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Research Article

Performances Study of Eco-friendly Binary Azeotropic Mixtures Used as Working Fluid in Three Refrigeration Cycles

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Abstract

According to the European (F-gas) regulation, all refrigerants with a global warming potential (GWP) above 150 will be out by 2030. Searching for alternative refrigerants that are environmentally friendly has become an urgent challenge for the refrigeration and air-conditioning sector. Based on their environmental advantages and good thermo-physical properties, azeotropic mixtures have recently gained special interest as substitutes for conventional refrigerants. This study aims to compare the performance of three eco-friendly azeotropic mixtures with the common refrigerant R134a in three refrigeration cycles: the basic cycle (BC), the ejector-expansion refrigeration cycle, and the ejector sub-cooled cycle. The mixtures under study are R1234ze+R600a, R1234yf+R600a, and R1234yf+R290. These mixtures have global warming potential (GWP) of 5.668, 3.8688, and 3.2865 respectively, whereas R134a has a GWP of 1430.
To reach this objective a numerical program was developed using MATLAB software to evaluate the coefficient of performance (COP), and the cooling capacity of the three refrigeration cycles using the studied eco-friendly mixtures and were compared with those of the commonly used R134a refrigerant. The entrainment ratio was also compared for the two ejector cycles using these refrigerants. The simulation was realized for condensing temperatures (Tc) selected between 30 and 55°C and evaporation temperatures (Te) ranging between -10 and 10°C. The results have shown that the eco-friendly azeotropic mixture R1234yf+R290 (GWP=3.51) has the best performances compared to the two other mixtures and they are close to those of R134a. On the other hand, the ejector expansion refrigeration cycle has exhibited a high coefficient of performance compared to the basic cycle and ejector sub-cooled cycle, and a high entrainment ratio compared to the ejector sub-cooled cycle for all used refrigerants. However, the ejector sub-cooled cycle gave a better cooling capacity than the other cycles. According to the obtained results, the azeotropic mixture R1234yf+R290 apart from its excellent environmental properties yields better performances in most of cases, this confirms that it could be a suitable substitute for conventional working fluid R134a which has a great global warming potential.

Keywords: *Azeotropic mixtures; ejector cycle; entrainment ratio; global warming potential*.

1. Introduction

Finding high-performing, environmentally friendly working fluids with low environmental effects is a major challenge facing the refrigeration sector. The European F-gas regulation mandates the phase-out of refrigerants with GWPs greater than 150 by 2030 [1].

The usually used refrigerants in the applications of thermodynamic machines such as heat pumps, air-conditioning, and refrigeration systems have a good performance but a high global warming potential (GWP), which has a significant negative influence on the environment and adds significantly to atmospheric greenhouse gas concentrations [2]. In refrigeration engineering, finding effective refrigeration systems is also crucial. The introduction of the technology of ejector expansion in the cooling systems to improve cooling efficiency was proposed for the first time by Kornhauser [3] in 1990. Several studies have focused on the amelioration of the conventional vapor-compression refrigeration cycle.

Xing et al. [4] have proposed a novel vapor-compression refrigeration cycle with mechanical sub-cooling using an ejector to improve the performance of a conventional single-stage vapor-compression refrigeration cycle. Their simulation results have shown the novel cycle displays volumetric refrigeration capacity improvements of 11.7% with R404A and 7.2% with R290 when the evaporator temperature ranges were from -40 to -10 °C, and the condenser temperature was 45 °C on the other hand, the novel cycle has achieved COP improvements of 9.5% with R404A and 7.0% with R290. In addition, they deduced that the improvement of the COP and cooling capacity of this novel cycle largely depends on the operation pressures of the ejector.

Yang et al. [5] studied a novel combined power and ejector-refrigeration cycle using a zeotropic mixture, and it was compared with a conventional combined power and ejector-refrigeration cycle. It was found that the cycle exergy

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achieves a maximum value of 10.29% with a mixture of isobutane/pentane (40%/60%), and the thermal efficiency gets a maximum value of 10.77% with a mixture of isobutane/pentane (70%/30%). The mixture of isobutane/pentane (80%/20%) has given a maximum temperature glide in the evaporator of 15.09 K.

In another study, based on the first and second laws of thermodynamics, a theoretical analysis of the performance of this new cycle was carried out by Yari and Sirousazar [6]. It was found that the COP and second law efficiency values of the new ejector-vapour compression refrigeration cycle are on average 8.6 and 8.15 % higher than that of the conventional ejector-vapour compression refrigeration cycle with R125. It was also shown that the COP of the new cycle is 21 % higher than that of conventional vapor compression.

Disawas and Wongwises [7] conducted an experimental study comparing the performance of a two-phase ejector refrigeration cycle with that of a conventional refrigeration cycle. The results showed that the coefficient of performance of the two-phase ejector refrigeration cycle was higher than that of the conventional refrigeration cycle across all experimental conditions. However, it was observed that as the heat sink temperature increased, the growth became relatively smaller. A literature review on two-phase ejectors and their applications in compression refrigeration systems and heat pumps was conducted by Sarkar [8]. The review revealed that both theoretical and experimental studies have confirmed that using an ejector as an expansion device can significantly improve the performance of subcritical and transcritical refrigeration and heat pump cycles. The review also showed that the improvement in energetic or exergetic performance by using an ejector is greatly influenced by the cycle operating conditions, the working fluids used, and the ejector geometries.

Besagni et al. [9] have presented a literature review on ejector refrigeration systems and working fluids. They deeply analyzed ejector technology and behavior, refrigerant properties and their influence on the ejector performance, and all of the ejector refrigeration technologies, with a focus on past, present, and future trends. They concluded that the use of heat-driven ejector refrigeration systems could be a promising alternative to traditional compressor-based refrigeration technologies for energy consumption reduction.

The incorporation of an ejector into the vapor compression cycle leads to improving the COP by reducing the throttling loss associated with the expansion device. A numerical simulation using a one-dimensional model based on mass balances was made by Nehdi et al. [10]. According to the simulation results of the improved cycle, it has been shown that the geometric parameters of the ejector design have considerable effects on the system's performance. A comparison of four different refrigeration cycles using ternary mixtures was proposed by Maalem et al. [11]. Their results showed that the cycle with booster and ejector gave better performance than the other studied cycles.

Many researchers have studied the use of pure refrigerants as working fluids in the ejector refrigeration cycle. Sarkar [12] conducted a comparative analysis of the performance of three natural refrigerants using the ejector expansion refrigeration cycle. The findings indicated propane yields a maximum COP improvement of 26.1 % followed by isobutane (22.8 %) and ammonia (11.7 %) for studies ranges. A thermodynamic analysis of an air conditioning system is conducted by Aisyah and Ariyadi [13] to assess the performance of R1224yd and compared to R123 and R245fa. The system is analyzed from a thermodynamic perspective and key performance indicators such as the Coefficient of Performance and exergy efficiency. The results are then compared to R245fa and R123. Results showed that R1224yd offers better performance than R245fa which has 1-3% higher performance value and exergy efficiency and has comparable performance to R123.

Rostamnejad and Zare [14] proposed a new ejector-expansion refrigeration cycle, and a comparison was made with the standard ejector-expansion refrigeration cycle and conventional vapor compression refrigeration system. Six environmentally friendly refrigerants were utilized as working fluids. Their results showed that, among the six investigated refrigerants, R1234ze is the best one for which the proposed system has 5.7% and 15.5% higher exergy efficiency values than the standard ejector-expansion refrigeration cycle and conventional vapor compression refrigeration, respectively, at a condensing temperature of 40 °C and evaporation temperature of 5 °C.

Ma et al. [15] conducted a numerical study of the fundamental refrigeration cycle with an ejector, using several hydrocarbon refrigerants, including propane, butane, isobutane, and propylene, as working fluids. The findings have shown that the ejector-expansion refrigeration cycle using hydrocarbons has greater COP, volumetric cooling capacity, and exergy efficiency, as well as lower exergy destruction compared with the standard refrigeration cycle. In another hand, they have noticed that propane and propylene have better performance than isobutene and butane.

A numerical model based on the energetic and exergetic methods has been developed by Maalem et al. [16] to compare the eco-friendly refrigerant R13I1 with the usually used fluid R134a in the ejector-expansion refrigeration cycle. The thermodynamic performances studied include the coefficient of performance (COP), the entrainment ratio (μ), the exergy destruction, and the exergy efficiency. Their results have indicated that the R13I1 has a better performance in terms of the entrainment ratio and the coefficient of performance, as well as lower exergy destruction compared to R134a.Li et al. [17] made a theoretical study on the performance characteristics of the ejector-expansion refrigeration cycle using R1234yf as refrigerant have been investigated. They showed that the EERC R1234yf has better performance than the standard refrigeration cycle, and the improvement is more important under the conditions of a higher condensation temperature and a lower evaporation temperature. The coefficient of performance and volumetric cooling capacity improvements of the ejector-expansion refrigeration cycle over the standard refrigeration cycle are also greater than that of the cycle using R134a as a working fluid.

Lucas et al. [18] have presented an ejector operation characteristic for a CO_2 ejector based on experimental data, which is designed to be used in system simulations such as the refrigeration cycle. Based on experimental data, correlations for the ejector efficiency and the driving mass flow rate were determined and used. The correlation for the ejector efficiency, which uses dimensionless coefficients, has predicted the experimental data within 10%.

Ozone depletion potential (ODP) and global warming potential (GWP) have recently become the two fundamental parameters in new refrigerants investigation, finding environmentally friendly refrigerants is therefore essential.

Binary azeotropic mixtures emerge as the most viable alternatives to conventional refrigerants in the vapor compression refrigeration cycle, Calleja-Anta et al [19] have studied experimentally the mixture RE170/R600 as a potential drop-in refrigerant of R600a. For a given operating condition, the energy performance of RE170/R600 mixtures with a maximum proportion of 27.5 % of RE170 has been tested using a water-to-water single-stage compression cycle, measuring COP increments from 10.1 to 17.6 % in relation to R600a. Then, the blend RE170/R600 (15/85 %), considered as the potential drop-in fluid of R600a, has been tested in a wide range of operating conditions, concluding that it offers the same cooling capacity as R600a but with COP increments from 12.6 % to 17.6 %.

Benbai et al. [20] have numerically studied six azeotropic mixtures, R1234yf + R290, R1234yf + R152a, R1234yf + R600a, R134a + R290, R134a + R600a, and R1270 + R134a, in single-stage steam compression refrigeration system. The effect of the entrainment ratio on the coefficient of performance has been investigated for the six refrigerants. The simulation results showed that the R1234yf + R290 mixture has given the highest coefficient of performance and entrainment ratio.

Using a constant-pressure two-phase ejector model, a numerical study was realized by Zhao et al. [21] to study the performance evolution of the ejector-expansion refrigeration cycle, the zeotropic mixture R134a/R143a was selected as working fluid. The simulation results reveal that the cycle COP increases first and then decreases as the mass fraction of R134a increases. The COP has reached a maximum value of 4.18 with a mass fraction of 0.9 and has yielded a minimum value of 3.66 with a mass fraction of 0.5. With mixture 0.9/0.1, the COP improvement has reached a maximum value of 10.47%.

The performance of the refrigeration cycle with ejector using four zeotropic binary mixtures based on R1234yf (R1234yf + R152a, R1234yf + R134a, R1234yf + R32, and R1234yf + R125) was investigated by Mehemmai et al. [22].The effects of key operating parameters such as evaporation temperature, condensation temperature, and mass fraction were also analyzed. Their results showed that the COP of the two mixtures (R1234yf + R152a and R1234yf + R134a) was not affected by the change in mass fractions on the other hand the mass fractions variation had a significant effect on the COP of the two other mixtures: R1234yf + R32 and R1234yf + R125. Among studied mixtures and fractions used, the mixture R1234yf + R152a has given the highest COP with a mass fraction of 0.75.

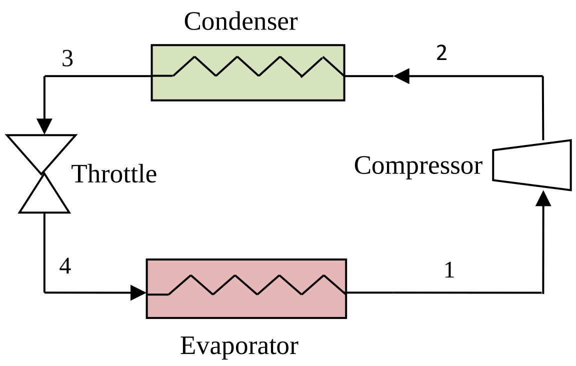
To investigate the performance of an ejector refrigeration cycle using three CO_2-based mixtures (CO_2+R290, CO_2+R1234yf, CO_2+R600a) in subcritical mode, and CO_2+R116 in transcritical mode, Abdou et al. [23] have developed a simulation program. Results have shown that the suction nozzle pressure drop influences significantly the cycle performance, but does not affect the entrainment ratio of the ejector. On the other hand, they found that the maximum performance of refrigeration cycles, in sub or trans-critical mode, was proportional to the evaporation temperature and was inversely proportional to the temperature of the condenser-gas cooler.

Liu et al. [24] established a model for a refrigeration cycle with an ejector using zeotropic refrigerants as a working fluid. Different mixtures were studied (R123/R245fa, R245fa/R141b, R141b/RC318, R245fa/R134a, R245fa/R22, R141b/R134a, R245fa/R143a, and R141b/R22). The performance of the ejector refrigeration cycle was studied, and results have indicated that the outlet temperatures of both the generator and evaporator can be increased using a zeotropic mixture compared to that using a pure refrigerant, contrariwise the average temperature difference of the heat transfer in the condenser is larger than that of the pure refrigerant. Among the studied refrigerants, using R245fa/R22 (0.3/0.7) as the working medium yields the refrigeration cycle with the best COP (0.293), which is 4% and 22% higher than those using R22 and R245fa, respectively. The results have revealed the advantages of zeotropic refrigerants.

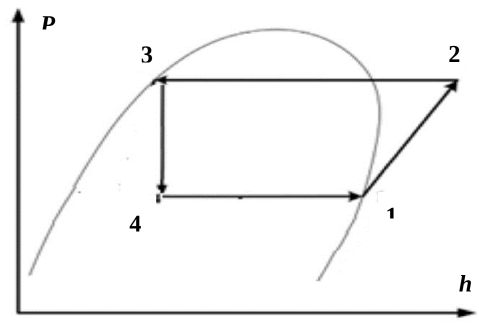
From the literature review, it was noticed that the types of cycles as well as the used refrigerants have an important influence on the final performances of the refrigeration Machinery.This work aims to investigate the performances of three different refrigeration cycles using environmentally friendly binary mixtures and compare them with the usually used refrigerant R134a. The selected azeotropic mixtures are R1234ze+R600a, R1234yf+R600a, and R1234yf+R290, they have respectively a global warming potential (GWP) equal to 5.668, 3.8688, and 3.2865.

2. Studied Cycles

In this study, performances of three refrigeration cycles using azeotropic eco-friendly mixtures as working fluids are investigated.



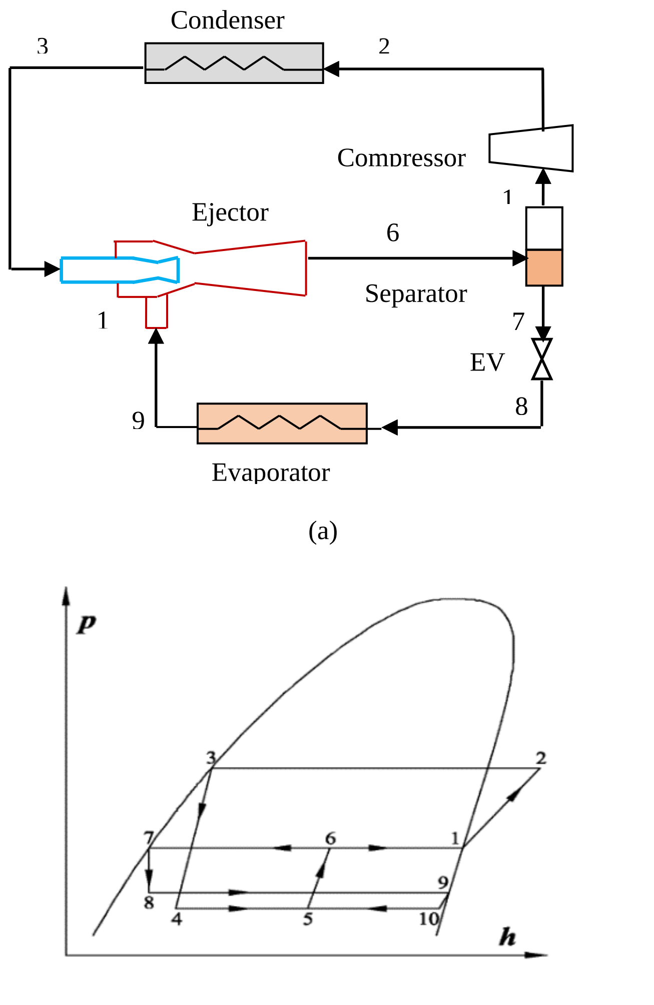
(a)



(b)

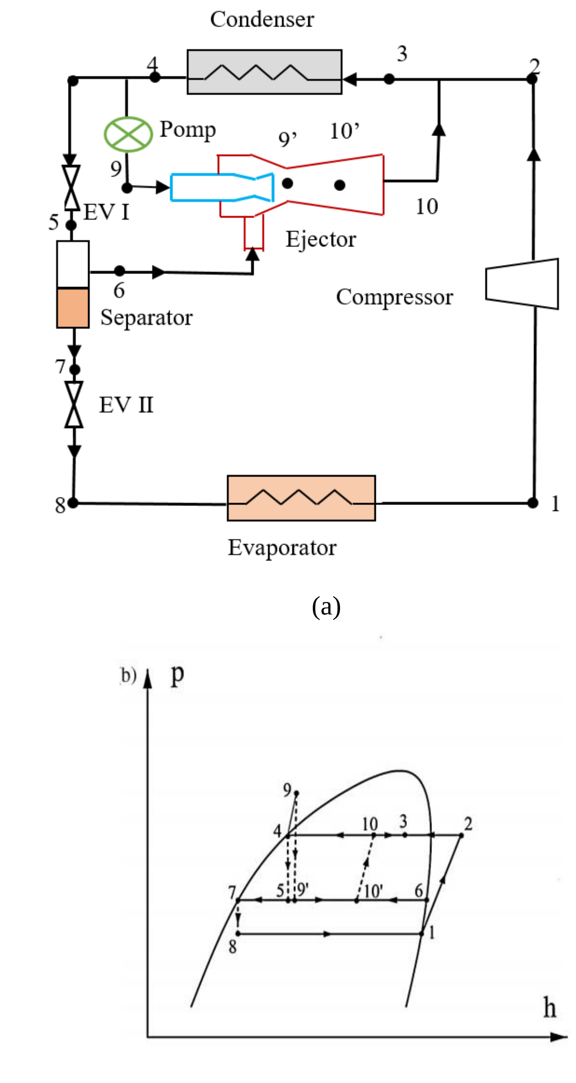
Figure 1. Schematic cycle (a) and P–h diagram (b) of the Basic Cycle [23].

The schematic representation and P-H diagrams of the concerned cycles are represented in Figure.1 for the basic cycle (BC), in Figure.2 for the ejector-expansion refrigeration cycle (Configuration 1), and in Figure.3 the ejector sub-cooled cycle (Configuration 2).



(b)

Figure 2. Schematic cycle (a) and P–h diagram (b) of the ejector-expansion refrigeration cycle (Configuration 1) [16].



(b)

Figure 3. Schematic cycle (a) and P–h diagram (b) of the ejector sub-cooled cycle (Configuration 2) [3].

Since the ejector is the more important component in the ejector cycles, it determines the performance of the cycle. The ejector model is classified into two types: constant pressure mixing model and constant area mixing model. According to previous studies (Sumeru et al [25]; Khalil et al.[26]; He et al. [27]; Li et al. [17]; Sarkar [8]; Xing et al. [4]) the constant-pressure mixing model is better than the constant-area mixing model. In this study, this model was adopted. For the ejector, the mathematical model is detailed below.

3. Mathematical Modelling and Simulation

3.1 Ejector Modelling [17] [4]

The entrainment ratio μ is an important parameter for assessing the ejector's performance it is defined as following:

$$\mu = \frac{m_s}{m_p} \quad (1)$$

Where m_p and m_s represent respectively, the mass flow rates of the primary and secondary flows.

The thermodynamic analysis has been conducted on the following common assumptions [17].

- Steady-state conditions are assumed.
- The throttling process in the expansion valve is isenthalpic.
- The azeotropic composition of each binary mixture remains constant throughout the process.
- Neglecting the pressure drop in heat exchangers and connecting pipes.
- The entry and exit velocities of the ejector are neglected.
- Engine flow and suction flow reach the same pressure at the inlet of the constant-area mixing section of the ejector, and no mixing occurs between the two flows before the inlet of the mixing section.
- The refrigerant leaving the condenser and evaporator ports is saturated.
- Heat does not transfer with the environment surrounding the system except in the condenser.
- The compressor has isentropic efficiency.

3.2 Motive Nozzle Outlet

The velocity of the motive fluid in the exit nozzle is given by:

$$u_{p,out} = \sqrt{2(h_{p,in} - h_{p,out})} \quad (2)$$

$$h_{p,out} = h_{p,in} - \eta_m(h_{p,in} - h_{p,out,is}) \quad (3)$$

Where$h_{p,in}$, is the inlet specific enthalpy of the primary fluid, $h_{p,out,is}$ is the exit enthalpy through an isentropic expansion process in the nozzle.

3.3 Suction Nozzle

For the ejector-expansion refrigeration cycle, at the suction nozzle outlet, the following equations can be applied:

$$h_{s,out} = h_{s,in} - \eta_s(h_{s,in} - h_{s,out,is}) \quad (4)$$

$$u_{s,out} = \sqrt{2(h_{s,in} - h_{s,out})} \quad (5)$$

For the ejector sub-cooled cycle at the suction nozzle the velocity is neglected Xing et al [4]

3.4 Mixing Chamber

The velocity exiting the mixing chamber for the ejector-expansion refrigeration cycle is given by:

$$u_{mix,out} = \sqrt{\eta_m}\ \left(\frac{1}{(1+\mu)}u_{p,out} + \frac{\mu}{(1+\mu)}u_{s,out}\right) \quad (6)$$

And for the ejector sub-cooled cycle it is given by:

$$u_{mix,out} = \sqrt{\eta_m}\ \left(\frac{1}{(1+\mu)}u_{p,out}\right) \quad (7)$$

The enthalpy and the entropy of the refrigerant at the mixing chamber for the ejector-expansion refrigeration cycle are given by:

$$h_{mix,out} = \frac{1}{(1+\mu)}\left(h_{p,out} + \frac{u_{p,out}^2}{2}\right) + \frac{\mu}{(1+\mu)}\left(h_{s,out} + \frac{u_{s,out}^2}{2}\right) - \frac{u_{mix,out}^2}{2} \quad (8)$$

$$s_{mix,out} = s(h_{mix,out}, p_{mix,out}) \quad (9)$$

The enthalpy of the refrigerant at the mixing chamber for the ejector sub-cooled cycle is given by:

$$h_{mix,out} = \frac{h_{p,in} + \mu\, h_{s,in}}{(1+\mu)} - \frac{u_{mix,out}^2}{2} \quad (10)$$

3.5 Diffuser Section Model.

The following relation calculates the specific enthalpy at the diffuser outlet:

$$h_{d,out} = h_{mix,out} + \frac{u_{mix,out}^2}{2} \quad (11)$$

And for the ejector sub-cooled cycle it given by:

$$h_{d,out,is} = h_{mix,out} + \eta_d(h_{d,out} - h_{mix,out}) \quad (12)$$

The ideal specific enthalpy of the mixed fluid at the diffuser's output can be obtained by using the definition of the diffuser's isentropic efficiency η_d for the ejector-expansion refrigeration cycle can be written as:

$$h_{d,out,is} = h_{mix,out} + \eta_d(h_{d,out} - h_{mix,out}) \quad (13)$$

For the ejector sub-cooled cycle, the specific enthalpy at the diffuser outlet is given by:

$$h_{d,out,is} = h(P_d, s_{out}) \quad (14)$$

The pressure and vapor quality of the refrigerant outlet the ejector-expansion refrigeration cycle are expressed by:

$$p_{d,out} = p(h_{d,out,is}\ , s_{mix,out}) \quad (15)$$

$$x_{d,out} = x(h_{d,out}\ , p_{d,out}) \quad (16)$$

To verify the preliminary input value for the entrainment ratio (μ), the following conditions must be satisfied:
For the ejector-expansion refrigeration cycle

$$x_{d,out} = \frac{1}{1+\mu} \quad (17)$$

From the above equations neglecting the diffuser outlet velocity, the entrainment ratio for the ejector sub-cooled cycle is summarized:

$$\mu = \sqrt{\frac{\eta_n\eta_m\eta_d(h_{p,in} - h_{p,out})}{(h_{d,out,is} - h_{mix,out})}} - 1 \quad (18)$$

The refrigeration COP (coefficient of performance) of the cycles can be expressed as:

$$COP = \frac{q_{col_{evap}}}{w_c} \quad (19)$$

$$COP = \frac{q_{col_{evap}}}{w_c + w_P} \quad (20)$$

The Operational parameters for both configurations are presented In Table 1, and Table 2 shows the equations used to model each component of the cycles.

Table 1. Operational Parameters for Both Configurations (Li et al [17] *Maalem et al* [11]*; Xing et al* [4]*).*

Parameter	Symbol	Value
Evaporating temperature	Te (°C)	5
Condensing temperature	Tc (°C)	40
Motive nozzle efficiency	η_n (%)	85
Suction nozzle efficiency	η_s (%)	85
Mixing section efficiency	η_m (%)	95
Diffuser efficiency	η_d (%)	85
Efficiency pump	η_p (%)	75

Based on the mathematical model built, a computer program was developed in MATLAB and the refrigerants thermodynamic properties were obtained using REFPROP Version 9.0 to investigate the performance of the studied mixtures used as working fluid in three refrigeration cycles.

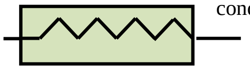
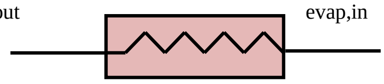
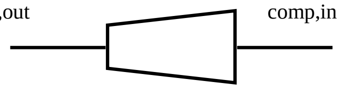
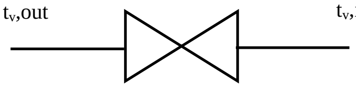
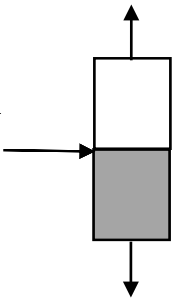
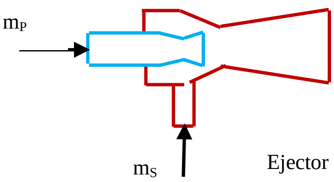
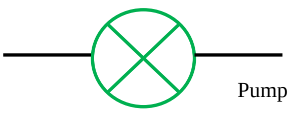
The detailed flowchart for the two ejector cycles calculation procedure is presented in Figure 4.

4. Results and Discussion

4.1 Environmental and Critical Properties of Working Fluids.

The environmental properties (GWP and ODP) have become the most important properties in the research and development of alternative working fluids. The critical and environmental properties of the azeotropic mixtures concerned in this study are presented in Table 3.

Table 2. Components Characteristic Equations.

Component	Characteristic Equations
 Condenser	BC $q_{col} = h_{cond,out} - h_{cond,in}$ Configuration1. $q_{col} = \frac{h_{cond,out}-h_{cond,in}}{1+\mu}$ Configuration 2. $q_{col} = 2(h_{cond,out} - h_{cond,in})$
 Evaporator	(BC): $q_{col} = h_{evap,out} - h_{evap,in}$ Configuration 1 $q_{col} = \frac{\mu\left(h_{evap,out}-h_{evap,in}\right)}{1+\mu}$ Configuration 2 $q_{col} = m'_1\left(h_{evap,out} - h_{evap,in}\right)$
 Compressor	BC : $w_c = h_{c,out} - h_{c,in}$ $\eta_c = 0.874 - 0.0135\pi$ [28] Configuration 1 $w_c = \frac{h_{c,out}-h_{c,in}}{1+\mu}$ $h_{c,out} = h_{c,in} + \frac{h_{c,out,is}-h_{c,in}}{\eta_c}$ Configuration 2 $w_c = m_{c,in}(h_{2is} - h_1)/\eta_c$ [4]
 Throttle valve	For BC, Configuration 1 and 2 $h_{tv,out} = h_{tv,in}$
 Separator	$h_{Sp,L,out} = h(p_{d,out}, x_{d,out} = 0)$ $h_{Sp,V,out} = h(p_{d,out}, x_{d,out} = 1)$
 Ejector	Calculation algorithm proposed by Li et al. [17] for Configuration 1. And by Xing et al [4] for Configuration 2
 Pump	$w_p = m_p(h_{9s} - h_4)/\eta_p$

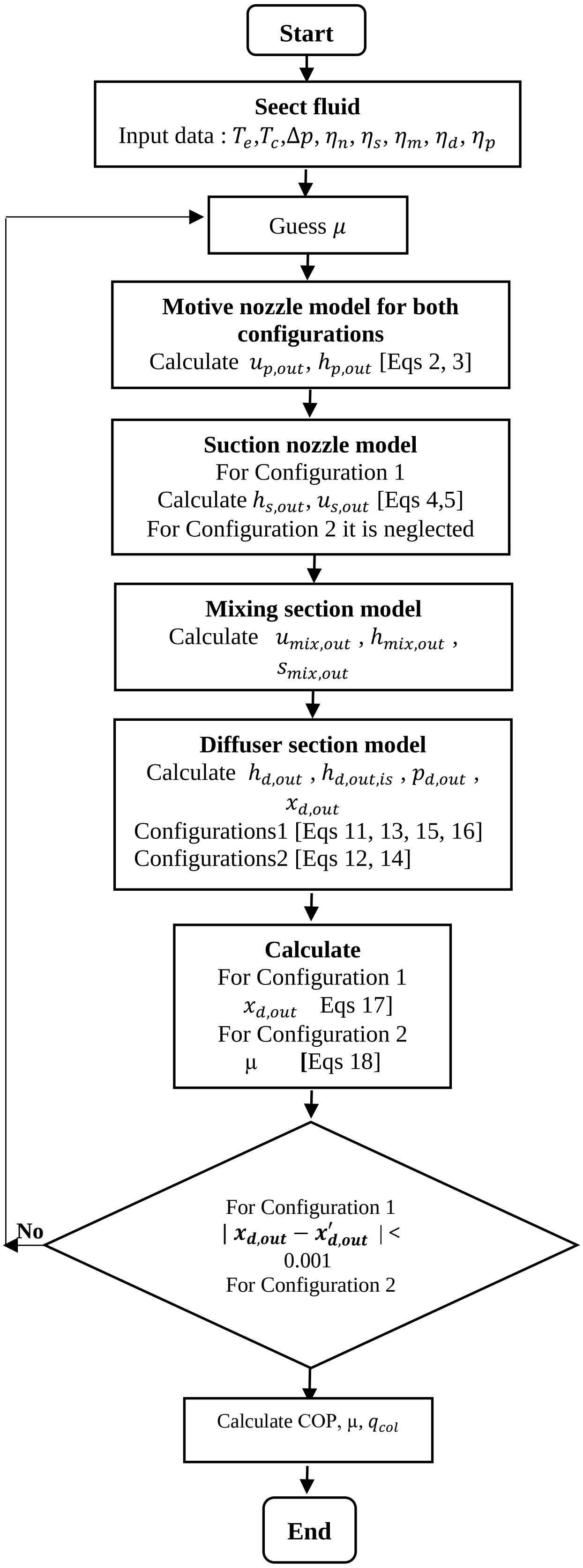


Figure 4. Flowchart for both ejector cycles calculation procedure.

The different values are obtained using the equations below:

Critical temperature T_c

$$T_{c_{mixture}}(k) = \sum_i^n x_{pure,\ i} * Tc_{pure,\ i} \tag{21}$$

Critical pressure P_c

$$P_c(k)_{mixture} = \sum_i^n x_{pure,\ i} * Pc_{pure,\ i} \tag{22}$$

Where $x_{pure,\ i}$ represents the molar fraction of component *i* in the mixture.

Table 3. Environmental and critical properties of working fluids.

Working Fluids	GWP	Critical Temperature $T_c(k)$	Critical Pressure $P_c(MPa)$	Mass Fraction*	Molar Fraction
R1234ze +R600a	5.40	390.940	3.6365	0.7972/ 0208	0.6670/ 0.33
R1234yf +R600a	3.93	373.098	3.4161	0.9285/ 0.0715	0.8686/ 0.1314
R1234yf +R290	3.51	369.301	4.0003	0.4879/ 0.5121	0.2887/ 0.7113
R134a	1430	374.210	4.0593	/	/

* For those values of the mass fractions, the binary mixtures are azeotropic.

Global warming potential of mixtures relation [11]:

$$GWP_{mixture} = \sum_i^n GWP_{pure,\ i} * w_{pure,\ i} \tag{23}$$

Where $w_{pure,\ i}$ represents the mass fraction of component *i* in the mixture.

From values of the Table 3, it can be seen that the three azeotropic mixtures exhibits a global warming potential lower than six, while the phase-out R134a has high GWP (GWP=1430).

4.2 Validation of Developed Code.

Before using the developed program to compare the performances of azeotropic mixtures with the traditional single fluid R134a in the three refrigeration cycles concerned in this study, the developed program was validated with found studies in the literature.

The developed program has been validated under the same operating conditions, by comparing the values of the entrainment ratio with the results reported by Maalem et al. [11] and the results of volumetric refrigeration capacity reported by Xing et al. [4].

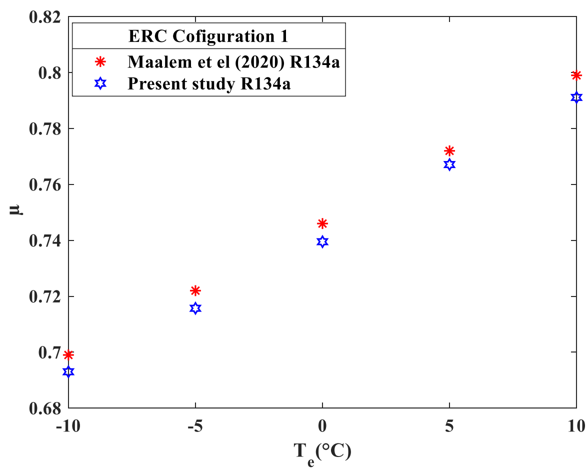


Figure 5. Validation of developed code with li el al [16].

From Figure 5 the entrainment ratio evolves from 0.699 to 0.799 when Te varies between -10 to 10 °C for Maalem et al and in this study, it varies between 0.693 and 0.791 for the same Te variation. Similarly, for Figure 6 the volumetric refrigeration capacity varies between 799.556 kj/m^3 and 2877.959 kj/m^3 when Te evolves from -40 to -10 °C for Xing et al and in this study, it varies between 797.748 kj/m^3 and 2876.283 kj/m^3 for the same Te variation. A good agreement can be noted between the results obtained from the developed program and those of the references.

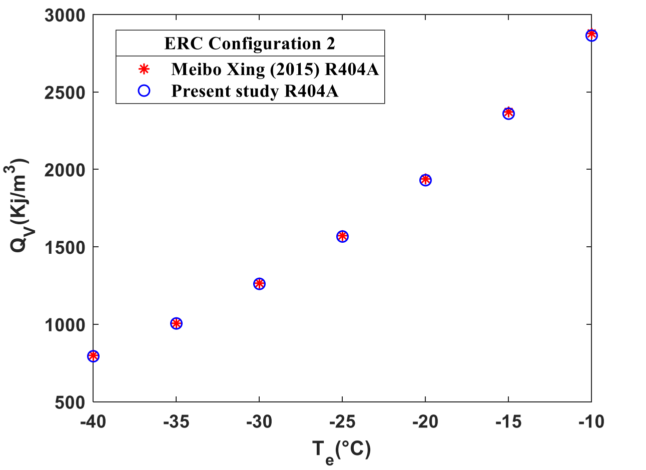


Figure 6. Validation of developed code with Xing et al [3]. For T_c=25°C and pressure ratio =1.14.

4.3 Performances Computation of Working Fluids

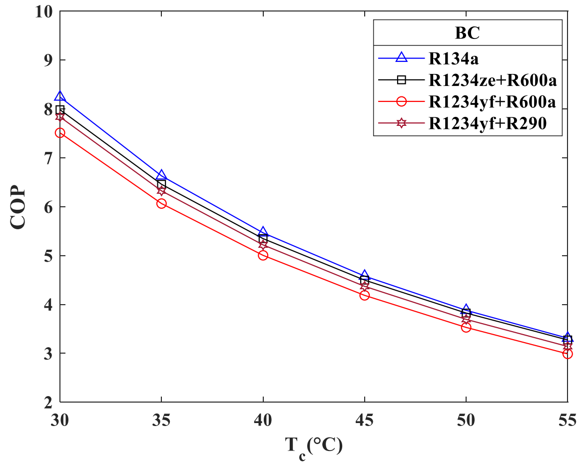
4.3.1 Influence of Condensing Temperatures on Performances

In this section, the effects of condenser temperature (Tc) for a constant evaporation temperature (Te) of 5 °C on the performances of the basic cycle, the ejector-expansion refrigeration cycle, and the ejectorsub-cooled cycle using the investigating fluids are presented.

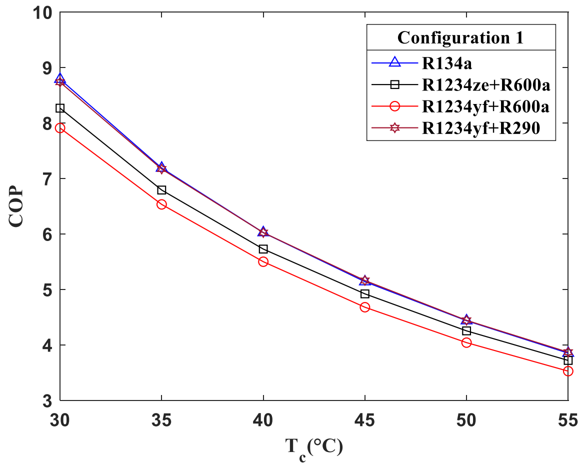
With the same condenser outlet temperatures (30 to 55) °C, Figure 7 shows the simulated results of the variation of the coefficient of performance (COP) values of the three refrigeration cycles using the investigating fluids. It was noted that the COP decreases in all refrigeration cycles with the increase in condenser temperature. This decrease is directly linked to the variation in enthalpy of the condenser in the refrigeration cycles, while the enthalpy at the evaporator outlet of cooling cycles remains constant.

For the basic cycle (Figure.7 (a)), it was observed that at low condensation temperatures, the COP value obtained with the single working fluid R134a is higher than those of the three investigated working fluid mixtures it decreases from (8.24 to 3.31). However, at high condensation temperatures the COP value obtained with the working fluids R1234ze+R600a are very close (8.04 to 3.3) of those of R134a and higher than the two other working fluid mixtures. The COP values of R1234yf+R600a and R1234yf+R290 decreases from (7.81 to 3.27) and (7.82 to 3.05), respectively, as the condensation temperatures increase from 30 to 55 °C.

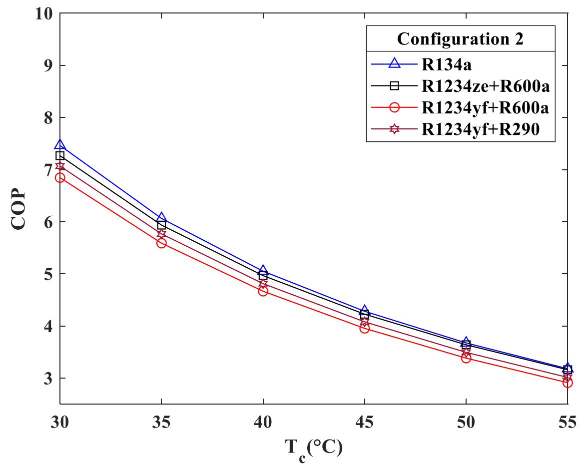
For the ejector-expansion refrigeration cycle (as shown in Figure.7 (b)), it was noticed that the COP values obtained with both working fluids R134a and R1234yf+R290 are nearly identical. They decrease from (8.7875 to 3.8501 and (8.8605 to 3.8828), respectively and they are higher than those of R1234ze+R600a and R1234yf+R600a which the COP decreases from (8.2626 to 3.7417) and (7.5778 to 3.4566), respectively, for all the condensation temperatures. In the case of the ejector sub-cooled cycle (Figure.7 (c)), the COP values obtained with both working fluids R134a and R1234ze+R600a are close, they decrease from (7.4632 to 3.1808) and (7.3193 to 3.1901), respectively. The differences in the COP values between the three cooling cycles can be explained due to the different architectures of each cycle, where the thermodynamic losses (irreversibility) are lower in the architecture of the ejector-expansion refrigeration cycle than the other cooling cycles, which explain the high COP of this cycle.



(a)



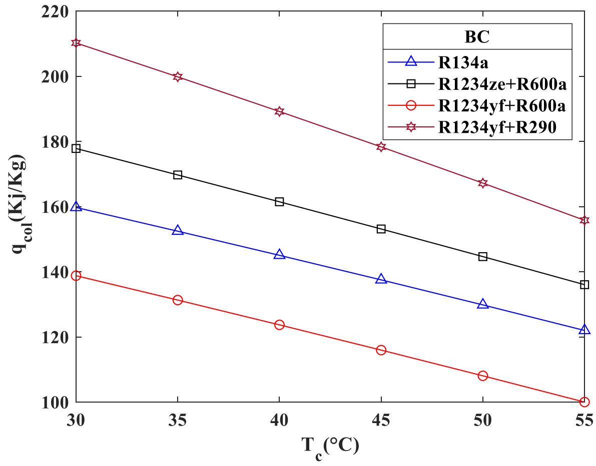
(b)



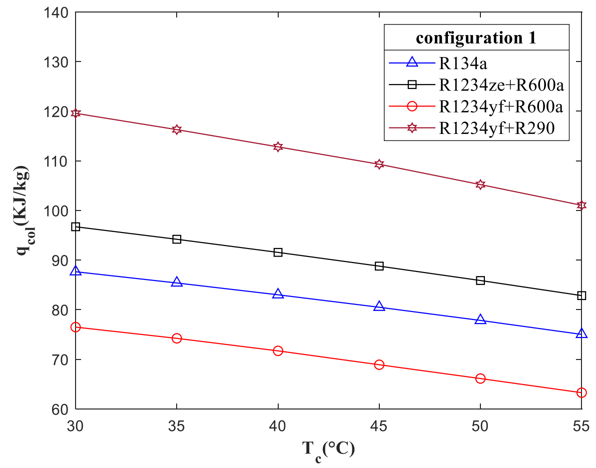
(c)

Figure 7. Influence of condensing temperatures on the COP of refrigeration systems.

Figure 8 shows the effect of the condensing temperatures on the cooling capacity of investigating working fluids in the three refrigeration systems. From the obtained numerical results, it is noticed that the cooling capacity decreases with the condensing temperatures. These results can be explained by the fact that the increase of the condensation temperature implies an increase in enthalpy at the condenser outlet of cooling cycles, while the enthalpy at the evaporator outlet of cooling cycles remains constant, which causes a reduction in cooling capacity and consequently, reduction in COP of cooling cycles.



(a)



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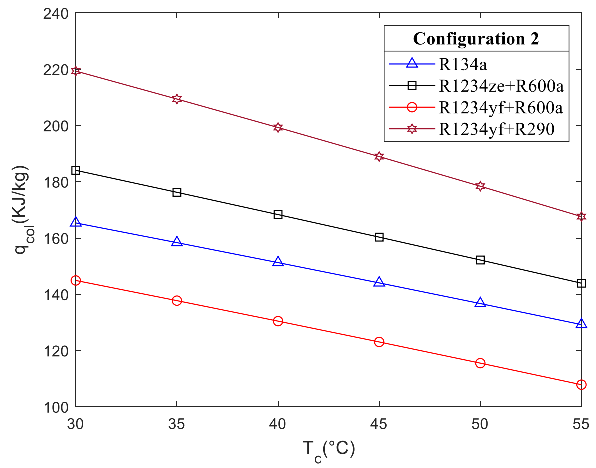


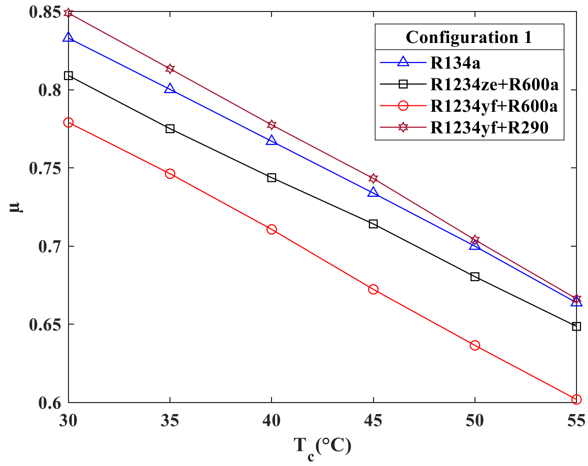
(c)

Figure 8. Influence of condensing temperatures on the cooling capacity of refrigeration systems.

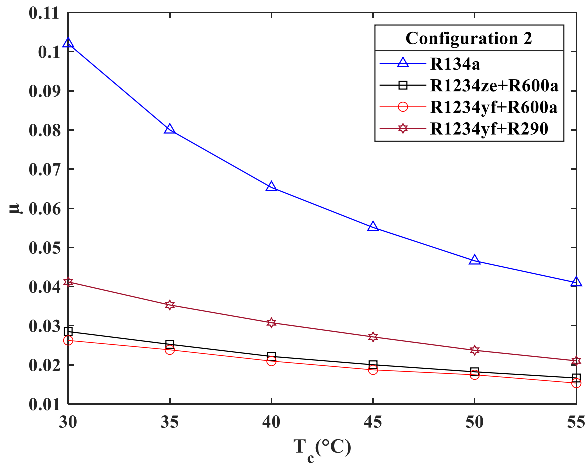
The results showed that the maximum values of the cooling capacity in all three cycles are obtained where they work with the mixture R1234yf+R290. It can be also noted that the two mixtures R1234yf+R290 and R1234ze+R600a give a better cooling capacity than R134a.

By comparing the cooling capacity values of the three studied cooling cycles, it can be seen that the ejector sub-cooled cycle (Figure.8 (c)) exhibits the highest cooling capacity (261.847 to 202.1223 kJ/kg), followed by the basic cycle (244.4832 to 188.4839 kJ/kg) (Figure.8 (a)), and finally the ejector-expansion refrigeration cycle(140.5819 to 119.1757 kJ/kg) (Figure.8 (b)).

Figure 9 shows the effect of condensing temperatures on the entrainment ratio (μ) of the ejection expansion cooling cycle and the ejection sub-cooling cycle using the investigated working fluids R1234ze+R600a, R1234yf+R600a, R1234yf+R600a, R1234yf+R290, and R134a. From the simulation results obtained, it can be seen that the entrainment ratio decreases with increasing condenser temperature in both cooling cycles. This can be explained by the fact that when the condensing temperatures increase from (30 to 55°C), the primary mass flow of the motive fluid increases, and the secondary mass flow of entrained fluid decreases, and hence the entrainment ratio of both cycles decreases.



(a)



(b)

Figure 9. Influence of condensing temperatures on entrainment ratio.

Figure 9(a) and Figure 9(b) show that the working fluid R134a in the two ejector cycles gives a higher entrainment ratio compared to the mixtures. On the other hand, among the three studied mixtures, the R1234yf+R290 gives the higher entrainment ratio, it vary between 0.85047 and 0.66921in the ejector-expansion refrigeration cycle.

By comparing the entrainment ratio values of both cycles, it can be seen that the ejector expansion refrigeration cycle (Figure.9 (a)) exhibits the highest entrainment ratio for all working fluids when the condenser temperature ranges from 30 to 55 °C and the evaporator temperature is 5 °C. This is because the primary flow of the motive fluid leaving from the condenser in the ejector-expansion refrigeration cycle is lower than the primary flow of the motive fluid leaving from the pump in the ejector sub-cooled cycle.

4.3.2 Influence of Evaporating Temperatures on Performances

In the following section, the effects of evaporator temperatures (Te) for a constant condensation temperature (Tc) of 40°C on the performances of the three studied cycles using the investigated mixtures are presented.

Figure 10 shows the variation of the coefficient of performance (COP) of the three refrigeration cycles using the investigated mixtures as working fluids at evaporator temperatures ranging from -10 to 10 °C.

In Figure 10 (a, b, and c), it is evident that the coefficient of performance (COP) of all mixtures increases as the evaporator temperature rises while maintaining a constant condensation temperature of 40°C. This increase is directly related to the change in enthalpy of the evaporator in the refrigeration cycles, while the enthalpy at the condenser outlet of the cycles remains constant. It is noticed from the results that the COP values obtained with the mixture R1234yf+R290 (3,7759 to 7,2792) are higher than those of R134a (3,6956 to 7,2625) and better than the two other mixtures.

Based on Figure 11, it is observed that the cooling capacity of the studied working fluids increases as the evaporation temperatures rise. This is due to that higher evaporation temperatures imply an increase in enthalpy at the evaporator outlet of the cooling cycles, while the enthalpy at the condenser outlet remains constant, which causes an increase in cooling capacity and consequently, the increase in the COP of cooling cycles.

The results showed that the maximum value of the cooling capacity in the three cycles is obtained with the working fluid mixture R1234yf+ R290 (GWP=3.51) followed by R1234ze+R600a, and R134a. The mixture R1234yf+R600a gives the lower values of the cooling capacity in the three cycles. When the evaporation temperature increases from -10 to 10 °C, the cooling capacity of the fluid mixture R1234yf+R290 in the basic cycle increases from 176.1074 to 193.4466 kJ/kg while in the ejector-expansion refrigeration cycle, and in ejector sub-cooled cycle, it increases from 111.5706 to 113.1995 kJ/kg and 187.588 to 205.0074 kJ/kg, respectively.

The effects of evaporator temperatures on the entrainment ratio of the ejector-expansion refrigeration cycle and ejector sub-cooled cycle using the studied refrigerant as working fluids are shown in Figure 12(a) and Figure 12(b), respectively.

Results revealed that the entrainment ratio in the ejector-expansion refrigeration cycle increases with increasing evaporation temperature for all the examined working fluids. This can be explained by the fact that when the evaporation temperatures increase from (-10 to 10°C), the primary mass flow of the driving fluid decreases, and the secondary mass flow of the trapped fluid increases, thus the entrainment ratio in the ejection expansion cooling cycle increases.

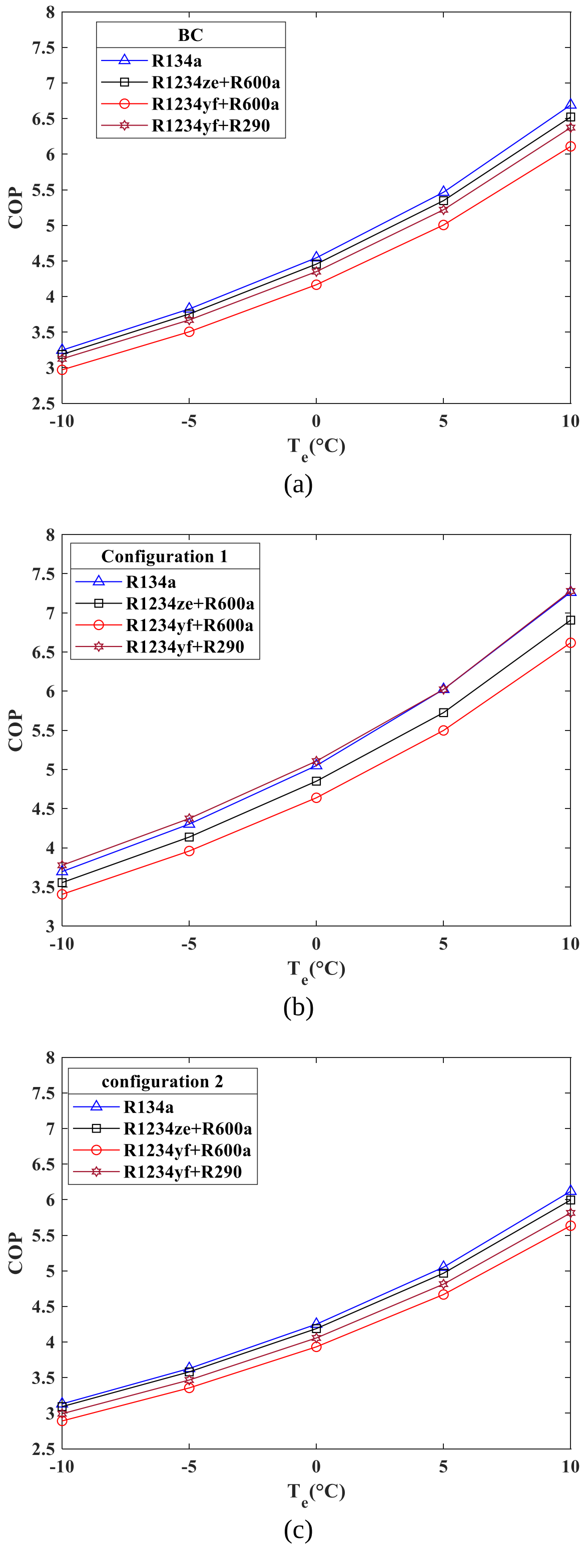
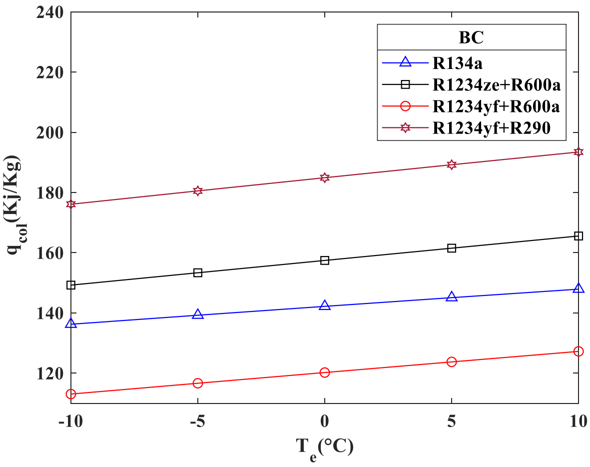


Figure 10. Influence of evaporating temperatures on the COP.

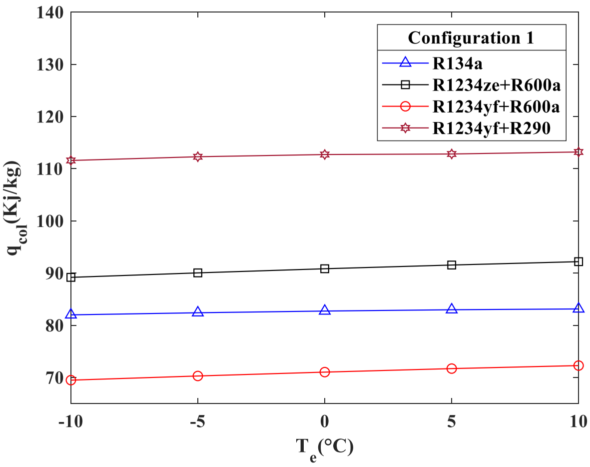
The effects of evaporator temperatures on the entrainment ratio of the ejector-expansion refrigeration cycle and ejector sub-cooled cycle using the studied refrigerant as working fluids are shown in Figure 12(a) and Figure 12(b), respectively.

Results revealed that the entrainment ratio in the ejector-expansion refrigeration cycle increases with increasing

evaporation temperature for all the examined working fluids. This can be explained by the fact that when the evaporation temperatures increase from (-10 to 10°C), the primary mass flow of the driving fluid decreases, and the secondary mass flow of the trapped fluid increases, thus the entrainment ratio in the ejection expansion cooling cycle increases.



(a)



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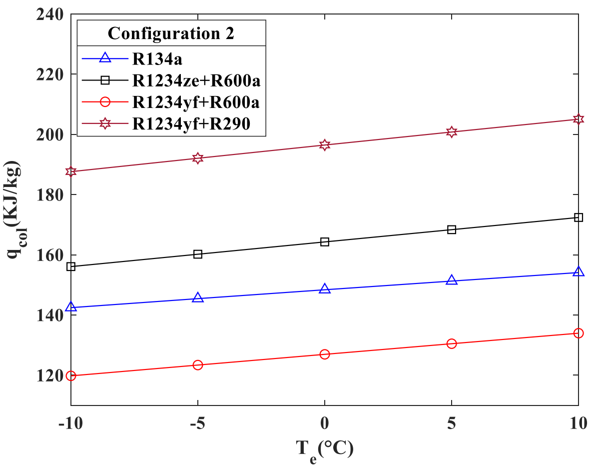
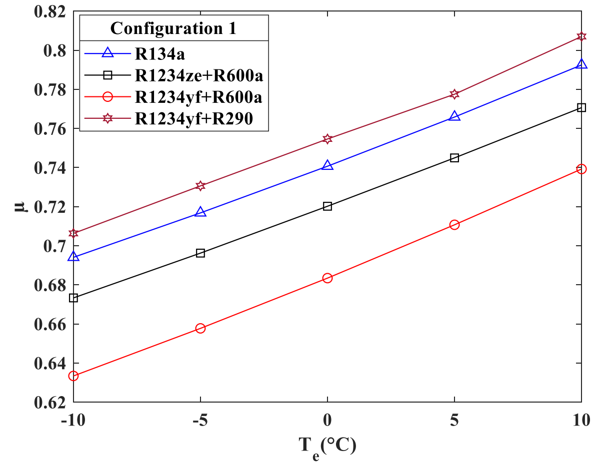


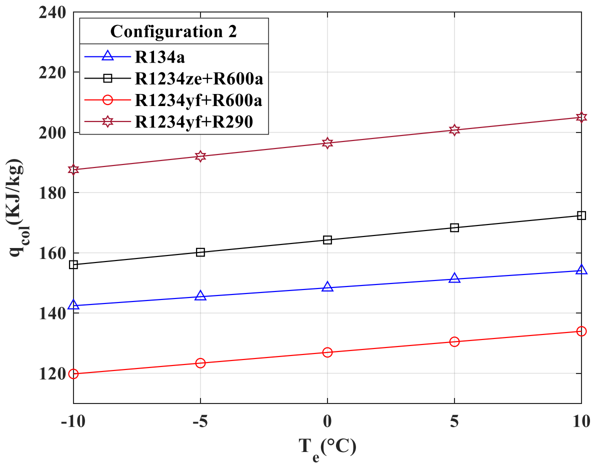
(c)

Figure 11. Influence of evaporating temperatures on the cooling capacity

However, for the ejector sub-cooled cycle (Figure 12(b)), the entrainment ratio remains constant because the evaporator has no direct interaction with the ejector.



(a)



(b)

Figure.12. Influence of evaporating temperatures on entrainment ratio.

5. Conclusions

In this work, the performances of three eco-friendly azeotropic mixtures were compared with the usually used refrigerant R134a, which has good performances but is environmentally unfriendly with a great warming potential. The studied refrigerants were used as working fluid in three refrigeration cycles: basic cycle, ejector-expansion refrigeration cycle, and ejector sub-cooled cycle.

A numerical program was developed using MATLAB software to evaluate the coefficient of performance (COP), the cooling capacity, and the entrainment ratio of the studied cycles.

The refrigerants thermodynamic properties were obtained using REFPROP Version 9.0. The main conclusions are listed below

- The ejector expansion refrigeration cycle exhibits a high coefficient of performance compared to the basic cycle and ejector sub-cooled cycle;
- The ejector expansion refrigeration cycle exhibits a high entrainment ratio compared to the ejector sub-cooled cycle;
- The ejector sub-cooled cycle exhibits a high cooling capacity compared to the basic and cycle ejector expansion refrigeration cycle;
- There is a decrease in COP, cooling capacity, and entrainment ratio with the increase in condenser temperature for all studied cycles;

- There is an increase in COP and cooling capacity with the increase in evaporator temperature for the studied refrigeration cycles.
- The entrainment ratio increases with the evaporator temperature for the basic cycle and the ejector-expansion refrigeration cycle but remains constant for the ejector sub-cooled cycle.
- The eco-friendly azeotropic mixture R1234yf+R290 gives the best performances compared to the two other mixtures ;
- The performances of the eco-friendly azeotropic mixture R1234yf+R290 are close to those of R134a.

According to the obtained results, the azeotropic mixture R1234yf+R290 apart from its excellent environmental properties (GWP= 3.51) yields better performances in most of the cases, this confirms that it could be a suitable substitute for conventional working fluid R134a in the studied refrigeration systems.

Nomenclature

Symbols

h	*Specific enthalpy [kJ kg^{-1}]*
m	*Mass flow rate [kg s^{-1}]*
s	*Specific entropy [kJ kg^{-1} K^{-1}]*
u	*Velocity [m s^{-1}]*
q_{col}	*Cooling capacity [kJ kg^{-1}]*
t_v	*Throttle valve*
W	*Specific work [kJ kg^{-1}]*
P	*Pressure [kPa]*
T	*Temperature [°C or K]*
x_d	*Vapor quality*
δp	*Pressure drop [kPa]*

Greek letters

μ	*Entrainment ratio of ejector*
η	*Efficiency*
Π	*Compression ratio*

Subscripts

cond	*Condenser*
D	*Diffuser*
c	Compressor
evap	*Evaporator*
p	Pompe
is	*Isentropic process*
in	*Inlet*
out	*Outlet*
1-10,9',10'	*State point*

Refrigerants name

R134a	*1,1,1,2-tetrafluoroethene*
R1234ze	*trans-1,3,3,3-tetrafluoropropene*
R600a	*Isobutene*
R1234yf	*2,3,3,3-tetrafluoropropene*
R290	*propane*

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