Simulation and performance of single solar Lithium-Bromide absorption system

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Abstract— **this paper present thermodynamic modelling and simulation of solar Lithium-bromide single stage cooling system using flat plate solar collector. The model based on masse and energy balance for each element of the cycle, in generator, absorber, condenser, evaporator, valve, solution pump and heat exchanger. A computer programme has been done in C language to predict of effect of heat source input temperature and cold source temperature on performance of the system. We found that the coefficient of performance COP and the efficiency of the system increased with temperature of evaporator and decreased with generator. The effect of efficiency of heat exchanger also found that has a considerable influence on performance of the system.**

Keywords— **absorption system, solar cooling, Lithium-Bromide solution, simulation, coefficient of performance.**

I. INTRODUCTION

The energy demand for refrigeration and air-conditioning to control temperature and humidity and for the provision of fresh air has increased continuously during the last decade and especially in Algeria it will still increase in future. This increase is caused amongst other reasons by increased thermal loads, occupant comfort demands, and architectural trends.

This has been responsible for the escalation of electricity demand and especially for the high peak loads (in summer) due to the use of electrically driven vapor compression machines. Moreover, the consumption of primary energy and the emissions of greenhouse gases associated with electricity generation from fossil fuels lead to considerable environmental consequences and monetary costs.

Conventional energy will not be enough to meet the continuously increasing need for energy in the future. In this case, renewable energy sources will become important.

An alternative solution for this problem is solar energy, available in most areas especially in Algeria [1] and representing a good source of thermal energy.

The absorption cooling cycle is usually a preferable alternative since it uses thermal energy collected from the sun without the need to convert this energy into mechanical energy as required by the vapour compression cycle. In addition the absorption cycle uses thermal energy at a lower temperature than that dictated by the vapour compression cycle.

Most of the absorption cooling cycles use either LiBr-H_2O or NH_3-H_2O solutions. In solar applications the LiBr-H_2O system is superior to the NH_3-H_2O system for several reasons. Among these reasons are that the LiBr- H_2O system is simpler in design and operation, and cheaper in cost as compared to the NH_3-H_2O system. Also, the LiBr-H_2O system can operate at a low generator temperature and this range of temperature the system has a COP (Coefficient of Performance) higher than that of the NH_3-H_2O system.

The LiBr-H2O system operates at a generator temperature in the range of 343 to 368 K with water used as a coolant in the absorber and condenser. The COP of the system is between 0.6 and 0.9 [2]

Considerable research experimental and simulation has been carried out to develop an efficient and an economic coupling between solar energy collection and the absorption unit LiBr – H2O.

A.Kececiler et al. [3] performed experiments on LiBr/ H2O system in lab conditions used low temperature geothermal energy as a powering source. They concluded that low-heat geothermal sources cannot be used efficiently in electricity generation, however could be used economically for refrigeration storing at 4-10ºC.

Da-Wen Sun [4] simulated LiBr /H2O absorption refrigeration systems. Detailed thermodynamic design data and optimum design maps were produced as a source of reference for developing new cycles and searching for new absorbent/refrigeration pairs.

Hammad and Audi [5] described the performance of a non-storage, continuous, solar operated absorption refrigeration cycle. The maximum ideal coefficient of performance of the system was determined to be equal to 1.6, while the peak actual coefficient of performance was determined to be equal to 0.55.

Ameel et al. [6] gives performance predictions of alternative low-cost absorbents for open cycle absorption using a number of absorbents. The most promising of the absorbents considered was a mixture of two elements, lithium chloride and zinc chloride. The estimated capacities per unit absorber area were 50–70% less than those of lithium bromide systems.

V Mittal et al [7] have simulated a solar-powered, single stage, absorption cooling system, using a flat plate collector and water–lithium bromide solution in india climate. They were found that the hot water inlet temperature affect in the surface area of some of the system components. Coefficient of performance (COP) is studied and it is found that high reference temperature increases the system COP and decreases the surface area of system.

In this work, thermodynamic modelling and simulation of solar Lithium-bromide single stage cooling system using flat plate solar collector. The model based on masse and energy balance for each element of the cycle. A computer programme has been done in C language to predict of effect of heat source input temperature and cold source temperature on performance of the system. We found that the coefficient of performance COP and the efficiency of the system increased with temperature of evaporator and decreased with generator. The effect of efficiency of heat exchanger also found that has a considerable influence on performance of the system.

II. Description of Solar Single Stage Absorption Cycle

A schematic representation of the solar single-stage absorption cycle is shown in figure1. The cycle consists of generator, absorber, a condenser, an evaporator, heat exchanger, circulating pumps and solar collector. The cycle working between two pressure levels: low pressure at the evaporator-absorber and high pressure at condenser-generator. The emitted vapours in the generator are constituted by pure water when in LiBr salt remain in the solution. Here, the solar energy is gained through the collector and is accumulated in the storage tank. Then, the hot water in the storage tank is supplied to the generator to boil off water vapour from a solution of Lithium Bromide and water. The water vapour is cooled down in the condenser and then passed to the evaporator where it again is evaporated at low pressure, thereby providing cooling to the required space.

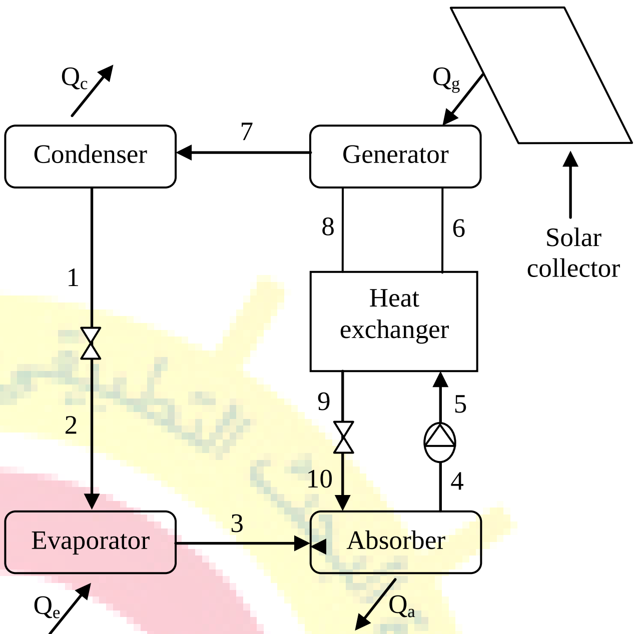

Fig.1 Schematic of the single-stage absorption cycle

III. Mathematical Modelling of Solar Single Stage Absorption Cycle

In order to analyse the cycle in figure1 thermodynamic properties of the internal state points need to be determined first. As it can be difficult to evaluate those state points, the following simplifying assumptions are commonly made:

- The pressure drops and heat losses in and across the chiller components are considered negligible.
- The liquid refrigerant leaving the condenser (state point 8) and the vapour leaving the evaporator (state point 10) are assumed to be saturated at their respective saturation temperatures.
- The strong solution leaving the absorber (state point 4) and the weak solution leaving the generator (state point 8) are assumed to be saturated and at equilibrium conditions at their respective temperatures. With this assumption, enthalpies at these state points can easily be obtained from correlation for the thermodynamic data of the H_2O/ LiBr absorption working pair in ASHARE[8].
- The refrigerant vapour leaving the generator (state pint 7) is assumed superheated at the generator temperature.
- Flow restrictors (i.e. the refrigerant and solution expansion valves) and the pump are considered adiabatic and isentropic.
- All liquid in the evaporator is evaporated and only vapour goes to the absorber.

A. Thermodynamic proprieties

The equilibrium pressure of the vapour phase or liquid P_s (kPa) is determined according to the water saturation temperature T_s (°C) by using the equation proposed by Patek [9]:

$$P_{eq} = P_c \exp\left(\frac{T_c}{T_k}\sum\alpha\right) \quad (1)$$

Were:

$$\alpha = \frac{T_c}{T_k}\begin{pmatrix} -7.85823T_0 + 1.8399T_0^{1.5} - 11.781T_0^3 + \\ 22.6705T_0^{3.5} - 15.9393T_0^4 + 1.77516T_0^{7.5} \end{pmatrix} \quad (2)$$

$T_c = 647.14°C$; $T_k = T + 273.15°C$; $P_c = 22064 Kpa$

$$T_0 = 1 - \frac{T_k}{T_c} \quad (3)$$

The enthalpy of saturated liquid and saturated vapour water are given by equations 4 and 5 successively [8]:

$$h_{liq} = Cp.T \quad (4)$$

$$h_{vap} = -125397.10^{-8}T^2 + 1.88060937.T + 2500.559 \quad (5)$$

The enthalpy of superheated vapor in point state 7 is given by [8] as:

$$h_{sv}(T1, T2) = 1.925.T1 - 0.125T2 + 2365 \quad (6)$$

Were : *T1* temperature of superheated vapour at pressure equal to the pressure of saturation at *T2*.

In the case of the solution, the equilibrium pressure of LiBr/H2O mixture is given by [8]:

$$Log\ P = C + \frac{D}{T'} + \frac{E}{T'^2} \quad (7)$$

T' is refrigerant temperature (K)

$$T' = T_r + 273.15$$

$$T_r = \frac{-2E}{D + \left[D^2 - 4E(C - Log\ P)\right]^2} - 273.15\ (°C) \quad (8)$$

The temperature of LiBr/H2O solution is given as follow:

$$T = \sum_0^3 B_n X^n + T_r \sum_0^3 A_n X^n \quad (°C) \quad (9)$$

Were:

C=7.05 ; D=-1596.49; E=-104095.5

$A_0 = -2.00755$ | $B_0 = 124.937$
$A_1 = 0.16976$ | $B_1 = -7.71649$
$A_2 = -3.133362E-03$ | $B_2 = 0.152286$
$A_3 = 1.97668E-05$ | $B_3 = -7.95090E-04$

And , the specific enthalpy *h (kJ/kg)* can be calculated according to the solution temperature T and the mass fraction X of the solution by [8]:

$$h = \sum_0^4 A_n X^n + T\sum_0^4 B_n X^n + T^2\sum_0^4 C_n X^n \quad (10)$$

Were the constants A_n , B_n and C_n are mentioned in ASHARE[8].

B. Modelling

The simulation of the system is based on mass and energy balances for each element of the cycle, supposed in permanent regime.

1) Generator

- The masse balance:

$$\dot{m}_s = \dot{m} + \dot{m}_w \quad (11)$$

$$\dot{m}_s = \dot{m}_6,\ \dot{m} = \dot{m}_7,\ \dot{m}_w = \dot{m}_8$$

$$\dot{m}_s X_s = \dot{m} + \dot{m}_w X_w \quad (12)$$

Were $\dot{m}$ is masse flow rate of pure water vapour at the outlet of generator and $\dot{m}_s$, $\dot{m}_w$ are masse flow rates of solution at inlet and outlet of generator respectively.

X the concentration of the aqueous LiBr solution with the subscripts, *s* and *w* representing the strong and weak solution (strong implies rich in refrigerant and week poor in refrigerant).

- The energy balance

$$Q_g = \dot{m} h_7 + \dot{m}_w h_8 - \dot{m}_s h_6 \quad (13)$$

$$Q_g = \dot{m} h_7 + (1-\lambda)\dot{m} h_8 - \lambda \dot{m} h_6 \quad (14)$$

Q_g : Heat input to the generator at temperature Tg

h7: specific enthalpy of superheated vapour at temperature Tg and high pressure (P_h).

h8 and h7: specific enthalpies of liquid solution at temperature T6 and T8 respectively and high pressure(P_h).

λ is the circulation ration, that describes the rate between the mass flow rate of solution delivered by the pump and water vapour desorbed in generator as follow :

$$\lambda = \frac{\dot{m}_s}{\dot{m}} \tag{15}$$

And it can be defined with function of concentration of rich solution *Xs* and poor solution *Xw* by:

$$\lambda = \frac{X_w}{X_w - X_s} \tag{16}$$

2) *Condenser*

$$\dot{m}_7 = \dot{m}_1 = \dot{m} \tag{17}$$

$$Q_c = \dot{m}(h_1 - h_7) \tag{18}$$

Were:

Q_c Heat rejected by condenser

h_1 specific enthalpy of saturated liquid refrigerant at temperature *Tc* and high pressure (P_h).

3) *Evaporator*

$$\dot{m}_3 = \dot{m}_2 = \dot{m}_1 = \dot{m}_7 = \dot{m} \tag{19}$$

$$Q_e = \dot{m}(h_3 - h_2) \tag{20}$$

Q_e Heat absorbed by evaporator

4) *Absorber*

$$\dot{m}_s = \dot{m} + \dot{m}_w \tag{21}$$

$$\dot{m}_s = \dot{m}_4 = \dot{m}_6\ ;\ \dot{m} = \dot{m}_7 = \dot{m}_1 = \dot{m}_3 = \dot{m}_2\ ;\ \dot{m}_w = \dot{m}_{10}$$

$$\dot{m}_s X_s = \dot{m} + \dot{m}_w X_w \tag{22}$$

$$Q_a = \dot{m} h_3 + \dot{m}_w h_{10} - \dot{m}_s h_4 \tag{23}$$

$$Q_a = \dot{m} h_3 + (1-\lambda)\dot{m} h_{10} - \lambda \dot{m} h_4 \tag{24}$$

5) *Solution pump*

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_6 = \dot{m}_s \tag{25}$$

$$W_p = \dot{m}_s (h_5 - h_4) \tag{26}$$

6) *Heat exchanger*

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_s \tag{27}$$

$$\dot{m}_8 = \dot{m}_9 = \dot{m}_w \tag{28}$$

$$T_9 = T_5 \varepsilon + T_8 (1-\varepsilon) \tag{29}$$

ε Describes the efficiency of the heat exchanger

$$h_6 = h_5 + \frac{\dot{m}_8}{\dot{m}_5}(h_8 - h_9) \tag{30}$$

Were :

$$\dot{m}_8 = \dot{m}\frac{X_s}{X_s - X_w} \tag{31}$$

$$\dot{m}_5 = \dot{m}\frac{X_w}{X_w - X_s} \tag{32}$$

7) *performance of the cycle*

Coefficient of performance of absorption cycle defined as a ratio between heat absorbed in evaporator (refrigeration effect) and heat supplied in generator and work of solution pump which can be written by:

$$Cop = \frac{Q_e}{Q_g + W_p} \tag{33}$$

The efficiency of the cycle is the ratio of real coefficient of performance and the maximal coefficient of performance (Carnot)

$$\eta = \frac{Cop}{COP_c} \tag{34}$$

$$COP_c = \left(\frac{T_g - T_a}{T_g}\right)\left(\frac{T_e}{T_c - T_e}\right) \tag{35}$$

IV. Results and Discussion

The temperatures of heating source (generator) and cooling source (evaporator) have considerable influence on the performance of the cycle. This is illustrated in figure 2 and 3 witch shows the variation of performance of the system with temperature of heating source Tg and cooling source Te.

In presenting the results, one temperature is changed while the others are held constant. Figure 2 and 3 show the variation of Cop with generator temperature, the temperature of condenser and absorber are fixed at 30°C. It can be seen that the COP decreases with increase of generator temperature. in isothermal Te=2 °C, the COP varies from 0.865 at Tg=65°C to 0.822 at Tg =95°C, so it decreases by 4 % in the range of temperature from 65 to 95°C(figure2) , this is caused by increasing the enthalpy of superheated vapour while the cooling load is constant. In other hand, the Cop increases with

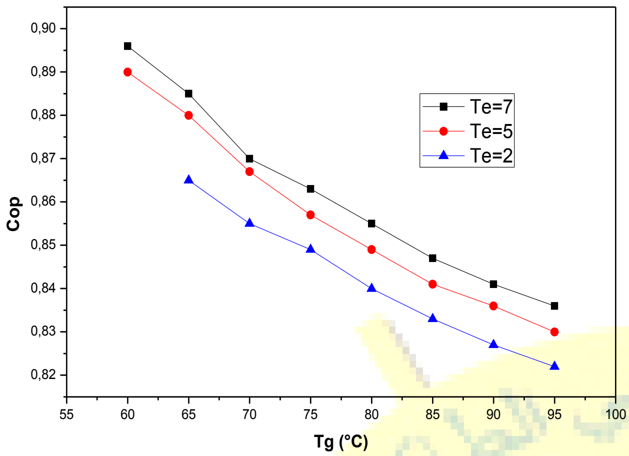

Fig. 2 Variation of the Cop of the system with temperature of generator Tg (Ta and Tc fixed at 30 °C)

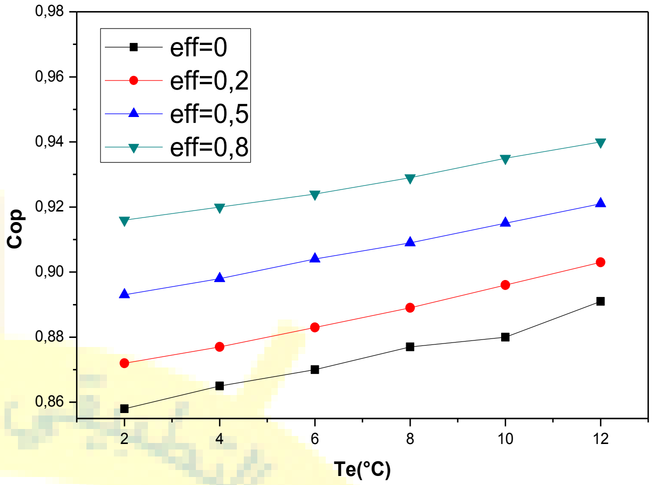

Fig. 4 Effect of efficiency of heat exchanger on coefficient of performance of the system (Tg=70, Tc= 30).

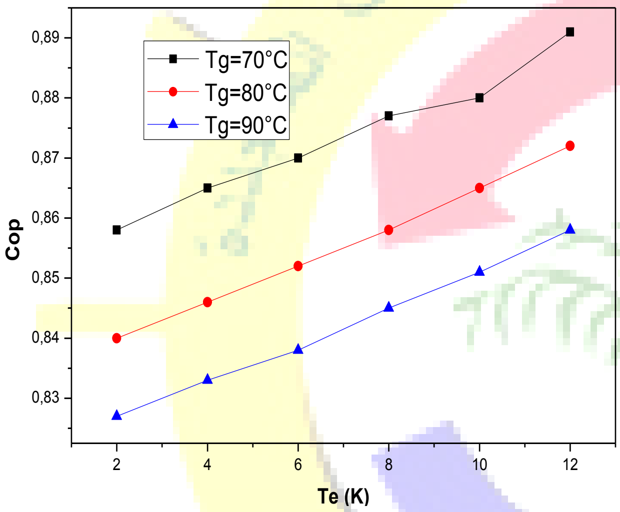

Fig. 3 Variation of the Cop of the system with temperature of evaporator Te (Ta and Tc fixed at 30 °C)

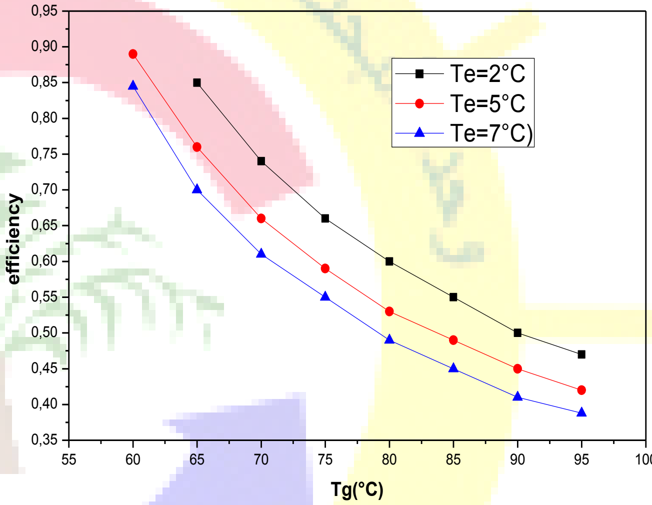

Fig .5 Effect of evaporator temperature on the efficiency of the system (Tc=Ta=30°C).

increasing of cooling source temperature this is showed in figure 3. In isothermal Tg=90The COP increases from 0.825 at Te=2 to 0.857 at Te =12°C, so the COP increases by 3.7%.

Figure 4 shows the effect of effectiveness of solution heat exchanger on the coefficient of performance COP. The generator temperature fixed at Tg =70°C and evaporator temperature at Ta=30°C. It can be seen that effectiveness of heat exchanger has important influence on performance of the system. The COP increases with increase of effectiveness, it rises by 5.8% from 0.885 to 0.94 at temperature Te= 12 °C. This is caused by heat exchanged between the strong solution and week solution.

Figure 5 illustrates the efficiency of the system with variation of generator temperature. In this case the temperature of condenser and absorber are fixed 30°C.the efficiency of the system decreases with increasing of generator temperature, by contrast, increases with increasing of evaporator temperature, it rises by 5 % from Te =2°C to Te=7°C.

The effect of evaporator temperature on circulation ratio is presented in figure 6. The temperature of absorber and condenser are fixed at 30 °C while Te varies from 2 to 12 °C. The temperature of evaporator effects negatively on circulation ration λ this is caused by the decreasing of the low pressure with increasing of temperature. When pressure decreases the strong solution and increases the week solution.

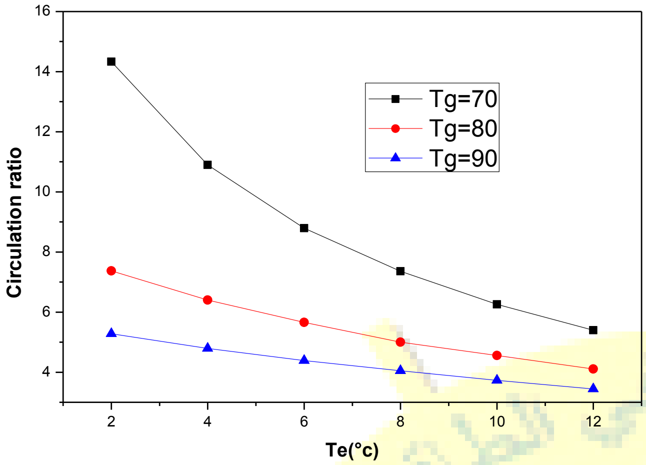

Fig. 6 Effect of generator temperature on the circulation ration λ (Tc=Ta=30°C).

V. Conclusion

In this work, thermodynamic model and has been done to simulate a solar Lithium-bromide single stage cooling system using flat plate solar collector. The model based on masse and energy balance for each element of the cycle. A computer programme has been done in C language to predict of effect of heat source input temperature and cold source temperature on performance of the system.

From the above study the following results can be drawn.

- The increase of temperatures of heating source (generator) decreases the COP of the system.
- The increase of cooling temperature source increases the COP and efficiency of the system.
- The increase of efficiency of solution heat exchanger has a considerable influence on the performance of the system.

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