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High Influence of the Modified Ejector on Performance of a Solar Absorption Refrigration System (NH_3/H_2O)

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***Abstract*— Solar absorption cooling system powered by the sun is a very attractive topic for researchers of the solar thermal energy domain. In this paper, the novel design of an ejector was added to the basic cycle of a solar (ammonia-water) absorption cooling system. However, the potential kinetic energy of an ejector was affected to enhance refrigeration efficiency. Nevertheless, thermodynamic analysis was used to investigate the performance of working fluid pair (NH3/H2O). Moreover, the mathematical model is introduced to evaluate system performance, exergy loss of each component and whole system configurations. Thus, the results show that, the proposed cycle enhances cycle peak under all operating conditions. Therefore, the improvement in the modified cycle has enhanced the COP by 50–71 % under condenser temperature range, and by 26–43% under evaporator temperature range, in comparison with the basic cycle.**

Keywords— Solar energy, absorption, NH_3/H_2O, modified ejector, exergy, (C.O.P).

I. INTRODUCTION

Nowadays, research and development have been focused on utilization of renewable energy sources such as solar energy, wind energy, geothermal energy, hydropower energy, hydrogen energy and etc. Solar energy is one of these sources which is an energy source for cooling applications and has highly popular advantages such as, clean, free, direct, continuity, renewable and easy usability [1]. However, the prices of energy are increasing exponentially worldwide, due to high demand on the industrial refrigeration systems which is one of the most energy consuming sector. By producing solar absorption refrigeration system, leading to cutting down energy costs and have no impacts on ozone layer for preserving the environment. Therefore, solar cooling system attracts many attentions of researchers and engineers of solar energy to achieve very high level of performance in terms of modified system configuration [2]. This study refers to summarize the different working fluids of solar absorption cooling systems and adsorption cooling systems, providing various results with their advantages and limitations, though the coefficient of performance of absorption cooling systems is better than that of adsorption systems [3]. Hence, Absorption cooling systems are mature technologies that proved their abilities to

provide clean cooling with the use of low grade solar and waste heat. In this paper, it can be compared that, modeling and simulation study of a 70 kW Yazaki absorption cooling machine working with water-lithium bromide mixture. The influence of different parameters (Heat exchanger efficiency, Generator, absorber and condenser temperatures) on the system performance is performed. The simulation results showed that, the COP of the cycle increases with increasing of the generator and evaporator temperatures, while, it decreases with the increase of the condenser and absorber temperatures. The COP of the system reached its maximum value of 0.77 with a generator temperature of Tg = 92 °C that is close to the value given by the Yazaki manufacturer [4]. On the other hand, the modification of the absorption refrigerator machine functioning with LiBr-H2O, by determining energy and mass states of each cycle element supposed in the permanent system. It can be determined the COP of the solar single-stage absorption refrigerator as a function of the temperature in the different components of the cycle. Therefore, the dimension of the cycle was changed by introducing double-line heat exchanger (between generator-condenser and between evaporator-generator). In this study, it can be seen, the use of a double line heat exchanger improves the COP by 4 % as compared to its value using a single-line heat exchanger, for both cycles [5]. The aim of this work is to study the refrigeration absorption system which using ammonia-water pair purpose of determining its performance and suggest improvements. This improvement in the coefficient of performance "COP" was achieved by increasing the number of boilers and absorbers (three boilers, condenser, evaporator and two absorbers). The results showed that, the possibility of operating with temperatures vary around 70 °C and elimination of the distillation column which is explained by the reduction in the overall cost of the facility, which leads to improve COP of solar absorption refrigeration [6]. However, computer simulation model has been developed to predict the performance of solar absorption refrigeration system using different working fluid. The model is based on detailed mass, energy balance, heat and mass transfer for the cycle component. Detailed thermodynamic properties for ammonia- water, ammonia-lithium nitrate and ammonia-sodium thiocyanate are expressed in polynomial equations and used in cycle simulation. The performances of these three cycles against various generator, evaporator, and condenser temperatures are compared. The results show that the ammonia-lithium nitrate and ammonia-sodium thiocyanate cycles give better performance than the ammonia-water cycle. Therefore, it can be used as a source of reference for developing new cycles and searching for new working fluids pairs. They can also be used in selecting operating conditions for existing systems and achieving automatic control for maintain optimum operation of the system [7]. 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 $LiBr/H_2O$ as a working fluid. In this paper, improvement of the system is achieved by utilizing the potential kinetic energy of the ejector to enhance refrigeration efficiency. 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 conditions [8]. As mention above, recent and previous studies have indicated to improve (C.O.P) of a solar absorption cooling system in different configurations for several cycles which working with fluids pair, as well as attempt to facilitate design process. Therefore, the work of this study will be focused on the novel design of an installed ejector which was added to the basic cycle of a solar (NH_3/H_2O) absorption refrigeration system and it's highly impacts on (C.O.P) enhancement.

A. Aims and Objectives

This article focuses on the evaluation of system performance (C.O.P) of a solar (NH_3/H_2O) absorption refrigeration cycle, which is operated by a novel design of modified ejector to be added as one of the system configurations. On the other hand, the main objective of study will lead to achieve suitable size and weight of the system in comparison to other cycles or studies for COP enhancement.

II. SYSTEM DISCRIBTION

A. Discribtion of a solar absorption cycle

According to the solution regeneration and thermal operation cycle, the absorption systems can be divided into three categories: single-effect, half-effect and double-effect solar absorption cycles. The single-effect and half-effect chillers require lower temperatures with respect to a double effect-chiller [4]. There are also two other absorption refrigeration systems DAR (diffusion absorption refrigeration) and (hybrid systems) that can achieve better performance. Hence, the most commonly one is a single effect absorption cycle.

B. Discription of a single effect solar absorption cycle

The main components of the cycle are generator, absorber, condenser, evaporator, heat exchanger, circulating pumps and solar collector. The cycle operating under two pressure levels: low pressure at the evaporator-absorber and high pressure at condenser-generator. The cycle commences working at the absorber. The absorber receives the vapor-refrigerant from the refrigerator and creates a rich-mixture. The pump discharges this mixture to the generator or the high-pressure zone. In the generator, the working fluid (NH_3/H_2O) separates from the absorbent by the heat generated via the solar collector. Using a pressure-relief valve, the weak-solution then returns to the absorber. A SHX (solution heat-exchanger) is prepared to recover the internal heat.

C. Description of Modified single effect solar absorption refrigeration cycle

Figure.1 demonstrates the main components of the single effect absorption cooling system. The solution heat exchanger is a crucial part of the cycle, due to highly effect for cooling down the working fluid pair approaching from the generator, which therefore heats up the working fluid following to the generator. An ejector was added and installed between the generator and the condenser. The primary high-pressure water vapor 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, and then entering the condenser. In this novel design, the ammonia-water mixture leaves the absorber (state 1) in the phase of a saturated solution at low pressure. The flow continuously pumps to the system at high pressure (state 3). The generator works from a high temperature source to separate the binary solution of water and Ammonia (strong solution comes from absorber). Thus, two-phase mixture are separated, and the weak liquid flows through SHE (state 4 to state 5), then transferred to the low pressure system and sprayed into the absorber (state 6). On the ejector, the secondary flow (water vapor from evaporator (state 13A)) and the primary flow of water vapor from the generator are mixed and passed to the condenser (state 10).

III. ASSUMPTIONS FOR ANALYSIS OF MODIFIED CYCLE

- The system operates under a steady state condition.
- The refrigerant leaving the condenser and evaporator is saturated (state points 11 and 13).
- NH_3/H_2O solution at the generator, heat exchanger and the absorber are assumed to be in the equilibrium state at their respective pressure and temperature to be 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 9 and 13B in Figure. 1a respectively). The velocity of the mixed flow leaving the ejector (at state 8) is also neglected.

IV. THERMODYNAMIC ANALYSIS

The thermodynamic analysis mainly aimed at assessing the thermodynamic imperfections and suggested possible ways of improving these imperfections. Here the system is analyzed 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 having inlet and outlet flow, work interactions and heat transfer. For analyzing ammonia-water 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.

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

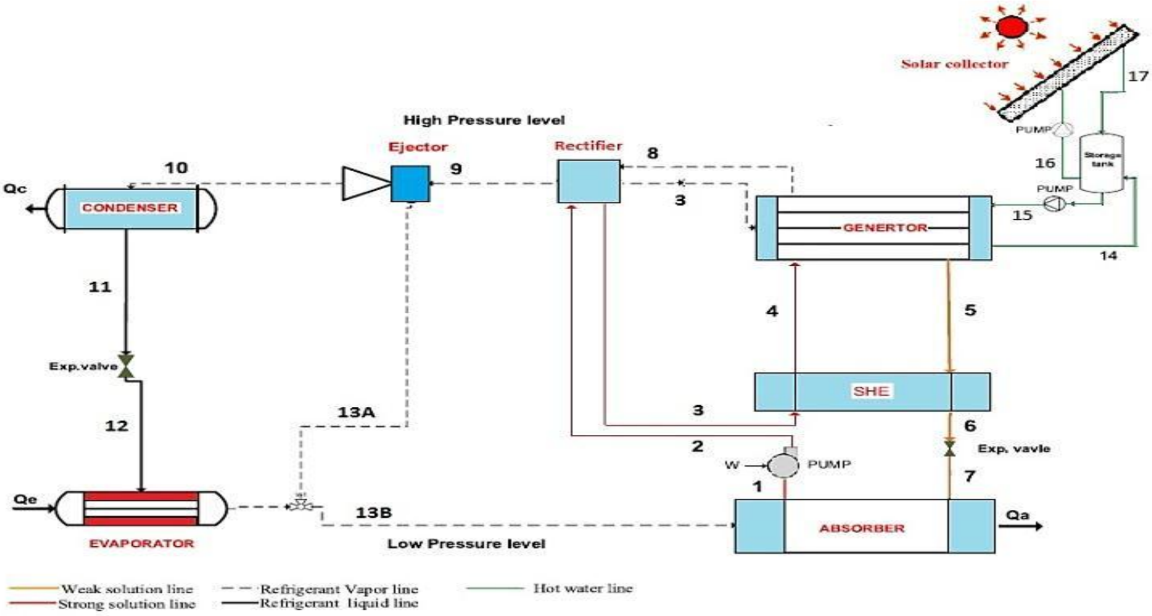

Fig. 1. System schematic of combined ejector absorption cooling system.

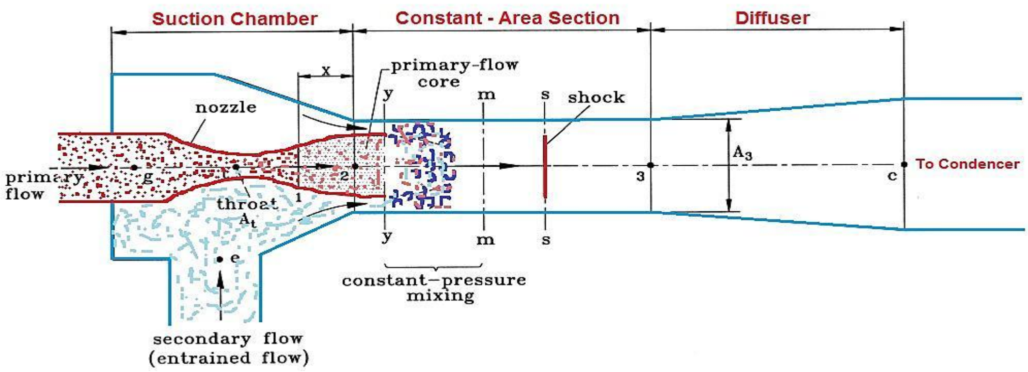

Fig. 2. Schematic diagram of the ejector.

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.

1) For the Generator – SHE – Absorber Loop:

The mass and energy balances around the generator

$$\dot{m}_4 + \dot{m}_3 = \dot{m}_5 + \dot{m}_8 \rightarrow \dot{m}_3 x_3 + \dot{m}_4 x_4 = \dot{m}_5 x_5 + \dot{m}_8 x_8 \tag{3}$$

Where: $x_4 = x_1$, $x_5 = x_7$

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 ϵ_{she}.

$$Q_{gen} = \dot{m}_5 h_5 + \dot{m}_8 h_8 - \dot{m}_3 h_3 - \dot{m}_4 h_4 \tag{4}$$

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

The liquid weak solution at state (4): $T_5 = T_{gen}, P_5 = P_{gen}$

Exergy balance formulation for generator can be wr itten as:

$$\Delta E_G = -\dot{m}_{ws}(h_4 - T_0 s_4) + \dot{m}_{ss}(h_5 - T_0 s_5) - \dot{m}_r(h_3 - T_0 s_3) + \dot{m}_r(h_8 - T_0 s_8) + \dot{m}_G(h_{14} - T_0 s_{14}) - \dot{m}_G(h_{15} - T_0 s_{15}) \tag{5}$$

$$\epsilon_{she} = \frac{T_5 - T_6}{T_5 - T_3} \tag{6}$$

$$C_{hot} = \dot{m}_5 \left(\frac{h_5 - h_6}{T_5 - T_6} \right), \quad C_{cold} = \dot{m}_2 \left(\frac{h_4 - h_3}{T_4 - T_3} \right) \tag{7}$$

$$Q_{hx} = \dot{m}_3 (h_4 - h_3) \quad , \quad Q_{hx} = \dot{m}_5 (h_5 - h_6) \quad (8)$$

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:

$$\Delta E_{SHX} = \dot{m}_{ws}(h_3 - T_0 s_3) - \dot{m}_{ws}(h_4 - T_0 s_4) + \frac{Q_{SHX}}{T_b} \quad (9)$$

3) Rectifier:

$$\dot{m}_2 + \dot{m}_8 = \dot{m}_3 + \dot{m}_{3''} \quad (10)$$

$$\dot{m}_2 x_2 + \dot{m}_8 x_8 = \dot{m}_3 x_3 + \dot{m}_{3''} x_{3''} \quad (11)$$

$$\Delta E_{Rect} = \dot{m}_{ss}(h_2 - T_0 s_2) + \dot{m}_8(h_8 - T_0 s_8) - \dot{m}_{3''}(h_{3''} - T_0 s_{3''}) - \dot{m}_3(h_3 - T_0 s_3) \quad (12)$$

4) Solution Expansion Valve Model:

Exergy balance formulation for expansion valve can be written as:

$$\Delta E_{REXP} = \dot{m}_{ss}(h_6 - T_0 s_6) - \dot{m}_{ss}(h_7 - T_0 s_7) \quad (13)$$

$$\Delta E_{SEXP} = \dot{m}_{ss}(h_{11} - T_0 s_{11}) - \dot{m}_{ss}(h_{12} - T_0 s_{12}) \quad (14)$$

where: $h_6 = h_7$, $\dot{m}_6 = \dot{m}_7$, $x_6 = x_7$

5) Pump Calculation:

$$h_2 = h_1 + \frac{W_{pump}}{\dot{m}_1} \quad (15)$$

$$W_{pump} = \dot{m}_6 v_1 \frac{P_{hight} - P_{low}}{1000} \quad (16)$$

Exergy balance formulation for pump can be written as:

$$\Delta E_{pump} = \dot{m}_{ws}(h_1 - T_0 s_1) - \dot{m}_{ws}(h_2 - T_0 s_2) + \dot{W}_P \quad (17)$$

6) Absorber:

$$Q_{abs} = \dot{m}_{13} h_{13} + \dot{m}_7 h_7 - \dot{m}_1 h_1 \quad (18)$$

Exergy balance formulation for absorber can be written as:

$$\Delta E_{abs} = \dot{m}_r(h_{13} - T_0 s_{13}) + \dot{m}_{ss}(h_7 - T_0 s_7) - \dot{m}_{ws}(h_1 - T_0 s_1) + \frac{Q_{abs}}{T_b} \quad (19)$$

7) Condenser:

Exergy balance formulation for condenser can be written as:

$$Q_{cond} = \dot{m}_{10}(h_{10} - h_{11}) \quad (20)$$

$$\Delta E_{cond} = \dot{m}_r(h_{10} - T_0 s_{10}) - \dot{m}_r(h_{11} - T_0 s_{11}) + \frac{Q_{cond}}{T_b} \quad (21)$$

8) Refrigerant Valve:

$$h_{11} = h_{12} \quad , \quad x_{11} = x_{12}$$

9) Evaporator:

$$Q_{eva} = \dot{m}_{12}(h_{13} - h_{12}) \quad (22)$$

Exergy balance formulation for evaporator can be written as:

$$\Delta E_{eva} = \dot{m}_r(h_{12} - T_0 s_{12}) - \dot{m}_r(h_{13} - T_0 s_{13}) + \frac{Q_{eva}}{T_b} \quad (23)$$

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

$$\Delta E_{Sys} = \Delta E_G + \Delta E_{abs} + \Delta E_{eva} + \Delta E_{cond} + \Delta E_{SHX} + \Delta E_{REXP} + \Delta E_{SEXP} + \Delta E_P \quad (24)$$

10) Ejector analysis:

A one-dimensional mathematical model for prediction of ejector performance of Huang et al [10], is used to analyze the effect of mixing chamber and entrainment ratio on the performance of system. The schematic diagram of the ejector used in the present study is shown in Fig.2.

11) Nozzle equations:

$$A_t = \frac{\dot{m}_p \sqrt{T_{gen}}}{P_{gen}\sqrt{\eta_P}\sqrt{\frac{k}{R}\left[\frac{2}{K+1}\right]^{\frac{K+1}{K-1}}}} \quad (25)$$

Where, η_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 Mp1 and the exit cross section area AP1 and pressure Pp1 are, using isentropic relations as showing in the following equations:

$$\frac{A_{p1}}{A_t} = \frac{1}{M_{p1}}\left[\frac{2}{k+1}\left(1 + \frac{k-1}{2} M_{p1^2}\right)\right]^{\frac{K+1}{K-1}} \quad (26)$$

$$\frac{P_g}{P_{p1}} = \left[1 + \frac{k-1}{2} M_{p1^2}\right]^{\frac{K}{K-1}} \quad (27)$$

12) Mixing section equations

The Mach number Mpy of the primary flow at the y–y section follows the isentropic relations as an approximation

$$\frac{P_{Py}}{P_{p1}} = \frac{\left[1 + \frac{k-1}{2} M_{p1^2}\right]^{\frac{K}{K-1}}}{\left[1 + \frac{k-1}{2} M_{py^2}\right]^{\frac{K}{K-1}}} \quad (28)$$

The entrained flow reaches chocking condition at the y–y section, i.e. M_{sy} =1. Similarity to the primary nozzle, the equivalent form of Mach number the secondary fluid at the nozzle exit plane is given as:

$$\frac{P_{low}}{P_{sy}} = \left[1 + \frac{k-1}{2} M_{sy^2}\right]^{\frac{K}{K-1}} \quad (29)$$

The entrained flow rate at choking condition follows

$$m_s = \frac{A_{sy} \times P_{low}}{\sqrt{T_{low}}} \sqrt{\frac{K}{R}\left[\frac{2}{k+1}\right]^{\frac{K+1}{K-1}}} \times \sqrt{\beta_s} \quad (30)$$

where β_s is the coefficient related to the isentropic efficiency of the entrained flow.

The geometrical cross sectional area at section y–y is A_3 that is the sum of the areas for the primary flow A_{Py} and for the entrained flow A_{sy}. That is,

$$A_{py} + A_{sy} = A_3 \quad (31)$$

The temperature and the Mach number of the two streams at section y–y can be derived as;

$$\frac{T_2}{T_{py}} = 1 + \frac{k-1}{2} M_{py^2} \tag{32}$$

$$\frac{T_{low}}{T_{sy}} = 1 + \frac{k-1}{2} M_{sy^2} \tag{33}$$

13) Mixed flow across the shock from section m–m to section 3–3

A sharp pressure rise occur at section s–s due to a supersonic shock will take place at this section in the constant area mixing chamber. Assuming that the mixed flow after the shock undergoing an isentropic process, the flow inside the constant area mixing chamber between the section m–m and section 3-3 has a uniform pressure P_3. Therefore, the pressure ratio across the mixing chamber can be obtained from:

$$\frac{P_3}{P_m} = 1 + \frac{2k}{k+1} (= M_m^2 - 1) \tag{34}$$

$$M_m^2 = \frac{1 + \frac{k-1}{2} M_m^2}{k M_m^2 - \frac{k-1}{2}} \tag{35}$$

14) Diffuser equations

The pressure at the exit of the diffuser follows the relation, assuming isentropic process

$$\frac{P_c}{P_3} = \left[1 + \frac{k-1}{2} M_3^2\right]^{\frac{K}{K-1}} \tag{36}$$

V. RESULTS AND DISCUSSTION

Figures 3-6 show the effect of the operating temperatures on the COP of basic cycle and the modified absorption cycle. The effect of generator temperature on COP for (basic and modified cycles) is shown in Figure 3. It can be seen 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 minimum generating temperature above which the operation of the cycle is possible. This temperature is called cut in/cut off temperature, whereas the cycle incapable to operate under this level of generator temperature. This is an important point for the utilization solar energy, since the fluid temperature for solar collector is generally below 100 °C. Therefore, the COP for the modified cycle is higher than that of the basic cycle. The effect of evaporator temperature is shown in Figure 4. In general, the COP values changed from 0.33 to 0.58 when the evaporator temperature was varied between 3 and 16°C. If the evaporator temperature rises, the concentration of the weak solution increase whiles the circulation ratio decrease. They cause a decrease in both generator and absorber thermal load. Thus, the COP increases almost linearly with evaporator temperature. It also can be seen that the highest value of the COP is obtained from the modified cycle. The effect of condenser temperature on the COP is shown in Figure 5. The COP values decrease with increasing condenser temperature. It can be seen that the maximum COP of the cycle in the order of 0.53 when the combined ejector is added. Figure 6 illustrates the comparison of COP value vs. absorber temperature for two cycles. Thus, increasing of absorber temperature causes reduction of COP for each cycle.

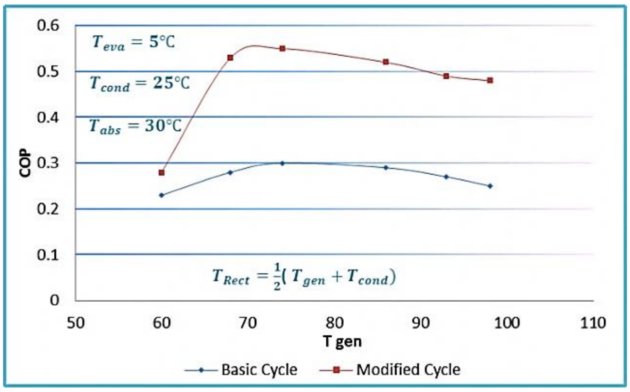

Fig. 3. Variation of COP with generator temperature.

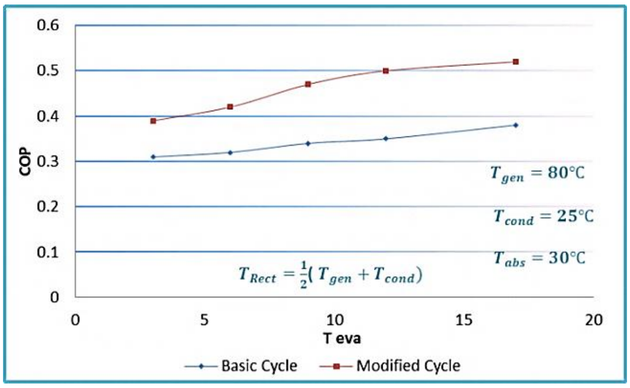

Fig. 4. Variation of COP with evaporator temperature.

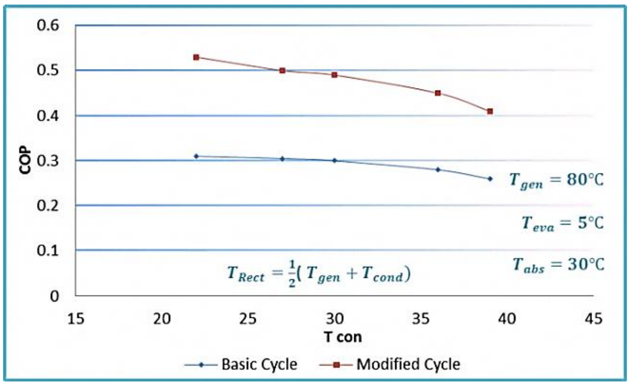

Fig. 5. Variation of COP with condenser temperature.

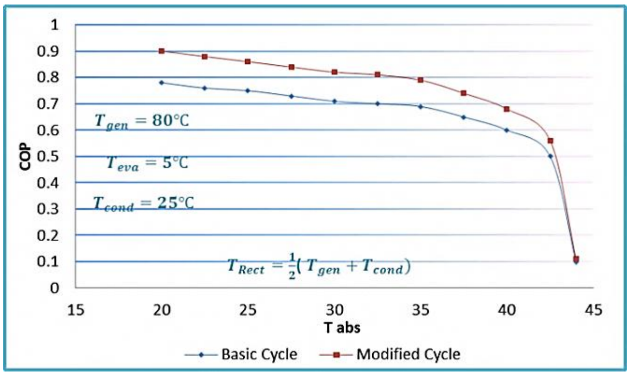

Fig. 6. Variation of COP with condenser temperature.

Variation of COP and total exergy destruction rate of system with generator temperature depicts in Figure 7. It is observed that COP of system increases with generator temperature up to certain value and then very negligible variation is observed. Also, it is found that with increasing generator temperature, total exergy destruction of system increases, so it is necessary to optimize the generator temperature for minimization of exergy destruction of system and maximization of COP.

VI. EVALUATE SYSTEM PERFORMANCE AT DIFFERENT OPERATING TEMPERATURE

The temperatures of the components that evaluate system performance falls within the following ranges 60-100°C for the generator temperature;20-40°C for water-cooled condensers; 2-17°C for evaporators; and 20-45 for the absorber temperature. Figures 3-6 illustrates the comparison between the modified and the basic cycle at different operating temperatures. The proposed cycle enhances cycle peak under all operating conditions, therefore, the improvement in the modified cycle increases the COP by 50–71 % under condenser temperature range, and by 26–43% under evaporator temperature range over the basic cycle.

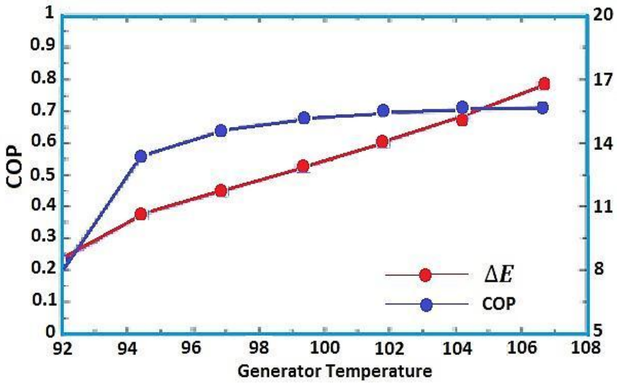

Fig. 7. Variation of COP and total exergy destruction of system with generator temperature.

VII. CONCLUSION

In this paper, results related to both basic and modified absorption refrigeration system driven by solar energy and operated with Ammonia/Water as its working fluid was conducted. The effects of operating temperatures are investigated. Thermodynamic analysis of ejector–absorption refrigeration system has been carried out, and the theoretical performance of the cycles was compared. The results show that the modified cycle provides potentially high COP than that of basic cycle over a wide range of operating temperatures. It also found that a solar absorption system can work only if the solar collectors outlet temperature is higher than the cut in/cut off temperature, and there exists an optimal value for this temperature at which a maximum value of the COP is obtained. The results indicated that the overall COPs increments of the modified cycle was 50-71 % at a condenser temperature of 20-40 °C, and by 26-43% at evaporator temperature of 2-17 °C over the basic cycle.

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