



# Energy and Exergy Analysis for Single and Parallel Flow Double Effect Water-Lithium Bromide Vapor Absorption Systems

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## ABSTRACT

The prices of energy have been increasing exponentially worldwide and industrial refrigeration is one of the most energy consuming sector. Absorption refrigeration systems are an alternative to vapor compression systems due to the increase of the environmental problems and electricity cost. In recent years, research has been increased to improve the performance of the absorption systems and the main way of improving efficiency is through thermodynamic analysis and optimization. Energy and exergy analysis of single and double effect systems is presented. The effect of different component temperature on the COP and exergetic efficiency is calculated. The results showed that the maximum COP and exergetic efficiency is achieved at lower values of the generator & the absorber temperature for both systems. The absorber temperature has the highest effect on the COP and energetic efficiency for both systems which shows that the absorber is the component that needs the improvement in design.

**Keywords:** Absorption, Thermal analysis, Exergy, COP, Absorber

## 1. INTRODUCTION

Vapor absorption systems using water as the refrigerant and the lithium-bromide as the absorbent represent the simplest idea in absorption refrigeration technology and are assuming greater importance due to their environmentally friendly operation. The cost of these systems is dependent on whether they are single effect or double effect. This in turns depends on the application and the source of heat available. [1].

In recent years, interest in absorption systems has been growing because they use friendly refrigerants and absorbents which don't deplete the ozone layer. They use cheap alternative energy sources, such solar energy or a waste byproduct heat source helping in control of global warming. Therefore, in recent years, research has been increased to improve the performance of the absorption refrigeration systems. The main way of improving efficiency is through thermodynamic analysis and optimization. [2]

In performing a thermodynamic analysis for a vapor absorption cycle, the principle of mass conservation and the first and second laws of thermodynamics are applied to individual components of the system. First and second law analyses of absorption systems for cooling and heating applications was also given by [3].

The energy analysis approach is of limited use in view of

the fact that it fails to determine the real losses in a

refrigeration system. The A.I.Shahata , M. M.

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analysis is a novel approach based on the second law of thermodynamics which assists in evaluation of irreversibility occurring in the system at local and global levels. Various studies investigated the performance analysis of absorption refrigeration systems using the energy analysis method based on the first law of thermodynamics by means of the coefficient of performance (COP). [4]

A modular computer code design was utilized to simulate the absorption systems and investigated the performance of single-effect cycle and various configurations of double-effect absorption refrigeration cycles using LiBr/H<sub>2</sub>O pair. [5]

A steady state computer simulation model was developed to predict the performance of an ARS using LiBr/H<sub>2</sub>O as a working pair. [6]

Exergy analyses have been made for absorption systems by different authors. A computer program was developed to predict the performance of an absorption refrigeration unit.[7] A single-stage refrigeration system operating with the water/lithium bromide mixture is optimized by making enthalpy and entropy balances in each one of the components of the system. An exergy analysis of double effect lithium bromide/water absorption refrigeration system was theoretically presented. [8]

The exergy analysis is carried out for each component of the system. The results showed that the highest exergy loss occurs in the HPG, which therefore makes the HPG the most important component of the double effect refrigeration system. The first and second law of thermodynamics was used to analyze the performance of a single-stage water/ lithium bromide absorption refrigeration system. [4]



The objective of this study is to develop a thermodynamic model including energy and exergy analyses for both single and parallel flow-double effect systems vapor absorption systems in order to determine which component has the great effect on the performance of the system.

**2. MODEL DESCRIPTION**

For thermodynamic analysis of the absorption system, the principles of mass conservation, first and second laws of thermodynamics are applied to each component of the system. Each component can be treated as a control volume with inlet and outlet streams, heat transfer and work interactions. In the system, mass conservation includes the mass balance of the total mass and each material species (e.g. refrigerant and absorbent in an absorption cycle). The governing equations of mass and type of material conservation for a steady state and steady flow system are given in equation (1&2) [7]:

Mass Conservation:

$$\sum(m)_{in} - \sum(m)_{out} = 0 \quad (1)$$

Species Conservation:

$$\sum(m.X)_{in} - \sum(m.X)_{out} = 0 \quad (2)$$

Where (m) is the mass flow rate and (X) is the mass fraction of Li-Br in the solution and the subscripts in and out are inlet and outlet flow in the control volume. The model solves the governing equations for the system's components and incorporates the chemical and thermodynamic properties of the water/lithium bromide solution. Single and double-effect systems configurations are shown in figures 1 and 2.

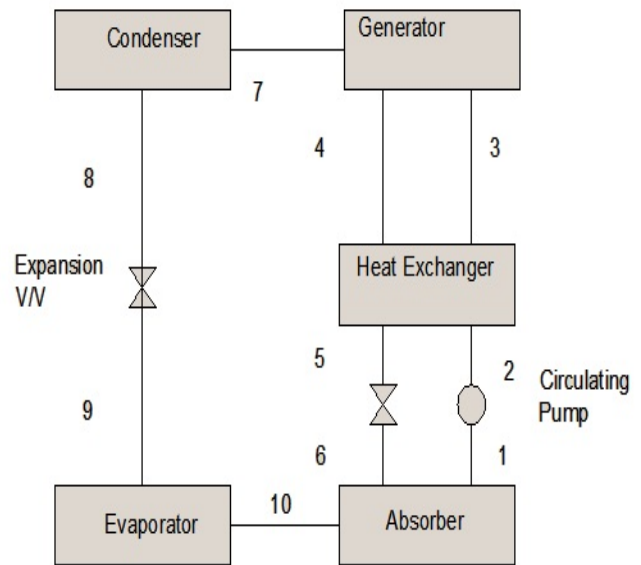


Figure (1) – single effect absorption cycle

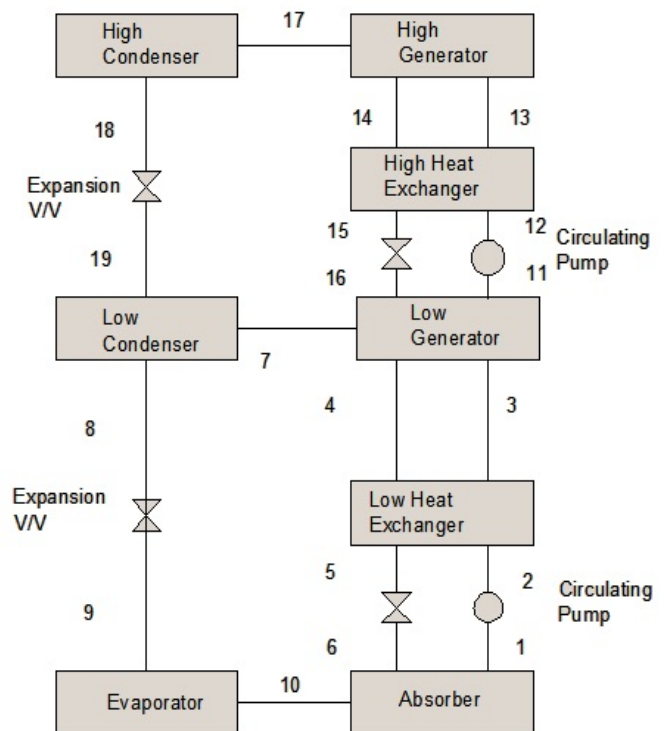


Figure (2) – parallel flow double effect absorption cycle

**2.1 First Law Analysis (Energy method)**

The first law of thermodynamics readily yields the energy balance of each component of the absorption system as follows (Conservation of energy):



$$\sum(\dot{m}h)_{out} - \sum(\dot{m}h)_{in} + [\sum \dot{Q}_{out} - \sum \dot{Q}_{in}] + W = 0 \quad (3)$$

Where the Q's are the heat transfer rates between the control volume and its environment, h is the specific enthalpy and W is the mechanical work transfer to or from the system. The heat balance over each component of the system gives the amount of the heat transfer to or from the component. Equations (4) to (9) represent the heat transfer for evaporator, condenser, absorber, generator, pump and the heat exchanger respectively.

So enthalpy data must be available for the working pair at all state points in the cycle. The enthalpies of the lithium bromide solution at the different points of the system are obtained in (kJ Kg<sup>-1</sup>) using the equations of the lithium bromide enthalpy-concentration diagram in tandem with the equations of the pressure-temperature equilibrium chart. [9]

$$h = \sum_{n=0}^4 A_n X^n + t \sum_{n=0}^4 B_n X^n + t^2 \sum_{n=0}^4 C_n X^n \quad (10)$$

Where t is the lithium bromide solution temperature in oC and X is the concentration in % Li-Br. Equation (10) could be used for the concentration range 40<X<70% Li-Br and Temperature range 15<t<165 oC. The enthalpies of both water vapor and liquid form flow in and out of the condenser and evaporator can be determined by using equations (11) and (12) respectively where t is the temperature of water. [10]

$$h = 2501 + cp.T \quad (11)$$

$$h_f = Cp_f.T \quad (12)$$

In order to perform the thermal calculations on the vapor absorption system, specific heats must be calculated for the lithium bromide solution at different points of the system. The specific heats of the water-lithium bromide solution were determined using the empirical formula given by equation (13). [11]

$$Cp = \sum_{n=0}^6 A_n X^n + t \sum_{n=0}^6 B_n X^n + t^2 \sum_{n=0}^6 C_n X^n. \quad (13)$$

Equations (14) and (15) could be used to calculate the refrigerant and absorbent temperatures. [12]

$$t = \sum_0^3 B_n X^n + t' \sum_0^3 A_n X^n \quad (14)$$

$$t' = (t - \sum_0^3 B_n X^n) / \sum_0^3 A_n X^n \quad (15)$$

Where t= solution temperature (° C)

t' = refrigerant temperature (° C)

X = the % of Li-Br-water.

The mass flow rate of the weak and strong aqueous Li-Br solution can be found by applying the mass balance on the absorber as shown in equations (16-18).

$$m_w = m_s + m_{ref} \quad (16)$$

$$m_w = m_{ref} \frac{X_s}{X_s X_w} \quad (17)$$

$$m_s = m_{w*} \frac{X_w}{X_s} \quad (18)$$

First law analysis of the absorption system leads to computing a coefficient of performance (COP), which is defined as the ratio of the useful energy obtained from the system in the evaporator to the primary energy input to the system in the generator.

$$COP = \frac{\text{useful energy output}}{\text{primary energy input}} = \frac{Q_E}{Q_G} \quad (19)$$

The solution circulation ratio is a useful parameter when dealing with design issues. It represents the ratio of the mass flow rates of the weak lithium bromide solution and the refrigerant as shown in equation (20)

$$CR = \frac{m_w}{m} = \frac{X_s}{X_s X_w} \quad (20)$$

## 2.2 Second Law Analysis (Exergy method)

The second law assists in evaluating the performance of the system based on exergy, which always decreases owing to irreversibility. Exergy analysis is the combination of the first and second laws of thermodynamics and can be defined as the maximum amount of work potential of a material or an energy stream, in relation to the surrounding environment [13]

Exergy balance for a control volume undergoing steady state process is expressed as [3]

$$EDi = \sum(m\dot{\mu})_m - \sum(m\dot{\mu})_{out} + \sum(Q(1 - \frac{T}{T})_m) + \sum(Q(1 - \frac{T}{T})_{out}) \pm \sum W \quad (21)$$

Where EDi represents the rate of exergy destruction or loss occurring in the process for each component. The first two terms on the right hand side represent the sum of the exergy input and the sum of the exergy output of the inlet and outlet streams of the control volume. The third and fourth terms are the exergy



associated with heat transferred  $Q$  from the source maintained at constant temperature  $T$ . The last term is the mechanical work transfer to or from the control volume. This term is negligible for absorption systems as the solution pump has very low power requirements. By neglecting changes in kinetic energy and potential energy and because there is no departure of chemical substances from the cycle to the environment, the chemical exergy is zero [14]. The specific exergy or the exergy of a fluid stream can be defined as [15]:

$$\psi = (h - h_o) - T_o (s - s_o) \quad (22)$$

Where  $\Psi$  is the exergy of the fluid at temperature  $T$ . The terms  $h$  and  $s$  are the enthalpy and entropy of the fluid, whereas,  $h_o$  and  $s_o$  are the enthalpy and entropy of the fluid at environmental temperature  $T_o$  (in all cases absolute temperature is used in K). In this study,  $T_o$  was taken as 298.15 K. The principle exergy losses in a process leading to these losses are due to dissipation (friction), heat transfer under temperature difference and unrestricted expansion. Enthalpy and entropy of reference for LiBr-H<sub>2</sub>O solution considered:  $T_0=25^\circ\text{C}$ ,  $p_0=101.325\text{ kPa}$  and  $x_{\text{LiBr}}=55.42\%$  are  $h_0=66.83\text{ kJ/kg}$  and  $s_0=0.1495\text{ kJ/kgK}$ .

The ratio of the exergy loss rate in each component to the total exergy loss rate of the system is defined as non-dimensional exergy loss of the component. The non-dimensional exergy loss of each component is written as follows:

$$\psi_i = \frac{ED_i}{ED_t} \quad (23)$$

By using Eq. (23), the significance of each component's contribution into the total exergy loss of the system can be found. Irreversibility in any component under consideration is expressed using the following equation: [16]

$$I_i = ED_i + EL_i \quad (24)$$

Where  $(EL_i)$  is the thermal exergy loss which takes place because of temperature difference between the control volume and the surroundings. It depends upon how the boundary of the system is selected. The thermal exergy loss rate is given by: [17]

$$EL_i = Q_i \left(1 - \frac{T_o}{T_i}\right) \quad (25)$$

Where  $Q_i$  is the heat rejected by the component under consideration and  $T_i$  is the temperature at its boundary. The second law efficiency of the absorption system is measured by the exergetic efficiency,  $E$ ; which

is defined as the ratio of the useful exergy gained from a system to that supplied to the system. The exergetic efficiency (rational efficiency) can be calculated as the ratio between the net exergy produced by the evaporator (exergy desired output) and the input exergy to the generator plus the mechanical work of the solution pump: [13]

$$\eta_{ex} = \frac{Q_e \left(1 - \frac{T_o}{T_e}\right)}{Q_g \left(1 - \frac{T_o}{T_g}\right) + W_p} \quad (26)$$

Where  $T_r$  is the temperature of the space to be cooled.

## 2.3 Theoretical Considerations

The mathematical model for this investigation has been developed with the following assumptions:

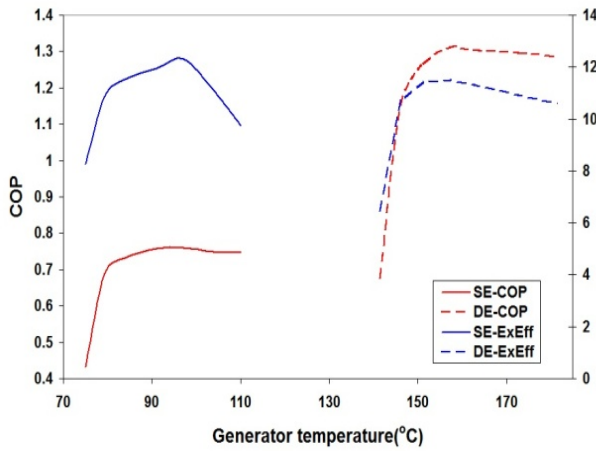
1. The heat losses and unintentional pressure drops in the pipelines and system components are negligible.
2. The processes at all throttle valves are assumed to be adiabatic, resulting in constant enthalpy processes.
3. Heat transfer to and from the surroundings to the system is negligible.
4. Circulating pump is isentropic.
5. For the double effect systems, both solution heat exchangers have same effectiveness.
6. The refrigerant states leaving the condenser and evaporator are assumed to be saturated.

## 3. RESULTS AND DISCUSSIONS

The results are presented by plotting the coefficient of performance and the exergetic efficiency versus the temperature for the different components of the vapor absorption system.

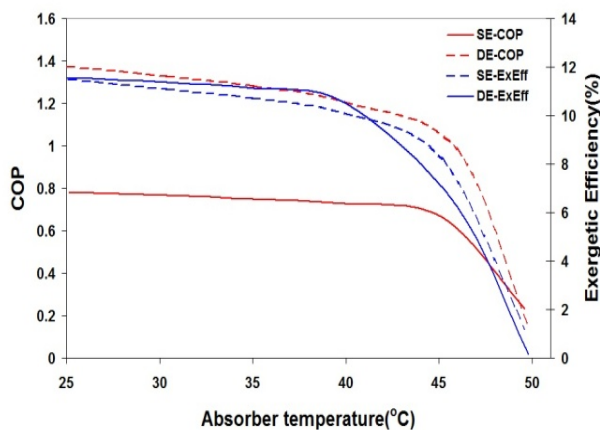
Figure 3 shows the effect of the generator temperature on the COP and the exergetic efficiency for both single and double-effect absorption systems. The following parameters are used: [ $Q_e=425.56\text{ kW}$ ,  $T_e=1.30^\circ\text{C}$ ,  $T_c=35^\circ\text{C}$ ,  $\epsilon=0.7$  and  $T_a=35^\circ\text{C}$ ] for single effect system while [ $Q_e=425.56\text{ kW}$ ,  $T_e=1.30^\circ\text{C}$ ,  $T_{lc}=35^\circ\text{C}$ ,  $\epsilon=0.7$  and  $T_a=35^\circ\text{C}$ ] for double effect system.

It is shown that there is an increase in both COP and exergetic efficiency initially for both single and double effect systems and then both COP and exergetic efficiency decrease as the generator temperature increases. The maximum COP and exergetic efficiency for single effect is 0.76 and 12.3 while the maximum COP and exergetic efficiency for double effect system is 1.31 and 11.48 respectively. The parallel flow-double effect system has a higher COP compared with the single effect system. On the other hand, the parallel flow-double effect system has a lower exergetic efficiency compared with the single effect system.



**Figure (3) COP and Exergetic Efficiency vs. Generator Temperature**

The increase in the generator temperature results in the increase of the solution exiting the generator, the absorber and the condenser temperatures. The similar trend of COP and exergetic efficiency was obtained by [5&17]. The maximum COP and exergetic efficiency is achieved at lower values of the generator temperature for both single and double effect systems. The percent of increase in the COP and exergetic efficiency for single effect is 37% and 23.95% while the percent of increase in the COP and exergetic efficiency for double effect is 41.4% and 39.7%.



**Figure (4) COP and Exergetic Efficiency vs. Absorber Temperature**

Figure 4 shows the effect of the absorber temperature on the COP and the exergetic efficiency for both single and double-effect absorption systems. The following parameters are used: [ $Q_e=425.56$  kW,  $T_e=1.30^\circ\text{C}$ ,  $T_c=35^\circ\text{C}$ ,  $\epsilon=0.7$  and  $T_g=91^\circ\text{C}$ ] for single effect system while [ $Q_e=425.56$  kW,  $T_e=1.30^\circ\text{C}$

$T_{lc}=35^\circ\text{C}$ ,  $\epsilon=0.7$  and  $T_g=161.2^\circ\text{C}$ ] for double effect system.

When the absorber temperature increases, the generator thermal load increase with a constant evaporator thermal load, therefore the COP and the exergetic efficiency decrease. It is shown that the decrease of the absorber temperature causes significant increase in both COP and the exergetic efficiency of the system. The maximum COP and exergetic efficiency for single effect is 0.78 and 11.48 while the maximum COP and exergetic efficiency for double effect system is 1.3736 and 11.5766 respectively. The maximum COP and exergetic efficiency is achieved at lower values of the absorber temperature. The double effect system has higher COP and exergetic efficiency compared with the single effect system. With the decrease of the absorber temperature, the percent of increase in the COP and exergetic efficiency for single effect is 64.8% and 85% while the percent of increase in the COP and exergetic efficiency for double effect is 85.1% and 97.2% respectively.

Figure 5 shows the effect of the evaporator temperature on the COP and the exergetic efficiency for both single and double-effect absorption systems. The following parameters are used: [ $Q_e=425.56$  kW,  $T_g=91^\circ\text{C}$ ,  $T_a=T_c=35^\circ\text{C}$ ,  $\epsilon=0.7$ ] for single effect system while [ $Q_e=425.56$  kW,  $T_g=161.2^\circ\text{C}$ ,  $T_a=T_{lc}=35^\circ\text{C}$ ,  $\epsilon=0.7$ ] for double effect system.

The increase in the evaporator temperature increases the COP and reduces the exergetic efficiency for both single and double effect systems. The parallel flow-double effect system has a higher COP compared with the single effect system. On the other hand, the single effect system has a higher exergetic efficiency compared with the single effect system.

When the evaporator temperature increases, the absorber and generator thermal loads decrease with constant cooling load ( $Q_e$ ) and thus the COP increase. The maximum COP is achieved at higher values of the evaporator temperature while the maximum exergetic efficiency is achieved at lower values of the evaporator temperature for both single and double effect systems. With the increase of the evaporator temperature, the percent of increase in the COP is 1.6% for single effect and 2.4% for double effect system. But the increase of the evaporator temperature causes a rate of decrease in the exergetic efficiency 1.63 for single effect system and 2.08% for double effect.



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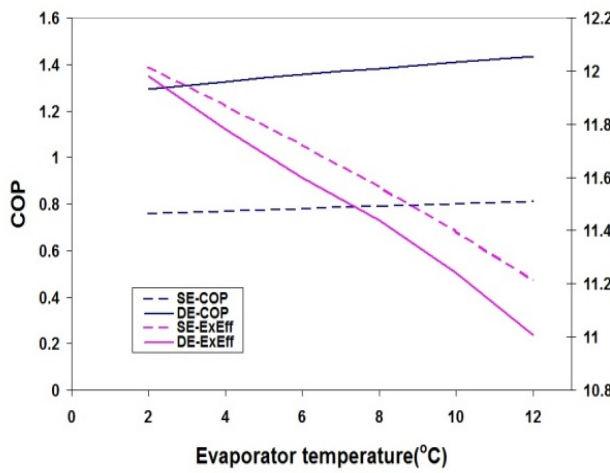


Figure (5) COP and Exergetic Efficiency vs. Evaporator Temperature

Figure 6 shows the effect of the condenser temperature on the COP and the exergetic efficiency for both single and double-effect absorption systems. The following parameters are used: [ $Q_e=425.56$  kW,  $T_g=91$ °C,  $T_e=1.3$ °C,  $\epsilon=0.7$ ] for single effect system while [ $Q_e=425.56$  kW,  $T_g=161.2$ °C,  $T_e=1.3$ °C,  $\epsilon=0.7$ ] for double effect system.

When the condenser temperature increases, the generator thermal load increases with constant evaporator thermal load and so COP and exergetic efficiency decrease. The maximum COP is achieved at lower values of condenser temperatures for both single and double effect systems. The maximum COP and exergetic efficiency for single effect is 0.7855 and 11.633 while the maximum COP and exergetic efficiency for double effect system is 1.31511 and 12.9955 respectively. The double effect system has higher COP and exergetic efficiency compared with the single effect system. With the decrease of the condenser temperature, the percent of increase in the COP and exergetic efficiency for single effect is 10.7% and 2.46% while the percent of increase in the COP and exergetic efficiency for double effect is 1.31% and 3.012% respectively.

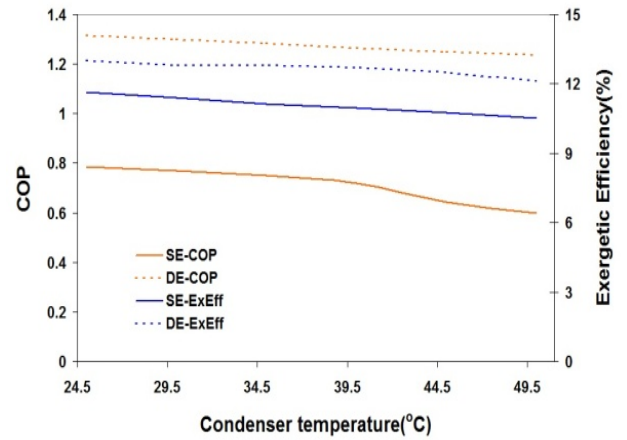


Figure (6) COP and Exergetic Efficiency vs. Condenser temperature

Figure 7 shows the effect of solution heat exchanger effectiveness on the COP and the exergetic efficiency for both single and double-effect absorption systems. As known, if ( $\epsilon$ ) increases, the heat exchange between the weak and strong solutions increases and as a result of this the temperature of the strong solution decreases and that of the weak solution increases. It is shown from the figure that the increase in solution heat exchanger effectiveness increases the COP and the exergetic efficiency for both single and double-effect absorption systems.

It is also shown that the double effect system (parallel flow) have a higher COP compared with the single effect system. On the other hand, the double effect system (parallel flow) has a lower exergetic efficiency compared with the single effect system. For the double effect system, it is observed that the effectiveness of solution heat exchanger '2' ( $\epsilon_2$ ) has greater effect on exergetic efficiency than the effectiveness of solution heat exchanger '1' ( $\epsilon_1$ ) in the range from (0.2-0.7) however the effectiveness of solution heat exchanger '1' ( $\epsilon_1$ ) has greater effect on exergetic efficiency than the effectiveness of solution heat exchanger '2' ( $\epsilon_2$ ) in the range from (0.7-1).

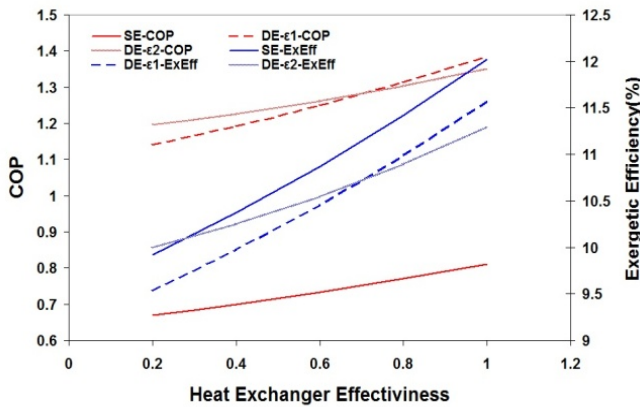


Figure (7) COP and Exergetic Efficiency vs. H. Exchanger Effectiveness

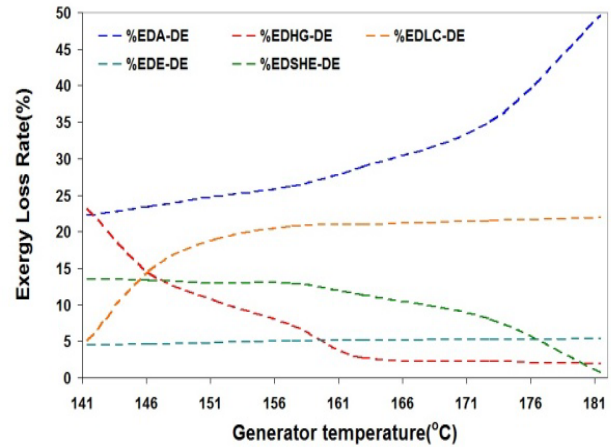


Figure (9) Exergy loss rate vs. Generator temperature for double effect systems

Figures 8 & 9 show the effect of the generator temperature on the ratio of the exergy loss in the different components to the total exergy loss of the system for both single and double effect systems. It is found that when the generator temperature increases, the concentration of the strong solution increases with constant weak solution concentration entering the generator and so the SCR decreases. Also when the generator temperature increase, the temperature of the solution returning from the generator to the absorber increase and the temperature of the vapor leaving the generator increase which increase the irreversibility in the absorber and the condenser and therefore the exergy loss in the absorber and the condenser increase with the increase in the generator temperature.

It was shown from the figure that the increase in the generator temperature has no effect on the exergy loss in the evaporator and a little effect on the exergy loss in the solution heat exchanger. So the increase in the generator temperature cause decrease in exergy loss rate of the generator and increase in the exergy loss rate of both condenser and absorber. The highest exergy loss rate was found in the absorber of 63.2 % for single effect at 110oC and 50% for double effect system at 180oC.

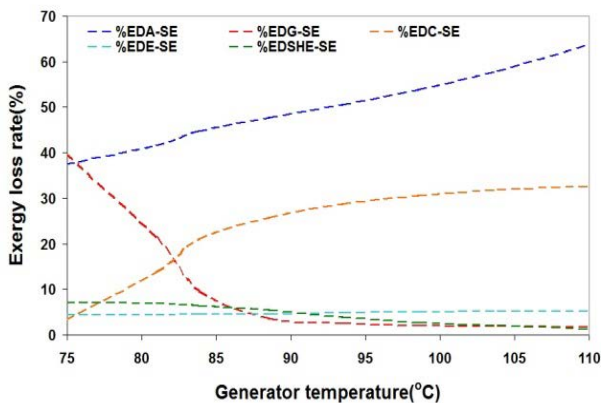


Figure (8) Exergy loss rate vs. Generator temperature for S.E system

Figure 10 shows the effect of the absorber temperature on the irreversibility percentage of the different components to the total irreversibility of the system for both single and double effect systems. It is found that when the absorber temperature increases, the concentration of the weak solution increases with constant strong solution concentration entering the generator and so the SCR increases.

Also when the absorber temperature increase, the temperature of the solution entering the generator increase and the temperature of the solution leaving the heat exchanger and entering the absorber increase which increase the irreversibility percentage of both the absorber and the generator as shown from the figure. It was also shown that the increase in the absorber temperature has a little effect on the evaporator. The highest irreversibility percentage was found in the absorber of 60.56 % for single effect and 49.4% for double effect system. The percent of increase of irreversibility in the absorber is 9.9% for single effect and 19.8% while the percent of increase of irreversibility in the generator is 39.6% for single effect and 40.6% for double effect system.

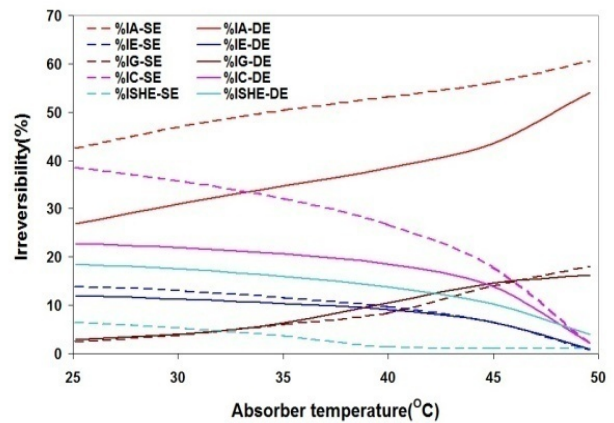


Figure (10) Irreversibility Percentage vs. Absorber temperature for both systems



Figures 11 & 12 show the effect of the generator temperature on the irreversibility percentage of the different components to the total irreversibility of the system for both single and double effect systems. As mentioned before when the generator temperature increases, the COP and the SCR increases. Also when the generator temperature increase, the temperature of the solution entering the absorber increase and the temperature of the solution leaving the heat exchanger and entering the generator increase which increase the irreversibility in both the absorber and the generator. It was shown from the figure that the increase in the absorber temperature has a little effect on the evaporator. The highest irreversibility percentage was found in the absorber of 64.56 % for single effect and 45% for double effect system.

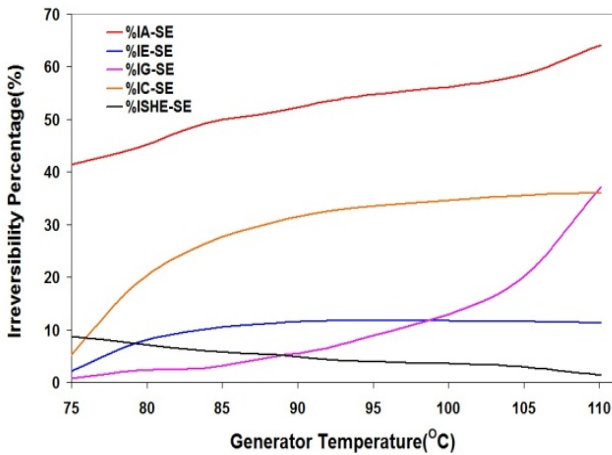


Figure (11) Irreversibility Percentage vs. Generator temperature for SE effect system

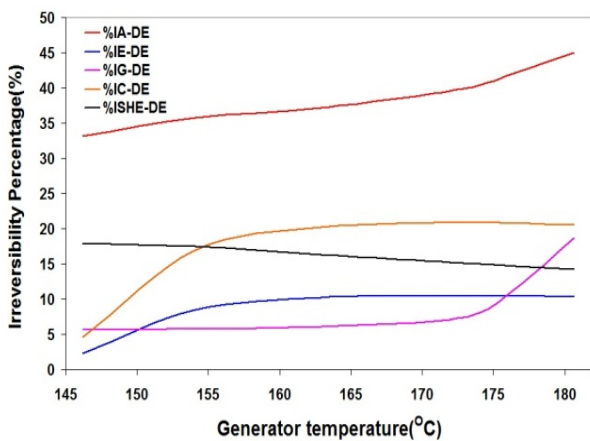


Figure (12) Irreversibility Percentage vs. Generator temperature for double effect system

Figure 13 represents the variation of the exergy loss rate of the different components for the base case of single and double effect systems. As shown in the figure, the absorber has highest exergy loss rate among the different components of 49.511% for single effect and

27.511% for double effect system. The worst component from the viewpoint of the exergy loss rate in the single-effect system is the absorber followed by the condenser, evaporator, solution heat exchanger and the generator. That shows that the absorber is the component that needs the maximum improvement in design so as to reduce the exergy loss rate and so reduces the irreversibility.

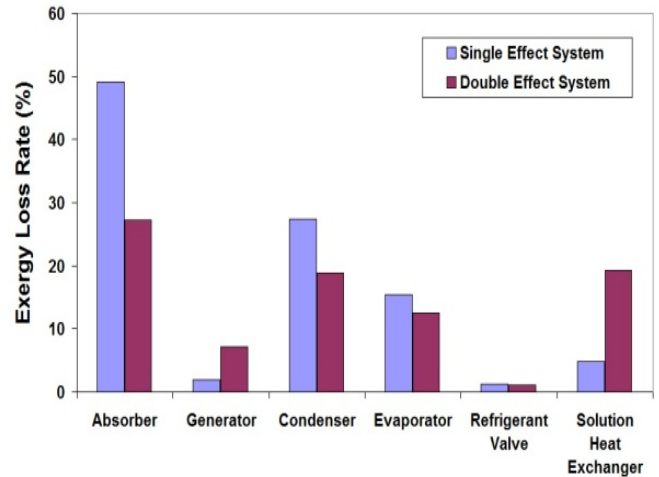


Fig 13 – Exergy Loss Rate vs. different components for single and double effect systems

#### 4. CONCLUSIONS

The maximum COP and exergetic efficiency is achieved at lower values of the generator and absorber temperatures for both single and double effect systems. The double effect system has higher COP and exergetic efficiency compared with the single effect system.

The maximum COP and exergetic efficiency for single effect is 0.78 and 11.48 while the maximum COP and exergetic efficiency for double effect system is 1.3736 and 11.5766. With the increase of the evaporator temperature, the percent of increase in the COP is 1.6% for single effect and 2.4% for double effect system. But the increase of the evaporator temperature causes a rate of decrease in the exergetic efficiency 1.63% for single effect system and 2.08% for double effect.

The increase in the absorber temperature increases the irreversibility percentage of both the absorber and the generator. The highest irreversibility percentage was found in the absorber of 60.56 % for single effect and 49.4% for double effect system. The absorber has highest exergy loss rate among the different components of 49.511% for single effect and 27.511% for double effect system which shows that the absorber is the component that needs the maximum improvement in design so as to reduce the exergy loss rate and so reduces the irreversibility.





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# Energy and Energy Analysis for Single and Parallel Flow Double Effect Water-Lithium Bromide Vapor Absorption Systems

## Nomenclature

$m_{ref}$	Refrigerant mass flow rate , kg/s
$X$	Concentration, %
$H$	Enthalpy, kJ/kg
$Q$	Heat transfer rate, kW
$W$	Pump Work, kW
$C_p$	Specific heat at constant pressure, kJ/kg
$E_{di}$	Exergy Destruction rate, kW
$I_i$	Irreversibility at component I, kW
$E_{li}$	Thermal Exergy, kW
$S$	Entropy, kJ/kgK
$T_r$	Temperature of space to be cooled, °C
$M_w$	Weak solution mass flow rate kg/s
$M_s$	Strong solution mass flow rate, kg/s
$T_{lc}$	Low condenser temperature, °C

### *Greek Symbols*

$Z_{ex}$	Exergetic Efficiency, %
$\Psi$	Exergy of the fluid stream, kJ/kg
$\varepsilon$	Heat Exchanger Effectiveness

### *Non-dimensional Numbers*

CR	Solution circulation ratio
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