

## APPOINTING THE PROPER OPERATING VARIABLES FOR EFFICIENT AIR-COOLED LiBr-WATER ABSORPTION REFRIGERATOR

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### 1- Abstract

The aim of the present study is to perform analytical simulation for the single-effect LiBr-water absorption refrigerating system (ARS) in order to scan for all possibilities of operating the cycle among most available operating variables to obtain the best performance and determine what are the proper parameter needed to be changed so that the refrigerator can operate using ambient air instead of water and cooling tower accessories to dissipate the heat at the condenser and absorber. The COPs of the cycle was obtained as a function of the different temperature of the cycle and solution concentration. The performance characteristics of the cycle were examined by changing the temperature of the heating source supplied at the generator, the temperatures at the condenser, absorber, and evaporator. It can be concluded that the concept of air cooled absorption chiller is feasible with coefficient of performance of 0.43 to 0.79 depending on the evaporator temperature for any cooling capacity in some applications, where the chilled water supply temperature is not necessary to be too cold,

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### الخلاصة

هدف الدراسة الحالية هو إجراء محاكاة تحليلية لدورة التبريد الامتصاصية ماء-بروميد الليثيوم أحادية التأثير لعمل مسح لكل إمكانيات تشغيل الدورة بعوامل التشغيل المتاحة و الأكثر ملائمة للحصول على افضل أداء وتقرير أي من هذه العوامل بحاجة إلى التغيير لتشغيل المنظومة بالاعتماد على الهواء الجوي بدلاً من الماء وملحقات أبراج التبريد لتبديد حرارة المكثف ووعاء الامتصاص. ظهر أن معامل أداء الدورة هو دالة لدرجات الحرارة المختلفة و لتركيز المحلول. تم اختبار خصائص أداء الدورة بتغيير درجة حرارة مصدر الطاقة المجهزة ودرجات الحرارة في المكثف ووعاء الامتصاص والمبخر. يمكن الاستنتاج من هذه الدراسة إن مبدأ تشغيل المنظومة الامتصاصية ماء-بروميد الليثيوم التي تبرد بالهواء ممكنًا وعلينا بمعامل أداء من 0.43 إلى 0.79 اعتمادًا على درجة حرارة التبخير لأي سعة تبريد لبعض التطبيقات عندما لا تكون هناك حاجة للماء المثلج بدرجات حرارة منخفضة جدًا.

### Nomenclature and Abbreviations

$i$	Specific Enthalpy	(kJ/kg)
$\dot{m}$	Mass Flow Rate	(kg/s)
$p$	Pressure	(kPa)
$\dot{Q}$	Heat Energy Rate	(kW)
$T$	Temperature	( $^{\circ}\text{C}$ )
$\epsilon$	Heat Exchanger Effectiveness	
$\xi$	LiBr Concentration	

### Subscripts

1,2,3, ...,10 State Points on Fig.1

$a$	Absorber
$c$	Condenser
$e$	Evaporator
$g$	Generator
$r$	Refrigerant
$ref.$	Reference
$s$	Strong Solution
$sat.$	Saturated condition
$w$	Weak Solution

### Abbreviations

ARS	Absorption Refrigeration System
COP	Coefficient of Performance
C.R.	Circulation Ratio
LiBr	Lithium Bromide

### 3- Introduction

The waste heat out of industrial operating processes can be converted to a useful refrigeration by using a heat operated refrigeration system, such as an absorption refrigeration cycle [1]

The early development of an absorption cycle dates back to the 1700's. It was known that ice could be produced by an evaporation of pure water from a vessel contained within an evacuated container in the presence of sulfuric acid [2 and 3]. The major problems of this system were corrosion and leakage of air into the vacuum vessel. In the 1950's, a system using lithium bromide/water as the working fluid was introduced for industrial applications. A few years later, a double-effect absorption system was introduced and has been used as an industrial standard for a high performance heat-operated refrigeration cycle.

American companies manufactured 100% of LiBr-Water absorption chillers worldwide [3], using the standard single-effect absorption cycle. In solar applications [4] the LiBr-water system is superior to the  $\text{NH}_3$ -water system for several reasons, among these, the LiBr-water system is simpler in design and operation, and can operate at low generator temperature at higher coefficient of performance. The goal of FREDRIC et al [5] in his work were to develop an absorption chiller 60% Lithium bromide solution in water driven by district heating water with a temperature down to 70  $^{\circ}\text{C}$ . The unit has a cooling capacity of more

than 1.15 MW, the COP however is only in the order of 0.6.

G. Grosman *etal.*[6], Carried out a performance simulation using a modular computing code. They compare the performance of a single effect LiBr-water ARS with triple effect systems as well as with the ideal Carnot cycle operating under the same condition *fig.*(1). They showed that COP of the single cycle did not exceed 0.8 (dashed curves), while, for triple effect cycle it closes to 1.27 for series flow (solid curves) and 1.73 for the parallel flow (dash-dot curves).

Bounaza and Mujtaba [7], carried out a simulation analysis on a LiBr-water ARS and concluded that the COP is very sensitive to the generator temperature for any evaporator temperature and varies linearly with  $T_g$  before reaching a maximal value ranging between 0.75 and 0.8 *fig.*(2).

Results of *fig.*(1) and *fig.*(2) will be used for comparison with the present work.

### 3-1 Single-effect absorption system

A single-effect absorption refrigeration system is the simplest and most commonly used design. There are two design configurations depending on the working fluids used; Ammonia-Water and Lithium bromide-Water systems. *Fig.*(2-a) shows the diagram of single-effect LiBr-water absorption cycle, where LiBr is absorbent (non-volatility absorbent), and water is

refrigerant. This mixture has been widely used; however, it contains some disadvantages. Since water is used as refrigerant, it is not possible to provide refrigeration at sub-zero temperatures. In the generator, solution is heated to vaporize the refrigerant (water), and separate from the solution, the vapor of refrigerant is condensed at the condenser delivering heat to a heat sink (cooling water/air), the condense liquid flows through an expansion valve to the evaporator where refrigerant is evaporated producing cooling effect. It should be noted from the property charts that the entire water-lithium bromide system operates under vacuum. The refrigerant vapor then goes to the absorber where is absorbed by high concentration solution coming from the generator and reject heat to a heat sink. Finally, the low concentration solution is pumped through heat exchanger to the generator.

A solution heat exchange is introduced as show in *Fig.*(2-a). The heat exchanger allows the solution from the absorber to be preheated before entering the generator by using the heat from the hot solution leaving the generator. This exchanger raises the cycles efficiency depending on its effectiveness as the heat input at the generator is reduced. Moreover, the size of the absorber can be reduced as less heat is rejected [7].

### 3-2 Crystallization

The pressure-temperature-mass fraction chart for LiBr-Water mixture (Fig.2-b), show lines marked as crystallization in the lower right section. The region to the right and below these crystallization lines indicates solidification of LiBr salt. In the crystallization region a two-phase mixture (slush) of water-lithium bromide solution and crystals of pure LiBr exist in equilibrium. The water-lithium bromide system should operate away from the crystallization region as the formation of solid crystals can block the pipes and valves. Crystallization can occur when the hot solution rich in LiBr salt is cooled (as within the process when the hot concentrated solution leaving the generator at point 4 (see Fig.(2-a) and Fig.(2-b), flowing toward the absorber and cooled via the heat exchanger and then passes through an expansion valve after which the solution is further cooled due to expansion). To avoid crystallization, the condenser pressure reduction below a certain value due to say, low cooling water temperature in the condenser should be avoided for a given generator temperature. Hence in commercial systems, the condenser pressure is artificially maintained high even though the temperature of the available heat sink is low. This actually reduces the performance of the system, but is necessary for proper operation of the system [8 and 9].

Air-cooled LiBr-water absorption chillers/coolers have been analyzed, designed, and prototype-tested, since at least the mid-1970s [10], primarily in Japan, the U.S., and Europe, for solar- and direct-fired applications. Today, only one air-cooled LiBr chiller is on the market (the Yazaki ACH-8) [11]. There is relatively little evidence of air-cooled LiBr absorption development efforts specifically targeting cooling, heating and power integrating (CHP) applications in light-commercial buildings. In the CHP application, chiller/cooler efficiency is less important relative to direct-fired applications. The big difference between water-cooled and air-cooled LiBr-water absorption chillers is the temperature of the absorber. With air-cooled, one cannot achieve a temperature of the solution in the absorber sufficiently low enough to maintain the evaporator pressure. The only way to compensate for the high absorber temperature is to increase the concentration of LiBr in the solution, but that brings it closer to crystallization. So, as characterized by previous investigators such as Foley, et al [12] and Kurosawa, et al [13]. The key barrier to air cooling of LiBr chillers in U.S. climates is crystallization of LiBr in the absorber

### 4- Steady Flow Analysis of Water-Lithium Bromide Systems

Figure (2-a) shows the schematic of the system indicating various state points. The corresponding points are also shown on the Pressure-Temperature-Concentration diagram Fig.(2-b). The steady flow analysis is carried out by applying mass and energy balance across evaporator, generator and heat exchanger, with the following assumptions:

1. The solution leaving the absorber (point 1) is saturated.
2. The strong solution and the water vapor, leave the generator are at the generator temperature  $T_g$ .
3. The enthalpy change across the pump is very small and can be omitted.
4. Perfectly insulated components.
5. No superheating in the evaporator, nor subcooling in the condenser.
6. Isenthalpic expansions.
- 7- the pumping capacity obeys the required cooling capacity and is a function of the heat supplied to the generator.

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_w \quad (1)$$

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_6 = \dot{m}_s \quad (2)$$

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_9 = \dot{m}_{10} = \dot{m}_r \quad (3)$$

Defining the ratio of pumping capacity (weak solution ( $\dot{m}_w$ ) kg/s) to the liberated vapor at the generator (refrigerant mass flow rate ( $\dot{m}_r$ ) kg/s) as circulation ratio (C.R)

$$C.R = \frac{\dot{m}_w}{\dot{m}_r} \quad (4)$$

#### 4-1 The evaporator :

The mass flow rate of the refrigerant at the evaporator for a given cooling capacity is:

$$\dot{m}_r = \frac{Q_e}{(h_{10} - h_9)} \quad (5)$$

where  $h_9$  and  $h_{10}$  (kJ/kg) are the enthalpies of saturated liquid water at condenser pressure and saturated water vapor at evaporator pressure respectively.

The dominated low pressure at the evaporator  $P_e$  (kPa) is the saturated pressure at evaporator temperature  $T_e$  ( $^{\circ}C$ ) and it is the same pressure at the absorber  $P_a$ ;

$$P_e = P_{sat}(T_e) = P_a \quad (6)$$

The enthalpy of pure water vapor and liquid at different temperatures and pressures can be obtained from pure water property data. For all practical purposes, liquid water enthalpy,  $h_{liquid}$  at any temperature  $T$  ( $^{\circ}C$ ) can be obtained from the equation [14 ].

$$h_{liquid} = 4.19(T - T_{ref}) \quad \text{kJ/kg} \quad (7)$$

$$h_{vapor} = 2501 + 1.88(T - T_{ref}) \quad \text{kJ/kg} \quad (8)$$

where  $T_{ref}$  is the reference temperature,  $0^{\circ}C$ .

#### 4-2 The generator:

$$\dot{m}_w = \dot{m}_s + \dot{m}_r \quad (9)$$

LiBr: mass balance through the generator is:

$$\xi_w \dot{m}_w = \xi_s \dot{m}_s \quad (10)$$

So,

$$\frac{\dot{m}_w}{\dot{m}_r} = \frac{\xi_s}{\xi_s - \xi_w} = C.R \quad (11)$$

The strong solution concentration  $\xi_s$  and the weak solution concentration  $\xi_w$  are needed, and can be calculated using an equation that related the solution concentration and solution temperature at required water vapor temperature as sited in ASHARE [15].

The heat input to the generator is given by:

$$\frac{Q_{gen}}{\dot{m}_r} = C.R(h_4 - h_3) - (h_4 - h_7) \quad (12)$$

The coefficient of performance Of the refrigeration cycle becomes:

$$COP = \frac{(h_{10} - h_9)}{C.R(h_4 - h_3) - (h_4 - h_7)} \quad (13)$$

#### 4-3 The solution heat exchanger:

Heat exchanger effectiveness is defined as the ratio of the actual heat transfer to the maximum possible heat transfer for given inlet conditions, and as LiBr-water solution specific heat reduced with increasing LiBr concentration [16,17], therefore the hot

concentrated stream (flow line from point 4 to 5), which has lower mass flow rate than the colder stream, represents the minimum heat capacity stream, accordingly the heat exchanger effectiveness can be written as :

$$\varepsilon = \frac{h_4 - h_5}{h_4 - h_2} \quad (14)$$

The heat transfer rate in the solution heat exchanger is given by;

$$\dot{m}_s(h_4 - h_5) = \dot{m}_w(h_3 - h_2) \quad (15)$$

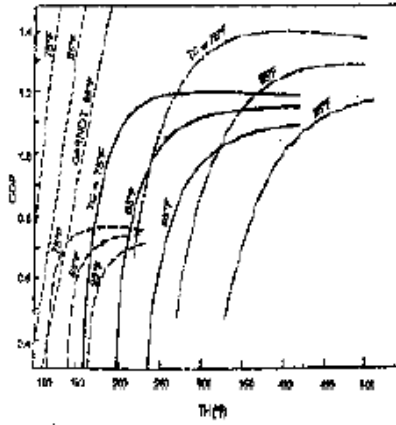
Joining equations 10, 14, and 15 the enthalpy of the weak solution entering to the generator  $h_3$  (kJ/kg) can be calculated as:

$$h_3 = h_2 + \varepsilon(h_4 - h_2) \left( \frac{\xi_w}{\xi_s} \right) \quad (16)$$

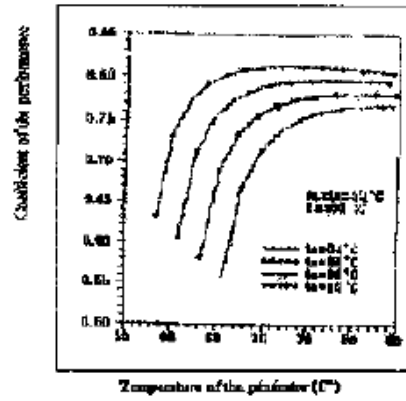
where, the specific enthalpy of the solution  $h_2$  and  $h_4$  (kJ/kg) can be calculated according to the solution temperature  $T$  ( $^{\circ}C$ ) and the mass fraction  $\xi$  of the solution LiBr-water [ 15 ]

$$h_2 = h_{sat}(T_s, \xi_w) \quad (17)$$

$$h_4 = h_{sat}(T_g, \xi_s) \quad (18)$$



(1-a) [6]



(1-b) [7]

Fig.(1); LiBr-Water absorption, refrigeration cycle performance

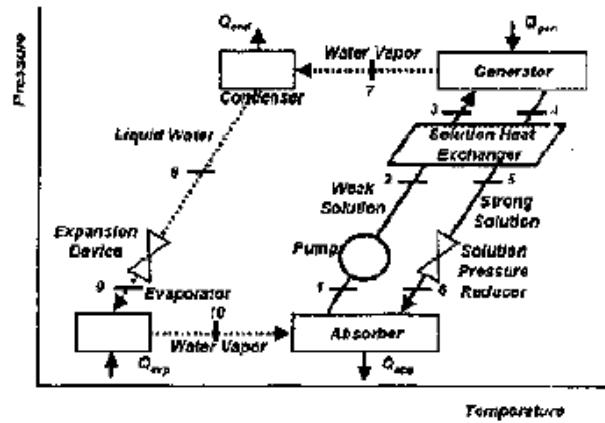


Fig.(2-a) single-effect LiBr-water absorption refrigeration system with a solution heat exchanger. [10]

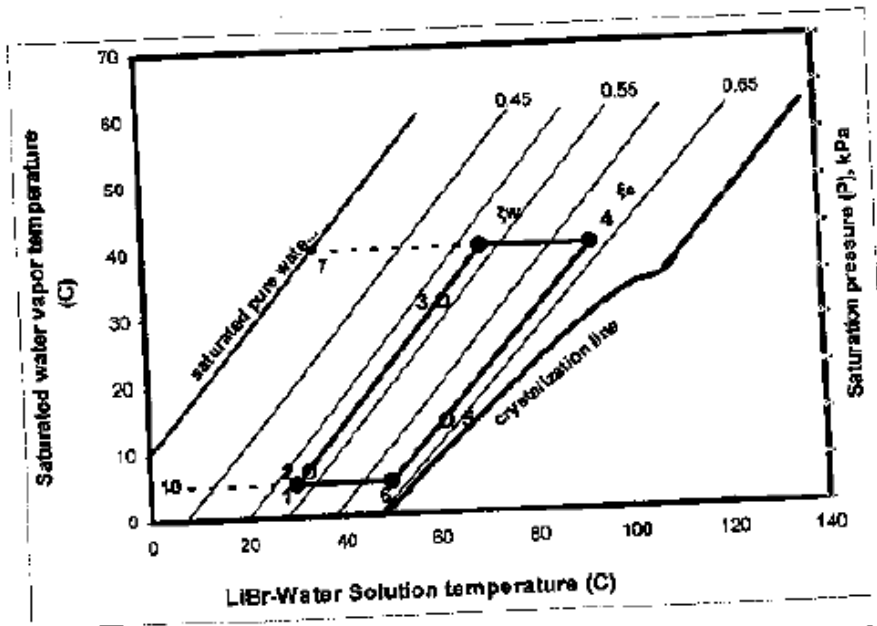


Fig.(2-b) Representation of the LiBr-Water single effect absorption cycle on P-T- $\xi$  chart

### 5- Result And Discussion

Four external operating temperatures affect the COP and capacity of an absorption chiller; these are generator, condenser, absorber and the required evaporator temperatures.

If the same cooling fluid is used to cool both the absorber and the condenser, either in series (as common in water cooled chillers) or in parallel, this is reduced to three different temperatures. In the most common plot of COP for this kind of systems the COP value is plotted against the generator temperature while the absorber and the condenser, as well as the evaporator temperature are held fixed. Assuming

$c=0.6$ , for known cooling capacity and given operating temperatures, the following parameters can be sequentially calculated:  $h_7$  and  $h_{10}$  from equation (8) at  $T_g$  and  $T_e$  respectively,  $h_9$  from equation (7) at  $T_c$ ,  $\xi_w = \xi_w(T_a, T_e)$  and  $\xi_s = \xi_s(T_g, T_e)$  from the correlations given in [15],  $m$ , and CR from equations (5) and (11) respectively,  $h_2 = h_{sat}(T_a, \xi_w)$ ,  $h_4 = h_{sat}(T_g, \xi_s)$  from the correlations given in [15],  $h_3$  from equation (16),  $Q_{gen}$  and COP from equations (12) and (13) respectively.

Fig.3 shows the variation of COP with generator temperature  $T_g$  for one kW cooling capacity at an evaporator temperature  $T_e$  of 3 °C of a single effect



LiBr-water absorption chiller at different condensing temperature  $T_c$ , maintaining absorber temperature  $T_a$  constant at  $30\text{ }^\circ\text{C}$ . It is evident, for any  $T_c$ , that the COP increases with generator temperature because higher  $T_g$  helps in liberation more amount of refrigerant. Comparing the result of fig.(3) (also of the going on figures; fig.(6) and fig.(7)), with the results of [6]; fig.(1-a) and [7]; fig(1-b), it is evident that the results exhibit the same typical, qualitative behavior, with the COP increasing sharply from zero at some minimum generator temperature then leveling off to some constant value at a higher  $T_g$  and even decreasing slightly with further increasing in  $T_g$ .

In fig.(3), COP continue to increase with  $T_g$  up to a point where  $\xi_s$  value about 0.63 for any set of operating conditions. A further increase of  $T_g$ , the cycle COP tends to level rather than continue to increase. That is because, at higher solution concentration more heat is required to liberate the same amount of refrigerant and the ratio of vapor generation to the rate of heat input remains just about the same. Fortunately this help to keep the cycle away from crystallization region, also one can appoint out the proper operating generator temperatures for any applied  $T_c$ , for example, at  $\xi_s = 0.63$  and

$T_c=45\text{ }^\circ\text{C}$  the proper  $T_g$  is  $98\text{ }^\circ\text{C}$  at which  $\text{COP}=0.77$  rather than maximum generator temperature of  $103\text{ }^\circ\text{C}$  which have the same value of COP. Also it is apparent that COP decreases with the condensing temperature because of the higher pressure which correspond to the condensing temperature. Appointing the minimum value of  $T_g$  in Fig(3) is clearly visible, below which no evaporation takes place and so no cooling capacity can be produced for each  $T_c$ , therefore COP is vanished for example at  $T_c=45\text{ }^\circ\text{C}$  with  $\xi_s=0.537$ , no evaporation occurs below  $78\text{ }^\circ\text{C}$  generator temperature.

In fig.(4), It can be seen that, at constant  $T_a$  and  $T_c$  (constant  $\xi_w$ ), the higher condensing temperature requires higher maximum generator temperature to maintain the same rate of evaporation,  $\xi_s$  remains constant ( $\xi_s=0.65$ ) (dashed-curves). In this case, so far  $T_g$  is increasing, COP still reducing because of the awful effect of increasing  $T_c$ , else the mass fraction  $\xi_s$  must be decrease as  $T_c$  increases (solid curves), this consequent, causes again a considerable drop of COP up to the end point where  $\Delta\xi=0$ , then the temperature in the generator would become lower than the saturation temperature of the strong solution

and if this happens no refrigerant can be distilled.

To insure the cycle operation away from crystallization region, there is also a maximum generator temperature that can be appointed in (Fig. 3) for each condensing temperature., for example at condensing temperature 45 °C, the maximum generator temperature should be not exceeds 103. °C . It seems for all the sets of operating conditions, there is no need to increase  $\xi_g$  more than 0.63 after which there will be no significant increase in COP.

At the low pressure side of the cycle and constant  $T_g$  and  $T_c$  (constant  $\xi_g$ ), a higher absorber temperature (Fig.5) requires a higher evaporation temperature to maintain the same rate of evaporation,  $\xi_w$  remains constant ( $\xi_w=0.537$ ) (dashed curves), keeping relatively good operating performance (COP), (however at higher  $T_e$ , above 15 °C, there will be a limited practical application of such system), else the mass fraction of LiBr in the absorber ( $\xi_w$ ) increases as  $T_a$  increases (solid curves), this consequent resulting in rapid decrease of COP which characterize the bad effect of increasing  $T_a$  on the system performance, that is because the consequence increase in  $T_a$  will increase  $\xi_w$

up to the end point where  $\Delta\xi = 0$  and no refrigerant can be distilled.

As in common water cooled absorption chillers, the same cooling fluid is used to cool both the absorber and the condenser in series, that's means same cooling temperature medium are used. Fig.(6) represents the present cycle analysis with unique temperature for both the condenser and absorber, applying the air as the cooling medium, at  $T_e=10$  °C (assuming air temperature 5 degrees above the condenser-absorber temperature). It can be seen that at higher condenser-absorber temperature there will be a need for higher generator temperature to start producing cooling with lower coefficient of performance, and any reduction in evaporator temperature will bring the cycle to a further reduction in COP, Fig.(7) shows that a reducing in  $T_e$  causes a drastically drop in COP and it is evident that at higher temperature climate, even producing cooling is not visible below 5 °C were  $T_e=T_a=50$  °C but COP equal to 0.67 and 0.79 at  $T_e=9$  °C and 20 °C respectively. Cooling can not be produced below  $T_e=9$  °C at  $T_c=T_a=55$  °C but COP equal to 0.43 and 0.76 at  $T_c=10$  °C and 20 °C respectively.

The mass fraction in the generator  $\xi_g$  and in the absorber  $\xi_w$  can be decreased and

increased only until they are equal (at which state the system stops working,  $\Delta\xi = \xi_s - \xi_w = 0$ ). This means that for every combination of condenser-absorber-evaporator temperatures there is a minimum value for the generator temperature (and for every combination of generator-condenser-evaporator temperatures there is a maximum

absorber temperature. In some applications, where the chilled water supply temperature is not necessary to be too cold, the concept of air cooled absorption chiller is feasible for any cooling capacity such that as shown in Fig.8, the cooling capacity is a function of heat supplied to the generator in connection with the pumping rate of solution.

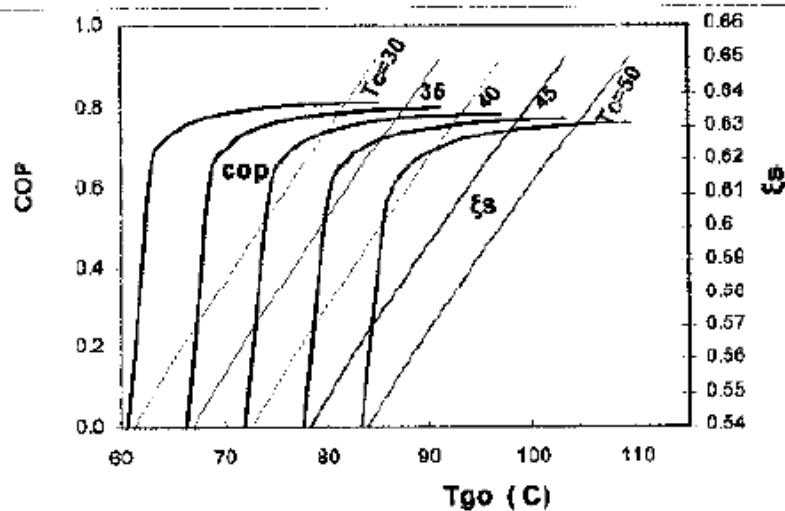
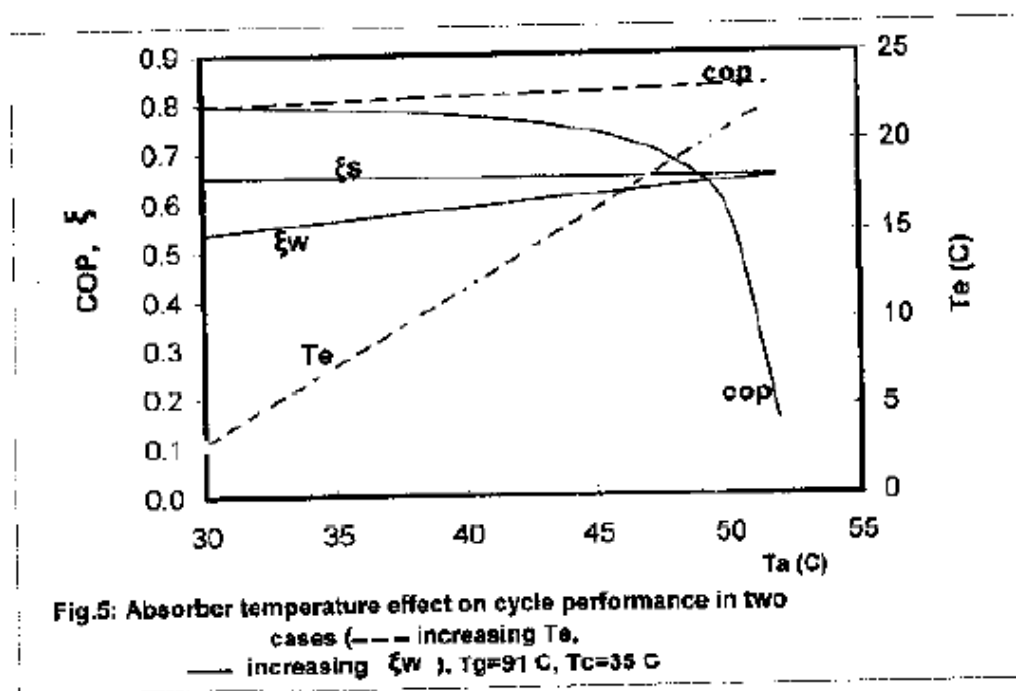
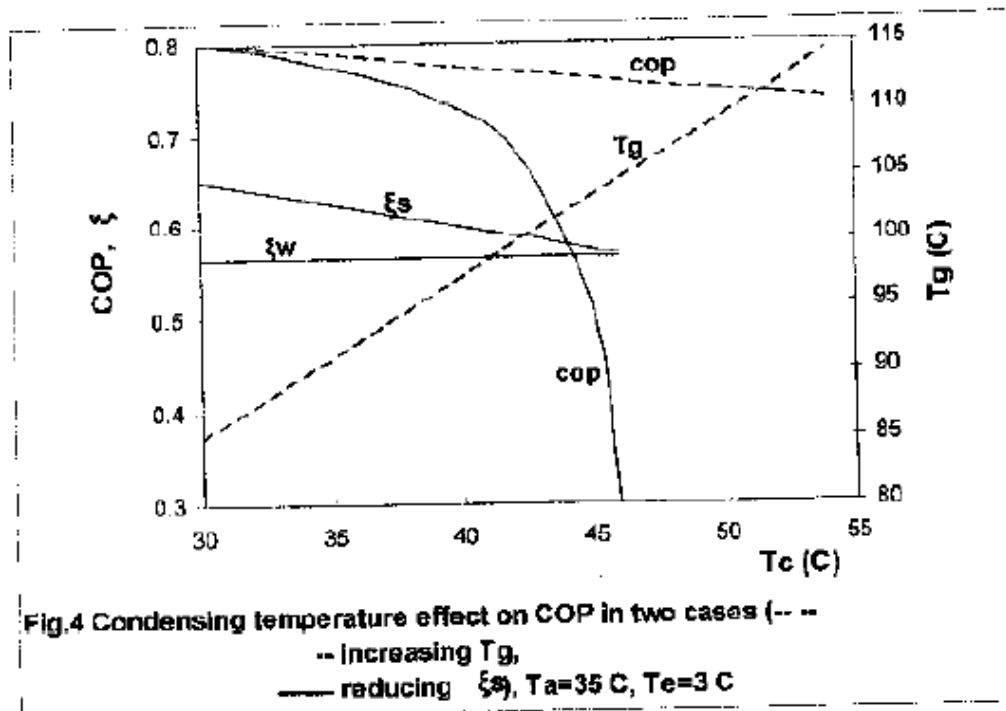


Fig. 3: COP variation with generator temperature ( $T_{go}$ ) at different  $T_c$ , ( $T_e=3$  C,  $T_a=30$  C,  $Q_e=1$  kW)



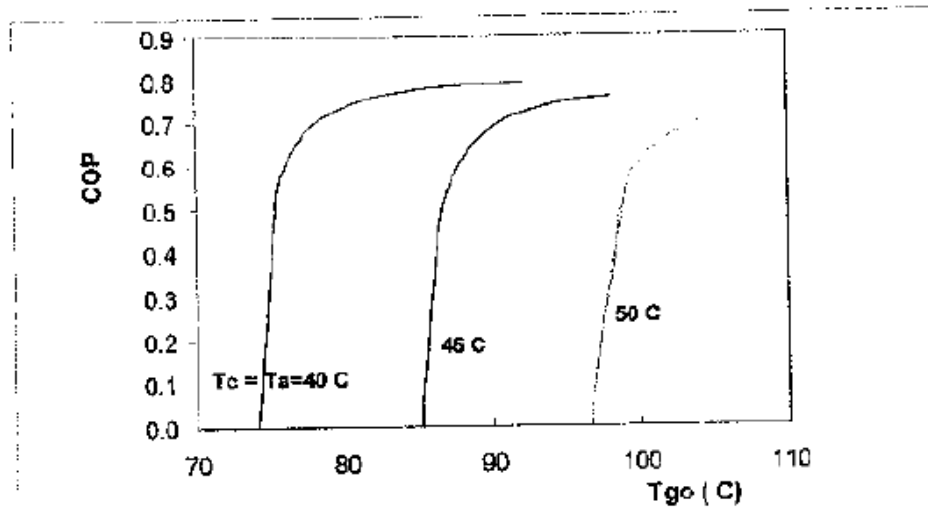


Fig. 6: COP variation with generator temperature ( $T_{go}$ ) at different equal  $T_c$  &  $T_a$  groups ( $T_e=10\text{ C}$ ,  $Q_e=1\text{ kW}$ )

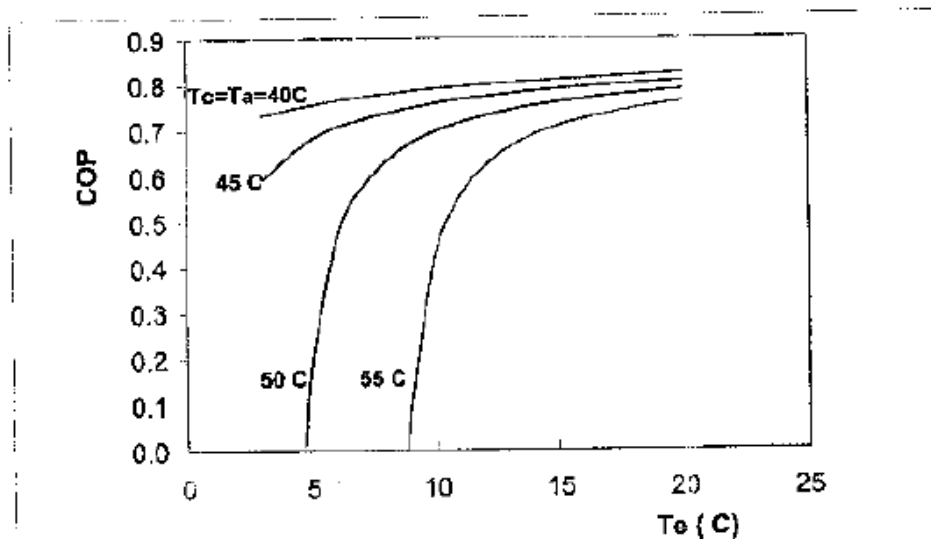


Fig.7: COP variation with evaporator temperature ( $T_e$ ) at different equal  $T_c$  &  $T_a$  groups ( $Q_e=1\text{ kW}$ )

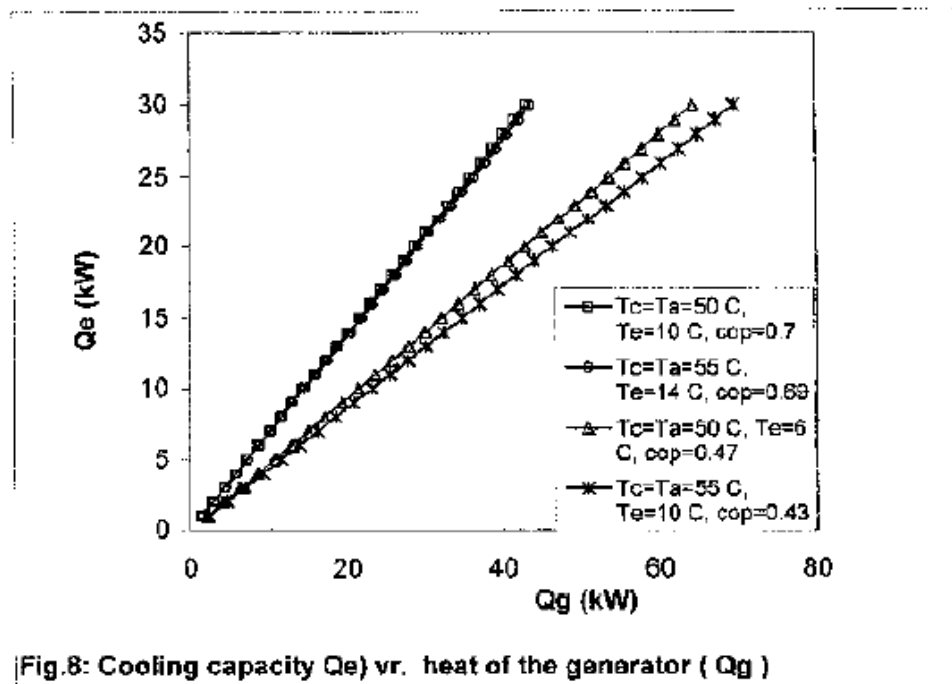


Fig.8: Cooling capacity ( $Q_e$ ) vs. heat of the generator ( $Q_g$ )

### 5- Conclusions

According to the discussed result it can be concluded that for producing cooling by an air cooled single effect LiBr-water absorption refrigerator:

- 1- The main parameters affecting the cycle performance are the temperature at generator, condenser, absorber and at the evaporator.
- 2- A higher condensing temperature requires a higher generator temperature
- 3- There is no need to increase  $\xi_v$  more than 0.63 after which there will be no significant increase in COP for any set of operating conditions.
- 4- A higher absorber temperature requires a higher evaporation temperature
- 5- For every combination of condenser-absorber-evaporator temperatures there is a minimum value for the generator temperature, and for every combination of generator-condenser-evaporator temperatures there is a maximum absorber temperature.
- 6- The concept of air cooled absorption chiller is feasible for any cooling capacity with coefficient of performance of 0.43 to 0.69 for a temperature range from 10

$^{\circ}\text{C}$  to  $14^{\circ}\text{C}$  when the condenser temperature is  $55^{\circ}\text{C}$

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