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Research Article

4E Analysis of Enhanced Ejector Compression Absorption Cascade Cycle Working with Low GWP and ODP Refrigerants

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Abstract

In this paper, a new ejector compression absorption cascade cycle is presented. The energy, exergy, economic, and environmental analyses of the enhanced ejector compression cascade cycle are carried out. The model of the ejector and the compression absorption cascade cycle are validated using numerical and experimental results from literature in the same operating conditions. The thermodynamic performance of 9 refrigerant fluids with low GWP and ODP are compared. Then comparison of the performance of the proposed cycle and the conventional compression absorption cascade cycle is presented and the effect of the same conception parameter on the performance of the proposed cycle is defined. The results show that the RE170 has a higher coefficient of performance and exergy efficiency and a lower annual cost of the proposed cycle than the other 8 refrigerants, further the RE170 has GWP equal to 0.1 and ODP equal to 0. The enhancement in the coefficient of performance and in the exergy efficiency of proposed cycle is 3.27 and 2.7 % respectively compared with conventional compression absorption cascade cycle. Also, the diminution of the annual cost and the equivalent mass emission of CO_2 of proposed cycle is 7.93, 2.3 % compared with conventional compression absorption cascade cycle. The analysis of obtained results allows the conclusion that there is a generation temperature in which the coefficient of performance and the exergy efficiency of the proposed cycle are at maximum value and its annual cost is at minimum value. The coefficient of performance and the exergy efficiency of the proposed cycle are positively affected by increasing the sub-cooling heat exchanger efficiency and both its annual cost and its equivalent mass of CO_2 emission are negatively affected, contrary to the inlet temperature of the absorption cycle section in the cascade heat exchanger. The heat exchanger components of the proposed cycle are responsible for the most the destruction of exergy. The performances of the proposed cycle are promoted.

Keywords: *Absorption cascade; ejector compression; enhanced ejector; COP; exergy efficiency.*

1. Introduction

The compression absorption cascade refrigeration cycle has become an attractive cycle because of its low consumption of electrical energy and its low environmental impact [1]. Moreover, in 2022, the European Commission restricted using of refrigerants with high global warming potential (GWP) to more than 150 except for the primary fluid of the cascade refrigeration cycle can be equal to 1500 [2].

Many studies were carried out about the compression absorption cascade cycle (CACC). Khelifa et al. [1] studied the conventional compression absorption cascaded cycle using different combinations like refrigerants R1234yf, R1234ze (E), and R1233zd (E) for compression cycle and ($LiCl-H_2O$) and ($LiBr-H_2O$) for absorption cycle. The geothermal energy of Guelma which is province in Algeria is used as the heat of generation. They found that the diminution of the electrical energy consumed by the cascade compression absorption cycle can reach 54.16% compared with the vapor compression refrigeration cycle in the same operating conditions. Du et al. [3] simulated the conventional compression absorption cascaded cycle working with different refrigerants for the compression cycle and ($LiBr-H_2O$) for the absorption cycle. They found that the coefficient of performance and the exergy efficiency of refrigerant RE170 are better than the other 15 refrigerants studied.

Many attempts are made for enhancing the performance of absorption cooling machine, Salek et al. [4] studied the effect of add rectifier to ammonia water absorption cycle to have maximum purity of ammonia refrigerant. Its analysis based on two approaches the first is energetic and the second is exergetic. They found that the use of rectifier for ammonia water absorption cycle enhances the coefficient performance and the exergy efficiency of cycle and reduced the exergy destruction total of cycle. B. Gurevich and A. Zohar [5] coupled an ammonia water absorption cycle with solar collector to study the variation of its performance with the daily and the monthly variation of the exterior temperature and same design parameter like solar collector area. They found that the coefficient of performance of cycle and the cooling production by the cycle are slightly affected by the

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growth of area of solar collector contrary for the generation heat.

The use of ejectors for enhancement of the performance of vapor compression is the subject of many articles, Maalem et al. [6] investigated the use of new refrigerant R1311 with zero GWP as substitute refrigerant R134a in vapor compression coupled with an ejector. They found that in the most case the cycle working with R1311 has better performance than the cycle use R134a.Fingas et al. [7] studied experimentally a vapor compression cycle used as a heat pump working with two different technology expansion valves and ejectors. They found that the heating coefficient of performance of the cycle using the ejector is higher by 38% than the cycle using the expansion valve.

Zou et al. [8] proposed a new vapor compression cycle working with both vapor injection and enhanced injector. The results show that the coefficient of performance of the proposed cycle is higher by 38.4 % than the conventional vapor compression cycle.

Also, the use of sub cooling heat exchanger is proposed by many researchers to improve the performance of the vapor compression cycle. Qi et al [9] introduced a sub-cooling heat exchanger to enhance the ejector vapor compression heat pump. They found a positive effect of the use of sub-cooling on the performance of enhanced ejector heat pumps. Moreover, Pitrach et al. [10-12] studied the experimental effect of the use of sub cooling heat exchanger on the performance of the vapor compression cycle using as heat pump. They found the use of sub-cooling can increase the coefficient of performance by 31 % compared with the vapor compression cycle without sub-cooling.

The previous studies on the absorption cooling cycle used principally two working pairs which are the NH_3-H_2O and H_2O-BrLi. Cimsit and Ozturk [13] compared the performance of CACC using two pairs. They found that the CACC used H_2O-LiBr has a higher coefficient of performance by 33% compared with CACC used NH_3-H_2O. Moreover, Seyfouri and Ameri [14] found that for CACC using NH_3-H_2O, the electrical energy consumed by the solution pump was high due to the high difference in pressure between the absorber and the generator. Therefore, the working pair choices for the absorption cycle section in this work are H_2O-LiBr.

In this paper, a new cycle of ejector compression absorption cascade cycle is presented. The energy, exergy, economic, and environment studies of enhanced ejector compression absorption cascade cycle are carried out. The model developed is validated using two steps, the first step is to compare the results obtained in this work with literature results of the compression absorption cascade cycle and the second step is the comparison of ejector model obtained in this work with experimental and numerical results found in literature in same operating conditions. The performance of the proposed cycle working with 9 low GWP and ODP refrigerants of the vapor compression section is compared the thermodynamic performance and the annual of proposed cycle cost are compared with the conventional compression absorption cascade cycle. Also, the effect of the generation temperature, the outlet temperature of the absorption section of the proposed cycle, and the sub-cooling heat exchanger efficiency are analyzed. The exergy destruction and the investment cost of the main component of the proposed cycle are studied.

2. The Proposed Solution to Develop Performance of Conventional Compression Absorption Cascaded Cycle

In this paper, a new solution is proposed to enhance the conventional compression-absorption cascaded performance cycle by incorporating an ejector, a sub-cooling heat exchanger, and an expansion valve, and separator to reduce the inlet mass flow of the compressor and its work by division into two parts the first enters the evaporator and continues to the compressor and the second compresses in the ejector and joined the first part in the compressor by using separator at a pressure equal to outlet ejector pressure and for enhancement the cooling production, the second part of refrigerant vaporizes in the sub-cooling heat exchanger after it expanded in expansion valve and the first part is sub cooling in the sub-cooling heat exchanger to maximize the cooling production in the evaporator.

3. Enhanced Ejector Compression Absorption Cascaded Cycle Description

Figure 1 illustrates the enhanced ejector compression absorption cascaded cycle (EECACC) which its components are a generator, an absorber, a condenser, a solution pump, a solution heat exchanger, a cascade heat exchanger, a liquid-vapor exchanger, a sub-cooling heat exchanger, an evaporator, an ejector, a separator tank, a compressor, three expansion valves, and a reducer pressure valve.

At point (7), the liquid-vapor mixture of H_2O enters the cascade heat exchanger absorbs the heat from the refrigerant of the enhanced ejector compression cycle, thus the absorption cycle refrigerant vaporizes and leaves the cascade heat exchanger in a vapor-saturated state at point (8). Then it heats in the liquid-vapor heat exchanger to reach point (9) which is the state of entering the absorber and absorbs by a weak solution coming from the reduced valve at point (15), the result is a strong solution cools at the absorber which it leaves at the point (10). The strong solution at point (10) undergoes an enhancement of pressure to condensation pressure at point (11) by passing the pump solution. The strong solution at point (11) heats by the weak solution leaving the generator at the solution heat exchanger to achieve point (12) which is the state of entering of the generator. The weak solution leaves the generator at point (13) is cooling in the solution heat exchanger by the strong solution and it leaves the heat exchanger at point (14) which undergoes an expansion in the reducer valve to reach point (15) and completes the cycle of solution. On the other hand, the saturated water vapor left the generator at point (16) condensed in the condenser, and left at point (5). The saturated water liquid is sub cooled in the liquid-vapor heat exchanger by water vapor left of the evaporator to achieve the point (6). The sub-cooling water liquid at point (6) expands in the expansion valve to close the absorption cycle at point (7).

For the enhanced ejector compression cycle, the cooling refrigerant vapor in the cascade heat exchanger at point (3) enters the ejector as primary fluid to entrain the secondary fluid of the ejector entering at point (4). The ejector compression fluid leaves the ejector at point (25) divided into two parts in the separator tank, the first is in a liquid state at point (26) cooled in the sub-cooling heat exchanger to reach a point (27) by the secondary ejector fluid which is a part of refrigerant of point (27) expands to the point (28) in the expansion valve EV03 and heats in the sub-cooling

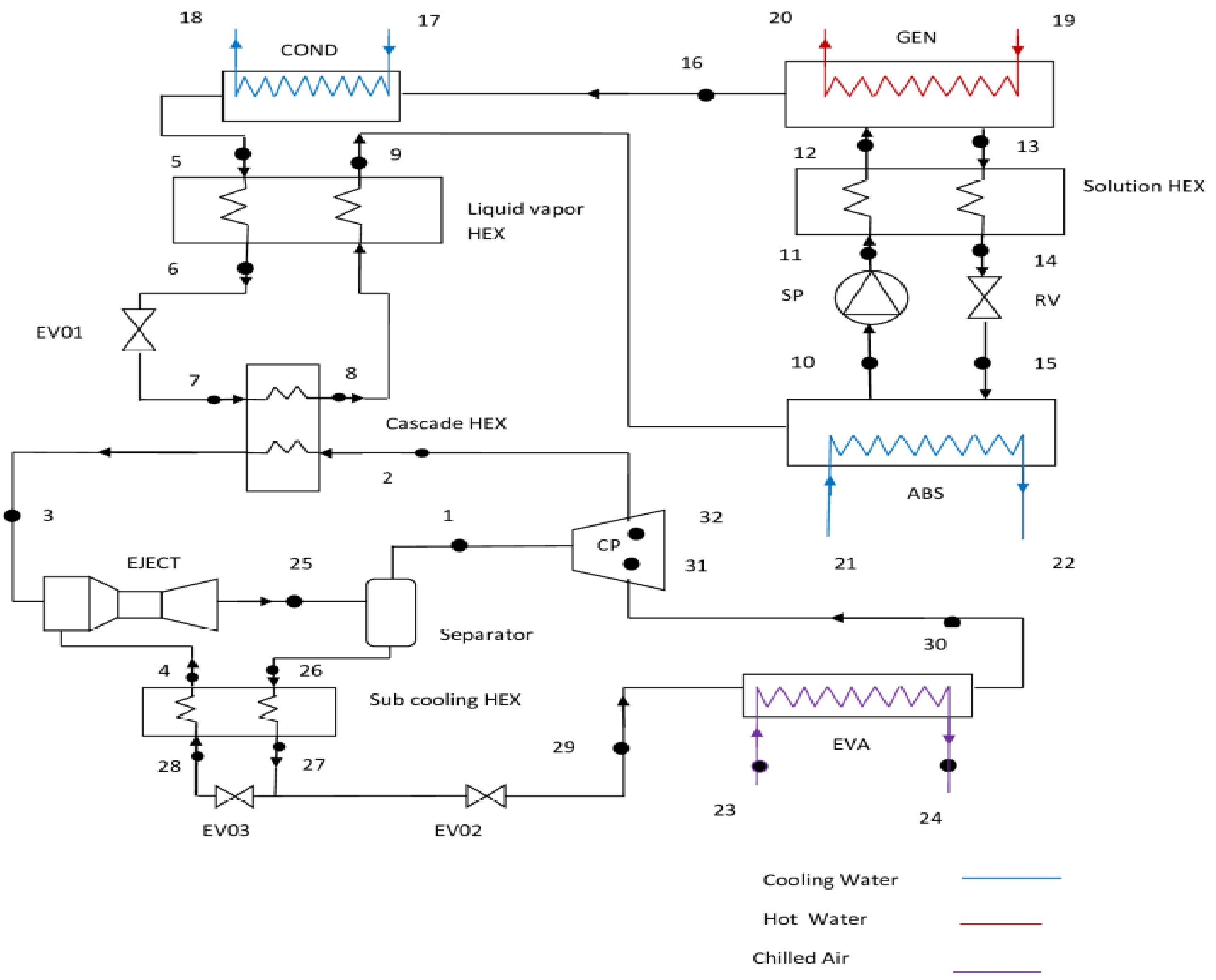


Figure 1. Schematic of Enhanced Ejector Compression Absorption Cascade Cycle (EECACC).

heat exchanger to the point (4). The other part of the refrigerant expands in the expansion valve EV02 to become a mixture of liquid vapor state at point (29) and evaporates in the evaporator and leaves in the saturated vapor state at point (30) and compresses in the compressor from the evaporation pressure to the outlet ejector pressure at point (31), than meets the second part of refrigerant left the separator tank at vapor saturated state at the point (1). The mixed superheated refrigerant vapor at point (32) is compressed at the compressor to point (2) and cooled in the cascade heat exchanger to point (3).

Figure 2 illustrates the conventional compression absorption cascade cycle (CCACC) which is explained by Du et al. [3]. The CCACC energy, exergy, and economic models are used in this paper for validation and comparison purposes.

4. Thermodynamic Cycle Modeling

4.1 Ejector Model

The ejector model used in this study is based on the model presented by Cheng et al. [15]. The type of model is the one-dimensional constant pressure of the mixing section. The following assumptions are taken in this study [15]:

- There is no exchange of heat between the ejector and exterior.
- The velocities of primary and secondary fluid of the ejector in the inlet of the ejector are neglected.
- The velocity of the fluid at the outlet of the ejector is neglected.
- The efficiency of the ejector nozzle, mixing, and diffuser section is considered constant.
- The refrigerant flow energy losses in the ejector are taken into account by using different ejector section efficiency.

The use of the energy, mass, and momentum conservation laws for every ejector section like as nozzle, mixing, and diffuser allows for defining the thermo physics parameters of the ejector as follows:

The outlet nozzle chamber velocity of primary fluid can be defined by the following equation [15]:

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

where, U is the primary fluid velocity, $h_{p,in}$ is the primary fluid enthalpy at the inlet of the nozzle chamber, η_n is the isentropic efficiency of the nozzle chamber, the subscript in is the inlet, out is the outlet and the is the isentropic expansion.

The outlet mixing chamber fluid velocity and enthalpy can be calculated as follows [15]:

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

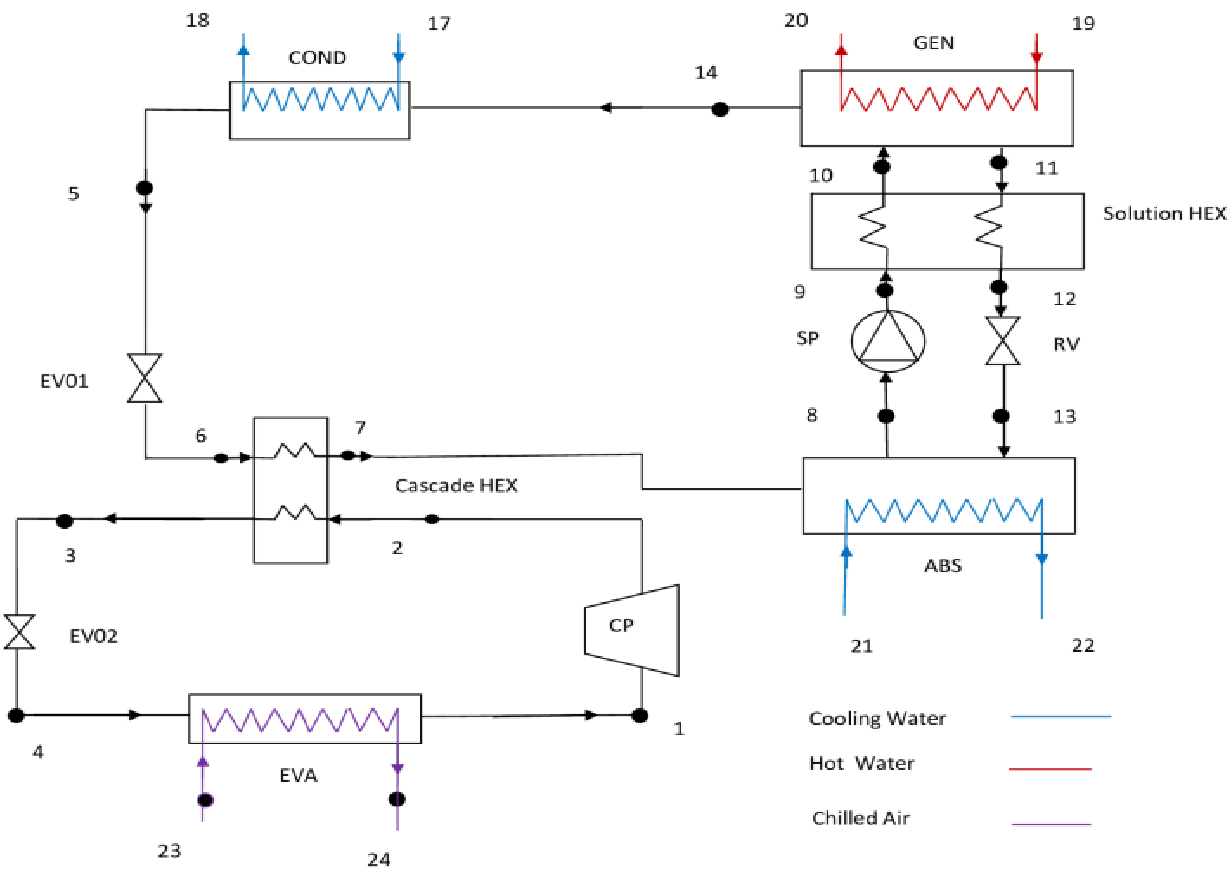


Figure 2. Schematic of Compression Absorption Cascade Cycle (CCACC).

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

where, $h_{s,in}$ is the secondly fluid enthalpy at the nozzle chamber inlet, η_m is the efficiency of the mixing chamber and μ represents the entrainment ratio which can be calculated by the following equation [15]:

$$\mu=\frac{m_p}{m_s} \quad (4)$$

The outlet diffuser chamber fluid enthalpy which is the fluid enthalpy at the ejector outlet calculated by the following equation [15]:

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

where, η_d is the efficiency of the diffuser chamber.

The entrainment ratio of the ejector can be evaluated using previous equations [15]:

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

4.2 Thermodynamic Model

Applying mass conservation, the first and second laws of thermodynamics for each component of the cycle allows defining the energy and the exergy balance of each component of cycle and the analysis of the thermodynamic performance of the cycle. Many assumptions must be made to simplify the study:

- The proposed cycle is under steady conditions [17], [16].
- Except in the ejector, the loss of pressure in all cycle components is negligible [16].
- There is no heat exchange between the cycle and environment except what is considered in the study [17].
- The outlet refrigerant state of the condenser, evaporator, and cascade heat exchanger are in saturated liquid, saturated vapor, and saturated vapor, respectively [17].

4.2.1 Mass Conservation

The mass conservation law can be defined as follows [15]:

$$\sum m_i-\sum m_o=0 \quad (7)$$

$$\sum m_i.x_i-\sum m_o.x_o=0 \quad (8)$$

where, m is the mass flow of refrigerant rate and x is the lithium bromide mass fraction in the solution, the subscript i is the inlet and o is the outlet.

4.2.2 The First Law of Thermodynamics

The first thermodynamic law which is the energy conservation law of component can be found by applying flowing equation [18]:

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

where, h is the specific enthalpy, Q is heat exchanged and W is the mechanical work to or from to component, the subscript i is the inlet and o is the outlet.

The energy balance equations of CCACC and EECACC are presented in Table 1.

4.2.3 The Second Law of Thermodynamics

The exergy study of the cycle allows us to optimize the performance and to understand the weak point from the point of view of finding the component that produces more exergy destruction to find a solution to decrease it in the future. In this study, only the physics exergy is taken into consideration [3]. Thus, the component exergy flow rate can be found by the following equation [15]:

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

Table 1. The energy balance of different components of cycle.

Cycle	Cycle component	The energy balance
CCACC	Generator	$Q_g = m_{14}.h_{14} + m_{10}.h_{10} - m_{11}.h_{11}$
	Absorber	$Q_a = m_7.h_7 + m_{13}.h_{13} - m_8.h_8$
	Condenser	$Q_c = m_{14}.h_{14} - m_5.h_5$
	Cascade heat exchanger	$Q_{chx} = m_7.h_7 - m_6.h_6$
	Evaporator	$Q_e = m_1.h_1 - m_4.h_4$
	Expansion valve EV01	$h_5 = h_6$
	Expansion valve EV02	$h_3 = h_4$
	Reducing valve	$h_{12} = h_{13}$
	Solution heat exchanger	$T_{12} = T_{11} - \varepsilon_{she}.(T_{11} - T_9)$ $h_{10} = \frac{m_{11}}{m_{10}}.(h_{11} - h_{12}) + h_9$
	Compressor	$W_{com} = m_1.\frac{h_{2,is} - h_1}{\eta_{is,com}.\eta_{el}}$
EECACC	Generator	$Q_g = m_{13}.h_{13} + m_{16}.h_{16} - m_{12}.h_{12}$
	Absorber	$Q_a = m_9.h_9 + m_{15}.h_{15} - m_{10}.h_{10}$
	Condenser	$Q_c = m_{16}.h_{16} - m_5.h_5$
	Cascade heat exchanger	$Q_{chx} = m_8.h_8 - m_7.h_7$
	Evaporator	$Q_e = m_{30}.h_{30} - m_{29}.h_{29}$
	Expansion valve EV01	$h_6 = h_7$
	Expansion valve EV02	$h_{27} = h_{29}$
	Expansion valve EV03	$h_{28} = h_{29}$
	Reducing valve	$h_{14} = h_{15}$
	Solution heat exchanger	$T_{14} = T_{13} - \varepsilon_{she}.(T_{13} - T_{11})$ $h_{12} = \frac{m_{13}}{m_{11}}.(h_{13} - h_{14}) + h_{11}$
	Liquid vapor heat exchanger	$T_6 = T_5 - \varepsilon_{che}.(T_5 - T_8)$ $h_9 = \frac{m_5}{m_7}.(h_5 - h_6) + h_8$
	Sub cooling heat exchanger	$T_{27} = T_{26} - \varepsilon_{sbhe}.(T_{26} - T_{28})$ $h_4 = \frac{m_{26}}{m_4}.(h_{26} - h_{27}) + h_{28}$
	Compressor	$W_{com} = m_{30}.\frac{h_{31,is} - h_{30}}{\eta_{is,com}.\eta_{el}} + m_2.\frac{h_{2,is} - h_{31}}{\eta_{is,com}.\eta_{el}}$

where, ex_i is the specific exergy at each state point of cycle which is defined as follows [18]:

$$ex_i = (h_i - h_0) - T_0.(s_i - s_0) \quad (11)$$

where, h_0, T_0 and S_0 are representing the specific enthalpy, temperature, and specific entropy of reference environmental state which are T_0 = 25 °C and P_0 = 101 kPa.

The different component exergy destruction of EECACC is presented in Table 2.

Table 2. The exergy destruction of different components of EECACC.

Cycle component	The destruction exergy
Generator	$Ex_g = m_{12}.ex_{12} + m_{19}.ex_{19} - m_{13}.ex_{13} - m_{16}.ex_{16} - m_{20}.ex_{20}$
Absorber	$Ex_a = m_9.ex_9 + m_{15}.ex_{15} + m_{21}.ex_{21} - m_{10}.ex_{10} - m_{22}.ex_{22}$
Condenser	$Ex_c = m_{16}.ex_{16} + m_{17}.ex_{17} - m_5.ex_5 - m_{18}.ex_{18}$
Cascade heat exchanger	$Ex_{chex} = m_2.ex_2 + m_7.ex_7 - m_3.ex_3 - m_8.ex_8$
Evaporator	$Ex_e = m_{23}.ex_{23} + m_{29}.ex_{29} - m_{24}.ex_{24} - m_{30}.ex_{30}$
EV 01	$Ex_{ev1} = m_6.T_0.(S_6 - S_7)$
EV 02	$Ex_{ev2} = m_{29}.T_0.(S_{27} - S_{29})$
EV 03	$Ex_{ev3} = m_{28}.T_0.(S_{27} - S_{28})$
Solution heat exchanger	$Ex_{hx} = m_{11}.ex_{11} + m_{13}.ex_{13} - m_{12}.ex_{12} - m_{14}.ex_{14}$
Separator tank	$Ex_{st} = m_{25}.ex_{25} - m_1.ex_1 - m_{26}.ex_{26}$
Liquid vapor heat exchanger	$Ex_{lvhex} = m_5.ex_5 + m_8.ex_8 - m_6.ex_6 - m_9.ex_9$
Sub cooling heat exchnager	$Ex_{Sbhx} = m_{26}.ex_{26} + m_{28}.ex_{28} - m_4.ex_4 - m_{27}.ex_{27}$
Compressor	$Ex_{com} = m_{30}.ex_{30} + m_{32}.ex_{32} - m_2.ex_2 - m_{31}.ex_{31} + W_{com}$
Ejector	$Ex_{ej} = m_3.ex_3 + m_4.ex_4 - m_{25}.ex_{25}$

The total destruction exergy of the EECACC which is the sum of the exergy production of each component can be found by following the equation [15]

$$Ex_{dt} = \sum Ex_i \quad (12)$$

The EECACC and the CCACC coefficient performance are calculated using the following equation [15]:

$$COP = \frac{Q_{ev}}{Q_g + W_{com}} \quad (13)$$

The ECACC and the CCACC exergy efficiency are calculated by following equation [15]

$$\eta_{ex} = \frac{Q_{ev} . \left(1 - \frac{T_0}{T_{ev}}\right)}{Q_g . \left(1 - \frac{T_0}{T_g}\right) + W_{com}} \quad (14)$$

4.3 The Economic Study

The economic analysis is the cost study of investment, operating, and maintenance of the cycle during the lifetime of operation but it is presented in the form of annual cost. The model used in this study is defined by Jain and al [19] and calculated by following equation:

$$C_T = t_{op} . (C_f Q_g + C_{ele} . W_{comp}) + C_r . M . \sum Z_i + C_{env} \quad (15)$$

where, C_T is the cost total of cycle, t_{op} is annual operation time, C_f is the cost fuel of heat generation, C_{ele} is the electrical energy cost, M is the maintenance factor, Z_i in the investment cost of cycle component, C_{env} is the social cost of CO_2 emission of cycle and C_r is the capital recovery factor which is defined by the following equation [4]:

$$C_r = \frac{i.(i+1)^L}{(i+1)^L - 1} \quad (16)$$

where, i is the interest annual rate and L is the cycle lifetime.

The investment cost of heat exchangers of cycle is depending to its exchange area which can be concluded using the following equation:

$$Q_i = U_i.A_i.LMTD_i \quad (17)$$

where, *LMTD* is the logarithmic mean temperature difference and it can be calculated with the following equation:

$$LMTD_i = \frac{\Delta T_i^H - \Delta T_i^c}{ln\left(\frac{\Delta T_i^H}{\Delta T_i^c}\right)} \quad (18)$$

In Table 3, the global heat exchange coefficient and *LMTD* of all heat exchangers of cycle are presented.

The investment cost of heat exchangers is defined by following equation [23]:

$$Z_{invi} = 516.62 x A_i + 268.45 \quad (19)$$

Table 3. The heat exchanger coefficient and LMTD of heat exchangers of cycle [24, 25].

Component	U(*kW/(m²·K)*)	LMTD
GEN	1.5	$LMTD_{gen} = \frac{(T_{19} - T_{13}) - (T_{20} - T_{16})}{ln\left(\frac{T_{19}-T_{13}}{T_{20}-T_{16}}\right)}$
ABS	0.7	$LMTD_{abs} = \frac{(T_{15} - T_{22}) - (T_{10} - T_{21})}{ln\left(\frac{T_{15}-T_{22}}{T_{10}-T_{21}}\right)}$
CON	*2.5*	$LMTD_{cond} = \frac{(T_5 - T_{17}) - (T_5 - T_{18})}{ln\left(\frac{T_5-T_{17}}{T_5-T_{18}}\right)}$
SHE	1	$LMTD_{SHE} = \frac{(T_{12} - T_{13}) - (T_{11} - T_{14})}{ln\left(\frac{T_{12}-T_{13}}{T_{11}-T_{14}}\right)}$
LVHE	1	$LMTD_{LVHE} = \frac{(T_5 - T_9) - (T_6 - T_8)}{ln\left(\frac{T_5-T_9}{T_6-T_8}\right)}$
SBHE	1	$LMTD_{SBHE} = \frac{(T_{26} - T_4) - (T_{27} - T_{28})}{ln\left(\frac{T_{26}-T_4}{T_{27}-T_{28}}\right)}$
CHE	0.55	$LMTD_{CHE} = T_3 - T_8$
EVA (EECACC)	1.5	$LMTD_{EVA} = \frac{(T_{23} - T_{29}) - (T_{24} - T_{29})}{ln\left(\frac{T_{23}-T_{29}}{T_{24}-T_{29}}\right)}$
EVA (CCACC)	1.5	$LMTD_{EVA} = \frac{(T_{23} - T_4) - (T_{24} - T_4)}{ln\left(\frac{T_{23}-T_4}{T_{24}-T_4}\right)}$

The compressor investment cost is calculated by following equation [2]:

$$Z_{inv\text{-}comp} = \left(\frac{573.m_2}{0,8996 - \eta_{is,com}}\right).\left(\frac{P_{32}}{P_2}\right).ln\left(\frac{P_{32}}{P_2}\right) + \left(\frac{573.m_{30}}{0,8996 - \eta_{is,com}}\right).\left(\frac{P_{30}}{P_{31}}\right).ln\left(\frac{P_{30}}{P_{31}}\right) \quad (20)$$

The ejector investment cost is defined by following equation [21]:

$$Z_{inv\text{-}eje} = 750.m_3.\left(\frac{T_3}{P_3}\right)^{0,05}.\left(\frac{P_{25}}{1000}\right)^{-0.75} \quad (21)$$

4.4 The Environmental Study

The environment regulatory requirement is more and more restrict; thus, the study of CO_2 emission is taken into consideration in this study. The annual emission of CO_2 is defined by following equation [22, 23]:

$$m_{co2} = ECF.t_{op}.W_{com} \quad (22)$$

where, *ECF* is emission conversation factor of electricity. The environment cost is defined by following equation [22,23]:

$$C_{env} = \mu_{co2}.\frac{m_{co2}}{1000} \quad (23)$$

where, μ_{co2} is the CO2 emission unit damage cost.

The parameters used in the economy and environmental model are cited in Table 4.

Table 4. The economic and environmental parameters used in this work.

Parameter	*Value*	*Ref.*
i	10%	[26]
L	15 *years*	[26]
M	1.06	[27]
t_{op}	6000 *h*	[28]
ECF	0.968 *kg. (kWh)⁻¹*	[29]
C_f	0.03785 *\$. (kWh)⁻¹*	[2]
C_e	0.0375 *\$. (kWh)⁻¹*	[2]
μ_{co2}	90 *\$.(ton)⁻¹*	[30]

4.5 The 9 Refrigerants Studied

The choice of refrigerant studied based on its impact on the environmental. Table 5 presents the different refrigerants studied and their GWP, ODP.

Table 5. GWP and ODP of 11 refrigerants used in this work [29], [31], [32].

Refrigerant	GWP	ODP
R600	20	0
R600a	20	0
R1234yf	4	0
R1243zf	<150	0
R744	1	0
R290	20	0
RE170	0.1	0
R1270	20	0
R152a	124	0

5. Results and Discussions

The performance estimation of CCACC and EECACC needs a simulation of functioning by developing a program in engineering equation solver EES based on the thermodynamic, economic, and environmental models given above.

The first step is to define the ejector entrainment ratio by applying an iterative method based on assuming its arbitrary value, then defining the different outlet fluid states of different ejector chambers which are the nozzle, mixing, and diffuser by applying Eqs. (1)-(5) and compare the value assumed with the new entrainment ratio value obtained by Eq. (6), if the difference between the values successive is under 10^{-4} then the algorithm will stop and the entrainment ratio taken value is the last one.

The known entrainment ratio allows for determining all flow mass rates of all state points by applying the mass and energy conservation laws. Then, all energy exchange of all cycle components cited in Table 1 and all exergy destruction cited in Table 2 can be calculated and concluded performance parameters of the cycle. In Figure 3, the simulation flow chart is presented.

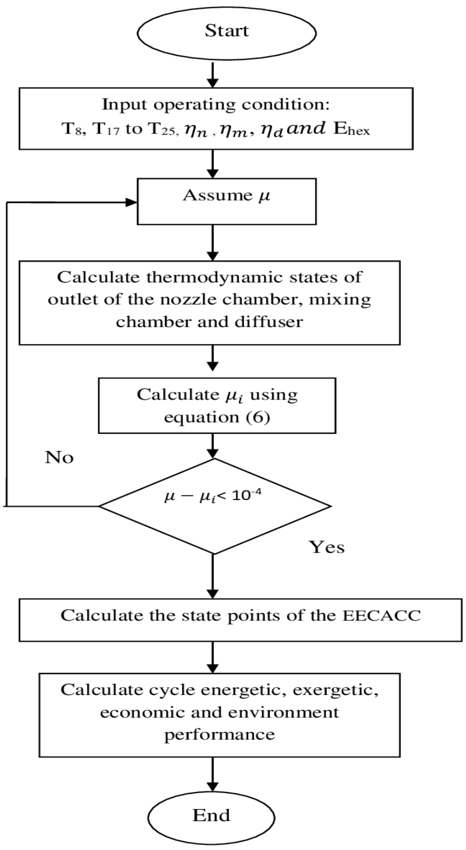


Figure 3. The simulation calculation flowchart.

The working conditions of studied cycle are presented in Table 6.

5.1 Validation of Model

The fact that lack of EECACC simulation results in the literature because it is a novel cycle, the validation of EECACC is based on the validation of its closer model which is the CCACC presented by [3], and the validation of the ejector model separately.

The validation of CCACC is based on the comparison of results obtained from this work and results obtained by [3] of the same cycle using the R134a, R744, R152a and R32 as refrigerants of the vapor compression section. The simulation results are presented in Table 7 in the same operating conditions presented in Table 6. It is clear that the results of the model developed for CCACC in this work are close to the results of [3].

5.2 Performance Comparison of EECAC working with 9 Refrigerant

The coefficient of performance and the exergy efficiency of EECACC working with 9 low GWP and ODP refrigerants are presented in Table 9.

The results show that the thermodynamic performances of all refrigerants studied are very close and two refrigerants R600 and RE170 had the same performance.

The choice of refrigerant of EECACC was studied based on the environmental impact of refrigerant and consequently, the fluid choice is RE170.

Table 6. Operating condition using in this study.

Parameter	Value	Parameter	Value
Q_{ev}	250 kW	ΔT_{eva}	8 K
T_g	363.15 K	ΔT_{con}	8 K
T_7	283.15 K	ΔT_{gen}	8 K
T_{17}	300.15 K	ΔT_{abs}	8 K
T_{18}	305.15 K	ΔT_{che}	8 K
T_{19}	381.15 K	ε_{she}	0.7
T_{20}	371.15 K	ε_{Lvhe}	0.7
T_{21}	300.15 K	ε_{sche}	0.7
T_{22}	305.15 K	$\eta_{is,\ com}$	0.8
T_{23}	271.15 K	$\eta_{el,\ com}$	0.9
T_{24}	266.15 K		

5.3 Performance Comparison Between EECACC and CCACC:

In this section, the thermodynamic performances and annual economic cost of EECACC and CCACC are compared according to the model presented above under the same operating conditions shown in Table 5.

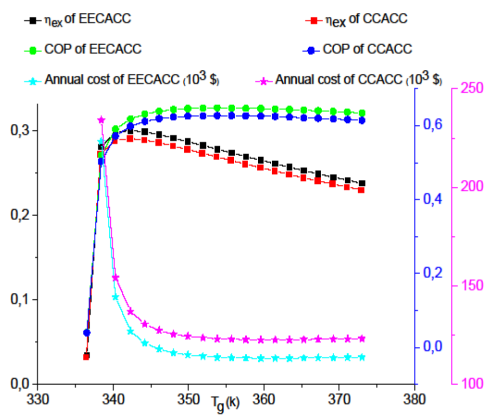


Figure 4. the variation of the exergy efficiency, COP and annual cost of EECACC and CCACC with the variation of generation temperature.

Figure 4 shows the effect of the generation temperature on the exergy efficiency, COP and the annual cost of EECACC and CCACC, the COP of the two systems increases from 0.04 and0.039 to 0.6435 and 0.6231, respectively with the increasing of generation temperature up to 348 K and then tends to stabilize around 0.64 and, 0.62 respectively and the coefficient of performance of EECACC is higher than the coefficient of performance of CCACC and the maximum value of the coefficient of performance of EECACC is 0.6477 at the generation temperature equal to 353.8 k. In addition, the coefficient of performance of the proposed EECACC cycle is 3.27% higher than the coefficient of performance of the CCACC cycle.

The exergy efficiency of EECAC and CCACC increases to the maximum values of 0.3 and 0.2921 at a generation temperature equal to 342.3 K, then decreases with increasing the generation temperature, and the EECACC exergy efficiency is higher than the CCACC by 2.7 %.

Table 7. Comparison of results obtained from this work and from [3] (COP_{ABS}: COP of absorption cycle, COP_{COM}: COP of vapor compression section, COP_{CCACC}: cop of CCACC).

Parameters	R134a/H_2O-LiBr		R744/H_2O-LiBr			R152a/H_2O-LiBr				R32/H_2O-LiBr		
	Ref [3]	This work	Ref [3]	This work	dev (%)	Ref [3]	Ref [3]	This work	dev (%)	Ref [3]	This work	dev (%)
Q_{GEN} (*kW*)	362.65	361.30	361.73	360.40	0.369	363.62	361.73	360.40	0.369	363.62	362.30	0.364
Q_{CHE} (*kW*)	287.74	287.70	287.02	287.00	0.006	288.52	287.02	287.00	0.006	288.52	288.50	0.007
Q_{ABS} (*kW*)	344.48	343.10	343.61	342.30	0.383	345.40	343.61	342.30	0.383	345.40	344.10	0.377
Q_{CON} (k*W*)	305.91	305.90	305.14	305.10	0.013	306.73	305.14	305.10	0.013	306.73	306.70	0.009
W_{COM} (k*W*)	41.91	41.93	41.13	41.14	0.024	42.800	41.13	41.14	0.024	42.800	42.80	0.000
COP_{ABS}	0.794	0.796	0.794	0.796	0.251	0.794	0.794	0.796	0.251	0.794	0.796	0.251
COP_{COM}	5.961	5.962	6.078	6.077	0.016	5.842	6.078	6.077	0.016	5.842	5.841	0.017
COP_{CCACC}	0.618	0.620	0.621	0.622	0.161	0.615	0.621	0.622	0.161	0.615	0.617	0.162
η_{ex}	0.234	0.247	0.236	0.249	5.508	0.232	0.236	0.249	5.508	0.232	0.245	5.603

Contrary to the COP, the annual cost of EECAC and CCACC decreases to 116 and 125.4 thousand dollars respectively at a generation temperature equal to 348 k then trends to stabilize around 113.5 and 122.5 thousand dollars respectively with increasing the generation temperature and the annual cost of EECAC is lower the CCACC annual cost by 7.93 %.

Table 8. Comparisons of entrainment ratio of the ejector model used in present study with experimental results of reference [16] and numerical results of reference [15].

T_g (°C)	T_c (°C)	Entrainment Ratio Exp. [16]	Entrainment Ratio Num. [15]	Entrainment Ratio Our model	Error Exp. (%) [16]	Error Num. (%)[15]
95	31.3	0.4377	0.4584	0.4473	-2.15	2.48
	33.0	0.3937	0.4114	0.4003	-1.65	2.77
90	33.8	0.3488	0.3614	0.3507	-0.54	3.05
	37.5	0.2718	0.2806	0.2700	0.67	3.93
84	33.6	0.3117	0.3286	0.3182	-2.04	3.27
	35.5	0.2880	0.2858	0.2754	4.58	3.78

It can explain the obtained results by increasing the generation temperature. The strong solution increases, including an enhancement in the difference between the strong and the weak solution, which includes a diminution in the solution mass flow and generation energy needed for heating. The strong solution trends to stabilize, for that reason all parameters above presented also trend to stabilize.

Table 9. Comparison of EECACC thermodynamic performance of 9 refrigerants studied in this work.

Refrigerant	COP	η_{ex}
R600	0. 6437	0.2576
R600a	0.6428	0.2568
R1234yf	0.6388	0.2534
R1243zf	0.6377	0.2525
R744	0.6054	0.2271
R290	0.6400	0.2545
RE170	0.6437	0.2576
R1270	0.6396	0.2541
R152a	0.6431	0.2571

The obtained results conclude that the EECACC has better performance compared with the conventional compression absorption cascade cycle and the generation temperature of EECACC must be not exceed the stabilization temperature 348 K.

The different state point of EECACC is presented in Table 10.

Table 10. The different properties of proposed cycle state point in operating condition cited in Table 5.

No. of state point	T (*K*)	P (*kPa*)	h (*kJ/kg*)	S (*kJ/kg.K*)	X (*g/kg*)	m (*kg/sec*)
1	*268*	*223.1*	*486.5*	*1.82*		*0.08643*
2	*307.9*	*482.8*	*537.2*	*1.867*		*0.6529*
3	*291.2*	*482.8*	*99.12*	*0.3659*		*0.6529*
4	*265.2*	*200.8*	*45.13*	*0.1744*		*0.534*
5	*308.2*	*5.629*	*146.6*	*0.5051*		*0.1169*
6	*290.7*	*5.629*	*73.45*	*0.2606*		*0.1169*
7	*283.2*	*1.228*	*73.45*	*0.2621*		*0.1169*
8	*283.2*	*1.228*	*2519*	*8.9*		*0.1169*
9	*323.8*	*1.228*	*2592*	*9.152*		*0.1169*
10	*308.2*	*1.228*	*76.13*	*0.2415*	*0.5219*	*0.6021*
11	*308.2*	*5.629*	*76.13*	*0.2415*	*0.5219*	*0.6021*
12	*339.2*	*5.629*	*132.9*	*0.446*	*0.5219*	*0.6021*
13	*363.2*	*5.629*	*239.7*	*0.483*	*0.6477*	*0.4852*
14	*324.7*	*5.629*	*169.2*	*0.2795*	*0.6477*	*0.4852*
15	*324.7*	*1.228*	*169.2*	*0.2795*	*0.6477*	*0.4852*
16	*363.2*	*5.629*	*2669*	*8.664*		*0.1169*
25	*268*	*223.1*	*74.79*	*0.2847*		*1.188*
26	*268*	*223.1*	*44.66*	*0.1724*		*1.102*
27	*266*	*223.1*	*40.02*	*0.155*		*1.102*
28	*265.2*	*200.8*	*40.02*	*0.1552*		*0.5354*
29	*263.2*	*186.1*	*40.02*	*0.1553*		*0.5665*
30	*263.2*	*186.1*	*481.3*	*1.831*		*0.5665*
31	*271.7*	*223.1*	*491.6*	*1.839*		*0.5665*
32	*271.2*	*223.1*	*490.9*	*1.836*		*0.6529*

5.4 Effect of the Inlet Temperature of Absorption Refrigerant Cycle to Cascade Heat Exchanger on the Performance of EECACC

The effect of varying the evaporation temperature of the absorption cycle (T_7 in Figure 1) on the coefficient performance, the exergy efficiency and the annual cost of EECACC is presented in Figure5. With increasing of $T_7$from 283 to 289 K, the coefficient of performance decreases from 0.6453 to 0.6304, the exergy efficiency decreases from 0.2592 to 0.2375 and the annual cost increases from 119.1 to 127.4 thousand dollars. The reason behind these results is the work of the compressor increases with increasing the compressor outlet pressure which follows the enhancement of T_7, and the annual cost of the system strongly depends on any variation in the work absorbed by the compressor.

It can be resumed the effect of varying the inlet temperature of absorption cycle refrigerant to the cascade heat exchanger from 283 to 289 K, a diminution in the coefficient of performance and the exergy efficiency of the proposed cycle by 2.36 and 9.14 %, respectively and enhancement in its annual cost by 6.97 %.

It can be concluded in the conception of EECAC cycle, the inlet temperature of absorption cycle to the cascade heat exchanger should be the lower possible.

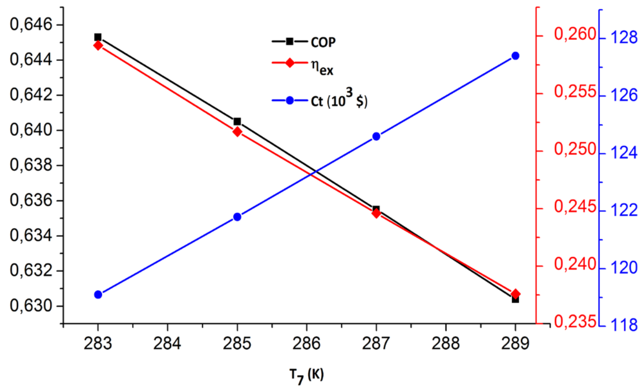


Figure 5. The effect of T_7 on the COP, the exergy efficiency, and the annual cost of EECACC.

Figure 6 shows the effect of sub-cooling heat exchanger efficiency on the coefficient performance, the exergy efficiency and the annual cost of EECACC. The increasing of the sub cooling heat exchanger efficiency from 0.6 to 0.9 includes an enhancement of the coefficient of performance from 0.6423 to 0.6501 and an enhancement of the exergy efficiency from 0.2578 to 0.2601 and a diminution in the annual cost from 119.6 to 118.7 thousand dollar.

These results can be explained by increasing the heat exchanger efficiency, the enthalpy of the ejector secondary fluid inlet increases including an increase in the entrainment ratio, the ejector secondary fluid mass flow as a consequence of a diminution in the evaporator mass flow, and the work of compressor of the first stage and in the whole compressor work.

5.5 Effect of Sub Cooling Heat Exchanger Efficiency on the Performance of ECACC

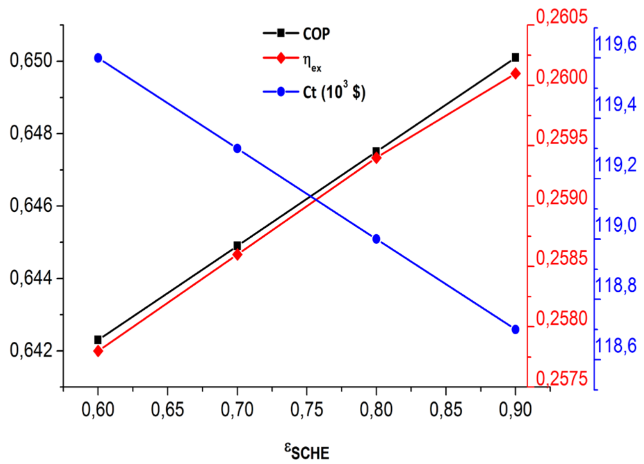


Figure 6. The effect of Sub cooling heat exchanger efficiency of the ejector on the COP, the exergy efficiency, and the annual cost of EECACC.

In can be resumed the effect of varying the sub cooling heat exchanger efficiency from 0.6 to 0.9 by an enhancement in the coefficient of performance and the exergy efficiency of proposed cycle by 0.5 and 0.89%, respectively and diminution in the its annual cost by0.76 %.

These results shows that the choice of sub cooling heat exchanger must be have an efficiency the higher possible.

5.6 Exergy Analysis of EECACC

The exergy destruction of EECACC components is presented in Figure 7. The cascade heat exchanger (CHE), the sub-cooling heat exchanger (SCHE), the absorber, and the generator are the main components destruction of exergy.

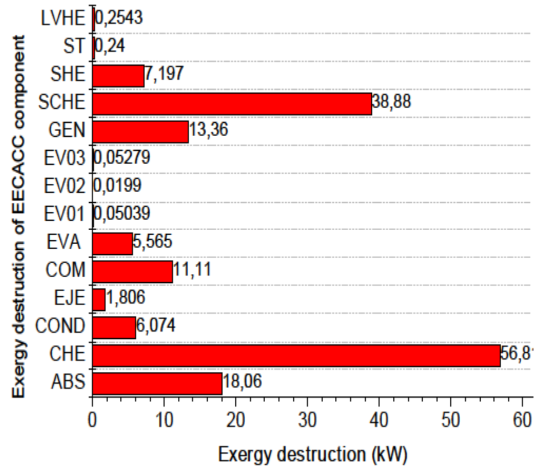


Figure7. The exergy destruction of different components of EECACC.

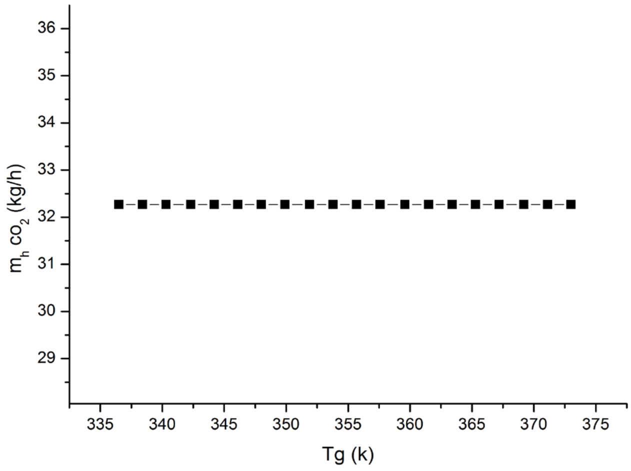
It's clear that the heat exchangers are responsible of 91.67 % of the total exergy destruction of cycle. Thus, for reduce the total exergy destruction of cycle and enhance its exergy efficiency, it's indispensable to develop heat exchange in the heat exchangers by using new materials that have thermo physical propriety more adaptive to heat exchange like as high thermal conductivity and low density or new mechanical construction to develop the convection heat exchange coefficient and the global heat exchange coefficient.

5.7 Environmental Analysis

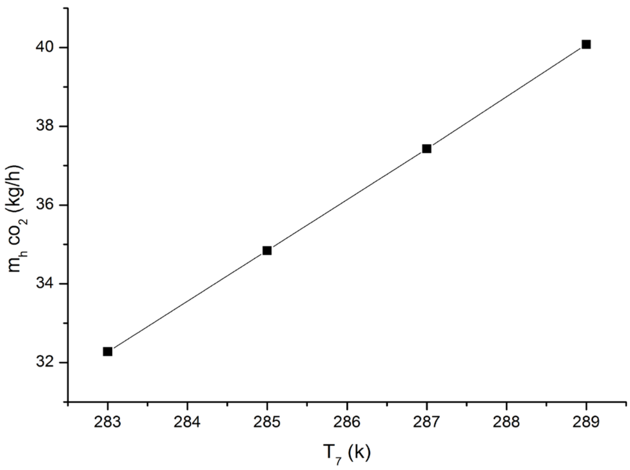
Environmental analysis based on the study of the effect of electrical power consumed by cycle on the environment using the equivalent CO_2 mass emission during one hour of service of the compressor.

Figure 8 presents the effect of the generation temperature, the inlet temperature of absorption cycle refrigerant in the cascade heat exchanger, and the sub-cooling heat exchanger efficiency on the equivalent mass emission of CO_2. The equivalent mass emission of CO_2 is constant and equal to 32.27 kg/h with variation in generation temperature. It can be explained this result by the mass of CO_2 emission is strongly dependent of the work of the compressor and the operating conditions of the vapor compression cycle were invariable with a variation of generation temperature, thus the work of the compressor is constant. But, the mass of CO_2 emission increased from 32.27 to 40.08 kg/h with increasing of the inlet temperature of absorption cycle refrigerant to cascade heat exchange from 283 to 289 K, and the mass of CO_2 emission decreased from 32.28 to 32.5 kg/h with increasing of sub-cooling heat exchanger efficiency from 0.6 to 0.9, it is clear that the variation equivalent emission mass of CO_2 follows the work absorbed by the compressor and the performance efficiency of the cycle.

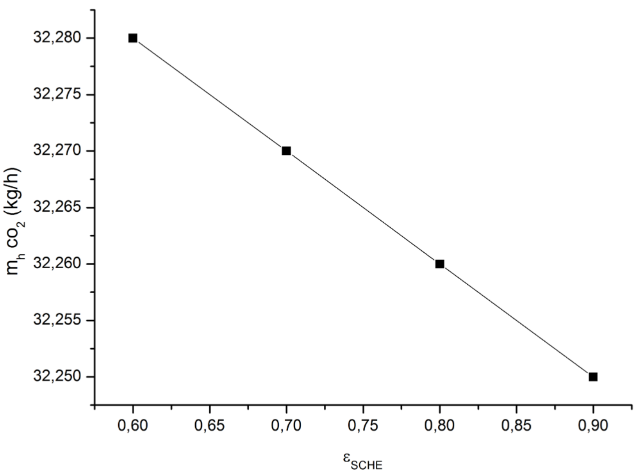
Figure 9 shows that the proposed EECACC cycle emits 32.27 kg/h of CO_2 into the air and the conventional CCACC cycle emits 33.01 kg/h. The reason is that in the proposed cycle, the mass flow rate at the compressor inlet is smaller than the mass flow rate at the compressor inlet of CCACC because the use of an ejector in the EECACC allows for compression of a part of the refrigerant which is the secondary fluid from the ejector to the pressure of exiting the ejector.



(A)



(B)



(C)

Figure 8. The effect of the generation temperature (A), the T_7 (B) and the sub cooling heat exchanger efficiency factor (C) on the hourly equivalent CO_2 mass emission

It can be concluded that the proposed cycle EECACC is more environmentally friendly because its equivalent mass emission of CO2 is lower by 2.29% than the conventional cycle.

From the point of view environmentally, the choice of the inlet temperature of absorption cycle refrigerant to cascade heat exchange should be lower possible and the sub-cooling heat exchanger efficiency should be the higher possible.

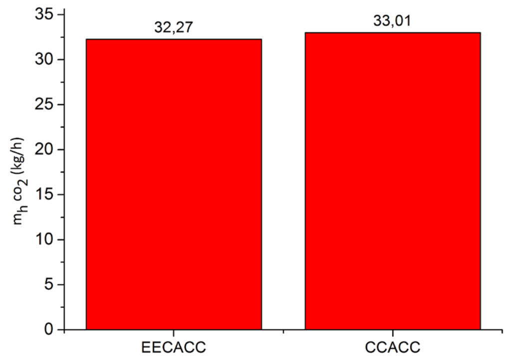


Figure 9. The comparison of equivalent CO_2 mass emission of EECACC and CCACC.

5.8 Investment Cost Analysis

Figure 10 depicts the investment cost of different components of EECACC. The almost investment cost is due to the cost of heat exchangers and the cost of heat exchanger is strongly depending to its area.

The added component the ejector and the sub cooling heat exchanger to the conventional compression absorption cascade cycle enhances in the investment cost by 3.14% in the total investment cost of EECACC.

Many recommendations can be made that the use of novel materials for heat exchangers with low prices and to develop the design of heat exchangers to enhance the global heat exchange coefficient could be diminution the investment price of EECACC.

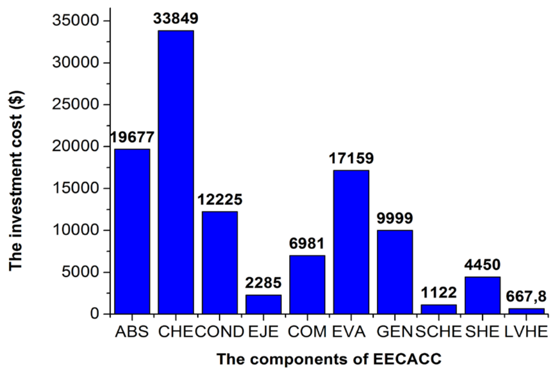


Figure 10. The investment cost of different components of ECACC.

6. Conclusion

In this work, a proposed cycle of enhanced ejector compression absorption cascade cycle is studied in different side energy, exergy, economic, and environmental. The 4E analysis of the enhanced ejector compression absorption cascaded cycle allows the conclusion of the following:

- The RE170 fluid has a higher coefficient of performance and the exergy efficiency of EECACC than 08 fluids studied with low GDP and ODP coefficients.
- The proposed cycle EECACC coefficient of performance and exergy efficiency are higher by 3.27 and 2.7 % than the coefficient of performance and the exergy efficiency

of the conventional compression absorption cascade cycle, respectively.
- The annual cost and the hourly equivalent CO2 emission of proposed cycle are lower by 7.93 and 2.29 % than the annual cost and the hourly equivalent mass of CO2 emission of conventional compression absorption cascade cycle.
- The enhanced ejector compression absorption cascade studied reaches maximum performance at generation equation equal to 348 K.
- There is a generation temperature at which the cycle performance is in maximum, for every operating condition.
- The increasing of the cascade heat exchanger inlet temperature of the absorption cycle refrigerant from 283 to 289 K negatively affected the coefficient performance and the exergy efficiency of the EECACC by 2.36 and 9.14 %, respectively, and positively affected the annual cost and the hourly equivalent CO2 emission of the cycle by 6.97 and 24.20%, respectively
- The cascade heat exchanger inlet temperature of the absorption cycle refrigerant should be the lower possible.
- The increasing sub-cooling heat exchanger efficiency from 0.6 to 0.9 positively affected the coefficient performance and the exergy efficiency of the EECACC by 0.5 and 0.9 %, respectively, and negatively affected the annual cost and the hourly equivalent CO2 emission of the cycle by 0.76 and 0.38 %, respectively.
- The sub cooling heat exchanger efficiency should be the higher possible.
- The cycle heat exchangers like as the cascade heat exchanger, the sub-cooling heat exchanger, the absorber and the generator are responsible for almost all exergy destruction of EECACC followed by the compressor and the ejector, respectively.
- The investment cost of EECACC strongly depends on the heat exchangers and their area.
- The proposed cycle investment cost is higher by 3.14 % than the conventional compression absorption cascade cycle.
- The results obtained in this work are promoted and it is recommended to use the EECACC system for the compression absorption cascade cycle.

The future projections of research can be:

- Simulation of a new alternative refrigerant for replacement purposes.
- Develop new cycle by using double ejector for minimizing the compressor mass flow to enhance efficiency of cycle and reduce the environmental impact.

Conflict of Interest

Author approves that to the best of their knowledge; there is not any conflict of interest or common interest with an institution/organization or a person that may affect the review process of the paper.

Credit Author Statement

Billal Mebarki: Conceptualization, Methodology, Software, Data Curation, Writing- Original draft preparation, Visualization, Investigation, Validation, Writing- Reviewing and Editing.

Nomenclature

COP	Coefficient of performance [-]
P	Pressure $[kPa]$
Ex_i	Exergy destruction $[kW]$
h	Enthalpy $[kJ/kg]$
T	Temperature [℃]
x	Mass fraction of lithium bromide $[g/kg]$
Q	Heat transfer rate $[kW]$
S	Entropy $[kJ/kg.K]$
W_{com}	Compressor work $[kW]$
m	Mass flow $[kg/s]$
Greek symbols	
η_{ex}	Exergy efficiency [-]
ρ	Mass density $[kg/m^3]$
ε	Efficiency [-]
μ	Entrainment Ratio [-]
Subscripts	
0	Reference value
abs	Absorber
con	Condenser
eva	Evaporator
ev	Expansion valve
rv	Reduce pressure valve
gen	Generator
com	Compressor
ejec	Ejector
st	Separator tank
sbhex	Sub cooling heat exchanger
shex	Solution heat exchanger
chex	Cascade heat exchanger
Lvhex	Liquid vapor heat exchanger
i	The ith chemical species
1, 2, .., 32	The state point number

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