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*Research Article*

# Performance Investigation of Ejector Assisted Power Cooling Absorption Cycle

*B.Mebarki

Laboratory of Built Environment, Department of Structures and Materials, Faculty of Civil Engineering, University of Science and Technology Houari Boumediene (USTHB), Bab Ezzouar, Alger, Algeria
E-mail: mebarkibillel88@gmail.com, billal.mebarki@usthb.edu.dz

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**Abstract**

In this paper, new cycle is developed to generate simultaneously electrical and cooling power by placing a turbine between the generator and ejector in the conventional ejector-assisted absorption cooling cycle. The aim of developed cycle is to increase the exergy efficiency of cycle by adding an electrical power generation made it more environmentally friendly and reduce its dependents of fossil energy sources. The first, second laws of thermodynamic, mass and energy balance is applied for each cycle component and the constant mixing pressure ejector model is used to develop a numerical model of proposed cycle. The results depict that the augmentation of generation temperature is positively affected the work produced in the turbine contrary for cycle coefficient of performance, for every working conditions there are a certain value of generation temperature which its exergy performance of cycle achieves the maximum, the augmentation of output pressure of turbine is positively affected the cycle coefficient of performance contrary of the work produced in the turbine and the cycle exergy efficiency and the augmentation of condensation temperature is positively affected the cycle exergy efficiency and the work produced in the turbine contrary for cycle coefficient of performance and the augmentation of evaporation temperature is positively affected the cycle coefficient of performance and the cycle exergy efficiency contrary for the work produced in the turbine The results also show that the improvement of exergy efficiency of proposed cycle is 29.41% and 46% compared with the absorption cooling cycle with double and triple effect under the same operating conditions.

***Keywords:*** *Absorption; ejector; COP; electrical power; cooling.*

## 1. Introduction

The friendly environmentally cycle needs more independent of conventional energy source as like fossil source, the idea is to develop a cooling cycle which simultaneously can be generated an electrical power using water vapor high pressure outlet of generator of conventional ejector assisted absorption cycle. The new cycle generates electrical energy and cooling simultaneously, which is expected to improve the poor exergy efficiency of the absorption cooling cycle [1]. The new proposed cycle is based on placing a turbine between the generator and the ejector in the conventional ejector-assisted absorption cycle to exploit the maximum pressure difference between the pressure of the steam leaving the generator and the pressure of the ejector's primary fluid entering the ejector, which can operate at a pressure below the generator pressure

In recent years, many works have proposed some cycles to dispute the irreversibility of power generation from low-grade heat source or waste heat. Meng et al [2] investigated the performance of a modified organic Rankine cycle combined with a flash process using a thermodynamic and techno-economic study. The results depict that the evaporator and the condenser are mainly responsible for the irreversibility of the cycle and the use of flash tank process reduces the irreversibility Cihan and Kavasoğulları [3] proposed a new organic Rankine cycle which produced both cooling and electrical power using an ejector and two turbine. The energy and exergy performance of proposed cycle working with various organic fluids R123, R600, R245fa, R141b, and R600a were analyzed under different evaporation and condensation temperature. The results show that the R141b is the more appropriate fluid of proposed cycle in point of view energy and exergy analysis.

Li et al [4] investigated an ejector assisted organic Rankine flash cycle using the first and second thermodynamic laws. The results show that the ejector-assisted organic Rankine flash cycle has better exergy and thermal efficiency than the conventional Rankine flash cycle in every working condition. Mendal et al [5] modified a conventional waste heat driven organic flash power cycle by replacing the low pressure throttle valve with an ejector. They found that the cooling performance is positively affected as the pressure in the flash tank increases, and the proposed cycle thermal efficiency is enhanced compared with conventional organic flash power plant. Mendel at al [6] proposed a compressor-ejector refrigeration cycle based on adding an ejector between the compressor and the condenser. The results show that there is a pressure ratio of the compressor corresponding to a maximum value of the coefficient of performance of the cycle under all operating conditions of the cycle. The coefficient of performance of the conventional ejector cycle and the developed ejector-compressor cycle can be improved by 25.7% and 37.2%, respectively, compared to the ejector-expansion cycle operating with R32 and R1234yf. Mendel et al [7] proposed a geothermal flash steam cycle with ejector support. The main modification of the proposed cycle is that the saturated water from the high pressure steam separator enters the ejector as the primary fluid compared to the dual flash steam cycle. The results show that the proposed cycle performance can be improved by 6.67% compared to the dual flash steam cycle. The cost is also 4.5% lower.

The use of the ejector to improve the exergy efficiency and coefficient of performance of the absorption cooling cycle has been discussed in a few papers. Azhar et al [8] proposed three various cycle of a dual ejector assisted absorption cycle using flash tank working with solar energy; the approach of their study is techno economic. They found that the cycle proposed N°3 is more 10 % cheaper then cycle proposed N°2 and 27% cheaper then cycle proposed N°2. They found also that the payback period and profit gain of cycle proposed N°3 against the conventional dual ejector assisted absorption cycle with flash tank are 16 years and 874 $. Hai et al [9] proposed and optimized a new bi evaporator cycle combined an absorption cycle and $CO_2$ ejector cycle. They found that the new cycle is better performance than the conventional $CO_2$ ejector cycle and the gas cooler. They concluded also that the generator is the main responsibility of irreversibility of cycle. Khalili et al [10] proposed a new double ejector multi pressure level absorption cycle with two schemes. The results depict that the proposal cycles developed cycle coefficient of performance by 108.35 % and 33.11 % comparing with conventional absorption cycle and the lower evaporation temperature can be reach -24°C. Al Hamed et al [11] proposed an ejector assisted absorption cycle using solar energy and geothermal energy for a small community. They found that the exergy efficiency of integrated system with two source of renewable energy reaches 55.98 % and the system coefficient of performance is equal to 63.60 %. Göktun [12] studied the effect of adding an ejector to simple effect absorption cooling cycle on its performance using the first law of thermodynamics. The results show that a growth up to 40% in the coefficient of performance of the ejector assisted absorption cycle can be achieved compared to the ordinary absorption cooling cycle. Majdi [13] designed an absorption refrigeration cycle combined with ejector which is placed between the generator and condenser to direct part of the steam coming from the evaporator to the condenser. He found that the coefficient of performance of the proposed cycle could be improved by 60% compared to the simple effect absorption cycle. Jiang et al [14] compared the ejector assisted absorption refrigeration cycle with the double effect absorption cooling cycle. They concluded that the performance coefficient of conventional double effect is slightly higher than that of the proposed cycle, which its annual operating cost is lower than that of the conventional double effect absorption cycle. Sirwan et al [15] developed a new ejector assisted absorption cooling cycle with flash tank. They found that most of the exergy destruction occurs in the evaporator, condenser and absorber, respectively. Sözen and Özalp [16] proposed ammonia absorption refrigeration cycle using liquid ejector which is placed between the generator and the absorber. They found that the coefficient of performance and exergy efficiency of proposed cycle improved by 49% and 56%, respectively, compared to a conventional absorption refrigeration cycle. Abed et al [17] developed an electrical plant using geothermal source combined with a single stage $CO_2$ trans-critical cycle. They concluded that the proposed cycle reaches an. exergy efficiency equal to 32% in the usual operating conditions rising to 39.21% with the Genetic Algorithm and 36.16% with the Nelder–Mead simplex method. Yi et al [18] developed an ejector absorption cycle before expansion to generate electric power using ocean thermal energy. They concluded that the work of the output turbine can be improved by 79% by using the pre-expansion process in the cycle. Bhowmick et al [19] can be proposed an ejector assisted absorption cooling cycle with regenerative Rankine cycle for both cooling and power generation using waste heat sources. The results show that the exergy efficiency of proposed cycle can be achieved 44.18%. Khalili et al [20] proposed a double liquid vapor ejector absorption cycle; the two ejectors are installed between the absorber and condenser. They found that the coefficient of performance and exergy efficiency of the proposed cycle can be improved by 108.35%, 33.11% and 31.61%, 46.62% compared to the basic absorption cycle and to the ejector assisted absorption cycle, respectively. Sioud et al [21] developed a dual effect ejector assisted absorption cycle. They concluded that the coefficient of performance of the cycle can be equaled 1.7 at a generation temperature equal to 340 °C. Vereda et al [22] proposed ejector assisted absorption cycle; the ejector is placed between the solution heat exchanger and the absorber to direct the refrigerant vapor coming from the evaporator with the rich solution coming from the generator. They concluded that the coefficient of performance of the proposed cycle is the same coefficient of performance as a single-effect absorption cycle with higher generation temperature of 9°C.Therefore; it can be used with a low grade temperature source.

The enhancement of ejector process is subject of many studies. Tang et al [23] developed a novel physical model of the mixing chamber of an ejector and concluded that the conventional laws of chamber mixing between primary ejector fluid and secondary ejector fluid do not accurately represent the mixing process, which is also true for the energy and momentum laws. Therefore, a complete understanding of the mixing process was presented in their work. Thongtip and Aphornratana [24] conducted an experimental study to determine the effects of nozzle chamber geometry on ejector performance. They concluded that for specific working conditions, there are certain geometry is required to achieve maximum performance, e.g., for low temperature generation, the use of a nozzle with the largest throat area is recommended to achieve maximum performance. Wang et al [25] performed a simulation of the primary nozzle with different geometries and surface roughness. They concluded that the entrainment ratio is high depending on the geometry of the primary nozzle and the surface roughness. Van Nguyen et al [26] performed an experimental and numerical study on an ejector with variable nozzle geometries. They concluded that a 24% improvement in the coefficient of performance can be achieved with using an ejector with variable nozzle geometries.

The ejector that has been used is various applications like refrigeration, lubricant and desalination [27], and it is the subject of many researchers which have focused on ejector modeling. Kavasoğulları et al [28] developed a dual ejector refrigeration cycle by adding second ejector and refrigeration pump to conventional ejector assisted compression refrigeration cycle. They found that the coefficient of performance and exergy efficiency of proposed cycle can be achieved 7.52 and 38.8%, respectively and the improvement in the coefficient of performance and exergy efficiency are significantly in the high condensing and low evaporation temperature Varga et al. [29] studied the effect of many design parameters on the performance of ejector like nozzle exit position, constant area length and the ratio of the area between the nozzle and the constant area. They concluded that for every working condition there is an optimal ratio which had the maximum performance of ejector. Sriveerakul et al. [30] developed a CFD model of ejector in refrigeration cycle, the model developed is validated by experimental data. They concluded that the CFD is an efficient method to represent the flow inside the ejector. Ariafar [31] used CFD method to study the effect of exit diameter of primary nozzle. He concluded that the increasing of motive pressure will increase the entrainment ratio and the increasing of Mach number in the exit of nozzle chamber and its area does not affect the ejector performance but it will affect the pressure distribution at outlet nozzle chamber.

Li et al. [32] developed a new ejector assisted high temperature compressor refrigeration cycle. They studied the effect of many ejector parameters like throat area and primary nozzle chamber length and the output area of nozzle chamber. They found that the new cycle using ejector could be reduced the cost of refrigeration machine from 59 795$ to 3311 $, thus an economical cost can be reach 94 % and the temperature outlet of compressor reduced from 184.8°C to 110 °C in comparing with conventional high temperature compressor refrigeration cycle. Tashtoush and Nayfeh [33] compared both constant and variable geometry ejector as the expansion component in the compression refrigeration cycle. They found that the variable geometry ejector is more suitable ejector for solar refrigeration cycle because of variation of its working condition.

However, the ejector design can be divided into two designs according to the existing nozzle chamber, if there is a constant area, therefore, the ejector is called the constant area ejector, if not, the ejector with constant pressure ejector is called Huang et al [34]. The constant pressure ejector theory developed by Keenan et al [35] is mostly used to modulate the pressure in the mixing chamber of the ejector because it has high stability and performance compared to the constant area ejector [26]. In this study, the ejector is a constant mixing pressure ejector to maximize the performance, and the model used is one-dimensional as it is very often used in thermodynamic studies of the cycle [36].

This study based on the use of high pressure of generator to rotate a turbine for electrical power generation and an ejector to entrain the water vapor coming from evaporator with water vapor coming from turbine. The aim of this paper is to investigate the performance of proposed cycle and the effect of various design parameters on its performance.

## 2. Cycle Description

The proposed ejector absorption cycle is shown in Figure 1. The refrigerant/absorbent pair used in this study is water/lithium bromide. The objective of this study is to thermodynamically analyze the proposed cycle. Our proposed cycle consists of a generator, a condenser, a turbine, an evaporator, a heat exchanger, an expansion valve, a reducing valve, an ejector, and a solution pump. In the evaporator, the refrigerant water is converted into steam, which then splits into two parts. The first part of the vapor enters the absorber and then meets the strong solution coming from the generator and passes through the heat exchanger and the reducing valve. The refrigerant out letting the evaporator enters into absorber and reaches the weak solution coming from reducing valve to produce strong solution. As the amount of lithium bromide dissolved in the water increases, the temperature of the solution decreases and leaves the heat as a condenser. The weak solution leaves the absorber and enters the pump, where its pressure increases to the pressure of the generator. Then the steam leaves the generator at high pressure and enters the turbine, where it expands to an intermediate pressure between the pressure of the generator and the pressure of the condenser. The expanded steam leaves the turbine and enters the ejector as the primary fluid, where it meets the second part of the steam leaving the evaporator in the mixing part of the ejector as the second fluid of the ejector, then the steam leaves the ejector under condenser pressure, enters the condenser and returns to the evaporator through the expansion valve to close the cycle.

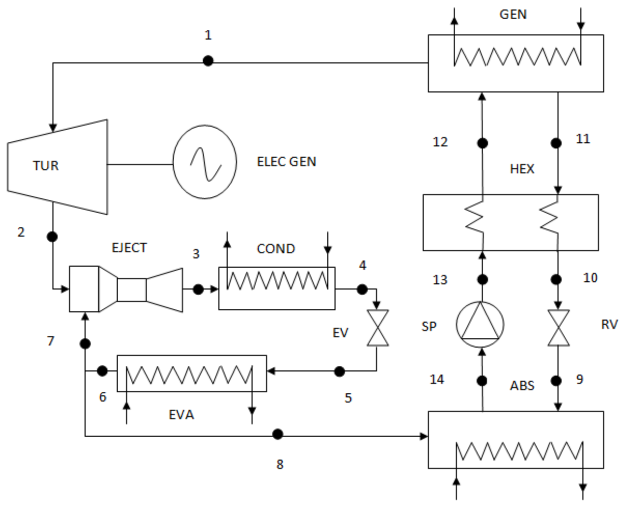


*Figure 1. The proposed cycle of ejector assisted absorption cycle.*

Where TUR is the turbine, ELEC GEN is the electrical generator, EJECT is the ejector, COND is the condenser, EVA is the evaporator, ABS is the absorber, HEX is the solution heat exchanger, GEN is the generator, SP is the solution pump, EV is the expansion valve and RV is the reducing valve.

In The state points 1, 2, 3, 6, 7 and 8, the state of fluid is water vapor but in the state point 4, the state of fluid is saturate liquid which became mixture (liquid vapor) in the state point 5. In the state points 9, 10 and 11, the solution of lithium bromide water is poor solution but became rich solution in the state points 12, 13 and 14.

## 3. Thermodynamic Cycle Model

### 3.1Ejector Model

The ejector model used in our study is a one dimensional model constant mixing pressure based on the following simplicity assumptions:

- An adiabatic one dimensional flow in the ejector [37].
- The velocities of primary and secondly fluid in the inlet and outlet of ejector are neglected [37].
- The efficiency of the nozzle, mixing chamber and diffusers are assumed to be constant [37].
- All losses of refrigerants flow are measured by using of different efficiency chambers [37].

Applying the mass, energy and momentum conservation in the different chambers of ejector and respect the above assumptions. The different parameters of ejector model are defined.

The velocity of primary fluid in the outlet of nozzle chamber is presented in the following equation by using of energy conservation and neglected of its velocity in the inlet of the nozzle chamber [37]:

$$U_{n,out}=\sqrt{\eta_n.\left(h_{p,in}-h_{n,out,is}\right).1000} \tag{1}$$

Where U is the velocity of primary fluid, $h_{p,in}$ is the enthalpy of primary fluid to nozzle chamber, $\eta_n$ is the nozzle chamber isentropic efficiency, in is the inlet of the nozzle chamber, out is the outlet of the nozzle chamber and the is the isentropic expansion.

Applying the momentum and the energy conservation in the mixing chamber and the neglecting of the secondly fluid velocity, the velocity and the enthalpy of mixing fluid in the outlet of mixing chamber are presented by following equations [37]:

$$U_{m,out}=\frac{U_{n,out}}{1+\mu}\sqrt{\eta_m} \tag{2}$$

$$h_{m,out}=\frac{h_{n,in}+\mu.h_{s,in}}{1+\mu}-\left(\frac{U_{n,out}{}^2}{2}\right)/1000 \tag{3}$$

Where $h_{s,in}$ is the enthalpy of secondly fluid in the inlet of the nozzle chamber, $\eta_m$ is the mixing chamber efficiency and the $\mu$ depicts the entrainment ratio where is defined by the following equation [37]:

$$\mu=\frac{m_p}{m_s} \tag{4}$$

Applying of energy conservation in the diffuser and assuming that the compression in the diffuser chamber is isentropic, the enthalpy of outlet of diffuser and the ejector can be presented by following equation [37]:

$$h_{d,out}=h_{m,out}+\frac{h_{d,out,is}-h_{m,out}}{\eta_d} \tag{5}$$

Where the $\eta_d$ is the diffuser efficiency.

Based on the previous equations the entrainment ratio can be calculated by following equation [37]:

$$\mu=\sqrt{\eta_n.\eta_m.\eta_d\frac{h_{p,in}-h_{n,out,is}}{h_{d,out,is}-h_{m,out}}}-1 \tag{6}$$

### 3.2Thermodynamic Model

The energy and exergy analysis are used in this study to investigate the thermodynamic performance of proposed cycle. According to many researchers the exergy analysis is the best way to evaluate the performance of thermodynamic process [1].

In this study, the mass conservation, the first and second laws of thermodynamics are applied to each component of proposed cycle.

Some simplify assumptions are made to simplify the study of our machine:

- The proposed cycle is under steady condition [1], [34].
- The drop of pressure in all component of system is negligible except in the ejector [1], [34].
- There is no loss or gain heat from or to system except what are considered in study [1].
- The expansion in the expansion valve, in the reducing vale and is isenthalpic [1].
- The expansion in the turbine is isentropic.
- The state of refrigerant of outlet of condenser and evaporator are in saturate liquid and saturate vapor, respectively [1].

### 3.3Mass Conservation

Mass conservation includes the mass balance of total mass and each material of the solution. The governing equations of mass and type of material conservation for a steady state system are [1]:

$$\sum m_i-\sum m_0=0 \tag{7}$$

$$\sum m_i.x_i-\sum m_0.x_0=0 \tag{8}$$

Where *m* is the mass flow rate and *x* is the mass fraction of lithium bromide in the solution. The mass fraction of the mixture at different points of the process (Figure1) is calculated using the corresponding temperature and pressure data.

### 3.4 The first Law of Thermodynamics

For each component of the proposed cycle, the first law of thermodynamics is applied as follows [1]:

$$\left(\sum m_i.h_i-\sum m_0.h_0\right)+\left(\sum Q_i-\sum Q_0\right)\quad +W=0 \tag{9}$$

Where *h* is the specific enthalpy, *Q* is heat exchanged and *W* is the mechanical work to or from to component.

The energy balance equations of different components of the proposed cycle are summarized in the table 1.

### 3.5The Second Law of Thermodynamics

In an open system and neglecting of kinetics energy, the exergy balance equation can be expressed follows as [1]:

$$Ex=\sum_j Q_j.\left(1-\frac{T_0}{T_i}\right)+\left(\sum_i m_i.ex_i\right)_{in}-\left(\sum_i m_i.ex_i\right)_{out}-W \tag{10}$$

Where $ex_i$ is the specific exergy of flow which can be defined as [1]:

$$ex_i=(h_i-h_0)-T_0.(S_i-S_0) \tag{11}$$

Where $h_0$, $T_0$ and $S_0$ are represent the specific enthalpy, temperature and specific entropy of reference environmental state of water which are $T_0$=25 °C and $P_0$= 101 KPa.

*Table 1. The energy balance of different cycle component.*

| Cycle component | The energy balance |
|---|---|
| Generator | $Q_g=m_1.h_1+m_{11}.h_{11}-m_{12}.h_{12}$ |
| Absorber | $Q_a=m_8.h_8+m_9.h_9-m_{14}.h_{14}$ |
| Condenser | $Q_c=m_4.h_4-m_3.h_3$ |
| Evaporator | $Q_e=m_6.h_6-m_5.h_5$ |
| Expansion valve | $h_5=h_4$ |
| Reducing valve | $h_{10}=h_9$ |
| Heat exchanger | $T_{10}=T_{11}-\varepsilon.(T_{11}-T_{13})$<br>$h_{12}=\frac{m_{11}}{m_{13}}.(h_{11}-h_{13})+h_{13}$ |
| Turbine | $W_t=m_1.(h_2-h_1)$ |
| Pump | $W_p=m_{14}.\frac{(P_{13}-P_{14})}{\eta_p.\rho_{14}}$ |

The destruction exergy of different component of proposed cycle are presented in the table 2:

*Table 2. The exergy destruction of different cycle component.*

| Cycle component | The destruction exergy |
|---|---|
| Generator | $Ex_g=Q_g.\left(1-\frac{T_0}{T_g}\right)-m_1.ex_1-m_{11}.ex_{11}+m_{12}.ex_{12}$ |
| Absorber | $Ex_a=-Q_a.\left(1-\frac{T_0}{T_a}\right)+m_8.ex_8+m_9.ex_9-m_{14}.ex_{14}$ |
| Condenser | $Ex_c=-Q_c.\left(1-\frac{T_0}{T_c}\right)-m_4.ex_4+m_3.ex_3$ |
| Evaporator | $Ex_e=Q_e.\left(1-\frac{T_0}{T_e}\right)+m_6.ex_6+m_5.ex_5$ |
| Expansion valve | $Ex_{ev}=m_4.T_0.(S_4-S_5)$ |
| Reducing valve | $Ex_{rv}=m_{10}.T_{10}.(S_{10}-S_9)$ |
| Heat exchanger | $Ex_{hx}=m_{11}.ex_{11}+m_{13}.ex_{13}-m_{10}.ex_{10}-m_{12}.ex_{12}h_{12}$ |
| Turbine | $Ex_t=m_1.ex_1-m_2.ex_2-W_t$ |
| Pump | $Ex_p=m_{14}.ex_{14}-m_{13}.ex_{13}+W_p$ |

The destruction exergy of the proposed cycle $Ex_{dt}$ is the sum of exergy destruction of each its component. It can be calculated by following equation [37]

$$Ex_{dt}=\sum Ex_i \tag{12}$$

The exergy entering in the proposed cycle can be evaluated by [35]

$$Ex_{in}=Q_g.\left(1-\frac{T_0}{T_g}\right)+W_p \tag{13}$$

The exergy efficiency of the presented cycle is calculated by following equation [37]

$$\eta_{ex}=1-\frac{Ex_{tot}}{Ex_{in}} \tag{14}$$

## 4. Results and Discussions

The performance simulation of the proposed cycle is based on a program developed in Matlab.

The thermodynamic properties of lithium bromide water and liquid water are calculated using the efficient calculation formulas developed by Patek and Klomfar [38], and the thermodynamic properties of steam are calculated using the formula developed by Patek and Klomfar [39].

Based on the thermodynamic model presented above, the flowchart of the simulation resolution equation is shown in Figure 2.

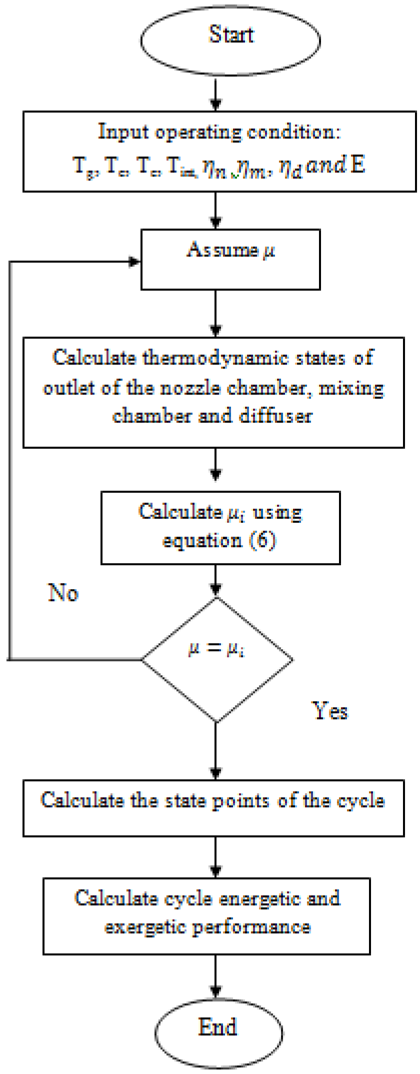


*Figure 2. The flowchart of simulation calculation.*

An iterative method is applied to define the entrainment ratio of ejector by assumed its value then calculate the different state of fluid in the outlet of the nozzle chamber, mixing chamber and diffuser of ejector applying Eqs. (1) – (5) and calculate the new value of entrainment ratio using Eq. (6), if the different between its two successive value calculated is under $10^{-4}$ then it is the real value of entrainment ratio.

After the known of entrainment ratio, it could be determinate the firstly ejector fluid mass flow which is equal to mass flow thronging the evaporator multiplied by entrainment ratio and the firstly ejector fluid mass flow which is equal to mass flow thronging the evaporator multiplied by (1- $\mu$). Then, it can be applied all Eqs. (7) – (14) to determine different mass and heat balance, exergy destruction in the different component, the cycle coefficient of performance and the cycle exergy efficiency.

The table 3 depicts the different state point properties of proposed cycle at optimum operating conditions.

### 4.1Model Validation

The ejector model validation based on the comparison between results available in the literature experimental study of ref [34], numerical study of ref [37] and results obtained from our proposed model in same working conditions, the evaporator temperature is equal to 8 °C, the working fluid is R141b and same different generator and condenser temperature. The results of comparison are resumed in table 4. It can be seen that the average of error of calculated results with experimental is 3.86% and 3.3% with numerical data of ref [37]. Thus, it can be used our model to predict ejector behavior.

*Table 3. The different properties of proposed cycle state point at optimum operating condition.*

| Number of state point | T (K) | P (kPa) | h (kJ/kg) | S (kJ/kg.K) | X (g/kg) | m (kg/sec) |
|---|---|---|---|---|---|---|
| 1 | 426 | 513.66 | 3361 | 7.91 | | 0.09791 |
| 2 | 333 | 19.80 | 2609 | 7.91 | | 0.09791 |
| 3 | 321 | 4.21 | 2589 | 8.563 | | 0.1253 |
| 4 | 303 | 4.21 | 125.1 | 0.4347 | | 0.1253 |
| 5 | 283 | 1.216 | 125.1 | 0.4445 | | 0.1253 |
| 6 | 283 | 1.216 | 2519 | 8.903 | | 0.1253 |
| 7 | 283 | 1.216 | 2519 | 8.903 | | 0.02739 |
| 8 | 283 | 1.216 | 2519 | 8.903 | | 0.09791 |
| 9 | 339.9 | 1.216 | 279.4 | 0.9152 | 0.1 | 0.2006 |
| 10 | 339.9 | 513.66 | 278.8 | 0.9147 | 0.1 | 0.2006 |
| 11 | 426 | 513.66 | 644.4 | 1.869 | 0.1 | 0.2006 |
| 12 | 392.7 | 513.66 | 942.9 | 0.8258 | 488 | 0.29851 |
| 13 | 303 | 513.66 | 398.8 | 0.2215 | 488 | 0.29851 |
| 14 | 303 | 1.216 | 59.73 | 0.2283 | 488 | 0.29851 |

*Table 4. Comparisons of ejector model results with experimental results of ref [34] and numerical results of ref [37].*

| $T_g$ (°C) | $T_c$ (°C) | Entrainment Ratio | | | Error | Error |
|---|---|---|---|---|---|---|
| | | Exp [34] | Num [37] | Our model | Exp (%) [34] | Num (%) [37] |
| 95 | 31.3 | 0.4377 | 0.4584 | 0.4473 | -2.15 | 2.48 |
| | 33 | 0.3937 | 0.4114 | 0.4003 | -1.65 | 2.77 |
| | 34.2 | 0.3505 | 0.3811 | 0.3701 | -5.30 | 2.97 |
| 90 | 33.8 | 0.3488 | 0.3614 | 0.3507 | -0.54 | 3.05 |
| | 36.7 | 0.3040 | 0.2967 | 0.284 | 7.04 | 4.47 |
| | 37.5 | 0.2718 | 0.2806 | 0.27 | 0.67 | 3.93 |
| 84 | 32.3 | 0.3883 | 0.3608 | 0.3504 | 10.82 | 2.97 |
| | 33.6 | 0.3117 | 0.3286 | 0.3182 | -2.04 | 3.27 |
| | 35.5 | 0.2880 | 0.2858 | 0.2754 | 4.58 | 3.78 |

### 4.2 Effect of Generation Temperature

The figure 3 depicts the effect of varying the generation temperature on the coefficient performance of cycle, the cycle exergy efficiency and on the work produced in the turbine. These results obtained in the following operating conditions, the condensation temperature has been assumed at 30°C, the evaporation temperature has been assumed at 10°C, the cooling capacity is equal to 300 kW and the outlet pressure of turbine is equal to 12.261 kPa. It can be seen that the coefficient of performance decrease with increasing the generation temperature contrary of work produced in the turbine. The exergy efficiency of the cycle increase to a certain value of generation temperature then decrease with increasing the generation temperature, in our operating conditions the maximum exergy efficiency is equal to 42.63% with 95.33 kW of work produced in the turbine at generation temperature equal to 152.3 °C. these results can be explained by an increasing in the generation temperature conducts an increasing in the water vapor desorbed in the generator and in the amount flow of turbine which is developing its work produced, contrary for coefficient of performance an increasing in the generation temperature conducts an increasing in the heat absorbed by generator from generation source with a constant cooling power produced equal to 300 kW, thus, the COP deteriorate. The exergy efficiency of cycle is increasing with increasing of generation temperature because the augmentation of exergy production which is the work produced in the turbine is more important than the augmentation of exergy destruction which is the heat absorbed by generator until a certain generation temperature which the exergy destruction in the generator become more important than the exergy production with augmentation of generator temperature.

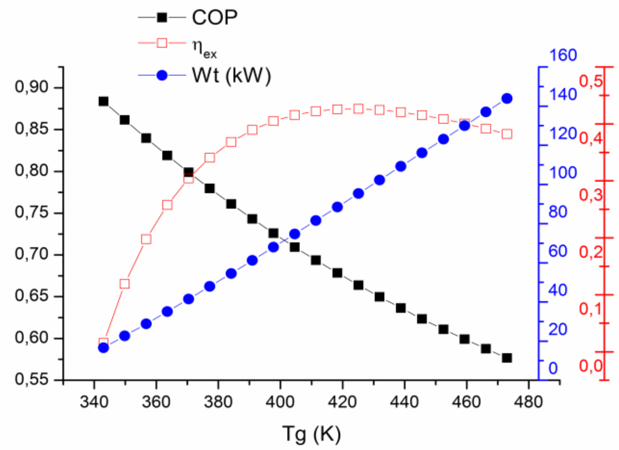


*Figure 3.The effect of generation temperature on the coefficient of performance, the exergy efficiency and on the turbine work produced.*

### 4.3 Effect of Outlet Pressure of Turbine

The figure 4 depicts the effect of varying the pressure outlet of turbine on the coefficient performance of cycle, the exergy efficiency and on the turbine work produced. The operating conditions of simulation are the condensation temperature is equal to 30°C, the evaporation temperature is equal to 10°C, the cooling capacity is equal to 300 kW and the temperature of generation is equal to 120 °C. It is clear that the coefficient of performance slowly increases but the work produced in the turbine slowly decrease with increasing the pressure outlet of turbine. The exergy efficiency is strongly deteriorated by the increasing the outlet pressure of turbine. These results can be explained by the augmentation of outlet pressure of the turbine conducts to decrease both the work produced of turbine and the exergy produced of cycle which is deteriorate the cycle exergy efficiency, on the contrary for the entrainment ratio, an increasing in the outlet pressure of the turbine which is the pressure of primary fluid of ejector conducts to develop the entrainment ratio which means that the mass flow of secondly fluid of ejector increased and the rest water vapor enters to absorber decreased which conducts to diminution of generation heat absorbed by generator and increases the coefficient performance of cycle COP.

The figure 5 depicts the effect of varying the pressure outlet of turbine on the entrainment ratio of ejector. The results depict that the entrainment ratio of ejector increases with an increasing of the pressure outlet of turbine. These results can be explained by the increasing of the outlet pressure of the turbine which is the primary fluid pressure of ejector conducts an increase in the secondly fluid mass flow of the ejector which is water vapor leaving the evaporator as consequently the entrainment ratio increases. Deng et all [40] found the similar phenomena in their study.

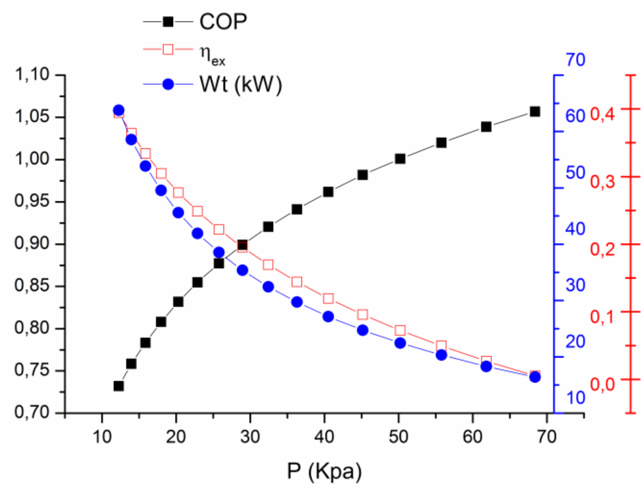


*Figure 4. The effect of pressure outlet of turbine on the coefficient of performance, the exergy efficiency and the work produced in the turbine.*

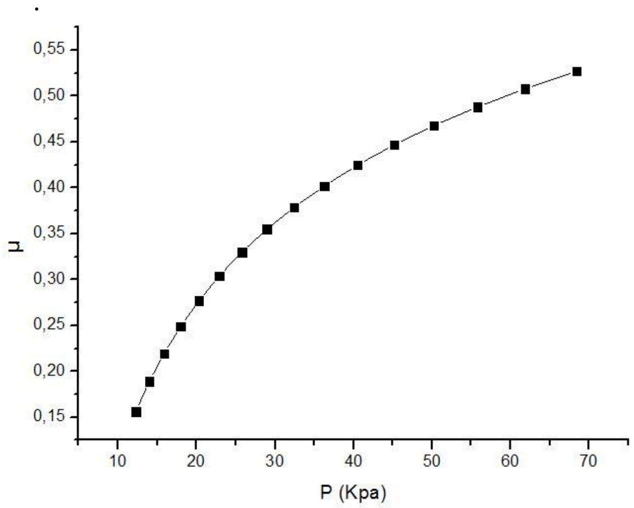


*Figure 5. The effect of pressure outlet of turbine on the entrainment ratio of ejector.*

### 4.4 The Effect of Condensation Temperature

The figure 6 depicts the effect of varying the condensation temperature on the cycle coefficient of performance, the cycle exergy efficiency and the work produced in the turbine. This simulation operated under following conditions the outlet pressure of turbine seems to be equal to 12.261 kPa, the evaporation temperature is equal to 10°C, the cooling capacity is equal to 300 kW and the temperature of generation is equal to 120 °C. It is clear that the increasing in the condensation temperature is positively affecting the work produced in the turbine and the exergy efficiency contrary of coefficient of performance. It can be explained these results by the condensation pressure is also the pressure of ejector diffuser which is inversely proportional to the second fluid mass flow of ejector as consequence the water vapor flow entering the absorber augmented correspondingly the rich solution flow entering to the generator which conducts an increasing in the heat absorbed by the generator.

The figure 7 depicts the effect of varying the condensation pressure on the entrainment ratio of ejector. It can be seen that an increasing in the condensation pressure conducts to diminution in the entrainment ratio. These results can be explained by the secondly fluid mass flow is negatively affected with an augmentation in the condensation pressure because of it is the pressure of water vapor leaving the ejector.

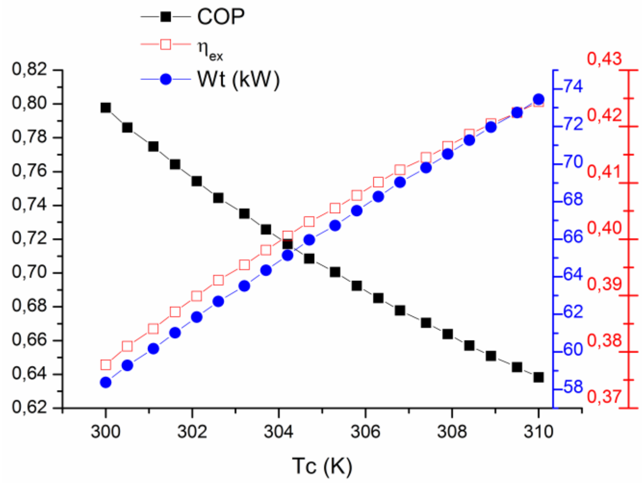


*Figure 6. The effect of condensation temperature on the coefficient of performance, the exergy efficiency and the work produced in the turbine.*

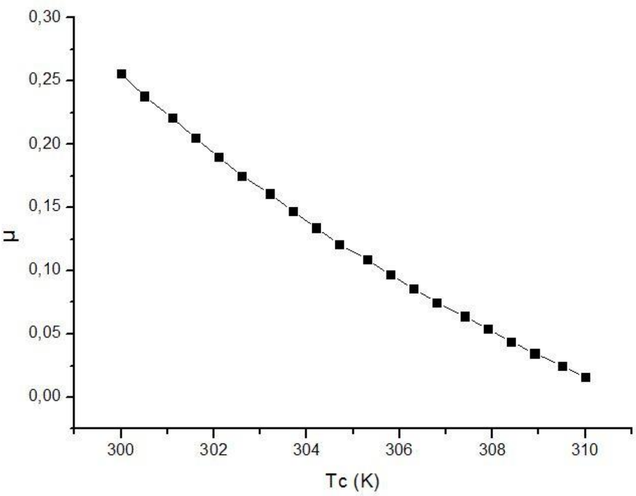


*Figure 7. The effect of condensation temperature on the entrainment ratio.*

### 4.5 Effect of Evaporation Temperature

The figure 8 depicts the impact of varying the evaporation temperature on the coefficient performance of cycle, the cycle exergy efficiency and on the work produced in the turbine under following operated conditions the outlet pressure of turbine is equal to 12.261 kPa, the condensation temperature is equal to 30°C, the cooling capacity is equal 300 kW and the temperature of generation is equal to 120°C. It is clear that an augmentation in the evaporation temperature positively affected the coefficient performance of cycle and the cycle exergy efficiency contrary the work produced in the turbine. It can be explained this results by the increasing the evaporation temperature conducts an increasing the pressure of evaporation which is the pressure of secondly fluid of ejector as consequence the secondly fluid flow of ejector augmented contrary for primary ejector fluid flow which conducts to diminution in the turbine work produced and in the heat absorbed in the generator.

Figure 9 represents the effect of varying the evaporation temperature on the entrainment ration. It is clear that the entrainment ratio is positively affected with an augmentation of evaporation temperature. These results can be explained by an increasing in the evaporation temperature conducts to augmentation of evaporation pressure which is the secondly fluid pressure of ejector. It well known that an augmentation of secondly fluid pressure of ejector conducts an augmentation of its mass flow. The same effect of evaporation temperature on the entrainment ratio is detected by Deng et al [40].

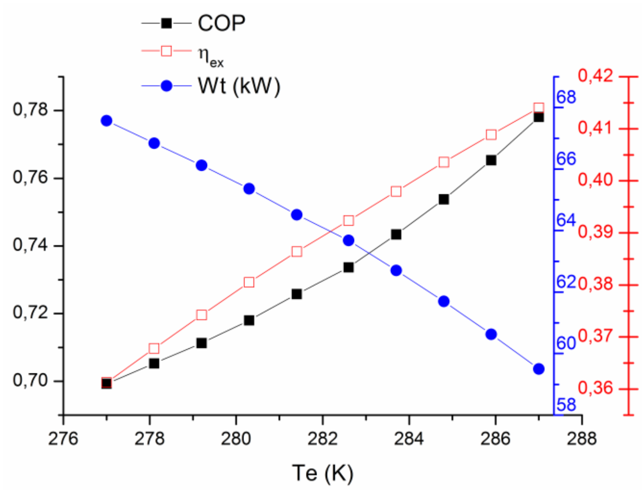


*Figure 8. The effect of evaporation temperature on the coefficient of performance COP, The exergy efficiency and on the work produced in the turbine.*

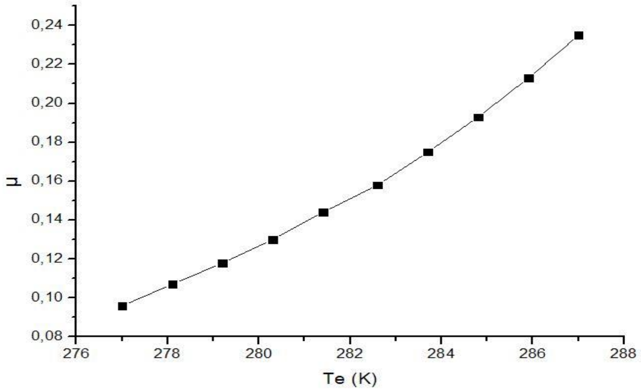


*Figure 9. The effect of evaporation temperature on the entrainment ratio.*

## 4.6 The Improvement of Exergy Efficiency

The proposed ejector power cooling absorption cycle efficiency improvement compared with the absorption simple effect, double effect and triple effect cycle Gomri [1] in various generation temperature, the evaporation temperature is equal to 4°C and the condensation temperature is equal to 39°C and the cooling capacity is equal to 300 kW presented in figure 10.

It is clear that for low grade temperature generation less than 100°C the proposed cycle has poor exergy efficiency because of low work produced in the turbine in comparing with the simple effect absorption cycle but the coefficient performance of proposed cycle is higher than its coefficient of performance. The proposed cycle has a high exergy efficiency in comparing with double and triple effect in middle range of generation temperature between 100 and 225 °C because of it has the same cooling capacity of double and triple effect absorption cycle adding up a respectable work produced in the turbine can be used to generate electrical power and the proposed cycle is very simple to realize end maintenance in comparing with double and triple effect absorption cycle, in point of view practical.

Table 5 represent the maximum improvement of proposed cycle exergy efficiency compared with double and triple effect absorption cycle of ref [1]

The reason behavior that the proposed ejector assisted absorption cycle has high exergy efficiency that the exergy production of proposed cycle is more important that the double and triple effect is the proposed cycle has many advantages as like produce a work in the turbine added to cooling power, use smartly the ejector to have a gratuitous compression of water vapor from evaporation pressure to condensation pressure and low number of cycle component and simplicity of cycle which are important factor to reduce the exergy destruction of cycle.

It can be concluded that the proposed cycle is recommended for cycle working with generation temperature of waste heat source of industries or geothermal or renewable energy source upper than 100°C.

*Table 5. The maximum improvement of proposed cycle exergy efficiency compared with double and triple effect absorption cycle.*

| $T_g$ (°C) | Double effect Absorption Cycle [1] | Tripple effect Absorption Cycle [1] | Proposed cycle | Exergy efficency Improvement |
|---|---|---|---|---|
| 132.6 | 0.2453 | - | 0.3174 | 29.41 % |
| 163 | - | 0.2529 | 0.3692 | 46 % |

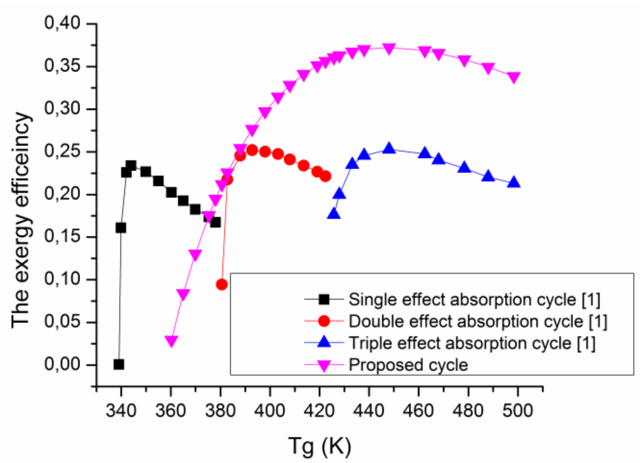


*Figure 10.The comparison between the proposed cycle and the simple effect, double effect and triple effect exergy efficiency from ref [1].*

## 5. Conclusions

A detailed numerical study of new proposed ejector assisted power cooling absorption cycle is carried out using first and second law of thermodynamic under often operating conditions used in the absorption cooling cycle. A constant mixing pressure model of ejector used in this simulation and its results validated with numerical and experimental data available in the literature. The analysis of simulation data of proposed cycle conducted to following conclusions:

- The increase of generation temperature is positively affected the work produced in the turbine contrary for coefficient performance of proposed cycle.
- For each operating conditions there is a certain value of generation temperature which is correspondence a maximum value of the cycle exergy efficiency.
- The increasing of pressure outlet of turbine is negatively

affected the work produced in the turbine and the cycle exergy efficiency contrary for the coefficient performance of proposed cycle and the entrainment ratio of the ejector.

- An increase in the condensing temperature leads to an increase in the work generated in the turbine and the cycle exergy efficiency and to a decrease in the coefficient of performance of the proposed cycle and the entrainment ratio of the ejector.
- An augmentation of evaporation temperature conducts an increasing of the coefficient performance, the exergy efficiency of proposed cycle and the entrainment ratio of ejector contrary for the work produced in the turbine.
- The proposed cycle is more efficiency in comparing with double and triple effect of absorption cycle for generation temperature superior of 100°C.
- The proposed cycle can be reachs 29.41 % and 46 % of exergy efficeincy improvement in comparaison with double and triple effect absorption cycle under generation temperature 132.6 °C and 162 °C respectively.
- Despite that the exergy efficiency of proposed cycle is lower than simple effect of absorption cycle but the coefficient of performance of proposed cycle is higher than its coefficient of performance for generation temperature under 100°C.

A thermodynamic comparaison of proposed cycle with simple, double and triple absorption effect is achieved in this paper but to make a final decision a technico economics study and analysis is required which will realize in the future works.

**Nomenclature**

| | |
|---|---|
| *COP* | coefficient of performance of absorption cooling machine |
| $\eta_{ex}$ | exergy efficeincy |
| *Ex* | exergy (kW) |
| *h* | enthalpy (kJ/kg) |
| *x* | mass fraction of lithium bromide by mass of solution (g/kg) |
| *Q* | heat transfert rate (kW) |
| *S* | Entropy (kJ/kg.K) |
| *T* | Temperature (K) |
| $W_t$ | The turbine work produced (kW) |
| $\rho$ | mass density (kg/m$^3$) |
| $\varepsilon$ | efficiency |
| $\mu$ | entrainment ratio |
| Subscripts | |
| 0 | reference value |
| a | absorber |
| c | condenser |
| e | evaporator |
| g | generator |
| i | the ith chemical species |
| 1,2,…,14 | the state point number |

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