. Other parameters employed in the thermodynamic analysis of the system cycle were based on the actual and ideal conditions of operation. However, standard charts /graphs were employed in determining some parameters such as vapour pressure of solution enthalpy of water etc. though the design was not fabricated at this time., the material selections were designed based on standard specifications. Therefore, this study is to design and analyze an absorption refrigeration system driven by solar energy making use of H_20 (refrigerant) and LiBr (absorbent).

Nigeria is abundantly blessed with a good amount of sunshine. It has been found that there is a estimated 3,000 hrs of annual sunshine [3]. Nigeria is located in the tropics between 3^0E to 15^0E of longitude and 4^0N and 14^0N of latitude [37]. This geographical location of Nigeria will help to appreciate meteorological data that will be used in the system analysis. Taking into account the climatic position of Enugu and Owerri (Lat 7.55^0N, Long 6.45^0; and Lat. 5.41^0N, Long 7.20^2C respectively), the limitation on maximum and minimum pressure was determined. Hence, the maximum pressure after condensation is designed at 4.3KPa at minimum temperature of 35^0C. While the minimum refrigeration temperature was designed at 5^0C with minimum pressure of 0.87KPa (properties table for R718 ASHRAE 2012, Fundamentals).

A. Design of System Components

As mentioned earlier the vapour absorption refrigeration system has four major components and in designing them standard parametric values and climatic position area of study were put into consideration.

1) Solar Energy Collector: Solar energy can be captured with the help of solar collector which are of two types (i) concentrating solar collector (ii) non-concentrating solar collector. In the non-concentrating type, the collector area that intercepts the solar radiation is same as the absorber area absorbing the radiation. The common types of non concentrating collectors are flat-plate and evacuated –tube collectors. Concentrating collector like the parabolic dish and parabolic trough collectors have a bigger interceptor than absorber. The flat-plate collectors is used in this design for heat absorption from solar radiation. Flat plate collectors use either air or liquid to transfer the collector heat to point of need, both working with the same principle. Majority of the flat-plate collectors have the following basic components in common (i) the absorber surface (ii) the heat transfer interface/fluid passage (iii) the glazing (iv) the insulation (v) the casing.

Considering the latitude and longitude position of Enugu and Owerri and the variation of suns position over the years (Table 1 and 2), the average incident solar beam radiation for a particular month can be calculated through the application of the following expression:

$$I_o = I_{sc}[\cos \emptyset . \cos \delta . \cos \omega - \sin \emptyset . \sin \delta] \qquad (1)$$

Equation (1) is the solar radiation on a horizontal plate collector,

Where I_{sc} = 1367 W/m^2

$\emptyset$ = latitude location (Enugu 7.55^0N, Owerri 5.41^0N)

δ = Sun's declination given as 23. 45 [360 (284 + n)/365]

n = day of the month at a particular time

w = time of day (e.g 1pm) x Latitude position

The thermal performance of a solar collector array depends upon a number of factors such as collector area, type, tilt angle, temperature of heat transfer fluid entering the collector and shading. The thermal performance of a collector can be calculated from a first law energy balance which for a flat-plate collector is given as:

$$Q_u = I_c A_c Y_s \propto_s - U_c A_c (t_c - t_o) \qquad (2)$$

Where Q_u is rate of useful energy gain, W

I_c is global insolation on plane of collector W/m^2

A_c is area of collector that absorbs radiation m^2

Y_s is not solar transmittance of glazing

α_s is solar absorptance of collector plate

U_c is overall heat loss coefficient W/m^{2}K

t_c is average collector plate surface temperature, ^{0}C

t_o is ambient air temperature, ^{0}C.

To improve the thermal performance of a solar collector, it is necessary to reduce either the overall heat loss coefficient or the area from which energy is lost [38]

Table 1: Meteorological data and global solar radiation for Enugu [3]

Month	T_M(°C)	$\frac{n}{N}$	$\frac{c}{C}$	$\frac{R}{100}$	$\bar{H}_0$ $(MJm^{-2}day^{-1})$	$\bar{H}_M$ $(MJm^{-2}day^{-1})$	$\frac{\bar{H}_M}{\bar{H}_O}$
JAN	35.81	0.5484	0.0310	0.46	35.82	16.09	0,4492
FEB	37.69	0.5579	0.0472	0.49	37.01	17.65	0.4769
MAR	37.30	0.5013	0.0571	0.61	37.54	18.05	0.4808
APR	35.57	0.5296	0.0652	0.71	36.44	18.56	0.5093
MAY	34.07	0.5247	0.0666	0.77	34.41	17.93	0.5211
JUNE	32.65	0.4294	0.0681	0.80	33.15	15.59	0.4703
JULY	31.62	0.3113	0.0712	0.82	34.85	14.23	0.4083
AUG	31.14	0.3088	0.0711	0.83	35.47	14.37	0.4051
SEPT	31.96	0.3702	0.0702	0.82	36.95	15.24	0.4124
OCT	32.91	0.5038	0.0669	0.78	37.73	14.58	0.3864
NOV	34.93	0.6459	0.0536	0.65	35.83	17.29	0.4826
DEC	35.49	0.6005	0.0385	0.52	35.22	16.46	0.4673

Table 2: Meteorological data and global solar radiation for Owerri [3]

Month	T_M(°C)	$\frac{n}{N}$	$\frac{c}{C}$	$\frac{R}{100}$	$\bar{H}_0$ $(MJm^{-2}day^{-1})$	$\bar{H}_M$ $(MJm^{-2}day^{-1})$	$\frac{\bar{H}_M}{\bar{H}_O}$
JAN	33.88	0.4275	0.0546	0.66	34.28	14.46	0,4201
FEB	35.11	0.4497	0.0568	0.69	36.06	15.51	0.4509
MAR	34.17	0.3967	0.0629	0.76	37.52	15.09	0.4041
APR	33.17	0.4255	0.0666	0.78	37.48	17.19	0.4450
MAY	32.14	0.4728	0.0671	0.78	36.24	16.28	0.4183
JUNE	30.87	0.3803	0.0644	0.83	35.13	14.54	0.3974
JULY	29.28	0.2564	0.0702	0.86	35.61	13.10	0.3506
AUG	29.37	0.2207	0.0692	0.87	37.05	13.42	0.3516
SEPT	30.28	0.2765	0.0691	0.84	37.26	14.43	0.3763
OCT	31.02	0.3388	0.0673	0.82	36.18	14.81	0.3949
NOV	32.84	0.4642	0.0580	0.77	34.38	15.41	0.4072
DEC	33.29	0.4611	0.0533	0.75	33.19	14.84	0.4329

Where T_M is the maximum temperature
$\frac{n}{N}$ is the fraction of sunshine duration
$\frac{c}{C}$ is the cloudiness index
R is the relative humidity
$\bar{H}_0$ is the monthly mean extraterrestrial solar radiation on horizontal surface
$\bar{H}_M$ is the measured monthly mean daily global radiation
$\frac{\bar{H}_M}{\bar{H}_O}$ is the clearness index

2) The Generator/Desorber: The generator delivers the refrigerant vapour to the rest of the system by separating refrigerant from the solution. The generator is designed to have the solution vertically fall over horizontal tubes with high temperature energy source, typically steam or hot water flowing through the tubes. The solution absorbs heat from the warmer steam or water, causing the refrigerant to boil (vaporize) and separate from the absorbent solution. As the refrigerant boils away, the concentration of the absorbent solution becomes stronger.

The generator is designed to be made of mild steel pipe, closed at one end and the other end with grooved flange. The flange cover supported the immersion heater used to simulate the solar energy supply to the generator. The heater is to be installed in the lower part of horizontal shell. The vapour line connection to the condenser is designed as a copper tube and the same material for weak solution inlet from the exchanger, likewise the strong solution outlet. The tube from the outlet is designed to protrude into the shell to ensure the immersion heat is always covered with solution to prevent element burn out.

3) The Condenser: This component is used to condense the refrigerant vapour and heat is extracted from refrigerant changes from vapour to liquid and the temperature of the refrigerant changes from 95^{0}C to 35^{0}C. The system condenser is designed as a spirally wound coiling coil made with copper tube coiled to about 10 turns. Cooling liquid flows inside the coil while the refrigerant vapour flows outside the coil. The coil is designed to be supported from the flange cover which is bolted to the grooved flange with an o – ring in place to maintain the vacuum required in the vessel. To facilitate heat rejection by the condenser, it is not insulated. Therefore, as heat transfers from the refrigerant vapor to the cooling liquid (may be water), The refrigerant condenses on the tube surfaces. The condensed liquid refrigerant is collected at the bottom of the condenser before proceeding to the expansion device.

4) The Evaporator: The evaporator containing a bundle of tubes, carry the system of circulating water to be cooled. In a typical LiBr system, water evaporates at an evaporating temperature of 5^0C. The corresponding saturated pressure is 0.87KPa in the evaporator. At this low pressure in the evaporator, the refrigerant gets evaporated by absorbing heat from the circulating water and the refrigerant vapours thus formed tend to increase the pressure in the vessel. Hence, with increase in pressure, the boiling temperature increases and the desired cooling effect is not obtained. Therefore, the refrigerant vapours are removed from the vessel into the lower pressure absorber. Most commonly the evaporator and absorber are contained inside the same shell, allowing refrigerant vapours generated in the evaporator to move continuously to the absorber.

5) The Absorber: The refrigerant vapour from the evaporator is absorbed by the solution inside the absorber. As the refrigerant vapour is absorbed, it condenses from a vapour to a liquid so that the heat it acquired in the evaporator is being released. With the help of an absorber pump an intermediate concentration solution is sprayed into the absorber. For a typical LiBr system a solution of 64.5% is cooled to an absorber temperature of 43^0C by a cooling water in the heat exchanger. During the absorption process the integral heat solution is removed by the cooling water. The diluted (weak) solution is collected at the bottom of the shell. The weak solution is mixed with the strong solution returning from the generator.

To increase the rate of absorption, the surface area of the solution, it is designed that the solution will be sprayed through nozzles in the absorber. The cooling water is used to remove the heat from the absorber. This will keep down the temperature of the solution, which depending upon the design it varies from 38 – 40^0C or 40 – 43^0C. The cooling water entering the tubes is about 30^0C and leaves at 35^0C absorbing heat from the condenser.

B Thermodynamic Properties of the System

The determination of the thermodynamic properties of each state in the cycle, the amount of heat transfers in each component, and the flow rates at different lines depend on the following [39],

Generator temperature t_g (^{0}C)
Evaporator temperature t_e (^{0}C)
Condenser temperature t_c (^{0}C)
Absorber temperature t_a (^{0}C)
Liquid –Liquid heat exchange effectiveness E_L
Refrigeration load Q_E (tons)

However, going by the assumption of neglecting the pump work and neglecting the pressure drop in components and lines and assigning saturation conditions to states number 1, 4,8 and 10 in Fig. 1, the properties of the system are determined as follows:

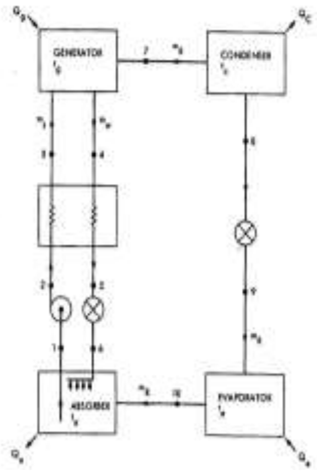

Fig 1: Flow Diagram for a Lithium Bromide - Water Absorption System [39]

Concentration of Absorber Solution:
This is determined by applying equation (3) as follows

$$X_1 = X_2 = X_3 = X_{stong\ solution} = \frac{49.04 + 1.125\ t_a - t_e}{134.65 + 0.47 t_a}$$

Concentration of Generator Solution:
This can be determined by applying the following equation:

$$X_4 = X_5 = X_6 = X_{weak\ solution} = \frac{49.04 + 1.125\ t_g - t_c}{134.65 + 0.47 t_g}$$

It may be noted that X_4 is always larger than X_1 and

$$X_7 = X_8 = X_9 = X_{10} = 0 \qquad (5)$$

Cycle Pressure Limits:
It is possible to evaluate the pressure in each line through the application of the follow expression:

$P_{evaporator}\ P_e = P_1 = P_6 = P_9 = P_{10}$ given by

$$\log_{10} P_e = 7.8553 - \frac{1555}{t_e + 273.15} - \frac{11.214 \times 10^4}{(t_e + 273.15)^2} \qquad (6)$$

The condenser pressure,
$P_c = P_2 = P_3 = P_4 = P_5 = P_7 = P_8$ is given by

$$\log_{10} P_c = 7.8553 - \frac{1555}{t_c + 273.15} - \frac{11.2414 \times 10^4}{(t_c + 273.15)^2} \qquad (7)$$

Flow Rates:
The enthalpy of saturated water (state 8) at the condenser temperature t_c is obtained as:

$$H_8 = (t_c - 25) \qquad (8)$$

The throttling process from 8 to 9 and that from 5 to 6 give:

$$\left.\begin{aligned} H_8 &= H_9 \\ H_5 &= H_6 \end{aligned}\right\} \quad (9)$$

Enthalpy of saturated water vapour (state 10) at the evaporator temperature is given as:

$$H_{10} = 572.8 + 0.417t_e \quad (10)$$

Therefore, applying the first law o thermodynamic to the evaporator we obtain that:

$$Q_E = m_R\,(H_{10} - H_9) \quad (11)$$

Where M_R is the refrigerant flow rate, equals the difference between the strong and weak solution rates, therefore applying equation (9):

$$M_R = \frac{Q_E}{H_{10} - H_8} \quad (12)$$

Also the LiBr mass balance in the absorber is:

$M_W\,X_6 + \; M_R X_{10} = M_8 X_1 = \;(M_W + M_R)\;X_1$

Hence,

$$M_w = \left(\frac{Q_E}{H_{10} - H_8}\right)\left(\frac{X_1}{X_4 - X_1}\right) \quad (13)$$

$$M_8 = \left(\frac{Q_E}{H_{10} - H_8}\right)\left(\frac{X_4}{X_4 - X_1}\right) \quad (14)$$

Liquid – Liquid Heat Exchanger Temperatures:

Once the heat exchanger effectiveness E_L, the mass flow rates (M_w, M_s), and the concentrations (X_4, X_1) are obtained, it is possible to determine the solution temperatures t_3 and t_5 [39]. Going by the weak solution side, which has the minimum heat capacity, the effectiveness E_L is defined by;

$$E_L = \frac{t_g - t_5}{t_g - t_a} \quad (15)$$

Or by the strong solution side we have:

$$E_L = \frac{(m_s.\;C_{x_1}).(t_3 - t_a)}{(m_w\,.\,C_{x_1})\,.\,(t_g - \;t_a)} \quad (16)$$

And

$$\left.\begin{aligned} C_{x_1}\;1.01 - 1.23X_1 + \;0.48X_1^2 \\ C_{x_4}\;1.01 - 1.23X_4 + \;0.48X_4^2 \end{aligned}\right\} \quad (17)$$

Equations (15) and (16) can be rewritten using equations (13) and (14) to obtain the temperatures t_3 and t_5 as:

$$t_5 = t_g - E_L\;(t_g - t_a) \quad (18)$$

And

$$t_3 = t_a + \left[E_L\;\left(\frac{X_1}{X_4}\right)\left(\frac{CX_4}{CX_1}\right)(t_g - t_a)\right] \quad (19)$$

Enthalpies H_1 and H_5 will be calculated by applying equation;

$H_{x,25} + C_x\;(t - 25)$

Or

$$42.81 - 425.92\;X + 404.67X^2 + (1.01 - 1.23X + 0.48X^2)\;(t) \quad (20)$$

Therefore;

$$H_1 = (42.81 - 425.92\;X_1 + \;404.67X_1^2) + \;(1.01 - 1.23X_1 + 0.48X_1^2).t_a \quad (21)$$

$$H_5 = (42.81 - 425.92\;X_4 + \;404.67X_4^2) + \;(1.01 - 1.23X_4 + 0.48X_4^2).t_5 \quad (22)$$

Heat Transfer in Condenser, Generator and Absorber:

The enthalpy of water vapour learning the generator and entering the condenser (state 7) is obtained as:

$$H_7 = 572.8 \; + \; 0.46t_g \; - \; 0.943t_c \quad (23)$$

The heat balance of the condenser is given as:

$$Q_c = M_R\;(H_7 - H_8) \quad (24)$$

or

$$Q_c = \frac{Q_g}{(H_{10} - H_8)}.\;(H_7 - H_8) \quad (25)$$

Heat balance for combined generator and heat exchanger control volume gives

$$Q_G = M_w H_5 + M_R H_7 - M_S H_Z \quad (26)$$

But as pump work is negligible, hence,

$$H_1 \approx H_2 \quad (27)$$

Applying (12) (13) (26) and (27), Q_G can now be written as:

$$Q_G = \frac{Q_E}{(H_{10} - H_8)}\left[\frac{X_1 H_5}{(X_4 - X_1)} + H_7 - \frac{X_4 H_1}{(X_4 - X_1)}\right] \quad (28)$$

Heat balance of the absorber gives Q_A as

$$Q_A = m_w H_6 + M_R H_{10} - \;m_s H_1 \quad (29)$$

Using (10), (12) (13) and (14), Q_A is rewritten as:

$$Q_A = \frac{Q_E}{(H_{10} - H_8)}\left[\frac{X_1 H_5}{(X_4 - X_1)} + H_{10} - \frac{X_4 H_1}{(X_4 - X_1)}\right] \quad (30)$$

Equations (25) and (30) are governed by the first law of thermodynamics in the form of:

$$Q_c + Q_A = Q_G + Q_E \quad (31)$$

Coefficient of Performance (COP):

This is usually defined as follows:

$$COP = \frac{refrigeration\;\;effect}{external\;\;heat\;\;input} = \frac{Q_E}{Q_G} \quad (32)$$

It can simply be derived from equation (28) as:

$$COP = \frac{(H_{10} - H_8)\,(X_4 - X_1)}{[X_1 H_5 + (X_4 - X_1)\,H_7 - X_4 H_1]} \quad (33)$$

Ideal Coefficient of Performance:

The maximum coefficient of performance of these designed absorption cycle is obtained as:

$$(COP)_{max} = \frac{T_e\;(T_g - T_a)}{T_g\;(T_c - T_e)} \quad (34)$$

Where T_e, T_a, T_c and T_g are the absolute temperature of evaporator, absorber, condenser and generator respectively. The relative performance ratio, which shows deviation from reversible cycle operation is given as

$$Ratio\; = \frac{(COP)_{actual}}{(COP)_{max}} \quad (35)$$

IV. PERFORMANCE ANALYSIS

The thermodynamics cycle performance of this absorption refrigeration system is analyzed based on the parameters designed on to the system as its typical running conditions. Meanwhile, this system is designed to function under the following conditions of basic operating parameters:

Generator/Desorber Temperature is 95^0C

Evaporator temperature tc is 5^0C

Condenser temperature tc is 35^0C

Absorber temperature ta is 45^0C

Effective Heat Exchanger E_L is obtained from equation (50) as 0.82

Evaporator rate of heat transfer Q_E = 1ton of refrigeration = 3024K cal/hr

Therefore, we go on to analyze each step of the cycle;

For solution concentration in absorber i.e. strong solution (X_1) we apply equation (3), thus;

$$X_1 = \frac{49.04 + 1.125\,(45-5)}{134.65 + 0.47(45)}$$

$$= \frac{94.04}{155.80}$$

$$= 0.603\ kg\ LiBr/kg\ solution$$

For solution concentration in the generator i.e. the weak solution (X_2) we apply equation (4), thus;

$$X_1 = \frac{49.04 + 1.125\,(95-35)}{134.65 + 0.47\,(95)}$$

$$= \frac{116.55}{179.30}$$

$$= 0.64\ kg\ LiBr/kg\ solution$$

For enthalpy of saturated liquid water (State 8) we apply eqn. (8), thus;

H_8 = (35-25)

= 10Kcal/Kg

For enthalpy of saturated water vapour (State 10) we apply eqn. (10), thus:

H_{10} = 572.8 + 0.417 (5^0C)

= 574.89Kcal/kg

The refrigerant flow rate (MR) in the evaporator can be obtained through equation (12)

$$M_R = \frac{3024}{574.89 - 10}$$

= 5.353 Kg/hr

The mass flow rate of water (Mw) in the absorber is obtained through application of eqn. (13), thus:

$$M_w = \left(\frac{3024}{574.89-10}\right)\left(\frac{0.603}{0.64-0.603}\right)$$
$$= 5.353\ x\ 16.297$$
$$= 87.24 Kg/hr$$

And the mass flow rate of Lithuim Bromide solution (Ms) is obtained through the application of eqn. (14)

$$M_w = \left(\frac{3024}{574.89-10}\right)\left(\frac{0.64}{0.64-0.603}\right)$$
$$= 5.353\ x\ 17.297$$
$$= 92.592 Kg/hr$$

For the temperature of liquid water (t_5) leaving the heat exchanger form the generator into the absorber we apply eqn. (18), hence:

$$t_5 = 95 - 0.82\,(95.45)$$
$$= 54^0 c$$

For the temperature of the solution (t_3) leaving the heat exchanger from the absorber back into the generator, we apply eqn. (19), hence:

$$t_3 = 35 + \left[0.82\left(\frac{0.603}{0.640}\right)\left(\frac{Cx_4}{Cx_1}\right)(95-35)\right]$$

But Cx_4 and Cx_1 which are the specific heat of weak and strong solution at concentrations X_4 and X_1 respectively are obtained from equation (17), thus:

$$Cx_1 = 1.01 - 1.23\,(0.603) + 0.48\,(0.603)^2$$
$$= 0.443\ Kcal/Kg^0C$$

and

$$Cx_1 = 1.01 - 1.23\,(0.640) + 0.48\,(0.640)^2$$
$$= 0.419\ Kcal/Kg^0C$$

Therefore;

$$t_3 = 35 + [0.82\ x\ 0.9422\ x\ 0.9458\ x\ 60]$$
$$= 78.85^0 C$$

Enthalpies at state 1 and state 5 is obtained through the application of eqns (21) and (22), thus:

$$H_1 = [42.81 - 425.92\,(0.603) + 404.67\,(0.603)^2] + [1.01 - 1.23\,(0.603) + 0.48\,(0.603)^2] \times 45^0 c$$
$$= -63.878 + 19.935$$
$$= -43.943\ Kcal/kg$$

And

$$H_5 = [42.81 - 425.92\,(0.64) + 404.67\,(0.64)^2] + [1.01 - 1.23\,(0.64) + 0.48\,(0.64)^2] \times 54$$
$$= -64.026 + 22.626$$
$$= -41.40\ Kcal/kg$$

The enthalpy of water vapour leaving the generator and entering the condenser (state 7) is eqn. (23), thus:

$$H_7 = 572.8 + 0.46(95) - 0.043\,(35)$$
$$= 614.995\ Kcal/kg$$

The rate of heat transfer in the condenser is (Q_c) is obtained by applying eqn. (24)

$$Q_c = 5.353\,(614.995 - 10)$$
$$= 3238.54 Kcal/kg$$

Heat transfer rate in the generator (Q_a) is obtained by applying eqn. (28), hence:

$$Q_G = \frac{3024}{(574.89-10)}\left[\frac{0.603\,(-41.40)}{(0.64-0.603)} + 614.995 - \frac{0.64\,(-43.943)}{0.64-0.603}\right]$$
$$= 5.353\,[-674.703 + 614.995 + 760.108]$$
$$= 3749.24\ Kcal/hr$$

The rate of heat transfer in the absorber (Q_A) is given by eqn. (30), hence:

$$Q_A = 5.353\,[-674.703 + 574.89 + 760.108]$$
$$= 3534.56\ Kcal/hr$$

Therefore, the coefficient of performance of this designed system is obtained through the application of eqn. (32), thus:

$$COP_{actual} = \frac{3024}{3749.24}$$
$$= \mathbf{0.807}$$

And the ideal coefficient of performance of the system (COP) $_{may}$ is obtained from eqn. (34), thus:

$$COP_{max} = \frac{(5 + 273.15)(95 - 45)}{(95 + 273.15)(35 - 5)}$$
$$= \mathbf{1.259}$$

Hence the relative performance ratio is:

$$\frac{0.807}{1.259}$$
$$= 0.6410$$

We can also compute the pressure limits in the system through equations (6) and (7) as follows:

Evaporator pressure

$$P_e = antilog\left[7.8553 - \frac{1555}{(5 + 273.15)} - \frac{11.2414\ x\ 10^4}{(5 + 273.15)^2}\right]$$
$$= 6.4834 mmHg$$
$$= \mathbf{0.8646 \approx 0.87 KPa}$$

Condenser Pressure

$$P_c = antilog\left[7.8553 - \frac{1555}{(35 + 273.15)} - \frac{11.2414\ x\ 10^4}{(35 + 273.15)^2}\right]$$
$$= 42.189 mmHg$$
$$= \mathbf{5.29 KPa}$$

V. DISCUSSION OF RESULTS

The analytical procedure followed in the analysis of design parameters of this VARS shows that the performance characteristics of a water/Lithium Bromide absorption system depends on many factors, these include the temperature of entering fluid to the generator/desorber, the cooling load at the evaporator, and the temperature of entering cooling fluid at the condenser or absorber. Should any of these designed parameters is altered or changed the coefficient of performance and efficiency will be altered positively or negatively the thermodynamic cycle. In this work justice has been done in the determination of various properties at the difference locations on the cycle. This system was designed to base on the evaporator which can cool up to 5^0C. Therefore, from the analysis of results it was obtained that the lower pressure in the evaporator and the higher pressure in the condenser are 0.87Kpa and 5.29KPa.

The generator temperature of up to 95^0C, the condensation temperature of 35^0C, the absorber temperature at 45^0C and the evaporator temperature of 5^0Cwhich were employed in the design, gave a very good coefficient of performance of 0.807 and a maximum COP of 1.259. When compared with that obtained by Lansing (2013), where 90^0C generator temperature, 40^0C condenser temperature, 40^0C absorber temperature and 7^0C evaporator temperature were used in running the system. The actual COP was 0.776 while the maximum COP was 1.1689. Therefore, the design parameters adopted by this study is quite recommendable for designing and construction of a solar powered vapour absorption refrigeration system for application in Nigeria and most sub-Saharan African nations.

VI. CONCLUSION

The United Nations Framework Convention on Climate Change (UNFCCC) reports that those who are least responsible for climate change are also the most vulnerable to its projected impacts. In no place is this more evident than in sub-saharan Africa, where greenhouse gas emissions are negligible from a global scale. Nigeria as a nation in this region face the same challenges as the rest of the world, with also the problem of growing fossil fuel energy demand. From many research analyses, it was seen that the level of global solar radiation recorded over this region especially Nigeria were high having least values. these values are adequate in citing solar driven systems as there is enough energy to power them all year round (Chineke, Okoro, and Igwilo, 2007; Chiemeka and Chineke, 2009). Therefore, designing and constructing solar driven preserving equipment, air conditioners, cookers, incubators to mention but a few that would be affordable but efficient for those living in the rural areas within Nigeria, becomes imperative as majority of the rural dwellers are below poverty level. This will in no small measure enhance the living conditions of these people which is one of the objectives of the Millenium Development Goals (MDG) of the United Nations of which Nigeria is a member.

In addition to the forgoing refrigeration systems that use environment friendly refrigerants provide a sustainability advantage when compared to other refrigerant selections. To minimize environmental impacts associated with refrigeration system operation, it is reasonable to evaluate the prospects of a clean source of energy. Hence, the designing of a H_2O-LiBr solar driven absorption refrigerating system under this study is unequivocally important. In this study the temperatures, pressures and concentration ration at different points were designed and evaluated. Thus the results indicate that a suitable solar vapour absorption refrigerating system can be designed keeping in view the climatic condition of a particular location. The methodology described in this study can be adopted to design and develop a suitable system that can be most effectively and efficiently used at maximum utilization of the solar power having in view the climatic conditions of temperatures.

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