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PERFORMANCE ANALYSIS OF A HOT WATER DRIVEN 5 TONS CAPACITY ABSORPTION REFRIGERATION SYSTEM FOR NIGERIAN APPLICATION

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ABSTRACT

Refrigeration by mechanical vapour compression system is an efficient method. However, the energy input is the shaft work, which is high-grade energy and, therefore, very expensive. And the work required is relatively large because of compression of the vapour which undergoes tremendous changes in specific volume. In order to achieve the raising of the refrigerant pressure from evaporator pressure to condenser pressure (without much altering its volume), the refrigerant vapour is dissolved in an inert liquid at the same pressure as the evaporator; and the solution so formed is pumped to a container, at condenser pressure. Thus, liquid which is practically incompressible and undergoes practically no change in specific volume, requires very little work for raising its pressure. Of all the absorbent-refrigerant pair available, the LiBr-H_2O combination is the most preferred because, among other reasons, it is environmentally benign. In this paper, a hot water driven, single stage, absorption cooling system, using a lithium bromide water solution, is analyzed. Four basic stages in the absorption cycle are generation, condensation, evaporation and absorption with ideally no moving part. A configuration of these four stages, having 5 Tons cooling capacity is determined and examined. The pressure parameter is varied in condenser and evaporator. Increasing the pressure of evaporator increases the COP; whereas, the COP is decreased by increase in the pressure of the condenser. Also, when the concentration of the LiBr solution moving into the absorber is lowered, the COP increases.

Keywords: Absorption Cycle, Solar Power, Configuration, System Energy, Coefficient of Performance.

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1. INTRODUCTION

A major application area of thermodynamics is refrigeration, which is the transfer of heat from a low temperature region to a high temperature one. Devices that produce refrigeration are called refrigerators, and the cycles on which they operate are called refrigeration cycles. The most frequently used refrigeration cycle is the vapour-compression refrigeration cycle in which the refrigerant is vaporized and condensed alternately, and is compressed in the vapour phase. Vapour - compression refrigeration dates back to 1834 when the Englishman Jacob Perkins received a patent for a closed-cycle ice machine using *ether* or other volatile fluids as refrigerants. Initially, vapour-compression refrigeration systems were large, and were mainly used for ice making, brewing, and cold storage. They lacked automatic controls and were steam-engine driven. In the 1890s, electric motor-driven smaller machines equipped with automatic controls started to replace the older units. By 1930, the continued improvements made it possible to have vapour-compression refrigeration systems that were relatively efficient, reliable, small, and expensive. Another well-known refrigeration cycle that has started gaining prominence in Nigeria, is the absorption refrigeration; where the refrigerant is dissolved in a liquid before it is *compressed*. This form of refrigeration becomes economically attractive when there is a source of inexpensive thermal energy at a temperature of 100 to 120 ^{o}C. As the name implies, absorption refrigeration systems involve the absorption of a refrigerant by a transport medium. The most widely used absorption refrigeration system is the ammonia-water system, where ammonia (NH_3) serves as the refrigerant and water (H_2O) as the transport medium. Other absorption refrigeration systems include water-lithium bromide and water-lithium chloride systems (Yunus & Michael 2011), where water serves as the refrigerant. The latter two systems are limited to applications such as air-conditioning, where the minimum temperature is above the freezing point of water. Among various absorbent and refrigerant pair, LiBr-Water is most promising in chiller application due to high safety, high volatility ratio, high affinity, high stability, and high latent heat (Anil, Bimal, Abhinaw, and Ashok, 2014). A vapour-absorption system operates with a condenser, a throttle valve, and an evaporator in the same way as a vapour-compression system, but the compressor is replaced by an absorber and generator (Eastop & McConkey, 1993), as shown in Figure. 1.

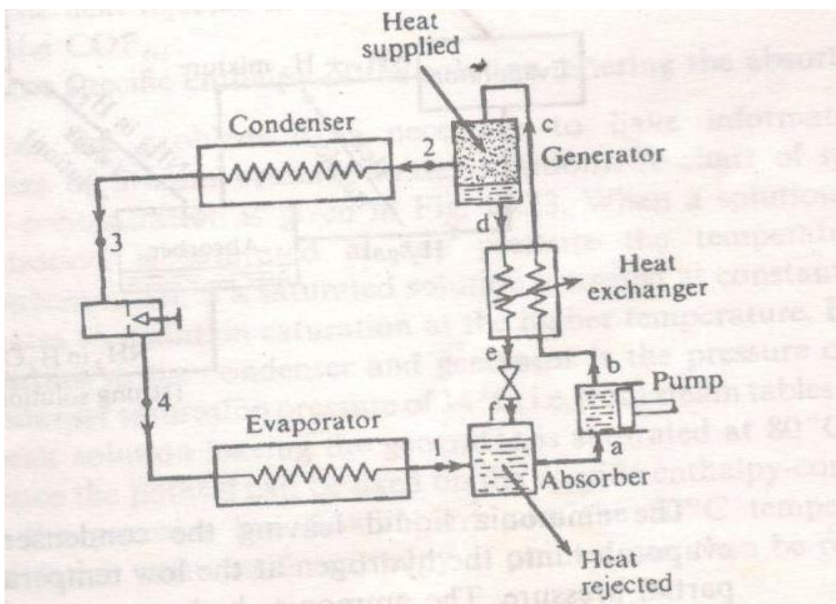

Figure 1: Diagrammatic arrangement of a vapour-absorption system

The refrigerant on leaving the evaporator is readily absorbed in a low-temperature absorbing medium, some heat being rejected during the process. The refrigerant-absorbent solution is then pumped to the higher pressure, and is heated in the generator. Due to the reduced solubility of the refrigerant-absorbent solution at the high pressure and temperature, refrigerant vapour is separated from the solution. The vapour passes to the condenser, and the weakened refrigerant-absorbent solution is throttled back to the absorber. A heat exchanger placed between the absorber and the generator makes the system more efficient, by transferring heat from the weak solution coming from the generator to the stronger solution pumped from the absorber (cf. Figure. 1). The work done in pumping the liquid solution is much less than that required to compress the vapour in the compressor of an equivalent vapour-compression cycle.

In the cycle, shown in Figure 2, water vapour evaporates and separates from aqueous LiBr solution in generator, and increases the concentration of LiBr in solution. At the same pressure, the water vapour from generator is condensed in condenser, and this condensed water is t of vaporization throttled to the evaporator, which is at low pressure. Due to reduced pressure, the water changes phase, and evaporates by taking latent heat of vaporization un the evaporator, at low temperature, and generates the cooling effect. At the same pressure, the vapour from evaporator is absorbed by LiBr aqueous solution supplied from the generator (having higher concentration of LiBr). The absorption of water vapour reduces the concentration of LiBr in aqueous solution; this solution is then passed to generator through pump, at higher pressure. The main energy input to the system is the heat supplied in the generator; this may be supplied in any convenient form such as a fuel-burning device, direct electrical heating, steam if already available, waste heat, or solar 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. Nigeria is blessed with abundant amount of solar radiated energy. It has been found that there is an estimated 3000hrs of annual sunshine. Obuka, Madu, & Onyechi, (2017) 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. Therefore, with abundant availability of solar energy in Nigeria, it becomes expedient to utilize this energy for cooling processes such as refrigeration and air-conditioning and other allied areas of applicability.

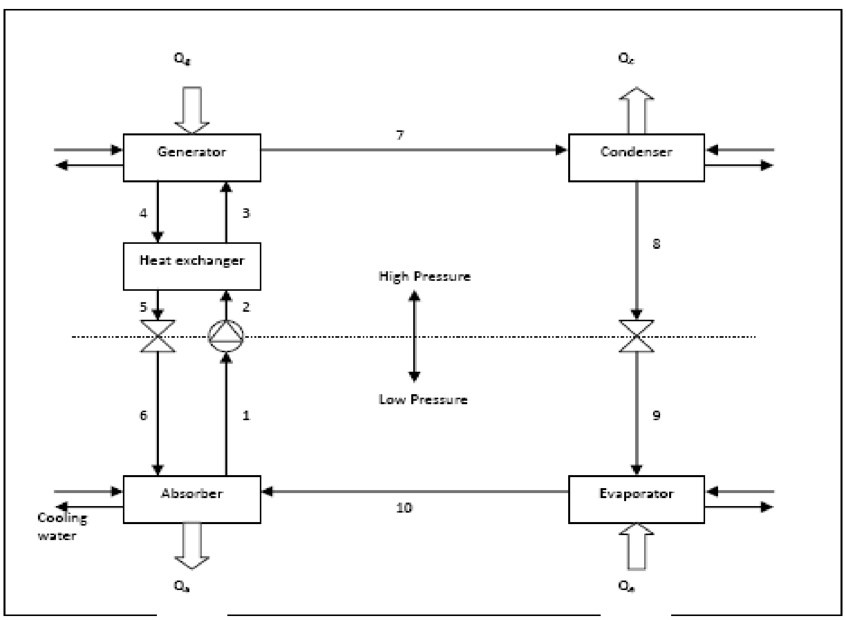

Figure 2: Schematic of a single effect liBr-water absorption system

Hwang, (2004). examined the potential energy benefits of integrated refrigeration system with micro-turbine and absorption chiller driven by solar energy. Bilgali (2011) conducted an hourly simulation and performance of solar electric vapour compression refrigeration system. Colona & Gabrielli (2003) applied a double generator principle in a solar powered system using ammonia–water absorption refrigeration system. Under solar driven system, Morine, Alonso, Palacin, & Guallar (2011) conducted a stationary analysis of a solar driven H_20-Libr absorption refrigeration system. Abdulateff, Sopian, Alghoul, Sulaiman, Zaharim, & Ahmad (2008) 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. Kaynakli (2014) 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). Sharma, Singh, & Gaur (2013) 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 Abdulla, El-Rahman, & El-Mahi (2005) developed a computer program for simulation of a solar powered ARS. Also, Shaikh, & Morabiya (2013). simulated the performance of a solar ARS using H_20 with a COP of 0.93.

In this paper, a 5 ton (17.585 kW) capacity absorption system is adapted for rural use in Nigeria. The different components of the cycle are configured using empirical *co-joints*, and their specification came by *gedanken* projections. In this contrivance, heat is released by allowing cooling water to circulate in absorber and condenser, at ambient temperature.

2. METHODOLOGY

To perform estimations of equipment sizing and performance evaluation of single-effect water-lithium bromide absorption cooler, basic assumptions and input values must be considered. With reference to Figure. 2, the basic assumptions are: (a) the steady state refrigerant is pure water, (b) there are no pressure changes except through the flow restrictors and the pump, (c) flow restrictors are adiabatic, (d) the pump is isentropic, and (e) there are no jacket heat losses (Soteris, George, Savvas, & Louis, 2001). The following nomenclatures fit into the heat and mass transfer calculation.

Table 1: Nomenclature

h	specific enthalpy (KJ/Kg)	Subscript	
m	mass flow rate (kg/sec)	i	Inlet
P	pressure (kPa)	e	exit
Q	Heat transfer rate (kW)	a	absorber
s	specific enthropy kJ/Kg^{o}K	g	generator
T	Temperature (^{o}K)	e	evaporator
X	LiBr mass fraction (%)	c	condenser
COP	Coefficient of Performance	1,2,3…	represent state point in Figure 2

2.1 Calculations: Heat and Mass Transfer

2.1.1 Mass Flow Rate Calculations

Mass flow rate in evaporator (m_9) = Load /Change in enthalpy

Mass flow rate for weak and strong solution

$m_4 . X_{ia} = m_3 . X_{i,g}$

Also, $m_3 = m_4 + m_7$

And, $m_1 = m_2 = m_{3;}$ $m_4 = m_5 = m_{6;}$ $m_7 = m_8 = m_9 = m_{10}$

Heat transfer rate in evaporator is taken as 5 Tons (Load). Based on this selected parameter, heat transfer rate at other components are calculated as follows;

2.1.2 Heat Transfer Rate at Condenser

$Q_c = m_7 (h_7 – h_8)$

h_7 = Enthalpy of superheated steam at saturation temperature of solution in generator and condenser pressure

h_8 = Enthalpy of water (saturated liquid) at condenser pressure and saturated temperature.

2.1.3 Heat Transfer Rate at Generator

$Q_g = m_4h_4 + m_7h_7 – m_3h_3$

h_4 = Enthalpy of solution at exit of generator temperature and $X_{i,a}$ Concentration

h_3 = Enthalpy of solution at inlet of generator temperature and $X_{i,g}$ concentration.

2.1.4 Heat Transfer Rate at Absorber

$Q_a = m_6h_6 + m_{10}h_{10} – m_1h_1$

h_6 = Enthalpy of solution at inlet of absorber temperature and $X_{i,a}$ concentration

h_{10} = Enthalpy of saturated water vapor at evaporator pressure and its saturation temperature.

h_1 = Enthalpy of solution at exit of absorber temperature and $X_{e,a}$ concentration.

2.2 Coefficient of Performance (COP)

COP (ideal) = Heat taken out in Evaporator / Heat supplied in generator = Q_e/Q_g

In the above ideal COP, work done by the pump, pressure drops and other losses are not included. Lower value of COP for absorption system compared to compression technology is because of using low grade energy rather than highly concentrated electric energy.

2.3 Configuration of a 5 TR System

2.3.1 Selection of Operating Parameters

After considering the optimum values in respect to ambient temperature, water properties and chilling requirements, following operating parameters are selected for various components of the cycle as shown in Table 2.

Table 2: Operating Parameters for Cycle

Pressure in generator and condenser	**7 kPa**
Pressure in absorber and evaporator	1 kPa
Saturation temperature in generator	82.863 ^{0}C
Saturation temperature in condenser	38.199 ^{0}C
Saturation temperature in evaporator	6.315 ^{0}C
Solution Temperature in absorber	36 ^{0}C
Solution Concentration at 3	55 %
Solution Concentration at 5	60 %
Evaporator capacity	5 Tons

2.3.2 Configuration of Heat Exchangers

Configuration for heat exchangers based on various empirical correlations is shown in Table 3. Standard dimensions of copper tubes had been selected from manufacturer's guide. Table 3 reveals that the absorber is having maximum dimensions among other and the compactness of complete system depends on the size of absorber.

3. RESULTS AND DISCUSSION

3.1 Coefficient of Performance Analysis

3.1.1 Effect of Solution Concentration

To check the solution concentration effectiveness, a constant difference of 5 % between the absorber inlet LiBr percentage ratio ($X_{i,a}$) and absorber exit ratio ($X_{e,a}$) are considered.

Table 3: Specifications of Components

	Absorber	Evaporator	Generator	Condenser
Inner diameter of tube	0.545 inch	0.43 inch	0.545 inch	0.43 inch
Outer diameter of tube	0.625 inch	0.5 inch	0.625 inch	0.5 inch
Number of tubes	38	18	16	8
Tube orientation	Vertical	Vertical	Horizontal	Horizontal
Length of tubes	1.2m	1.2 m	2.7 m (with one shell pass)	2.2m (with one shell pass)
Overall heat transfer	487.43	1507.07	719.35	1782.19

coefficient (W/m^2 ^{o}C)				
Water inlet temperature	28^0C Cooling	12^0C Chilled	92^0C Hot	28^0C Cooling
Water exit temperature	31^0C Cooling	8^0C Chilled	84^0C Hot	34^0C Cooling
Log mean temp. difference (LMTD)	18	13.5	17	19.61

The x-axis in Figure 3 represents the exit concentration of solution in absorber. The concentration of LiBr solution decrease while passing through the absorber due to absorption of water vapor. Enthalpy, which is a function of LiBr solution concentration and temperature at various state points as referred in Figure 2 are determined using properties of solution from reference (ASHRAE, 2007). Following conditions are assumed:

1. Evaporator capacity 5 Ton.
2. Solution heat exchanger exit temperature, 69.36 ^{0}C.
3. Generator solution exit temperature, 82.86 ^{0}C.
4. Absorber solution exit temperature, 55.86 ^{0}C.
5. Saturation temperature in condenser, 38.19 ^{0}C
6. Saturation temperature in evaporator, 6.315 ^{0}C.
7. Pressure in generator and condenser 7 kPa.
8. Pressure in absorber and evaporator 1 kPa.

3.1.2 **Effect of Evaporator Pressure on COP**.

To check this effect, the following conditions were assumed:

1. Evaporator capacity 5 Ton.
2. Solution heat exchanger exit temperature, 69.36 ^{0}C.
3. Generator solution exit temperature, 82.86 ^{0}C.
4. Absorber solution exit temperature, 55.86 ^{0}C.
5. Saturation temperature in condenser 38.19 ^{0}C.
6. Pressure in condenser 7 kPa.
7. Absorber inlet LiBr concentration (%) 60.
8. Absorber exit LiBr concentration (%) 55.

The correlation and variation between the COP and array of pressures, in the evaporator, is shown in Figure 4.

3.1.3 **Effect of Condenser Pressure on COP**

To check this effect, the following conditions were assumed:

1. Evaporator capacity 5 Ton.
2. Solution heat exchanger exit temperature, 69.36 ^{0}C.
3. Generator solution exit temperature, 82.86 ^{0}C.
4. Absorber solution exit temperature, 55.86 ^{0}C.
5. Saturation temperature in evaporator, 6.315 ^{0}C.
6. Pressure in absorber and evaporator 1 kPa.
7. Absorber inlet LiBr concentration (%) 60.
8. Absorber exit LiBr concentration (%) 55.

Figure 5 illustrates how the COP varies, and corresponds to different pressures, in condenser.

The specifications for various heat exchangers in the absorption cycle were determined in Table 2; this result was obtained using the empirical correlations related with heat and mass transfer. The size of the absorber affects the compactness of the overall system. The assumed temperature of cooling water availability depends on the ambient temperature, which can affect the overall dimensions of the system. The overall size of the system is reduced by the use of solution heat exchanger, which also increases the efficiency of the system. Figure 3 gives the COP variation with exit concentration of LiBr solution in absorber. The COP is more at the low exit concentration; and reduces as the exit concentration increase. The graph is plotted with constant difference of 5% in concentration. The concentration reduces due to affinity of solution to absorb water vapour. Increasing the value of pressure in evaporator gives more COP as shown in Figure 4.

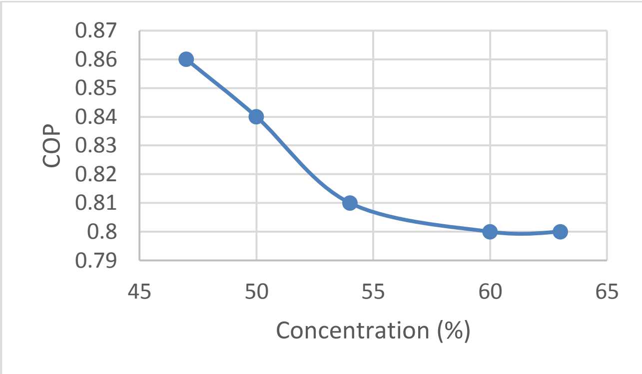

Figure 3: Concentration of LiBr solution in absorber

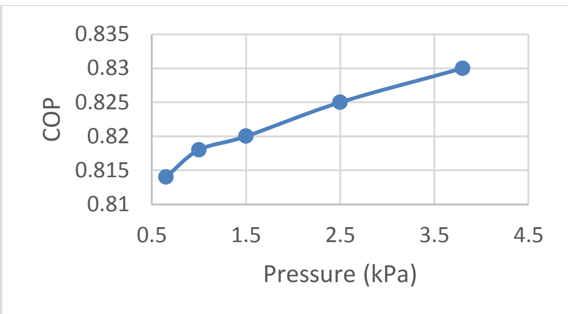

Figure 4: Value of pressure in evaporator

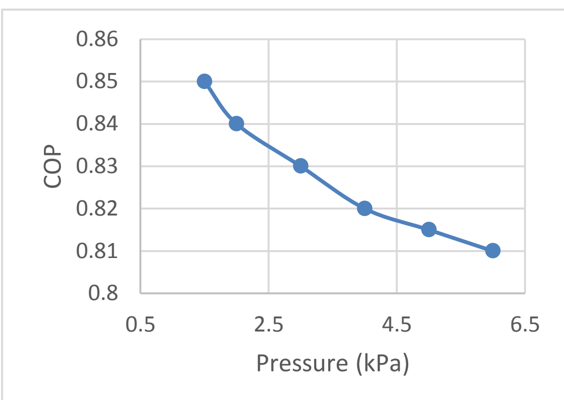

Figure 5: Value of pressure in evaporator

However, the required chilling water temperature must be kept in view, since at higher pressure, the saturation temperature of water increases. This condition makes the chilling ineffective compared to ambient temperature. Consequently, the atmospheric temperature is an important criterion for the selection of optimum pressures in the condenser as well as in the absorber. Similarly, condenser low pressure produces a high COP. The reduced saturation water temperature hampers the lowering of pressure in the condenser. This is due to cooling water availability from the ambient temperature. The driving effect of hot water is achieved specifically from flat solar panel mounted on house-tops or elevated planes, in tropical countries like Nigeria, where ambient temperature spans between 37 ^{0}C and 50 ^{0}C, in dry season.

4. CONCLUSION

Refrigeration systems that use environment friendly refrigerants, provide a sustainability advantage when compared to other refrigerant selections. To minimize hazardous environmental impacts associated with refrigeration system operation, it is reasonable to evaluate the prospects of a clean source of energy, and eco-friendly refrigerants. Hence, designing a H_2O-LiBr solar driven absorption refrigerating system under this study is unequivocally important. In this study the values of temperatures, pressures and concentration at different points, were estimated. The results indicate that a suitable solar vapour absorption refrigerating system can be designed keeping in view the climatic condition of Nigeria. Such modeling of thermal systems presents many advantages: viz., the elimination of the expense of building prototypes, the optimization of the system components, estimation of the amount of energy delivered from the system, and prediction of temperature variations of the system. The performance of the system as delineated, corresponds to parameters such as concentration, pressure and temperature; which helps in optimum selection of these parameters.

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