

# Performance Analysis of a Solar Driven Single Stage LiBr/H<sub>2</sub>O Absorption Refrigeration system

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## ABSTRACT

The solar-assisted combined ejector was configured with the basic cycle of solar absorption refrigeration system to evaluate the performance of this cycle which using LiBr/H<sub>2</sub>O as a working fluid and operating under steady-state conditions. In this paper, the improvement of the system is achieved by utilizing the potential kinetic energy of the ejector to enhance refrigeration efficiency. However, the first and the second law of thermodynamics are used to analyze the performance of a single-stage water-lithium bromide absorption refrigeration system (ARS), whereas some working parameters are varied. Moreover, a mathematical model based on the exergy method is introduced to evaluate the system performance, exergy loss of each component and total exergy loss of all the system components. As well as, Parameters connected with performance of the cycle—circulation ratio (CR), coefficient of performance (COP), exergetic efficiency are calculated from the thermodynamic properties of the working fluids at various operating conditions. In addition, Minimum generator temperature that required to operate the system was evaluated. The results showed that, the evaporator, condenser loads and post-addition of the ejector are found to be permanently higher than that in the basic cycle. As well as, The COP of the modified cycle is improved by up to 60 % compared with that in the basic cycle at the given condition.

**Keywords:** LiBr-H<sub>2</sub>O, COP, Exergy, Optimization, Generator temperature, Combined ejector absorption cycle.

## 1 Introduction

Absorption refrigeration systems (ARSs) have been gaining popularity because, firstly, they operate on environment friendly refrigerants. Secondly, they harness cheap alternative energy sources, such as geothermal, biomass, solar energy or a waste by product heat source. Therefore, in recent years, research has been devoted to the improvement of ARSs. The main way of improving efficiency is through thermodynamic analysis and optimization [1–4].



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The basis of thermodynamics is stated in the first and second laws. The first law describes the conservation of energy, while the second law is used to describe the quality of energy and material. The first law optimization should result in maximizing the coefficient of performance (COP), thus providing maximum heat removal for minimum power input, while the second law optimization should result in maximizing the exergetic efficiency and minimizing entropy generation within the system, hence providing maximum cooling for the smallest destruction of available energy (exergy).

The exergy method, known as the second law analysis, calculates the exergy loss caused by irreversibility, which is an important thermodynamic property that measures the useful work that can be produced by a substance or the amount of work needed to complete a process [5]. The exergy analysis is a powerful tool for thermodynamic analysis of energy-conversion systems. The concept of exergy is extensively discussed in the literature by Kotas [6], Szargut *et al.* [7] and Bejan [8]. Aphornratana and Eames [9] reported that the heat operated absorption refrigeration systems are attracting increasing interest, as they can be driven by low-temperature heat sources and employ environment friendly working fluids. Furthermore, they estimated the specific entropy field for temperatures ranging from 10 to 200 °C and concentrations ranging from 20% to 70%. Furthermore, some of the papers include economic analysis [10,11].

Many researches are recently interested in applications of solar-assisted absorption cooling/refrigeration systems and the improvement of their corresponding performance as it saves energy and is environmentally friendly. Solar energy is available in most areas and regarded as a good source of thermal energy. For many solar absorption cooling systems, LiBr/ Water and Ammonia/Water is a major working fluid pair in the context of these systems. The absorption performance of a cooling system is critically dependent on the chemical and thermodynamic properties of the working fluid [3]. In solar applications, the LiBr–H<sub>2</sub>O system is superior to the NH<sub>3</sub>–H<sub>2</sub>O system, due to its simpler design and operation and low cost. Moreover, it is functional under low generator temperature and perform better than that of NH<sub>3</sub>–H<sub>2</sub>O, due to its simpler design and operation and low cost. Moreover, it is functional under low generator temperature and perform better than that of NH<sub>3</sub>–H<sub>2</sub>O. The LiBr/water system has been widely used for many years and their properties are well established. Many types of absorption cycles have been developed, however, the system's complexities increased over a conventional single-effect absorption system. The double-effect absorption systems using lithium bromide/water seem to be a high performance system, which is available commercially. The (single-effect) cooling system using LiBr/H<sub>2</sub>O as its working fluid system can provide COP as high as a double-effect system at minimal increase of system complexity. Thus, research has focused on improving these absorption cooling systems by enhancing the coefficient of performance (COP) of these systems with respect to both heating and cooling applications [5,6]. Ventas *et al.* [7] numerically studied single-effect

absorption cycles and utilized ammonia-lithium nitrate solution as its working pair to determine the effect of mass flow rate recirculating through the absorber, as well as system performance. Numerous works also aim to optimize the operating parameters to enhance performance and minimize energy consumption in ARS

This work conceptualizes and analyse the single effect absorption refrigeration system combined with the ejector and an upgraded system that considerably lower the energy footprint. This paper aims to improve the system's performance by using the potential kinetic energy from the primary streamlines to drive the low pressure secondary streamlines to produce a quieter flow with more thrust and lower temperature. The proposed cycle will be evaluated via the entrainment ratio and the effect of the operation temperature on the thermal loads and the system's performance. A general description of the proposed system will be presented, and further analysis on its performance is conducted based on a corresponding mathematical model.

## **2 Description of Absorption Cycle**

There are many types of absorption cycles, single and multiple effect absorption cycles, the most commonly one is a single effect absorption cycle.

### **2.1 Description of Solar Single effect Absorption Cycle:**

The basic cycle consists of generator, absorber, a condenser, an evaporator, solution heat exchanger, circulating pumps, and a solar collector. The cycle works between two pressure levels: low pressure at the evaporator-absorber, and high pressure at condenser-generator. The emitted vapours in the generator consist of pure water, while LiBr sal remained in the solution.

### **2.2 Description of Modified Single Effect Combined Ejector Absorption Refrigeration system:**

The main part of the single effect absorption cooling system is shown schematically in Figure. 1.a The solution heat exchanger is advantageous, because it cools down the solution coming from the generator, which then heats up the solution entering the generator . An ejector was added between the generator and the condenser. The primary high-pressure water vapour from the generator enters the ejector, then from the secondary inlet of the ejector, the entrainment low pressure water vapor from the evaporator is mixed with the primary flow at the mixing chamber, passing the diffuser, then entering the condenser. In this new design, the lithium bromide mixture leaves the absorber (state 1) in the form of a saturated solution at low pressure. It is pumped to the system at high pressure (state 3). The generator operates from a high temperature source to separate the binary solution of water and Lithium bromide (strong solution comes from absorber). This two-phase mixture is separated, and the weak liquid flows through SHE (state 4 to state 5) than throttled to the low pressure system and sprayed into the absorber (state 6). On the ejector, the secondary flow (water vapour from evaporator (state

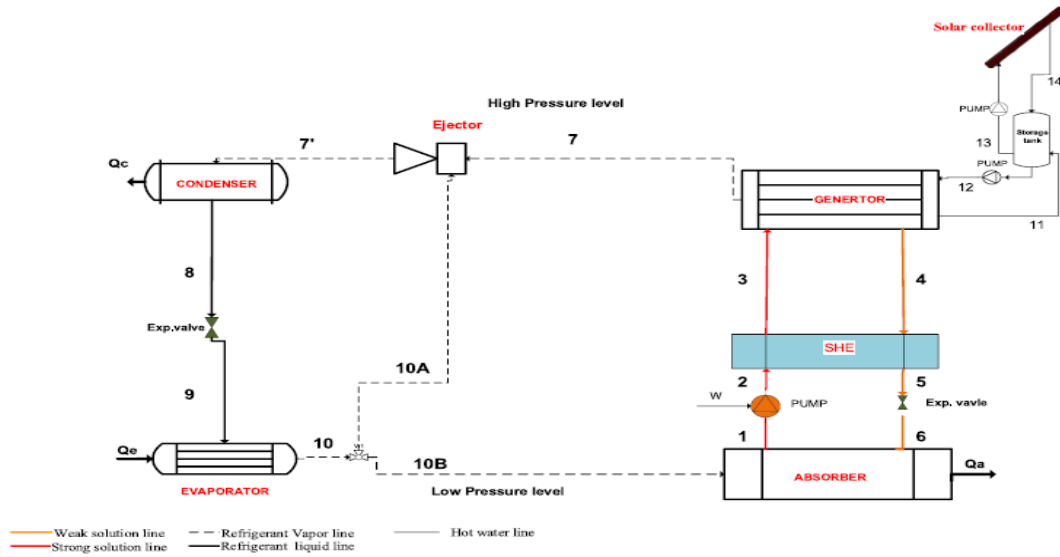
10A)) and the primary flow of water vapour from the generator are mixed and passed to the condenser (state 7).

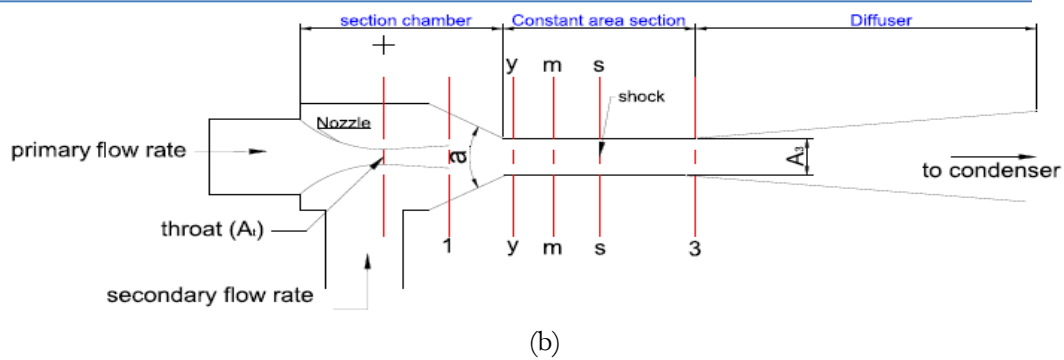
### 3 Simulation and Analysis of Proposed Cycle

For the purpose of simulation and analysis, the following assumptions are made:

- The system operates under a steady state condition.
- The refrigerant leaving the condenser and evaporator is saturated (state points 8 and 10).
- LiBr/H<sub>2</sub>O solution in the generator, solution heat exchanger, and absorber are assumed to be in the equilibrium state at their respective pressure and temperature and assumed saturated state.
- The frictional pressure drop in the cycle is neglected except through the expansion device.
- The flow inside the ejector is steady and one-dimensional. The ejector walls are adiabatic.
- The primary flow and the secondary flow are saturated and their velocities are negligible before entering the ejector (states 7 and 10B in Figure. 1a respectively). The velocity of the mixed flow leaving the ejector (at state 7') is also neglected.

In order to calculate the heat and mass balance for the proposed cycle, the thermodynamic properties (pressure, temperature, concentration, enthalpy and density) are necessary for the simulation. The binary mixture of LiBr/H<sub>2</sub>O and pure H<sub>2</sub>O are used in the proposed system. The detailed thermodynamic property equations of LiBr/H<sub>2</sub>O are found by Engineering Equation Software.





**Figure 1:** (a) System schematic of single effect combined absorption–ejector cooling system, (b) schematic diagram of the ejector

### 3.1 Thermodynamic Analysis

The thermodynamic analysis mainly aimed at assessing the thermodynamic imperfections and suggested possible ways of improving these imperfections. Here the system is analysed based on mass, energy and exergy balance. Each component of system can be assumed as control volume having inlet and outlet flow, work interactions and heat transfer. For analysing vapour absorption system circulation ratio is one of the most important parameter, it is defined as ratio of strong solution flow rate to refrigerant flow rate,

$$CR = \frac{\dot{m}_{ss}}{\dot{m}_r} = \frac{X_{SS}}{X_{SS} - X_{WS}} \tag{1}$$

The coefficient of performance (COP) is used to measure the system performance:

$$COP = \frac{Q_{eva}}{Q_{gen} + W_{pump}} \tag{2}$$

In order to use previous equation, mass and energy conservation should be determined at each component.

#### 3.1.1 For the Generator – SHE – Absorber Loop:

The mass and energy balances around the generator

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_7 \quad \rightarrow \quad \dot{m}_3 x_3 = \dot{m}_4 x_4 + \dot{m}_7 x_7 \tag{3}$$

Where:  $x_3 = x_1, \quad x_4 = x_6$

$$Q_{gen} = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_3 h_3 \tag{4}$$

The fluid properties in this loop can be derived and developed as:

The liquid weak solution at state (4):

$$T_4 = T_{gen}, \quad P_4 = P_{gen}$$

Exergy balance formulation for generator can be written as:

$$\Delta E_G = \dot{m}_{ws}(h_3 - T_0 s_3) + \dot{m}_{ss}(h_4 - T_0 s_4) - \dot{m}_r(h_7 - T_0 s_7) + \dot{m}_G(h_{11} - T_0 s_{11}) - \dot{m}_G(h_{12} - T_0 s_{12}) \tag{5}$$

### 3.1.2 Heat Exchanger:

SHE performance is expressed in terms of effectiveness  $\epsilon_{she}$ .

The solution and refrigerant heat exchanger performance, expressed in terms of an effectiveness  $\epsilon_{she}$

$$\epsilon_{she} = \frac{T_4 - T_5}{T_4 - T_2} \quad (6)$$

$$C_{hot} = \dot{m}_4 \left( \frac{h_4 - h_5}{T_4 - T_5} \right) \quad , \quad C_{cold} = \dot{m}_2 \left( \frac{h_3 - h_2}{T_3 - T_2} \right) \quad (7)$$

$$Q_{hx} = \dot{m}_2 (h_3 - h_2) \quad , \quad Q_{hx} = \dot{m}_4 (h_4 - h_5) \quad (8)$$

where,

$$\dot{m}_1 = \dot{m}_2 = 0.05 \frac{kg}{s} \quad , \quad T_1 = T_{abs} = T_2$$

Exergy balance formulation for solution heat exchanger can be written as:

$$\Delta E_{SHX} = \dot{m}_{ws}(h_2 - T_0 s_2) - \dot{m}_{ws}(h_3 - T_0 s_3) + \frac{Q_{SHX}}{T_b} \quad (9)$$

### 3.1.3 Solution Expansion Valve Model:

$$h_5 = h_6 \quad , \quad \dot{m}_5 = \dot{m}_6 \quad , \quad x_5 = x_6$$

Exergy balance formulation for expansion valve can be written as:

$$\Delta E_{REXP} = \dot{m}_{ss}(h_5 - T_0 s_5) - \dot{m}_{ss}(h_6 - T_0 s_6) \quad (10)$$

$$\Delta E_{SEXP} = \dot{m}_{ss}(h_8 - T_0 s_8) - \dot{m}_{ss}(h_9 - T_0 s_9) \quad (11)$$

### 3.1.4 Pump Calculation:

$$h_2 = h_1 + \frac{W_{pump}}{\dot{m}_1} \quad (12)$$

$$W_{pump} = \dot{m}_5 v_1 \frac{P_{high} - P_{low}}{1000} \quad (13)$$

Exergy balance formulation for pump can be written as:

$$\Delta E_{pump} = \dot{m}_{ws}(h_1 - T_0 s_1) - \dot{m}_{ws}(h_2 - T_0 s_2) + \dot{W}_p \quad (14)$$

### 3.1.5 Absorber:

$$Q_{abs} = \dot{m}_{10} h_{10} + \dot{m}_6 h_6 - \dot{m}_1 h_1 \quad (15)$$

Exergy balance formulation for absorber can be written as:

$$\Delta E_{abs} = \dot{m}_r(h_{10} - T_0s_{10}) + \dot{m}_{ss}(h_6 - T_0s_6) - \dot{m}_{ws}(h_1 - T_0s_1) + \frac{Q_{abs}}{T_b} \quad (16)$$

### 3.1.6 Condenser:

$$Q_{cond} = \dot{m}_7(h_7 - h_8) \quad (17)$$

Exergy balance formulation for condenser can be written as:

$$\Delta E_{cond} = \dot{m}_r(h_7 - T_0s_7) - \dot{m}_r(h_8 - T_0s_8) + \frac{Q_{cond}}{T_b} \quad (18)$$

### 3.1.7 Refrigerant Valve:

$$h_8 = h_9 \quad , \quad x_8 = x_9$$

### 3.1.8 Evaporator:

$$Q_{eva} = \dot{m}_9(h_{10} - h_9) \quad (19)$$

Exergy balance formulation for evaporator can be written as:

$$\Delta E_{eva} = \dot{m}_r(h_9 - T_0s_9) - \dot{m}_r(h_{10} - T_0s_{10}) + \frac{Q_{eva}}{T_b} \quad (20)$$

The total rate of exergy destruction of absorption system is the sum of exergy destruction in each component and can be written as:

$$\Delta E_{sys} = \Delta E_G + \Delta E_{abs} + \Delta E_{eva} + \Delta E_{cond} + \Delta E_{SHX} + \Delta E_{REXP} + \Delta E_{SEXP} + \Delta E_P \quad (21)$$

## 3.2 Nozzle Equations

For a given pressure  $P_{gen} = P_7$ , temperature  $T_{gen} = T_7$  and  $\dot{m}_p = \dot{m}_7$  the nozzle throat area required for choking condition follows the gas dynamic equation:

$$A_t = \frac{\dot{m}_p \sqrt{T_{gen}}}{P_{gen} \sqrt{\eta_p} \sqrt{\frac{k}{R} \left[ \frac{2}{k+1} \right]^{\frac{k+1}{k-1}}}} \quad (22)$$

where  $\eta_p$  is a coefficient relating to the isentropic efficiency of the compressible flow in the nozzle. The relations between the Mach number at the exit of nozzle  $M_{p1}$  and the exit cross section area  $A_{p1}$  and pressure  $P_{p1}$  are, using isentropic relations as showing in the following equations:

$$\frac{A_{p1}}{A_t} = \frac{1}{M_{p1}} \left[ \frac{2}{k+1} \left( 1 + \frac{k-1}{2} M_{p1}^2 \right) \right]^{\frac{k+1}{k-1}} \quad (23)$$

## 4 Results and Discussions

Table 1 shows that the highest heat load occurs in the generator (approximately 13kW) and the heat transfer rate of the refrigerant heat exchanger is lower than that of the solution heat exchanger due to mass flow rate and temperature difference between the fluids.

**Table 1: Heat transfer rates of components and performance parameters of the system**

Components	Heat transfer rates (KW)
Generator	12.985
Condenser	10.483
Evaporator	10.000
Absorber	12.502
Pump	0.00012
Solution heat exchanger	2.254
Refrigerant heat exchanger	0.165

Table 2 shows the simulation results of the second law analysis of the ARS. The generator has the highest exergy loss rate (approximately 2 kW), and the next largest exergy loss rate occurred in the absorber (approximately 1.5kW). Since exergy loss rates in the pump and expansion valves are very small, their effects on the total exergy loss rate are inconsiderable. The exergy losses of the generator and absorber are important fractions of the total exergy loss in the system.

**Table 2: Exergy loss rates of the components**

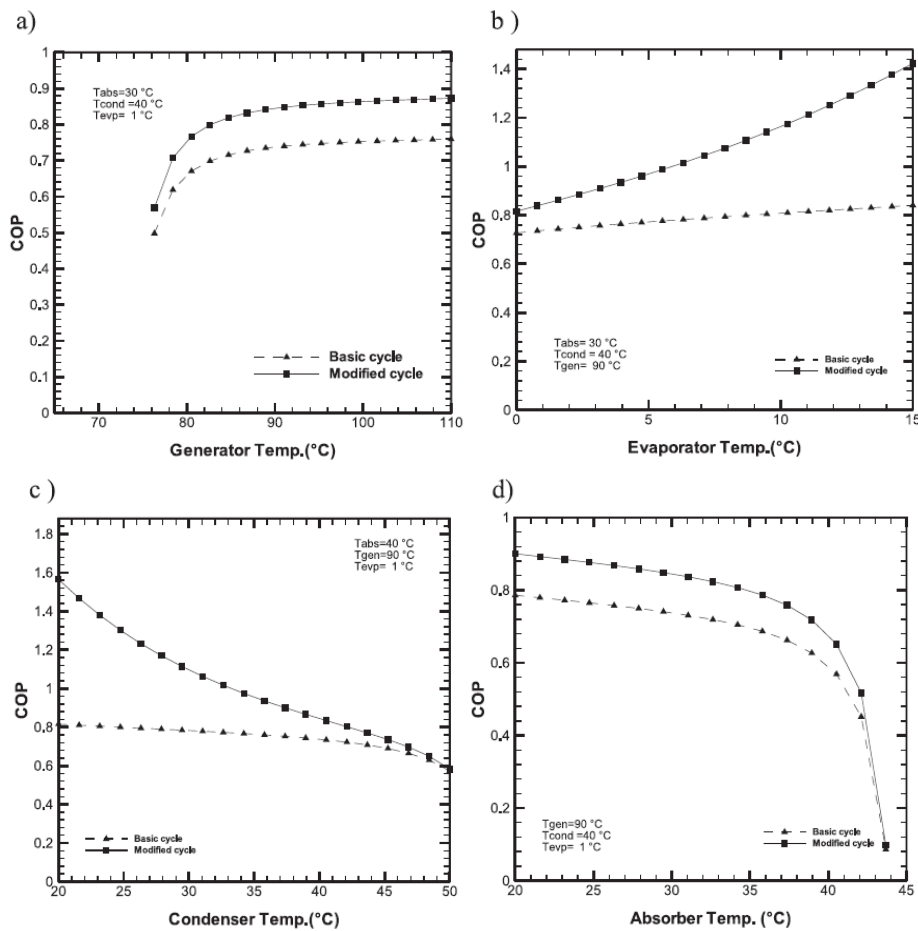
Component	Exergy loss rate (KW)
Generator	2.002
Condenser	0.048
Evaporator	0.685
Absorber	1.533
Pump	0.00015
Solution expansion valve	-----
Refrigerant expansion valve	0.013
Solution heat exchanger	0.093
Refrigerant heat exchanger	0.008
Total	4.383

### 4.1 Comparison Between the Basic and the Modified Cycle:

Figure. 2(a–d) depicts the COP of basic cycle and the modified absorption cycle under various operating temperatures. Comparison of COP values vs. generator temperature for (basic and modified cycles) are shown in Figure. 2(a). It is noticed that for the two absorption cycle, there is an optimum value of COP. This value of COP increases with generator temperature until it reaches the optimum value. This value depends on the type of the cycle. Moreover, there is a low generator temperature limit, where the cycle cannot operate at a generator lower than this. This is an important point for the utilization solar energy, since the fluid temperature for solar collector are generally below 100 °C. Therefore, the simulation exhibited in this figure reveals that modified cycle has a higher COP value than the basic cycle. The effect of evaporator



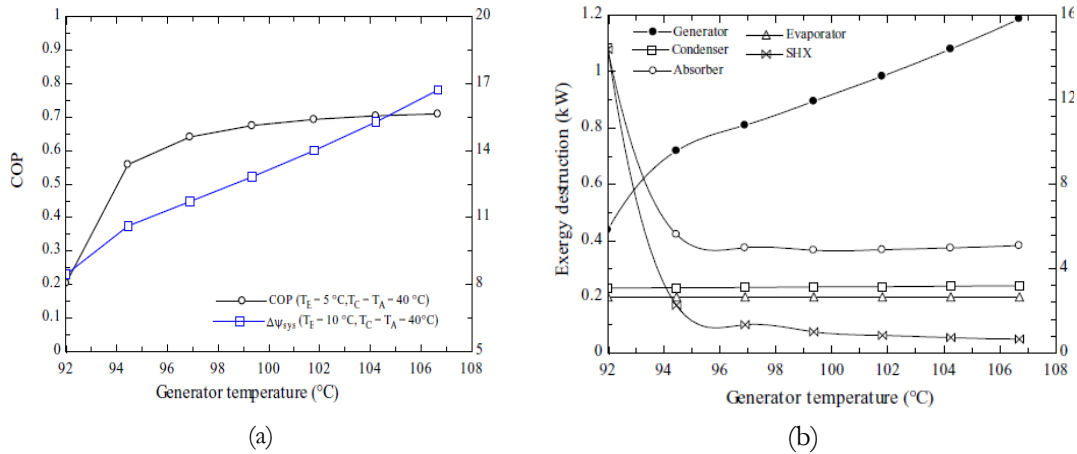
temperature on the COP of the cycles is shown in Figure. 2(b). It is evident that the COP value of two cycles increases as the evaporation temperature increases, as shown in Figure. 2(b). It also can be seen that the highest value of the COP is obtained from the modified cycle. Figure. 2(c and d). illustrate the comparison of COP value vs. condenser and absorber temperature for two cycles. Increase condenser and absorber temperature causes a decrease in COP for each cycle. For modified cycle operation under choked conditions, the cycle is independent of condenser pressure when the condenser pressure is less than a certain critical value; however, the cycle falls to similar values to the basic cycle when the condenser pressure is greater than this critical value, as shown in Figure. 2(c). It is observed that the absorber temperature of 44 °C and above the performance of the system tend to lean towards zero. This is due to the fact that the concentration of the liquid leaving the absorber has dropped to the same concentration as the liquid coming back from the generator. From this figure, it could be determined that the COP of the modified cycle is higher than that of the conventional cycles at all simulated working conditions. This is due to the fact that the performance of a refrigeration system is influenced by adding an ejector to the basic cycle.



**Figure 2:** (a–d) Comparison of influence of operation temperature on the COP values for basic and modified cycle

## 4.2 Effect of Generator Temperature:

Variation of COP and total exergy destruction rate of system with generator temperature is depicted in Figure.3.a. It is observed that COP of system increase with generator temperature up to certain value and then after very negligible variation is observed. With increasing generator temperature, strong solution concentration increases hence mass flow rate of weak and strong solution decreases while refrigerant mass flow rate remains unchanged. Enthalpy of superheated refrigerant stream marginally increases with generator temperature while enthalpy of weak and strong solution increases rapidly hence lowering generator heat load. Therefore, COP of system increases with generator temperature.



**Figure 3: a)** Variation of COP and total exergy destruction of system with generator temperature, **b)** Variation of exergy destruction of various component system with generator temperature

From Figure.3.a. it is found that with increasing generator temperature, total exergy destruction of system increases. This can be understood by Figure.3.b. It is observed that exergy destruction in condenser and generator increases with increasing generator temperature while in SHX and absorber it decreases. It is clear that with increasing generator temperature total exergy destruction of system increases rapidly so it is necessary to optimize the generator temperature for minimization of exergy destruction of system. From Figure.3.a it is observed that optimum generator temperature corresponding to minimization of exergy destruction rate found lower than that for maximization of COP.

## 5 Conclusion

The results show that COP of the cycle increases with increasing generator and evaporator temperatures but decreases with increasing condenser and absorber temperatures. Moreover, exergy losses in the expansion valves, pump and heat exchangers, especially in refrigerant heat exchanger, are very small fractions of the total exergy loss in the ARS. Three components that obtained the highest exergy loss are the generator, the absorber and the evaporator.

In the parametric analysis of the system, it is shown that a low condenser temperature yields a higher cooling COP and higher exergetic efficiency. The system operating with relatively high evaporator temperatures has better cooling COP and experiences smaller exergetic efficiency than the one having low evaporator temperatures. Increasing the generator temperature can improve the cooling COP of the absorption system, but as the heat source temperature further increases, the COP of the system levels off. This negative effect of increasing the heat source temperature is more dominant on the exergetic efficiency of the system. Consequently, this negative result on the exergetic efficiency and the COP removes the beneficial effect of a high heat source temperature.

In this study, an improved system of the single-stage absorption cycle operated with LiBr/H<sub>2</sub>O as its working fluid was conducted. Thermodynamic analysis of ejector-absorption refrigeration system has been carried out, and the theoretical performance of the cycles were compared. The results show that the modified combined absorption cycle is superior to the basic cycle over a wide range of operating conditions. The results also show that the thermal load of the condenser and the evaporator increase as the generator and evaporator temperature increases, while it decreases as the condenser temperature increases. The results indicated that the overall COPs increments of the modified cycle was 8–60 % at a condenser temperature of 25–45 °C, and by 30–85% at evaporator temperature of 0–10 °C over the basic cycle.

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