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Research Article

Energetic performance investigation of ejector air conditioning cycles using the environment friendly gas R161 (Fluoroethane) as substitute to the phase-out R22 (Chlorodifluoromethane)

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ABSTRACT

The present research aims to conduct a comparative examination between two refrigerants, the phase-out R22, and the new eco-friendly R161. They were used in two different ejector air conditioning cycles (Standard ejector cycle (SEC) and Modified ejector cycle (MEC)) under a wide range of working conditions.
A numerical simulation has been carried out with MATLAB simulation code through the thermodynamic energy analysis to explore various thermodynamic performances ((primary (m_{pf}) and secondary (m_{sf}) mass flow rate), entrainment ratio (μ), pressure lift ratio (*PLR*), refrigeration effect (Q), compressor work (W) and coefficient of performance (*COP*)) of SEC and MEC working with both refrigerants under the same operating temperatures (condensing temperature (T_{cond}) varies from (30 to 55) °C and evaporating temperature (T_{evap}) varies from (-10 to 10) °C).
The tests show that under the same given operating temperatures, the (m_{pf} and m_{sf}), *COP*, μ and *PLR* of R161 are close to those obtained with R22 in both cycles. Moreover, it has been proved that MEC has a higher Q and *COP* than SEC. On the other hand, the thermodynamic analysis revealed that as T_{cond} increases, (m_{sf}, μ, Q and *COP*) decreases, and (m_{pf}, *PLR* and W) increases. However, as T_{evap} increases, the (m_{sf}, μ, Q and *COP*) increases and (m_{pf}, *PLR* and W) decreases. Overall, the simulated results confirm that R161 can be useful for air conditioning applications and can serve as a good alternative for the phase-out R22.

1. Introduction

In practical applications of thermal comfort in residential and commercial buildings, the vapor compression cycle (VCC), which mainly includes a compressor, a condenser, an expansion devices (such as throttle valve, capillary tube, expander, etc.), and an evaporator is one of the most widely

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Nomenclature

Symbols		*evap*	Evaporation process
COP	Coefficient of performance	*comp*	Compressor
PLR	Pressure lift ratio	*d*	Diffuser
h	Specific enthalpy, kJ kg^{-1}	*pf*	Primary flow
m	Mass flow rate, kg s^{-1}	*sf*	Secondary flow
s	Specific entropy, kJ kg^{-1} K^{-1}	*is*	Isentropic process
v	Velocity, m s^{-1}	*mn*	Motive nozzle
Q	Heat load, kw	sn	Suction nozzle
W	Work input, kW	*1,…,10*	Thermodynamic states in cycles
P	Pressure, kPa	***Abbreviations***	
T	Temperature, °C or K	VCC	Vapor compression cycle
x	Vapor quality	ACS	Air conditioning system
SNPD	Suction nozzle pressure drop, kPa	HCFC	Hydrochlorofluorocarbons
Greek symbols		HFC	Hydrofluorocarbons
μ	Entrainment ratio of two-phase ejector	GWP	Global warming potential
η	Component efficiency	ODP	Ozone depleting potential
Subscripts		SEC	Standard ejector cycle
cond	Condensing process	MEC	Modified ejector cycle

technologies used in air conditioning systems (ACS).

Due to its superior stability and safety properties, where the air conditioning application with this popular device is rapidly growing around the world due to the increased human thermal comfort need especially in hot climates.

The VCC contributes considerably in the energy consumption (electric power) in the residential and commercial buildings especially during the peak time due to the temperature difference between the heat exchangers (condenser and evaporator) of the system, which leads the compressor to work full time without auto switching [1].

The chlorodifluoromethane (R22) belonging to the hydrochlorofluorocarbons (HCFC) class which contains chlorine atoms is an important working fluid in the VCC system because of its excellent thermo-physical properties and high efficiency, hence it is the most widely fluid used in various cooling and air conditioning applications [2]. However, due to the ozone depletion and the global warming problems related to the popular working fluid R22 (ozone depletion potential (ODP) of 0.055 and high global warming potential (GWP) of 1800) in the case of the leakage of refrigerant, the Montreal (Ozone layer protection) and Kyoto (Climate protection) protocols established by the international community of protection of the environment have decided to phase-out R22 before the year 2030 in developed countries and in 2040 in developing countries to mitigate the ozone damaging and global warming [3-5]. Consequently, the air conditioning industry is carrying out extensive researches to find alternative working fluids to the phase-out R22 which can meet the requirements of environmental performance with zero ODP, very low-GWP and give a suitable system performance the same as the R22 [6].

On the other hand, in the context of recent developments in the field of energy engineering from building energy point of view, the air conditioning industry is looking to improve the efficiency of the VCC system of air conditioning system by the change of the architecture of the thermodynamic cycle and the reduction of the power required by the compressor to run the cycle.

In the VCC system, the traditional throttling devices (such as throttle valve, capillary tube, etc.), which use the isenthalpic process make the system performance to degrade [7]. Accordingly, many solutions have been conducted by the academic researchers to propose new ideas aiming to enhance the cycle performances of standard system and recover its energy losses.

One of the promising VCC modification which provides a perfect method to solve this problem is the use of ejector expansion technology [8] in the place of the traditional throttling devices because of its various advantages (no moving parts, simple structure, it is noise-free, long lifetime, low maintenance, etc.).

This new plays a vital role in generating isentropic condition where the entropy remains constant in the throttling process to recover part of the expansion losses from the VCC and increase the thermodynamic cycle efficiency [9-10]. Moreover, ejector expansion technology also reduces the compressor specific work input by raising its suction pressure, which aids to reduce energy source use (electric power) and consequently increases the coefficient of performance (COP) of the new system [11].

In this context, the evaluation of the performance characteristics of the VCC systems which use the ejector as an expansion device process where the enthalpy remains constant to reduce working fluid pressure from high-pressure at condenser to low-pressure at evaporator causes dropping in cooling capacity in evaporator unit where the thermodynamic losses (throttling irreversibilities) in the throttling with different working fluids (Pure Refrigerants and Mixture Refrigerants) have become the core topics of interest for research in recent years, where there have been numerous numerical and experimental studies in the open scientific literature about this issue.

For the first time, the use of ejector expansion technology in the VCC system was proposed and studied numerically by the work of Kornhauser [12] in 1990. Eight working fluids were used which are R11, R12, R113, R114, R500, R502, R22 and R717. According to his research, the working fluid R502 has given the highest coefficient of performance improvement and the COP improvement using R12 was 21 % over the standard system.

In another study, Nehdi et al [13] proposed of incorporating an two-phase ejector as an expander into VCC to improve the COP by reducing the throttling loss associated with the expansion device and compared various working fluids between twenty single working fluids (R115, R116, R123, R124, R125, R134a, R141b, R142b, R143a, R152a, R22, R227ea, R23, R236ea, R236fa, R245ca, R245fa, R32, R41 and RC318) and seventeen working fluid mixtures (R401A, R401B, R401C, R402A, R402B, R404A, R405A, R406A, R408a, R409A, R409B, R410A, R410B, R411A, R411B, R414B and R500). It has been found that the best performance of 22 % COP improvement was obtained with the working fluid R141b.

Disawas and Wongwises [14] carried out an experiment on ejector-expansion refrigeration cycle (EERC) with the working fluid R134a. It was proved that compared with the VCC system, the EERC had a larger COP under all the investigated conditions.

Sag et al [15] experimentally investigated ejector expander refrigeration systems using R134a refrigerant in terms of energetic and exergetic aspects, and displayed that the ejector expander system exhibits a lower total irreversibility in comparison with the basic system.

Sarkar [16] analyzed and compared three natural working fluids (R600a (isobutene), R717 (ammonia) and R290 (propane)) and showed that maximum performance improvement by using two-phase ejector can be achieved in case of isobutene, whereas minimum performance improvement can be achieved for ammonia.

In another study of Sarkar [17] with same working fluids, the author showed that the values of the optimum area ratio, the corresponding entrainment ratio and the pressure lift ratio in EERC depended on the used working fluid.

Sumeru et al [18] presented a numerical approach for determining the motive nozzle and constant-area of an ejector as an expansion device, based on cooling capacity of the split-type air conditioner using R22 as working fluid.

The results showed that the motive nozzle diameter is constant (1.14 mm) with variations of the condenser temperature, whereas the constant-area diameter decreases as the condenser temperature increases.

In other work of the same authors [19], the authors conducted a numerical and experimental study of an ejector as an expansion device in split-type air conditioner using R22 as working fluid for energy savings. The study introduces a novel ejector cycle based on modification of standard ejector cycle. The comparison between the numerical and experimental results of modification of ejector cycle showed poor agreement due to high difference in their entrainment ratio.

The performance of the EERC using a zeotropic blend (R134a/R143a) as working fluid, is investigated theoretically by Zhao et al [20]. They found that, for the EERC at the considered operating condition, a blend of 0.9/0.1 of R134a/R143a yields the maximum coefficient of performance of 4.18, which is 3.06 % higher than that for the basic system using single fluid R134a.

Hu et al [21] conducted an experimental and numerical analysis on an EERC with the zeotropic mixture R410A equipped with an adjustable liquid-gas ejector on an air conditioning system. They reported that, the ejector expansion with adjustable nozzle can improve the EERC performance and the EERC could increase the energy efficiency ratio by 9.1 % compared to the VCC system.

Zhang et al [22] found the substitution ejector for throttle valve in refrigeration system using R32 could increase the system coefficient of performance by (5.22 –13.77) % and exergetic efficiency by (5.13–13.83) % respectively through optimizing the value of the pressure difference between mixing section and evaporator. The reduction of overall exergy destruction was ranged from (8.84 to 15.84) %.

Lawrence and Elbel [23] conducted an experimental study on the EERC using R134a and R1234yf, and it showed maximum coefficient of performance (COP) improvements of 6 % with R1234yf and 5 % with R134a compared to a conventional cycle.

Pottker and Hrnjak [24] experimentally tested the working fluid mixture R410A in EERC system and reported improvements from (8.2 to 14.8) % over the VCC system.

Zhitong Ma et al [25] performed a study of the VCC system, including a two-phase ejector, and using hydrocarbons working fluids (R600, R600a and R1270) for an energy analysis and another for the exergy. The results show that the use of the two-phase ejector caused an increase in the coefficient of performance, where R290 reported the major increase in comparison with R600, R600a and R1270 under an evaporating temperature of 5 °C and a condensing temperature of 40 °C.

Li et al [26] theoretically found that the EERC using R1234yf as working fluid was superior to the corresponding VCC system, especially in extreme operating environments. It was also found that EERC using R1234yf has a lower coefficient of performance, but it possesses a higher coefficient of performance improving potential than the corresponding R134a system.

In other work, Deng et al [27] developed a thermodynamic model for the transcritical CO2 cycle with an ejector and discovered that the employment of this apparatus provided 22% improvement of COP compared with the simple throttle valve system.

Yu et al [28] theoretically studied the working fluid R32 in EERC with a two-stage suction ejector. The theoretical study showed that the developed cycle gives a higher cooling (heating) capacity and a higher coefficient of performance.

Maalem et al [29] evaluated and compared the thermodynamics performances of new three ternary azeotropic mixtures refrigerants (R600a/R1234ze/R13I1,R134a/R1234yf/R600a and (R134a/RE170/R600a) with the traditional single working fluid R134a in four vapor compression refrigeration configurations with and without ejector expansion under the same operating temperatures. The study showed that working fluid mixture (R134a/RE170/R600a) exhibited the highest performances in terms of coefficient of performance, environmental protection, and cooling capacity compared to traditional working fluid R134a, for the given evaporating and condensing temperatures.

Ajay Kumar Yadav and Neeraj [30] carried out a simulation on EERC with the working fluids R1234yf, R1234ze and R134a. Results show that the COP of the R1234ze is highest compared to R1234yf and R134a for the given evaporating and condensing temperature.

A modified EERC with zeotropic mixture (R290/R600a) for freezers is proposed by Yan et al [31] and a comparison with conventional ejector expansion cycle and basic throttling cycle is carried out. They showed that, under the given operating conditions, performance improvement of the modified cycle in terms of volumetric refrigeration capacity over the basic throttling cycle could reach about 4.5 %.

Liu et al [32] presents a study of an ejector expansion CO_2 air conditioner system. A comprehensive analytical model for 3-ton air-to-air controllable ejector expansion transcritical CO_2 air conditioners was developed and validated. Parametric studies of the ejector expansion CO_2 systems show that the coefficient of performance (COP) and the cooling capacity reach maximum levels when the motive nozzle throat diameter becomes 2.8 mm; the maximum cooling COP and cooling capacity occurred for a mixing section constant-area diameter of between 4.1 and 4.2 mm; COP and cooling capacity are affected by the outdoor air temperature.

In another study of Liu et al [33], the authors present performance enhancement of a transcritical CO_2 air conditioner with a controllable ejector at variable operating conditions and variable compressor frequencies. Results showed that the COP of a CO_2 air conditioning system can be enhanced by using an ejector expansion device to replace a conventional expansion valve. In addition, the COP reached a maximum when the distance between motive nozzle exit and mixing section entrance was three times the mixing section diameter.

The performance of (EERC) with R134a alternative refrigerants (R152a, R1234yf, R404A, R407C, R507A and R600a) for automobile air-conditioning application is investigated numerically by Idan Al-Chlaihawi et al [34]. The study showed that R152a and R1234yf have the closest performance to R134a and can be considered the most suitable alternative refrigerants for R134a.

In another work, Idan Al-Chlaihawi et al [35] presented a study of the performance characteristics of an ejector-expansion refrigeration cycle and standard cycle using six low GWP alternative refrigerants (R1234ze, R1234yf, R290, R600a, R152a and RE170) for R134a through the first and second laws of thermodynamics. The study showed that R152a outperforms R134a in terms of coefficient of performance (COP) and exergy efficiency (η_{ex}).

Gao et al [36] presented a modified dual-evaporator ejector expansion refrigeration cycle (MDEEC) in which a two-phase ejector is used as the expansion device to recover the expansion work and the R290 used as working fluid. The results show that COP of the modified cycle is improved by about 10%.

Maalem et al [37] evaluated and compared the cooling performances of the eco-friendly refrigerant R13I1 with the commonly used R134a, which has good performances in the ejector-expansion refrigeration cycle (EERC) under the same operating temperatures. The study showed that the use of pure refrigerant R13I1 as a working fluid in the EERC system exhibited a higher coefficient of performance, entrainment ratio, and exergy efficiency, as well as lower exergy destruction compared with traditional working fluid R134a under the same operating temperatures.

Aghazadeh Dokandari et al [38] evaluated and compared the thermodynamics performances of nitrous oxide (N_2O) and CO_2 in ejector-expansion refrigeration cycle. The results show that (N_2O) can be considered alternative refrigerant for the working fluid CO_2.

The authors Tao Bai and Yu [39] proposed applying the ejector-expansion refrigeration system (ERS) to develop a (-50 °C) low-temperature freezer using R290 as working fluid. The results demonstrated that the ERS-based freezer could operate stably with the appropriate nozzle throat diameter and compressor displacement.

Hacipasaoglu and I.Tekin Ozturk [40] presented a numerical simulation for determining the performance of the ejector expansion refrigeration cycle (EERC) by exploring the optimum pressure drop for three refrigerants (R134a, R600a, and R290).

The results showed that R290 is the most appropriate refrigerant for ejector expansion refrigeration cycle) among the refrigerants investigated.

C.Aktemur, İ.Tekin Öztürk [41] evaluated and compared the thermodynamic performance of pure refrigerant and nano-refrigerant (R1270/CuO) systems in ejector expansion vapor compression refrigeration system with constant area mixing theory for low-temperature applications. The results pointed out that the system using nano-refrigerant is the best one for which the proposed system has 8 % lower discharge temperature of the main compressor, 0.59% lower ejector area ratio, 3.23% lower entrainment ratio, 8.93% higher exergy efficiency, 8.96% higher COP (Coefficient of Performance) and 21.23% lower total exergy destruction than pure refrigerant-based system at a condenser temperature of 45 °C and an evaporator temperature of −30 °C.

Based on the literature survey about the evaluation of the performance characteristics of the different working fluids (Pure Refrigerants and Mixture Refrigerants) in the VCC systems, which use the ejector as an expansion device cited in the above review, it was noted that the pure refrigerant Fluoroethane (R161) applied in the ejector air conditioning technologies was not found in the previous investigation published in the open literature. Despite, there is a renewed interest in the use of the R161 refrigerant as working fluid in the refrigeration engineering in the last years [2, 42], where it could be a possible option for near future refrigerant candidates to replace the conventional working fluids in refrigeration, air-conditioning and heat pump applications.

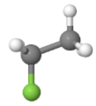
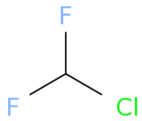
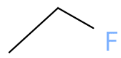
The working fluid R161 belonging to the hydrofluorocarbons (HFC) class is considered as a viable eco-friendly refrigerant does not containing chlorine atoms (ODP=0) and has very low global warming potential (GWP=12) [43], which gives an excellent life cycle climate performance. Moreover, it has excellent thermo-physical properties (normal boiling point, critical parameters, etc.) like to those of the phase-out R22 [44-45], which implicate similar working condition in air conditioning system (ACS) of the residential and commercial buildings. Thus, considering the strict environmental legislation imposed by the Montreal and Kyoto protocols established by the international community of protection of the environment, the Fluoroethane (R161) can be selected as a future working fluid candidate in the ejector air conditioning cycles and can be proposed as a perfect alternative to the phase-out R22 in air conditioning systems with the ejector as an expansion device, which has not been used before.

With the best of author's knowledge, no previous study in the scientific literature has considered this alternative to investigate and compare the performances of the environment friendly gas R161 with the phase-out R22 in the ejector air conditioning cycles. Therefore, the present research study aims to investigate theoretically the thermodynamic performances ((primary (m_{pf}) and the secondary (m_{sf}) mass flow rate), entrainment ratio (μ), pressure lift ratio (*PLR*), refrigeration effect (Q), compressor work (W) and coefficient of performance (*COP*)) of two ejector air conditioning cycles (Standard ejector cycle (SEC) and Modified ejector cycle (MEC)) operating for the first time with the environment friendly gas R161 a substitute to the most widely working fluid R22, where the thermodynamic performances of the two cycles using eco-friendly R161 as working fluid have been investigated in comparison with those using traditional working fluid R22 under the same operating parameters.

The effects of operating temperatures (condensing temperature and evaporating temperature) on the thermodynamic performances of the two air-conditioning cycles (SEC and MEC) are also evaluated and discussed for the investigated working fluids.

The environment properties (Ozone depleting potential (ODP) values and Global warming potential (GWP)) and the physical properties of the working fluids that are subject to this research are given in Table 1 [44, 46].

Table 1 Basic physical and environmental properties of R22 and R161

Specifications	**Unit**	**R22**	**R161**
Component properties			
Fluid name	-	Chlorodifluoromethane	Fluoroethane
Type	-	HCFC	HFC
Molecular structure	-		
Chemical structure	-		
Molecular formula	-	$CHClF_2$	C_2H_5F
Cas No	-	75-45-6	353-36-6
Basic physical properties			
Molar mass	(kg/kmol)	86.468	48.06
Critical temperature	(K)	369.30	375.25
Critical pressure	(MPa)	4.990	5.046
Normal boiling point	(K)	232.34	235.60
Basic environment properties			
ODP	-	0.055	0
GWP	-	1800	12

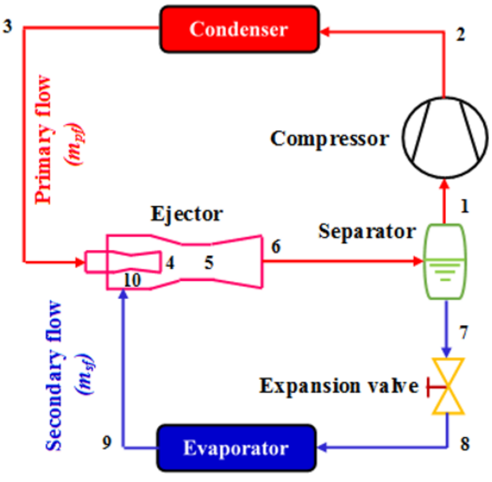
To reach the objectives of the present project, the contents of this article are organized as follows: The description of the two ejector air conditioning cycles (Standard ejector cycle (SEC) and Modified ejector cycle (MEC)) is discussed in detail in section 2. In section 3, the methodology of the theoretical analysis of the studied ejector air conditioning cycles used in this work is developed. Then, in section 4, after the validation of the model of simulation with the results published in the literature, the thermodynamic performance results obtained (such as COP, entrainment ratio, etc.) of the selected working fluids (R22 and R161) are analyzed and compared under the same operating conditions. Finally, section 5, contains the main conclusions of the present study and its perspectives.

2. Description of Cycles (SEC and MEC)

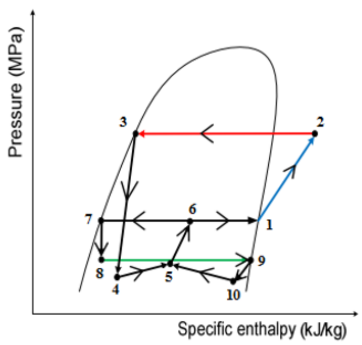
2.1 Standard ejector cycle (SEC)

The schematic view of a standard ejector cycle (SEC) and its presentation in the pressure-enthalpy (P-h) diagram are displayed in Fig.1 (a) and Fig.1 (b), respectively. The SEC is a promising type of alternative air conditioning system in refrigeration engineering, where is consists of six basic components that are: a compressor, a condenser, an ejector, an evaporator, a separator and an expansion valve.

The SEC operates as follows: the primary flow (m_{pf}) leaving from the condenser in the form of saturated liquid (state 3) at high-pressure and the secondary flow (m_{sf}) leaving from the evaporator in the form of saturated vapor (state 9) at low-pressure are expanding through motive and suction nozzles, respectively (3→4 and 9→10), where the high-pressure primary flow is expanded through the motive nozzle (convergent–divergent nozzle) in the two-phase ejector to produce high velocity flow (state 4), which entrains the vaporized working fluid from the evaporator in the suction nozzle. Then, the two flows are mixed together in the mixing chamber at constant pressure. The mixed flow at the exit of the mixing section (state 5) enters the diffuser section, where its velocity drops and pressure increases by discharged through (5→6).Then, the mixed flow leaves the ejector (state 6) and enters into the gas–liquid separator where is separated in forms of saturated vapor (state 1) and saturated liquid (state 7). The liquid circulates through the expansion valve (7→8) where its pressure and temperature drop to the evaporator condition (state 8) before enters the evaporator to absorbs heat (8→9) from the refrigerated space by boiling a working fluid, whereas the vapor flows into the compressor in which its pressure is raised to superheated (1→2) and condensed back to a liquid in the condenser (2→3), with the heat of condensation rejected to a relatively high-temperature heat sink (ambient air for most refrigeration and air-conditioning systems) to complete the cycle.



(a) Schematic diagram of SEC

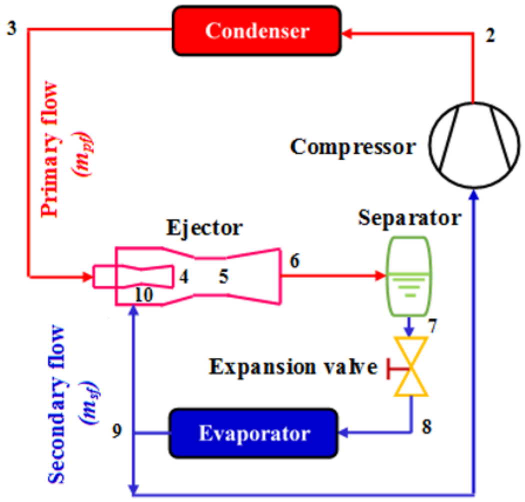


(b) P-h diagram of SEC

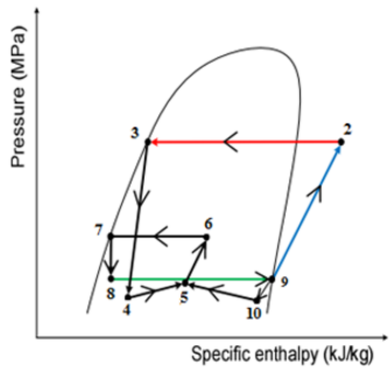
Fig.1 Configuration (a) and corresponding (P-h) diagram (b) of SEC.

2.2 Modified ejector cycle (MEC)

The schematic view of the modified ejector cycle (MEC) and its presentation in the pressure-enthalpy (P-h) diagram are presented in Fig.2 (a) and Fig.2 (b), respectively.
The MEC includes the same components of the SEC and his operating principle is identical to the SEC. The only difference between MEC and SEC is in the type of separator used.



(a) Schematic diagram of MEC



(b) P-h diagram of MEC

Fig.2 Configuration (a) and corresponding (P-h) diagram (b) of MEC.

In the SEC, the separator has an inlet that flows the working fluid from the ejector, and two outlets that flow out the vapor refrigerant to compressor suction and liquid refrigerant to the evaporator. Meanwhile, the modified ejector cycle has a separator that only has an inlet and one outlet. The outlet is connected to the evaporator device, while the refrigerant is sent to the suction inlet of ejector expansion and to the compressor after exchange of heat in the evaporator.

3. Thermodynamic Analysis of Studied Ejector Air Conditioning Cycles (SEC and MEC)

3.1 Thermodynamic assumptions

For simplification of the analysis of the studied ejector air conditioning cycles (SEC and MEC), the following thermodynamic assumptions were made:

- One-dimensional homogeneous equilibrium flow in the ejector expansion is considered;
- The inner wall of the ejector expansion is adiabatic;
- The ejector components have constant values of efficiencies;
- The mixing pressure in the mixing chamber of ejector is constant;
- Isenthalpic process in the expansion valve (h_8=h_7);
- The compression process in the compressor is irreversible and has a given isentropic efficiency;
- Kinetic and potential energy variations are neglected in the ejector air conditioning cycles;
- Heat losses in the cycles are neglected;
- Refrigerant pressure drop in the condenser, evaporator, separator and the connection tubes are neglected;
- Saturation conditions apply at exit of the evaporator and condenser of ejector air conditioning cycles;
- Saturation condition for both liquid and vapor at the separator exit.

According to the previous considerations, the thermodynamic equations for the both ejector air conditioning cycles (SEC and MEC) are developed in the following subsection.

3.2 Mathematical models of ejector expansion technology

A typical ejector expansion technology consists of a motive nozzle, a suction nozzle, a mixing section and a diffuser. The thermodynamic performances of the ejector air conditioning cycles depend largely on ejector performances. Note that there are two models can be used to simulate the ejectors, where the ejector expansion can be categorized into two classifications according to the position of the motive nozzle exit plane:

- If the motive nozzle exit placed inside the suction chamber, the mixing of the primary and the entrained fluids occurs inside the suction chamber with a constant pressure, and the ejector is classified as **"*a constant-pressure mixing ejector*"**;
- If the motive nozzle exit placed inside the constant-area section, the mixing of the primary and the secondary fluids occurs inside the constant-area section and the ejector is classified as **"*a constant-area mixing ejector*"**.

Both ejector types have been extensively in previous studies [8-11, 47-48]. The constant-pressure ejector has a better performance than the constant-area ejector and is consequently widely used in the most studies [8-11]. Therefore, in this paper, the constant pressure ejector model (Fig.3) is adopted and developed to investigate the thermodynamic performances of the ejector air conditioning cycles (SEC and MEC) using the ecofriendly R161 and phase-out R22 as working fluids.

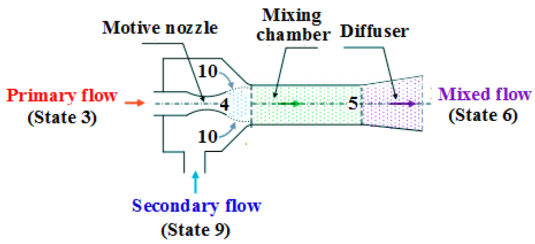


Fig.3 Representation of the different sections of ejector expansion technology.

For air conditioning applications, the entrainment ratio (μ) and the pressure lift ratio (PLR) are the two main factors that affect the ejector performance and therefore influence the performance of ejector air conditioning cycles. These factors are defined by Eq. (1) and Eq. (2), respectively:

$$\mu = \frac{m_{sf}}{m_{pf}} \tag{1}$$

$$PLR = \frac{P_6}{P_9} \tag{2}$$

where, (m_{pf} and m_{sf}) are the primary and the secondary mass flow rate of the ejector expansion, respectively. The P_9 is the pressure of the secondary flow and the P_6 is the pressure of the two-phase flow at the diffuser outlet.

For the thermodynamic modeling of the ejector expansion technology, the algorithm presented by Zhang et al [22] and Maalem et al [37] is used in the present study. The modeling of the ejector expansion technology of study ejector air conditioning cycles (SEC and MEC) starts with determining the thermodynamic properties (such as pressures, enthalpies and entropies) of the primary and secondary flows at the outlets of the condenser and evaporator (states 3 and 9), respectively as follows:

At the condenser outlet:

$$p_3 = p_{cond} = p(T_3, x = 0) \tag{3}$$

$$h_3 = h(T_3, x = 0) \tag{4}$$

$$s_3 = s(T_3, x = 0) \tag{5}$$

At the evaporator outlet:

$$p_9 = p_{evap} = p(T_9, x = 1) \tag{6}$$

$$h_9 = h(T_9, x = 1) \tag{7}$$

$$s_9 = s(T_9, x = 1) \tag{8}$$

For ejector modeling, an initial value for the entrainment ratio (μ_0) and a given pressure drop in the suction nozzle (*SNPD*). Also, the efficiencies for motive and suction nozzles (η_{mn} and η_{sn}) as well as the diffuser (η_d), should be given.

The governing equations for ejector expansion technology modeling are given below.

3.2.1 Model of primary flow through nozzle

The thermodynamic equations for the primary flow through nozzle can be given as:

$$p_4 = p_9 - SNPD \tag{9}$$

$$s_4 = s_3 \tag{10}$$

$$h_{4,is} = h(p_4, s_4) \tag{11}$$

Applying the isentropic efficiency of the motive nozzle, the enthalpy of the high-pressure primary fluid is given as:

$$h_4 = h_3 - \eta_{mn}(h_3 - h_{4,is}) \tag{12}$$

Applying the conservation of energy through the expansion, the velocity of the primary fluid at the motive nozzle outlet is given as:

$$v_4 = \sqrt{2(h_3 - h_4)} \tag{13}$$

3.2.2 Model of secondary flow

The relevant equations of the secondary flow are given by:

$$p_{10} = p_9 - SNPD \tag{14}$$

$$s_{10} = s_9 \tag{15}$$

$$h_{10,is} = h(p_{10}, s_{10}) \tag{16}$$

Applying the isentropic efficiency of the suction nozzle, the enthalpy of the secondary fluid is given as:

$$h_{10} = h_9 - \eta_{sn}(h_9 - h_{10,is}) \tag{17}$$

Applying theenergy conservation equation between the inlet and the exit of the suction nozzle, the velocity of the secondary fluid at the suction nozzle outlet is is given as:

$$v_{10} = \sqrt{2(h_9 - h_{10})} \tag{18}$$

3.2.3 Model of mixing process

The thermodynamic equations of the mixed flow at the exit of mixing chamber are:

$$p_5 = p_4 = p_{10} \tag{19}$$

Applying the momentum conservation equation between the inlet and the exit of mixing chamber, the velocity of the mixture at the exit of the mixing chamber, is given as:

$$v_5 = (v_4 + \mu v_{10})/(1 + \mu) \tag{20}$$

Applying the energy conservation equation, the enthalpy of the mixture at the exit of the mixing chamber, is given as:

$$h_5 = \frac{h_3}{(1+\mu)} + \frac{\mu h_9}{(1+\mu)} - \frac{v_5^2}{2} \quad (21)$$

$$s_5 = s(p_5, h_5) \quad (22)$$

3.2.4 Model of mixed flow through the diffuser

In the diffuser section, the mixed flow converts the kinetic energy into pressure energy.

The thermodynamic properties of mixed fluid when leaving the diffuse can be obtained as:

$$h_6 = h_5 + \frac{v_5^2}{2} \quad (23)$$

$$h_{6,is} = h_5 + \eta_d (h_6 - h_5) \quad (24)$$

$$p_6 = p(h_{6,is}, s_5) \quad (25)$$

$$x_6 = s(p_6, h_6) \quad (26)$$

To verify the preliminary input value for the entrainment ratio, the following relationship of the quality ($x_6^{'}$) of the two-phase mixture exited from the ejector must be satisfied by an iterative calculation to adjusting the value of (μ):

$$x_6^{'} = \frac{1}{1+\mu} \quad (27)$$

Once the ejector performance is obtained, the other parameters of components in the SEC and MEC can be found in ordinary fashion.

3.2 Systems coefficient of performance

The governing equations of the systems are developed as follows:

3.2.1 Model of the evaporator

The heat absorbed (Q_{SEC} and Q_{MEC}) by the working fluid in the evaporator of SEC and MEC is calculated using the following equations:

$$Q_{SEC} = m_{sf} (h_9 - h_8) \quad (28)$$

$$Q_{MEC} = (m_{sf} + m_{pf})(h_9 - h_8) \quad (29)$$

Where, h_9 is the enthalpy of the working fluid at the exit of evaporator and h_8 is the enthalpy of the working fluid at the inlet of evaporator in SEC and MEC.

3.2.2 Model of the compressor

The input power (W_{SEC} and W_{MEC}) of SEC and MEC is given respectively by:

$$W_{SEC} = m_{pf} (h_2 - h_1) \quad (30)$$

$$W_{MEC} = m_{pf} (h_2 - h_9) \quad (31)$$

Where, h_2 is the enthalpy of the working fluid at the exit of compressor in (SEC and MEC), and ($h_1$and h_9) are the enthalpies of the working fluid at the inlet of compressor in SEC and MEC, respectively.

In the SEC, the actual enthalpy of state 2 is expressed by:

$$h_2 = h_1 + (h_{2,is} - h_1)/\eta_{comp} \quad (32)$$

Where, η_{comp} is the isentropic efficiency of the compression process.

The isentropic efficiency (η_{comp}) has been calculated by the empirical relation of Brunin et al [49]:

$$\eta_{comp} = 0.874 - 0.0135 (p_2 / p_6) \quad (33)$$

In the MEC, the actual enthalpy of state 2 is expressed by:

$$h_2 = h_9 + (h_{2,is} - h_9)/\eta_{comp} \quad (34)$$

3.2.3 Model of the condenser:

The heat release (Q_{SEC} or MEC) by the systems to ambient can be expressed as:

$$Q_{SEC_{OR} MEC} = m_{pf} (h_3 - h_2) \quad (35)$$

Where, h_3 is the enthalpy of the working fluid at the exit of condenser and h_2 is the enthalpy of the working fluid at the inlet of condenser in SEC and MEC, respectively.

3.2.4 COP of SEC and MEC

The efficiency of the ejector air conditioning cycles (SEC and MEC) are assessed by the help of COP (coefficient of performance), defined as the ratio between the cooling effect and energy input of the cycle.

The coefficient of performance (COP) for the both cycles are calculated respectively as follows:

The COP of the SEC is:

$$COP_{SEC} = \frac{Q_{SEC}}{W_{SEC}} \quad (36)$$

The COP of the MEC is:

$$COP_{MEC} = \frac{Q_{MES}}{W_{MEC}} \quad (37)$$

3.4 Simulation process

Based on the assumptions and manipulation of Eqs. (2)-(37) described above, a computer simulation program based on iterative procedures using MATLAB software has been developed in order to comprehensively compare the thermodynamic performances of R161 to that of R22.

By the given the operating conditions (T_{evap} , T_{cond} , $SNPD$, η_{mn} , η_{sn} and η_d), the software gives the thermodynamic performances of both working fluids in the thermodynamic cycles (SEC and MEC).The simulation process was conducted in steps as shown in Fig.4.

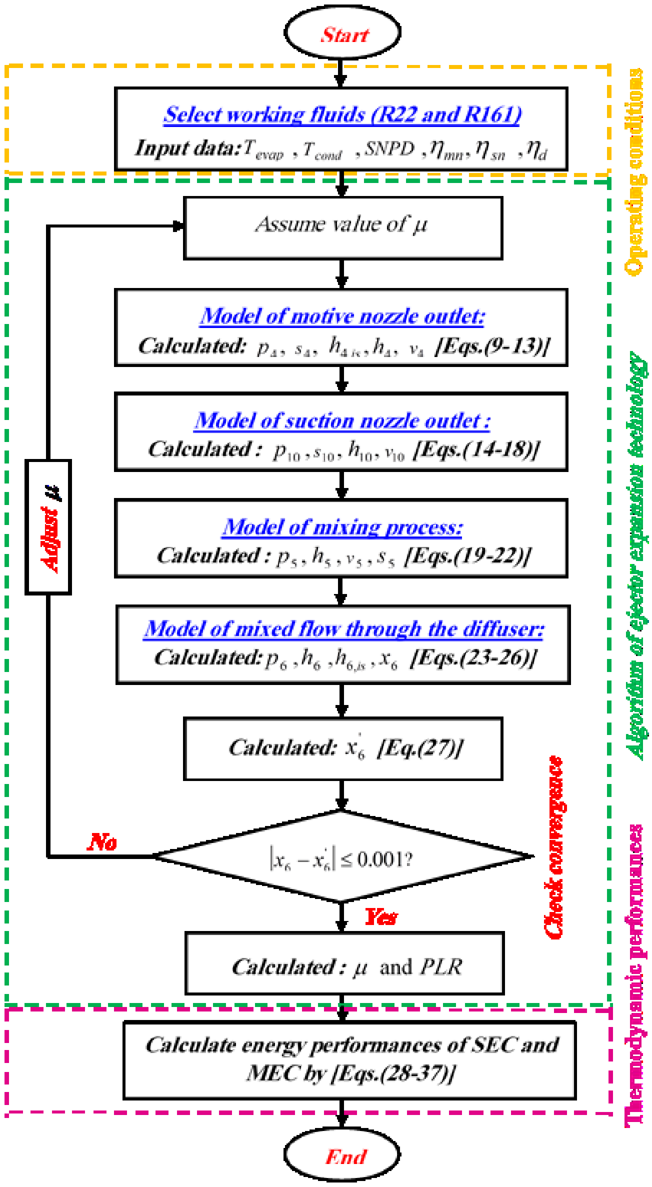


Fig.4 Flow chart of the simulation program.

4. Simulation Results and Discussion

4.1 Validation of the computer simulation model

Before using the developed computer simulation program to investigate the performance of the ejector air conditioning cycles (SEC and MEC) using the working fluids (R161 and R22), the simulation code was validated by comparing the values of the maximum coefficient of performance (COP) with the data available in the literature in order to check its validity.

The values of the maximum coefficient of performance (COP) reported by Sarkar [16] and Li et al [26] using the hydrocarbon (Isobutane (R600a)) as working fluid in ejector-expansion vapor compression refrigeration system are used in this work in order to check the developed program under the same operating conditions (condensation temperatures (T_{cond}) varying from (35 to 55) °C, the constant evaporation temperature (T_{evap}) of 5°C and ejector component efficiencies (motive nozzle efficiency, suction nozzle efficiency and diffuser efficiency) are kept constants at values ($\eta_{mn} = \eta_{sn} = \eta_d = 0.85$)). The simulation results are illustrated in Fig.5 for different condenser temperatures.

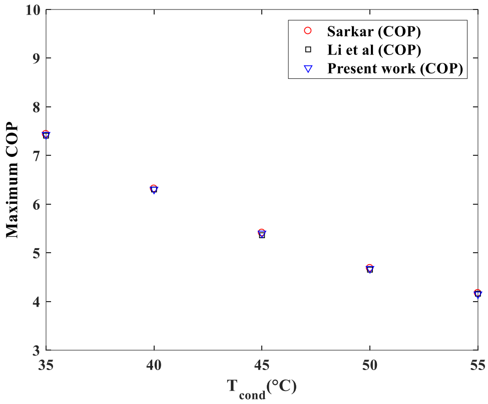


Fig.5 Validation of present simulation code with Sarkar [16] and Li et al [26] results using R600a fluid.

As given in Fig.5, a good agreement can be revealed between the values of the maximum COP calculated using the computer simulation model and the values reported from reference's results [16, 26], which confirms the validity of our simulation model.

It is obvious that the deviations of the COP with the reference's results [16, 26] are ranged from (0.13 to 0.47) % with a mean value of 0.24 % and (0.27 to 0.21) % with a mean value of 0.11 %, respectively. These deviations are very acceptable and indicate that the developed model can calculate the thermodynamic performances.

The thermodynamic performances of the ejector air conditioning cycles (Standard ejector cycle (SEC) and Modified ejector cycle (MEC)) are significantly affected by operating temperatures such as the condensing temperature (T_{cond}) and the evaporating temperature (T_{evap}).

Therefore, the investigations focus on discussing the thermodynamic performances of the cycles under the following operating conditions:

The condensing temperature, T_{cond}, between (30 and 55) °C, the evaporation temperature, T_{evap}, varying from (-10 to 10) °C and the ejector expansion is assumed to have the following efficiencies:

($\eta_{mn} = \eta_{sn} = \eta_d = 0.85$).

The simulation results of the comparative evaluation of the investigated working fluids (R161 and R22) in the Standard ejector cycle (SEC) and Modified ejector cycle (MEC) will be discussed in the next sections.

4.2 Influence of operating conditions on performances of SEC and MEC

4.2.1 Effect of the evaporator outlet temperature on performances of SEC and MEC

In this section, the following results are obtained when the evaporator outlet temperature varies from (-10 to 10) °C and the condenser outlet temperature is fixed at 40 °C.

Fig.6 reveal the variation of the primary and the secondary mass flow by varying the evaporating temperature (T_{evap}= -10 to 10) °C) at the constant condensation temperature (T_{cond}) of 40 °C for the air conditioning cycles (SEC and MEC) operating with the working fluids (R22 and R161).

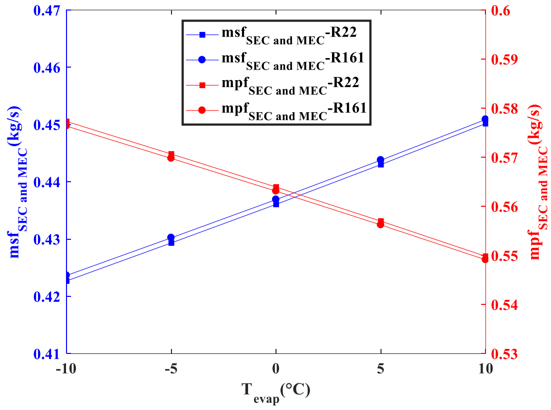


Fig.6 Variation of $m_{sf,SEC\ and\ MEC}$ and $m_{pf,SEC\ and\ MEC}$ with evaporation temperature.

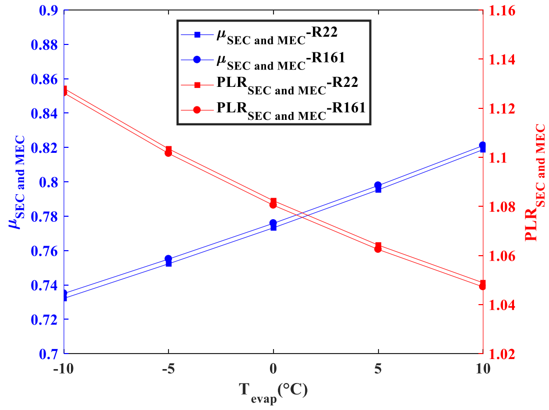


Fig.7 Variation of $\mu_{SEC\ and\ MEC}$ and $PLR_{SEC\ and\ MEC}$ with evaporation temperature.

From the results obtained, it is seen that when the (T_{evap}) raises from (-10 to 10) °C), the secondary mass flow ($m_{sf,SEC\ and\ MEC}$) leaving from the evaporator of both refrigerants R161 and R22 increases gradually due to an increase in the secondary flow velocity by the increasing evaporator pressure, however, the primary mass flow ($m_{pf,SEC\ and\ MEC}$) leaving from the condenser of the R161 and R22 decreases gradually with the (T_{evap}) due to a decrease of the primary flow velocity by the increasing evaporator pressure.

On the other hand, it can be easily inferred from the curves that the secondary mass flow rate of the eco-friendly R161 would be more than using R22 in the SEC and MEC at a low evaporation temperature (-10 °C) or high evaporation temperature (10 °C). However, the primary mass flow rate of the eco-friendly R161 would be less than using R22 in the SEC and MEC at a low evaporation temperature (-10 °C) or high evaporation temperature (10 °C).

Within the studied range of evaporating temperatures, it is found that the secondary mass flow ($m_{sf,SEC\ and\ MEC}$) of the studied refrigerants R161 and R22 varies from (0.4238 to 0.4509) kg/s) and from (0.4228 to 0.4502) kg/s), respectively, as the (T_{evap}) increases from (-10 to 10) °C).

On the other side, the primary mass flow ($m_{pf,SEC\ and\ MEC}$) of the R161 and R22 varies from (0.5762 to 0.5491) kg/s) and from (0.5772 to 0.5498) kg/s), respectively, as the (T_{evap}) increases from (-10 to 10) °C).

Fig.7 depict the simulated results of the variations of the entrainment ratio ($\mu_{SEC\ and\ MEC}$) and pressure lift ratio ($PLR_{SEC\ and\ MEC}$) with the evaporating temperature (T_{evap}) varied from (-10 to 10) °C) for both SEC and MEC operating with the phase-out R22 and the proposed eco-friendly R161.

From the simulated results, it can be observed that the entrainment ratio ($\mu_{SEC\ and\ MEC}$) curves of the both working fluids (R22 and R161) increases with the increasing of the evaporator temperature (T_{evap}). However, the pressure lift ratio ($PLR_{SEC\ and\ MEC}$) of (R22 and R161) decreases with the increasing of the evaporator temperature (T_{evap}). The ejector capacity is evaluated by the entrainment ratio ($\mu_{SEC\ and\ MEC}$), which is defined as the ratio between the secondary mass flow rate (entrained vapor) leaving from the evaporator and primary mass flow rate (motive fluid) leaving from the condenser. As discussed in the previous discussion, when the evaporation temperature rises from (-10 to 10) °C), the primary mass flow rate decreases and the secondary mass flow rate increases, and consequently the entrainment ratio ($\mu_{SEC\ and\ MEC}$) increases. At the same time, when the entrainment ratio ($\mu_{SEC\ and\ MEC}$) increases, the pressure lift ratio ($PLR_{SEC\ and\ MEC}$) decreased, owing to the reduction in the conversion from pressure to kinetic energy with the decreased primary mass flow rate, leaving from the condenser and increased secondary mass flow rate, leaving from the evaporator. Obviously, this is beneficial to the reductions of pressure ratio and power consumption of the compressor.

Compared with the phase-out R22 which has high ODP and GWP, it is found that the eco-friendly R161 refrigerant offers values of the entrainment ratio ($\mu_{SEC\ and\ MEC}$) and pressure lift ratio ($PLR_{SEC\ and\ MEC}$) very close to those of R22 in both SEC and MEC under the same operating temperatures (T_{evap} and T_{cond}).

The ($\mu_{SEC\ and\ MEC}$) values calculated of the both working fluids R161 and R22 increases from (0.7354 to 0.8211) and from (0.7325 to 0.8188), respectively, as the (T_{evap}) increases from (-10 to 10 °C) and (T_{cond}) is set at (40 °C). On the other side, the ($PLR_{SEC\ and\ MEC}$) values calculated of the working fluids R161 and R22 decreases from (1.1261 to 1.0472) and from (1.1280 to 1.0489), respectively, as the (T_{evap}) increases from (-10 to 10 °C) and (T_{cond}) is set at (40 °C).

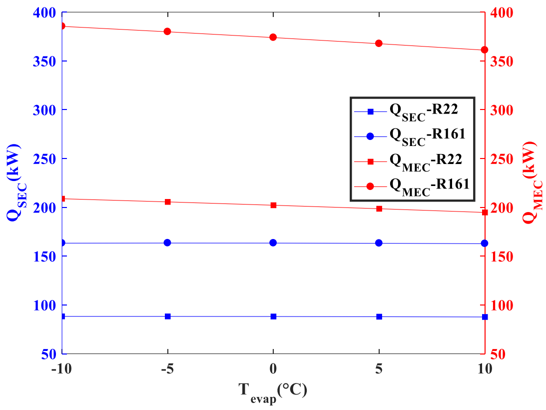


Fig.8 Variation of Q_{SEC} and Q_{MEC} with evaporation temperature.

The variation of the cooling capacity of the investigated working fluids (R161 and R22) in the SEC and MEC with the evaporating temperature, when the condenser temperature is kept constant, are shown in Fig.8.

From the curves plotted in Fig.8, it is observed that the cooling capacity increase for both working fluids, since the latent heat of vaporization of the working fluids increases as its evaporating temperature increases; hence, the refrigeration effect increases.

It is evident that working fluid R161 possesses the higher refrigeration effect compared to R22 in the both cycles (SEC and MEC) under the whole range of given evaporating temperatures. Since, R161 has a higher latent heat of vaporization compared to R22. The high normal boiling point has an effect on the latent heat of vaporization whereas the high normal boiling point makes a greater latent heat of vaporization and as a result, the refrigerating effect increases. In another, the R161 has a critical temperature much higher than that of R22 (see Table 1), which makes a greater heat transfer.

It can be also observed that the cooling capacity of both working fluids is higher in MEC than the SEC, when the evaporating temperature increases from (-10 to 10) °C). This is because all refrigerants flows through the evaporator of MEC. As a result, the cooling capacity that is produced by the MEC is higher than that of the SEC, which not all refrigerants flows into his evaporator, where part of the refrigerant flows through the compressor without passing through the evaporator, which makes a low heat transfer compared to MEC.

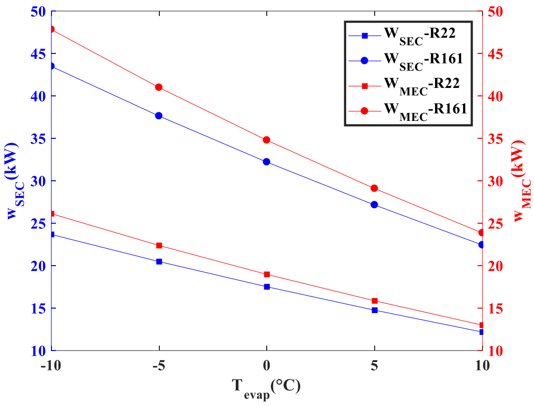


Fig.9 Variation of W_{SEC} and W_{MEC} with evaporation temperature.

Fig.9 portrays the variation of the input power in the compressor with evaporating temperature at a constant condensation temperature (T_{cond}) of 40 °C in the SEC and MEC.

From Fig.9, it is noticed that any increase in evaporating temperature leads to decrease in input power in the compressor. It can be seen that among the studied working fluids, R161 gives higher input power in the SEC and MEC at the same operating temperatures, but it also produced a significantly higher refrigerating effect (Fig.8) than the R22.

It is found that the input power in the compressor values calculated of both working fluids (R161 and R22) in SEC decreases from (43.4699 to 22.4269 kW) and from (23.6768 to 12.1789 kW), respectively, as the (T_{evap}) increases from (-10 to 10 °C) and (T_{cond}) is set at (40 °C). On the other side, the input power in the compressor values calculated of the R161 and R22 in MEC decreases from (47.8078 to 23.8464 kW) and from (26.1183 to 12.9924 kW), respectively, as the (T_{evap}) increases from (-10 to 10 °C) and (T_{cond}) is set at (40 °C).

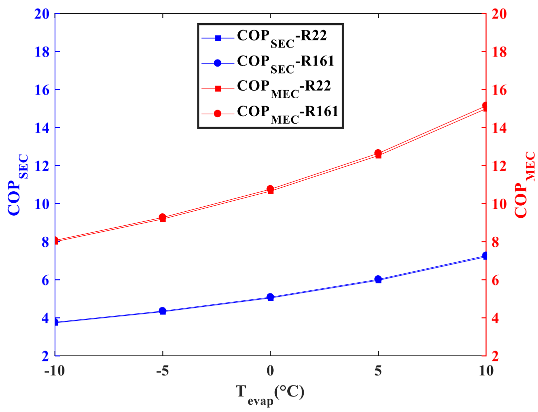


Fig.10 Variation of COP_{SEC} and COP_{MEC} with evaporation temperature.

Fig.10 shows the variations of the coefficient of performance (COP) when the temperature of the evaporator (T_{evap}) varies from (-10 to 10 °C) for the SEC and MEC operating with R22 and R161. The coefficient of performance it can be considered as an energy efficiency index of the equipment when it is operating with particular working fluid.

For both cycles (SEC and MEC), it can be observed that COP of the ejector air conditioning cycles (SEC and MEC) operating with R22 and R161 increases monotonously with the increase of the evaporating temperature (T_{evap}). This is because that the pressure of the secondary fluid increases with the evaporating temperature, leading to a less pressure ratio required for the compressor, and consequently less compressor power consumption, when the condensing temperature and pressure are maintained constant. It could be also observed that MEC gives a higher COP compared to the SEC with both working fluids under the same given conditions. This can be interpreted by the highest cooling capacity of the evaporator of MEC compared to the SEC, which results higher COP in the case of MEC.

Compared with the phase-out R22, the proposed eco-friendly R161 refrigerant offers very similar values of COP to those of R22 in both SEC and MEC at the same operating conditions, which will be useful to applied R161 in the SEC and MEC to replace the R22.

It is found that the coefficient of performance (COP) values calculated of the R161 and R22 in SEC increase from (3.7574 to 7.2583) and from (3.7287 to 7.1992), respectively, as the (T_{evap}) increases from (-10 to 10 °C) and (T_{cond}) is set at (40 °C).

On the other side, the coefficient of performance (COP) values calculated of the R161 and R22 in MEC increases from (8.0624 to 15.1397) and from (7.9949 to 14.9909), respectively, as the (T_{evap}) increases from (-10 to 10 °C) and (T_{cond}) is set at (40 °C).That indicates that the performance of the SEC and MEC can be further improved by R161.

4.2.2 Effect of the condenser outlet temperature on performances of SEC and MEC

In this part, the following results are obtained when the condenser outlet temperature ranges from (30 to 55 °C) and evaporator outlet temperature is fixed at 5 °C.

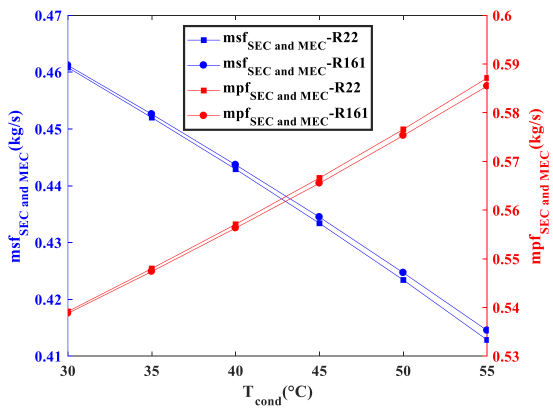


Fig.11 Variation of $m_{sf,SEC\ and\ MEC}$ and $m_{pf,SEC\ and\ MEC}$ with condensation temperature.

Fig.11 depicts the variation of the primary and the secondary mass flow by varying the condensing temperature (T_{cond}= 30 to 55 °C) at the constant evaporation temperature (T_{evap}) of

5 °C for the SEC and MEC operating with the working fluids (R22 and R161). From the results obtained, it can be observed that an increase in the condenser temperature leads to a gradual decrease in the secondary mass flow and a gradual increase in the primary mass flow of both refrigerants R161 and R22 due to a decrease of the secondary flow velocity and an increase of the primary flow velocity by the increasing condenser pressure.

On the other hand, it can be easily inferred from the curves that the secondary mass flow rate of the eco-friendly R161 would be more than using R22 in the SEC and MEC at a low condensation temperature (35°C) or high condensation temperature (55°C). However, the primary mass flow rate of the eco-friendly R161 would be less than using R22 in the SEC and MEC at a low condensation temperature (35°C) or high condensation temperature (55°C). Within the studied range of condensing temperatures, it is found that the secondary mass flow ($m_{sf,SEC\ and\ MEC}$) of the studied refrigerants R161 and R22 varies from (0.4613 to 0.4145 kg/s) and from (0.4607 to 0.4129kg/s), respectively, as the (T_{cond}) increases from (30 to 55 °C). On the other side, the primary mass flow ($m_{pf,SEC\ and\ MEC}$) of the R161 and R22 varies from (0.5387 to 0.5855 kg/s) and from (0.5393 to 0.5871 kg/s), respectively, as the (T_{cond}) increases from (30 to 55 °C).

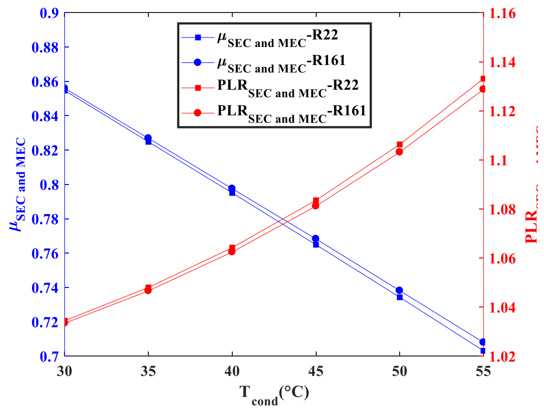


Fig.12 Variation of $\mu_{SEC\ and\ MEC}$ and $PLR_{SEC\ and\ MEC}$ with condensation temperature.

Fig.12 illustrates the simulated results of the variations of the entrainment ratio ($\mu_{SEC\ and\ MEC}$) and pressure lift ratio ($PLR_{SEC\ and\ MEC}$) with the condensing temperature (T_{cond}) varied from (30 to 55 °C) for both SEC and MEC operating with phase-out R22 and eco-friendly R161.

From the simulated results, it can be observed that the entrainment ratio ($\mu_{SEC\ and\ MEC}$) of the both working fluids (R22 and R161) decreases with the increasing of the condenser temperature (T_{cond}). However, the pressure lift ratio ($PLR_{SEC\ and\ MEC}$) of (R22 and R161) increases with the increase of the condenser temperature (T_{cond}). Since, when the condensation temperature rises from (30 to 35 °C), the primary mass flow rate increases and the secondary mass flow rate decreases, and consequently the entrainment ratio ($\mu_{SEC\ and\ MEC}$) decreases.

Compared with the phase-out R22, it is found that the eco-friendly R161 refrigerant offers values of the entrainment ratio ($\mu_{SEC\ and\ MEC}$) and pressure lift ratio ($PLR_{SEC\ and\ MEC}$) very close to those of R22 in both SEC and MEC under the same operating temperatures (T_{evap} and T_{cond}). The ($\mu_{SEC\ and\ MEC}$) values calculated of the both working fluids R161 and R22 decrease from (0.8563 to 0.7080) and from (0.8543 to 0.7032), respectively, as the (T_{cond}) increases from (30 to 55 °C) and (T_{evap}) is set at (5 °C). On the other side, the ($PLR_{SEC\ and\ MEC}$) values calculated of the working fluids R161 and R22 increase from (1.0334 to 1.1288) and from (1.0345 to 1.1331), respectively, as the (T_{cond}) increases from (30 to 55 °C) and (T_{evap}) is set at (5 °C).

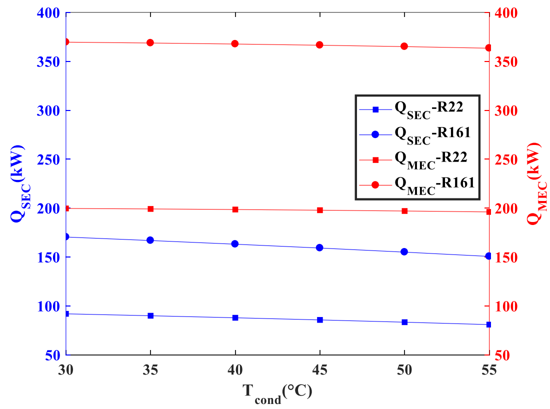


Fig.13 Variation of Q_{SEC} and Q_{MEC} with condensation temperature.

Fig.13 shows the variation of cooling capacity with the condensation temperature at an evaporating temperature of 5 °C for R22 and R161 refrigerants. As shown in the figure, the cooling capacity decreases as condensation temperature increases for all the investigating refrigerants. R161 exhibited much higher cooling effect than R22 over a wide condensation temperature range as clearly shown in Fig.13.

In comparison to the SEC and MEC, SEC shows a low cooling capacity over a wide condensation temperature range. This can be explained by the primary and secondary flows of the SEC, because not all refrigerants flows into the evaporator, where there is part of the refrigerant that flows through the compressor without passing through the evaporator. In the case of MEC, all refrigerants flow through the evaporator.

The cooling capacity values calculated of the both working fluids R161 and R22 in SEC decrease from (170.4865 to 150.6376 kW) and from (91.9855 to 80.9658 kW), respectively, as the (T_{cond}) increases from (30 to 55 °C) and (T_{evap}) is set at (5 °C). On the other side, the cooling capacity values of the R161 and R22 in MEC decrease from (369.5810 to 363.3976 kW) and from (199.6582 to 196.1105 kW), respectively, as the (T_{cond}) increases from (30 to 35 °C) and (T_{evap}) is set at (5 °C).

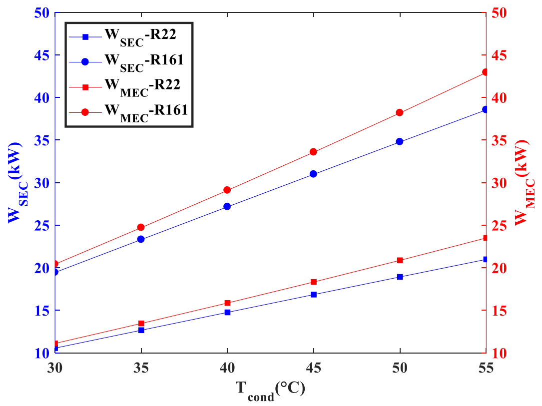


Fig.14 Variation of W_{SEC} and W_{MEC} with condensation temperature.

The effects of the condensation temperature on the power consumption of the SEC and MEC is shown in Fig.14.The figure revealed that the simulation results of power consumption increases as the condensation temperature increases for the both refrigerants. The figure also showed that R22 exhibited lower power consumption than that of R161 in both cycles.

In comparison to the SEC and MEC, SEC shows a very less power consumption over a wide condensation temperature range.

It is found that the input power in the compressor values calculated of both working fluids (R161 and R22) in SEC increases from (19.4400 to 38.5348 kW) and from (10.5564 to 20.9787 kW), respectively, as the (T_{cond}) increases from (30 to 35 °C) and (T_{evap}) is set at (5 °C). On the other side, the input power in the compressor values calculated of the R161 and R22 in MEC increases from (20.4119 to 42.9245 kW) and from (11.1109 to 23.4990 kW), respectively, as the (T_{cond}) increases from (30 to 35 °C) and (T_{evap}) is set at (5 °C).

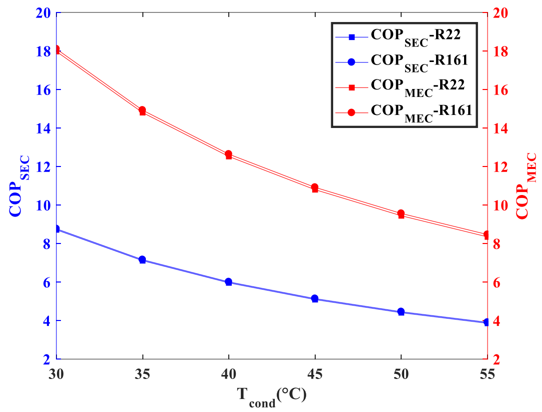


Fig.15 Variation of COP_{SEC} and COP_{MEC} with condensation temperature.

Fig.15 shows the variations of the coefficient of performance (COP) when the temperature of the condenser (T_{cond}) varies from (30 to 55 °C) for the SEC and MEC operating with R22 and R161. In Fig.15, the results show that the COP decreases with the condensing temperature monotonically. This is because the higher (T_{cond}) leads to an increase in the pressure ratio, which contributes to more power consumption of the compressor, when the evaporating temperature and pressure are maintained constant.

It could be also observed that MEC gives a higher COP compared to the SEC with both working fluids.

Compared with the phase-out R22, the eco-friendly R161 refrigerant offers very similar values of COP to those of R22 in both

SEC and MEC under the same operating temperatures (T_{evap} and T_{cond}). The coefficient of performance (COP) values calculated of the working fluids R161 and R22 in SEC decrease from (8.7699 to 3.9091) and from (8.7137 to 3.8594), respectively, as the (T_{cond}) increases from (30 to 55 °C) and (T_{evap}) is set at (5 °C). On the other side, the coefficient of performance (COP) values calculated of the working fluids R161 and R22 in MEC decrease from (18.1062 to 8.4660) and from (17.9695 to 8.3455), respectively, as the (T_{cond}) increases from (30 to 55 °C) and (T_{evap}) is set at (5 °C).

5. Conclusion

In this paper, an energy performances investigation was performed theoretically of two ejector air conditioning cycles (Standard ejector cycle (SEC) and Modified ejector cycle (MEC)) operating for the first time with the environment friendly gas R161 ((Fluoroethane), (ODP=0 and GWP=12)) as a substitute to the most widely used working fluid R22 (Chlorodifluoromethane), which will phase-out before the year 2030 due to its environmental problems (ODP=0.055 and GWP=1800).

The comparison of the thermodynamic performance simulation of both investigated working fluids (R161 and R22) in the SEC and MEC was carried out under the same air conditioning operating conditions for condensing temperature (T_{cond}) selected at (30 to 55 °C) and evaporation temperatures (T_{evap}) ranged between (-10 to 10 °C).

Based on the results of properties comparison, the following conclusions can be drawn from the present work:

- For both investigated working fluids (R161 and R22), the primary mass flow rate increases and the secondary mass flow rate decreases with the increase of the condenser temperature (T_{cond}), however, the primary mass flow rate decreases and the secondary mass flow rate increases with the increasing evaporation temperature (T_{evap});
- At the same operating temperatures (T_{cond} and T_{evap}), the secondary mass flow rate of the working fluid R161 is higher than that of R22 in the SEC and MEC, however, the primary mass flow rate of the working fluid R161 is lower than that of R22 in the SEC and MEC;
- The primary and the secondary mass flow rates of the investigated working fluids (R161 and R22) have a significant effect on the thermodynamic performances (COP, μ, and PLR) in the SEC and MEC, especially for the μ;
- At the same operating temperatures, the eco-friendly R161 refrigerant offers similar values of coefficient of performance (COP), entrainment ratio (μ), and pressure lift ratio (PLR) compared to those of R22 in both cycles (SEC and MEC);
- The PLR of the investigated working fluids (R161 and R22) in the SEC and MEC increases with the condenser temperature (T_{cond}) and decreases with the increasing of the increasing evaporation temperature (T_{evap});
- Under the same operating temperatures, R161 showed a higher cooling effect than R22 in the SEC and MEC, since; R161 has a higher latent heat of vaporization compared to R22. The high normal boiling point has an effect on the latent heat of vaporization whereas the high normal boiling point makes a greater latent heat of vaporization and as a result, the refrigerating effect increases. In another, the R161 has a critical temperature much higher than that of R22, which makes a greater heat transfer;
- The MEC showed a higher cooling effect than the SEC because all refrigerants flows through the evaporator of the MEC, which makes a greater heat transfer;
- The phase-out R22 offers lower power consumption than that of the eco-friendly R161 in the SEC and MEC;
- The COP and μ of the investigated working fluids (R161 and R22) in the SEC and MEC increases with the increasing evaporation temperature (T_{evap}) and decreases with the increasing of the condenser temperature (T_{cond});
- The COP of the MEC operating with R161 and R22 shows an improvement above that of the SEC under the same operating temperatures (T_{evap} and T_{cond}).

Finally, it can be concluded, from the above analyzing of the thermodynamic performances of both studied working fluids (R161 and R22) in ejector air conditioning cycles (SEC and MEC), that R161 offers similar values of thermodynamic performances to those of R22 in (SEC and MEC), which confirm that the eco-friendly working fluid R161 can replace the phase-out R22 in ejector air conditioning systems.

In the next step, it would be very interesting to make an energetic analysis in future works with both working fluids in other ejector air conditioning systems.

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