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NUMERICAL SIMULATION OF R1234ZE EJECTOR-EXPANSION REFRIGERATION CYCLE

Ezzaalouni yathreb[1], Lakhdher Kairouani [2] & Ezzedine Nahdi[3]

Unité de Recherche Energétique et Environnement, National Engineering School of Tunis, BP 37 Le Belvédère, 1002 Tunis, Tunisia

Email:ezzaalouni.yathreb26@gmail.com

Abstract : - Nowadays, the world faces a lot of challenges, mostly energy and environment crises. Refrigeration and air conditioning systems share with an enormous part in the world energy consumption. Reducing this energy consumption will not only contribute to solve energy crisis but also reduce the global warming by using environmentally friendly refrigerant R1234ze. Using a two phase ejector as an expansion device is a promising technique to reduce the power consumption of the traditional refrigeration systems.
A computer simulation of the improved cycle is carried out using a one-dimensional model based on mass, momentum and energy balances. Refrigerant characteristics were evaluated using NIST subroutines for equations of state solutions. According to the results of simulation of the improved cycle, it has been shown that the geometric parameters of the ejector design have considerable effects on the system's performance. The maximum COP is obtained for $Ø_{opt}$ whose value is around 9.9.Compared with the standard cycle the COP of the improved cycle shows an increase of about 18%.

Keywords: ejector, simulation, réfrigération, environnement**,** R1234ze

1. INTRODUCTION

The vapor compression refrigeration system (VCRS) is the most common used system in refrigeration, many researchers carried out many investigations to improve its performance. There are several methods to enhance performance of the vapor compression refrigeration cycle. The use of a heat exchanger for sub-cooling and superheating is a conventional method.
Recently, several researchers have used an inverter to regulate the motor rotation of the compressor according to cooling load in the cooled compartment.
However, thanks to no moving parts, low-cost, simple structure and low maintenance requirements, the use of two-phase ejector is a promising cycle modification. The use of the ejector as an expansion device by replacing the throttling valve in the vapor compression refrigeration cycle reduces the throttling loss. Moreover, the ejector also reduces the compressor specific work by raising its suction pressure and consequently increases the system coefficient of performance. The cycle that uses the two-phase ejector as an expansion device is called ejector expansion refrigeration cycle (EERC) **[2]**.
Kornhouser in 1990 **[1]** analyzed the thermodynamic performance of the ejector-expansion refrigeration cycle using R-12 as a refrigerant using constant mixing pressure model. He showed a theoretical COP improvement of

Vol.4 Issue 1, Pg.: 62-73
January 2016

up to 21% over the conventional cycle with expansion valve. Bilir and Ersoy**[3]** performed a computational analysis on the performance improvement of the ejector expansion refrigeration cycle over conventional VCRC similar to that of Kornhauser**[1]**. Using refrigerant R134a, the COP improvement of the expansion cycle over standard cycle was 10-22%. Moreover, the COP improvement increases when the condenser temperature increases. Hence, the use of the ejector instead of an expansion valve is more advantageous in the air-cooled condensers than that of water-cooled condensers.

Sarkar **[4]-[5]** investigated the performance improvement of three natural refrigerants namely, ammonia, propane and isobutane. The results revealed that a maximum performance improvement by using ejector can be achieved in the case of isobutane, where as a minimum performance improvement can be achieved for ammonia. Furthermore, the COP improvement over basic expansion cycle increases due to the increase in pressure lift ratio with the increase in condenser temperature and the decrease in evaporator temperature. The performance of the ejector expansion refrigeration cycle was theoretically studied using several synthetic refrigerants by Nehdi et al **[6]**. They performed the effect of geometric parameters such as the area ratio and the ratio of mixing chamber to primary nozzle throat area. The results revealed that maximum COP is obtained when the optimum area ratio is around 10. For the optimum area ratio, refrigerant R141b achieved the highest COP improvement over the conventional cycle 22%.

Due to the environmental concerns about ozone depletion and global warming, CFC, HCFC and HFC refrigerants are now being regulated [7].The new Solstice ze Refrigerant (HFO-1234ze) is the best medium pressure, low GWP refrigerant on the market when considering the balance of all properties. It is an energy-efficient alternative to traditional refrigerants in air-cooled and water-cooled chillers for supermarkets and commercial buildings, as well as in other medium temperature applications like heat pumps, fridges, vending machines, beverage dispensers, air dryers, CO_2 cascade systems in commercial refrigeration, etc.Multi-awarded by the industry, Solstice ze meets the criteria that are most important to refrigerants customers: Performance, Cost Effectiveness, Environmental Impact and Safety.

Field tests of air-cooled chillers in similar systems comparing the Solstice ze with the propane (R290) show significantly lower energy consumption. In addition, compared to traditional refrigerants, the properties and operating characteristics of Solstice ze are a very good match, but without the environmental penalty of high GWP HFCs.

According to compressors experts, performance with HFOs can be further improved with the optimization of compressor design.

Solstice ze exhibits similar performance to medium-pressure refrigerants like 134a, so only minor changes are required to use Solstice ze. It has lower discharge pressure results in less mechanical stress, thus extending the life of the compressor. It provides efficient cooling in all global climate zones. It is commercially available and has a GWP of 6, exceeding existing climate protection goals :

Also, atmospheric life is only 18 days, much lower than the 13 years of 134a.Solstice ze Refrigerant is significantly safer in use than alternatives such as hydrocarbons and ammonia, which are either extremely flammable or highly toxic.

Chemical name	Trans-1,3,3,3-tetrafluoropro-1-ene
Molecular formula	CHF=CHCF3
Molecular Weight	114 g/mol
ODP	0
GWP	6
Critical temperature	109.4°C
Critical pressure	36.36 bar

Table1: physical properties

In this study, a theoretical analysis of the R1234ze refrigeration cycle using a two phase ejector as expander device is carried out. The effect of the section ratio Ø of mixing chamber to primary nozzle throat area,

Vol.4 Issue 1, Pg.: 62-73
January 2016

evaporating and condensing temperature, has been considered. In this study the REFPROP IX thermodynamic characteristic routines were employed in simulation model to evaluate their performance.

2. THERMODYNAMIC ANALYSIS:

A schematic diagram of the system and its corresponding cycle states on pressure-enthalpy plot are given in Figures 1 and 2, respectively.

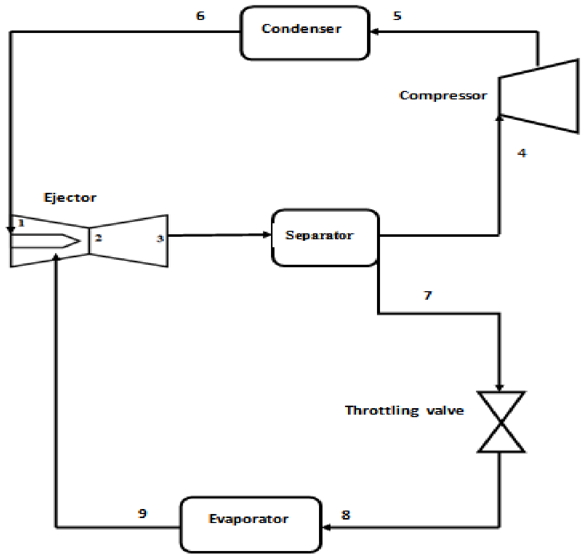

Figure1: Schematic of the improved system cycle

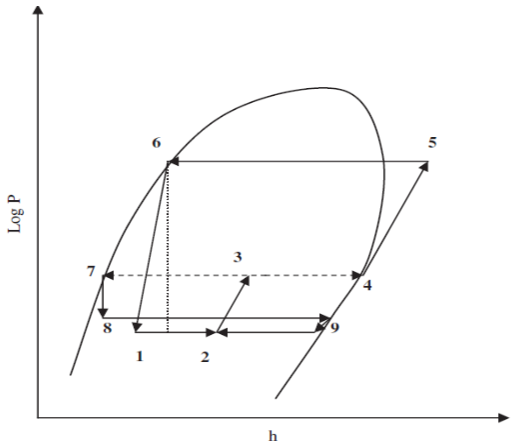

Figure 2: The log P–h diagram of the improved cycle.

As can be seen in Figure 1, the ejector is installed at the outlet of the condenser, and the motive fluid (liquid from the condenser) enters into the nozzle at a relatively high pressure. Reduction of the pressure of the liquid in the nozzle provides the potential energy for conversion to kinetic energy of the liquid. The driving flow entrains vapor out of the evaporator. The two phases are mixed in mixing chamber and leave it after a recovery of pressure in the diffuser part of the ejector. The liquid portion is directed to the evaporator through a small-pressure-drop expansion device while the vapor portion enters the compressor suction.

The ejector process on log P–h chart is shown in Figure 2. The lines from points 4 to 6 is a series process in the compressor and the condenser. The lines from points 7 to 9 is a series process in the expander and the evaporator. Points 1 and 2 are the state of the flow at the exit of the primary nozzle and in the mixing area of the ejector while point 2–3 is a compression process.

Governing equations based on the balance of mass, momentum and energy are derived for components of the system.

Vol.4 Issue 1, Pg.: 62-73
January 2016

2.1. Analysis of the ejector:

The ejector under consideration is shown in **Figure 3**. The motive flow coming from the condenser enters the ejector at a relatively high pressure and zero velocity, i.e. stagnation condition corresponding to state (0) and expands to a pressure at state (1). The secondary flow from the evaporator is then induced into the ejector by the low pressure flow at its nozzle exit. Both fluids mix together in the mixing chamber section. The mixed flow at the end of the mixing duct state (2) is discharged into a diffuser, and then the diffused flow exits from the ejector at section (3) to the separator. To simplify this analysis, the following assumptions are made in this study:

1. The refrigerant was at all times in thermodynamic quasi-equilibrium.
2. Characteristics and velocities were constant over cross section (one-dimensional model).
3. All fluid characteristics are uniform over the cross section after complete mixing at the exit of the mixing tube.
4. There is no external heat transfer.
5. There is no wall friction.

The control volume between sections (1) and (3) is divided into two regions and those are the control volume (1–2) and (2–3) as shown in Figure 3.

2.1.1. Flow nozzle:

The exit velocity from the nozzle is calculated from:

$$V_1 = \sqrt{2\eta\,(h_e - h_1)} \qquad (1)$$

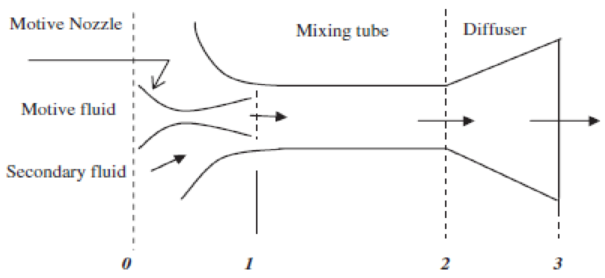

Figure 3: configuration of the ejector

h_1 is the enthalpy, at the outlet of the motive nozzle, for an isentropic process

$$h_1 = h\,(S_e, P_1\,) \qquad (1^{'})$$

$$h1a = h_1 - \eta\big(h_e - h_1\big) \qquad (2)$$

The density, at the outlet of the motive nozzle, is calculated from h_{1a} and P_1

$$\rho 1 = \rho\,(h_{1a}, P_1) \qquad (2^{'})$$

The mass flow rate is

$$\dot{m}' = \rho_1 V_1\, A_1 \qquad (4)$$

2.1.2. Flow in the mixing chamber:

Using the continuity equation, the total mass flow through the mixing tube is computed as

$$\dot{m}^{'} + \dot{m}" = \rho_2\, A_2 V_2 \qquad (5)$$

A momentum balance of the mixing tube yields

$$(P_2 - P_1)A_2 = \dot{m}'V_1 - (\dot{m}^{'} + \dot{m}")V_2 \qquad (6)$$

Combining the above equations, we can obtain the pressure rise in the mixing tube from

$$\frac{P_2-P_1}{\frac{1}{2}\rho_1 V_1^2} = 2\frac{A_1}{A_2} - 2(1+U)^2\frac{\rho_1}{\rho_2}\left(\frac{A_1}{A_2}\right)^2 \quad (7)$$

Where $U = \frac{\dot{m}''}{\dot{m}'}$ represents the flow entrainement ratio and the density ratio, $\frac{\rho_1}{\rho_2}$ can be approximated by Chen (1988) as

$$\frac{\rho_2}{\rho_1} = \frac{U}{1+U}\frac{\rho_v}{\rho_1} + \frac{1}{1+U} \quad (8)$$

ρ_v is the refrigerant's vapour density at the evaporator outlet.
The mixing velocity is defined as

$$V_2 = \frac{1}{1+U}V_1 \quad (9)$$

The velocity at the outlet section nozzle is insignificant.
At the outlet of the mixing section, by conservation of energy

$$h_2 = \frac{1}{1+U}h_c + \frac{U}{1+U}h_e - \frac{V_2^2}{2} \quad (9')$$

The entropy, at the outlet of the mixing section, is calculated from h_2 and P_2

$$S_2 = S_2(h_2, P_2) \quad (10)$$

2.1.3. Diffuser flow :

At the outlet of the diffuser, by conservation of energy

$$h_3 = h_2 + V_2^2 \quad (11)$$

The exit diffuser velocity is insignificant, so the exit diffuser enthalpy is written by

$$h_{3a} = h_2 + \eta_d\frac{V_2^2}{2} \quad (11')$$

The exit diffuser pressure is defined by S_2 and h_{3a}

$$P_3 = P(S_2, h_{3a}) \quad (12)$$

From P_3 and h_3, the exit diffuser intensive state is known (x_3, r_3. . .).
When the geometry parameters of the ejector are known, such as $\emptyset = \frac{A_2}{A_1}$; the efficiencies of nozzle and diffuser and the operating conditions, we can determine the outlet diffuser parameters such as P_3 and h_3.

2.2. Analysis of the improved cycle

The compressor undergoes a non-isentropic process for vapour compression. The power input to the compressor can be represented by the flowing equation:

$$W_{co} = (h_5 - h_4)/\eta_{comp} \quad (13)$$

Where h_4 is the saturated vapour enthalpy at P_3, h_5 is the isentropic enthalpy at the compressor outlet, η_{comp} is the isentropic compressor efficiency which determined by an empirical relation proposed by Brunin et al. (1997)

$$\eta_{comp} = 0.874 - 0.0135\tau \quad (14)$$

The cooling capacity is defined by

$$Q_e = \dot{m}''(h_9 - h_7) \quad (15)$$

Where: h_9 is the saturated vapor enthalpy at P_e, h_7 is the saturated liquid enthalpy at P_3.
The coefficient of the performance of the improved cycle system, COP_i, is determined by the following definition:

$$COP_i = \frac{Q_e}{W_{co}} = U\frac{h_9-h_7}{h_5-h_4}\eta_{co} \quad (16)$$

The relative performance of the ejector expansion cycle to the basic cycle is defined as

$$COP_r = \frac{COP_i}{COP_s} \quad (17)$$

2.3. Computational procedure

For the given geometry of the ejector and operating conditions, Equations (7), (6), (8), (9), (9'),(10), (11) and (11') are solved simultaneously. P_3 is evaluated by iteration assuming the entrainment ratio. First a value of U is guessed, P_3 and h_3 are determined.
By using REFPROP **[8]**, P_3 and h_3 give the vapor quality, x, at the diffuser outlet. Then, the value of x is compared to $\frac{1}{1+U}$.
This computation process is repeated till Equation (17) is satisfied

$$x = \frac{1}{1+U} \quad (18)$$

Finally, P_3 and h_3 are known and the COP_i and COP_r are calculated.

3. MODEL VALIDATION

A computer program in FORTRAN **[7]** is developed to simulate the thermodynamic performance of the improved cycle. The refrigerants properties are evaluated by using REFPROP V 9.0. The model is validated by comparing with the results of improved cycle using propane as a working fluid reported by **Sarkar [4]**. With T_c varying from 35 to 55 °C, the comparison for Te = 15 °C, $\eta_n = \eta_d = 0.85$, Ø=6.25 is presented in Fig. 4. It is found that the values of COP_i calculated from the present model agree well with that of **Sarkar [4]**. Hence, the validity of the mathematical model is confirmed.

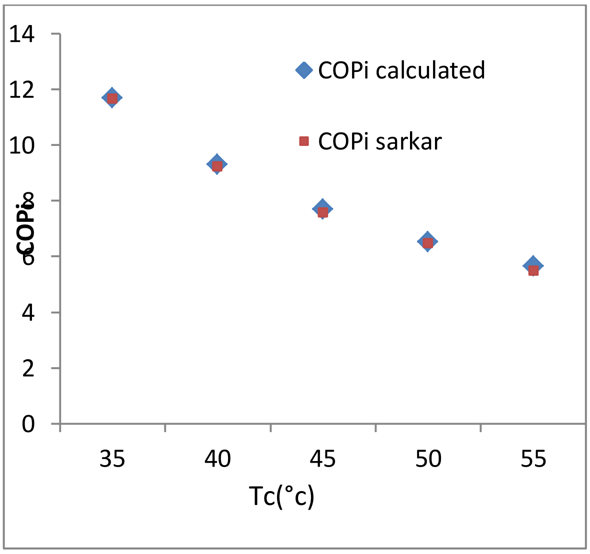

Figure 4.Comparison between the present model and Sarkar [2] on propane improvement cycle

4. RESULTS AND DISCUSSION

Based on the model developed above, the performance of the improved cycle using R1234ze is investigated for various condensing temperatures (30–55 °C) and evaporation temperatures (-15 to 0 °C). In this analysis, the ejector is assumed to have the following efficiencies: $\eta_n = \eta_d$ =0.85 [6 ,8].

4.1 Impact of condensing temperature:

Figures 5, 6, 7 and 8 depict the influence of the condensing temperature respectively on theCOP, the compressor work, the evaporator heat and the compression ratio between standard and improved cycle. The following results are obtained when simulation conditions are given as: evaporating temperatureTe = 5 °C and Ø=6.25.
Here, the condensing temperature varies between 35 and 55 °C with a step of 5 °C.The results in **Figure 5** show that the coefficient of performance decreases with the increasing condensing temperature. This is in agreement with the previous works such as [2,6].

Figure 5 shows that for T_c=55°C, Te=5°C and Ø=6.25, the COP_i value of the improved system is about 35% greater than those of the standard system (COP_{std}=2.3; COP_i =3.11; COP_r=1.35).
For the improved cycle, an addition of an ejector contributes to reducing the compression ratio. Consequently, the compressor work decreases(**Figure.6**) and the evaporator heat decreases(**Figure.7**)
Figure .8 shows that by increasing condensing temperature, the compression ratio increases and the COP_r increases.

4.2 Impact of evaporating temperature:

Figure 9, 10, 11 and 12 show the effect of condenser temperature respectively on the COP, the compressor work, the evaporator heat and the compression ratio between standard and improved cycle.The simulation conditions are given as condensing temperatureT_c = 35 °C and Ø=6.25.

Vol.4 Issue 1, Pg.: 62-73
January 2016

Here, the evaporator temperature varies between -15 and 0 °C with a step of 5 °C. From Figure 9, it can be seen that as the evaporator temperature increases, the COP decreases. This is in agreement with the previous works such as [2, 6].

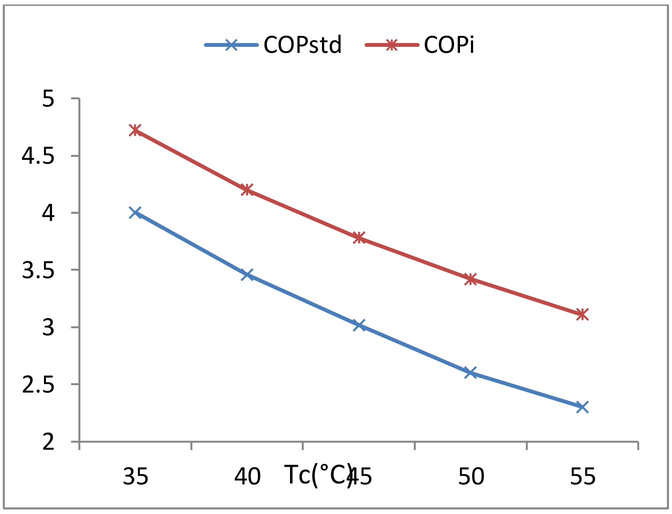

Figure 5.Comparison between COP of standard and improvedrefrigerationcycle operatingwith R1234ze versus condensing temperature Ø=6.25 and Te = 5 °C.

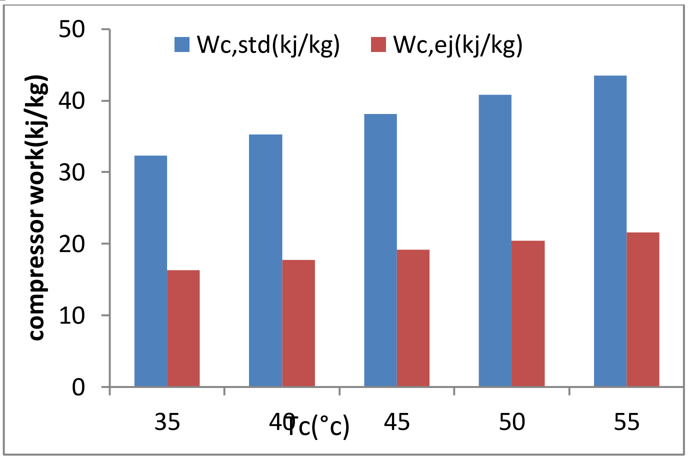

Figure 6.Compressor work for standard and improvedrefrigerationcycle operatingwith R1234ze versus condensing temperature Ø=6.25and Te = 5 °C.

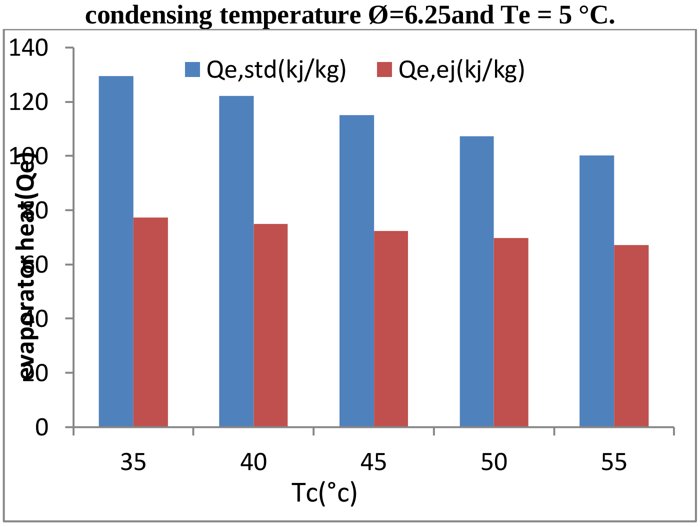

Figure 7.Comparison between COP for standard and improved refrigeration cycle operating with R1234ze versus condensing temperature Ø=6.25and Te = 5 °C.

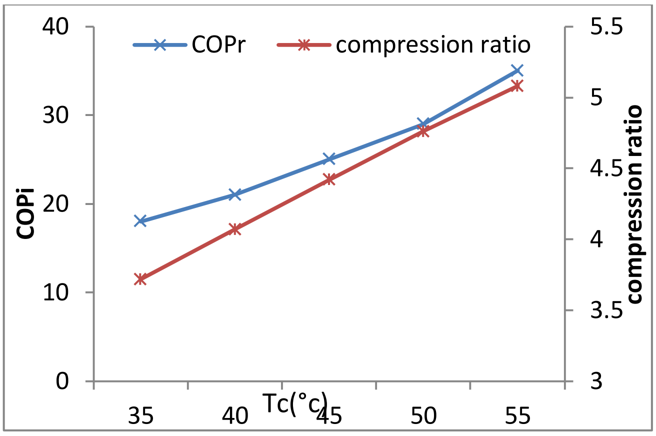

Figure 8.variation in compression ratio and the COP_r of improved refrigeration cycle operating with R1234ze versus condensing temperature Ø=6.25and Te = 5 °C.

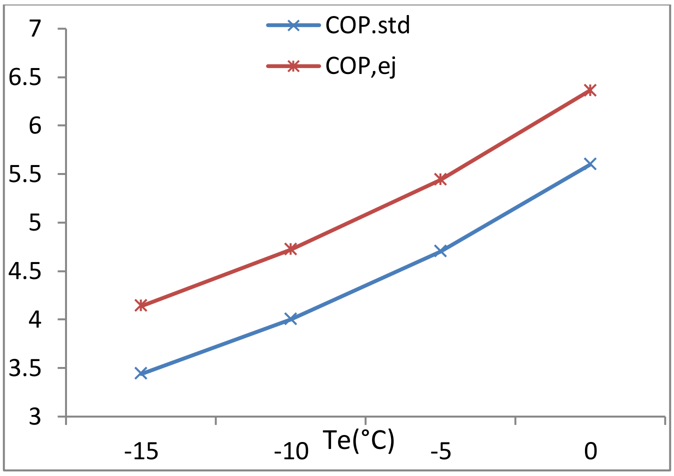

Figure 9.Variation in standard and improved refrigeration cycle COP operating with R1234ze versus evaporating temperature Ø=6.25and Tc = 35 °C.

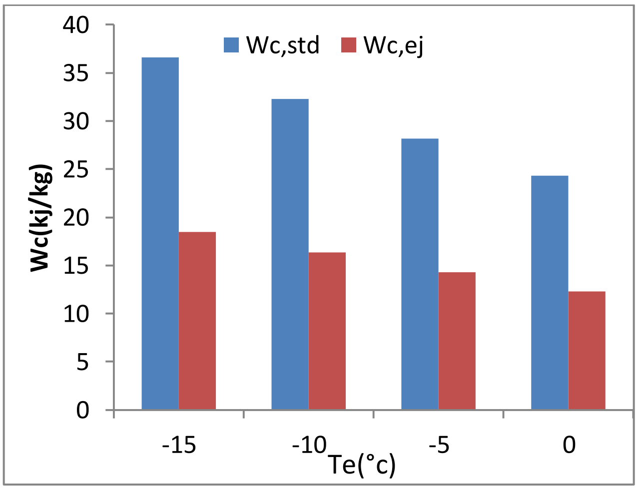

Figure 10.variation in standard and improved refrigeration cycle COP operating with R1234ze versus evaporating temperature Ø=6.25and Tc = 35 °C.

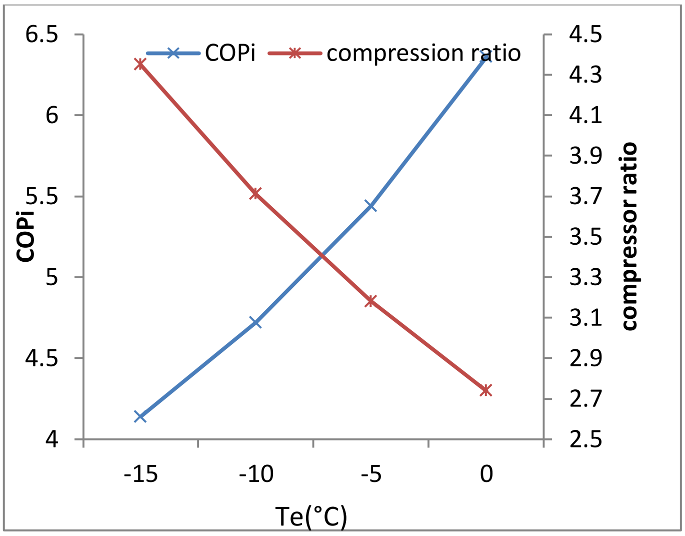

Figure 11.Variation in standard and improved refrigeration cycle COP operating with R1234ze versus evaporating temperature Ø=6.25and Tc = 35 °C.

4.3. Influence of geometric ratio Ø on the improved cycle

For given values of Tc, Te and a given refrigerant, an optimum ejector solution that satisfies the equation derived in the previous section is found. The curves on **Figure 12** and **Figure 13** indicate the effect of the area ratio on the coefficient of performance COP_i and compression ratio, it has been shown that for the improved cycle the COP increases until a maximum value is reached and then decreases with increasing area ratio Ø. The change in COP is associated with the change of the compression suction pressure, when the compression suction pressure increases, the load on the compressor decreases, and conversely. For fixed condenser and evaporator temperatures, there is only one area ratio Ø at which the COP_i has a maximum value. For example, as seen from **Figure 12**, the COP_i is at its maximum value at $Ø_{opt}$, for Tc=30°C and Te=-15°C; COPi is 4.7 for R1234ze.

The optimum of Ø was obtained experimentally by **Matsuo et al. [10]**, (Ø =15.7) and by **Nehdi et al. [11]**. In the Nehdi work**,** the performances of R11 ejectors have been studied over a large range of area ratio (from 4 to13). For given operating conditions, it was shown the existence of an optimum value of $Ø_{opt}$ =9.9, giving the maximum of COP.

Also the optimum of Ø was established theoretically by **Cizungu et al.[11]** (Ø =5.5) and by **Yapici [12]** (Ø =11.46) for different operating conditions. By selecting R1234ze as the refrigerant, Figure 12 shows that for Tc=30°C, Te=-15°C and Ø = $Ø_{opt,}$ the COPi value of improved system is about 18% greater than those of standard system (COP_r=1.18). For the improved cycle, an addition of an ejector contributes to reducing the compression ratio, consequently the load on the compressor decreases.

Figure 13 shows the variations of the compression ratio with Ø. It appears that the compression ratio decreases until a minimum value is reached and then increases with increasing area ratio. Therefore, there exists an optimum area ratio ($Ø_{opt}$), which means that the system has maximum performance, COP. If the ejector operates beyond ($Ø_{opt}$), some energy is wasted and consequently the compression ratio increases and the COP decreases.

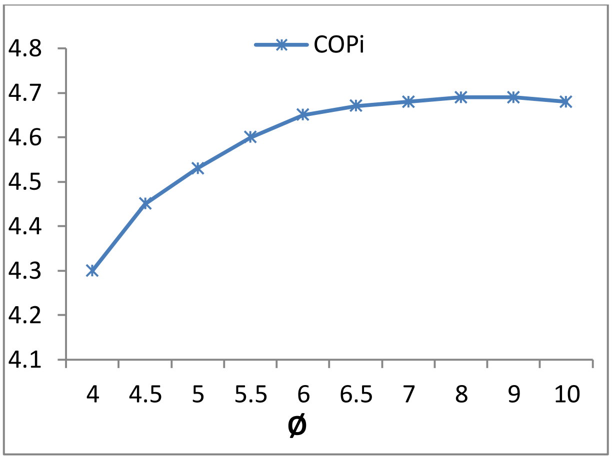

Figure12.Variation of improved cycle COP, versus geometric ratio for R1234ze (Tc=30°C and Te=-15°C).

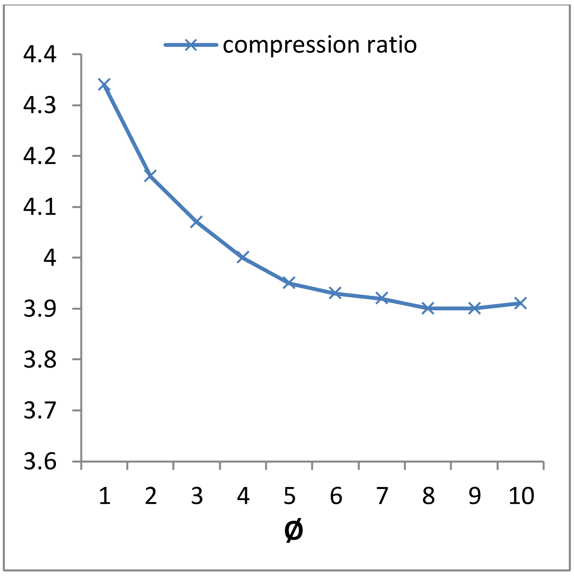

Figure 13.Variation of compression ratio, versus geometric ratio for R1234ze (T_c=30°C and Te=-15°C).

5. Conclusion

In the present study, a refrigeration cycle that combines an ejector cycle and compression cycle was described. Results have been computed for standard and improved cycle by using REFPROP.

It appears that the geometric parameters of the ejector design have considerable effects on the system's performance. The maximum COP is obtained for $Ø_{opt}$ whose value is around 9 (COP_{opt}=4.7).

Also the study shows that for a given evaporator temperature, the COP of the standard cycle decreases much more than that of the improved cycle, when the condenser temperature increases, and conversely.

NOMENCLATURE

A	=	section (m^2)
COP	=	coefficient of performance
h	=	specificenthalpy ($J\ kg^{-1}$)

m’	=	mass flow rate (kg s^{-1})
P	=	pressure (Pa)
Q	=	cooling capacity (W)
S	=	entropie (J kg1K^{-1})
T	=	température (K)
U	=	flow entrainement ratio
V	=	velocity (m s^{-1})
W	=	specific work (J kg^{-1})
W	=	power (W)
ϕ	=	geometric area ratio
ŋ	=	efficiency
r	=	density (kgm^{-3})
t	=	compression ratio

Subscripts

a	=	actual
c	=	condenser
co	=	compressor
d	=	diffuser
e	=	evaporator
i	=	improved
m	=	mixture chamber
n	=	nozzle
opt	=	optimal
r	=	ratio, relative
s	=	standard
v	=	saturated vapour at the evaporator outlet
’	=	primary
’’	=	secondary

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A Brief Author Biography

Ezzaalouni yathreb: with the diploma of engineer and master specialized engineering energetic at the national school of engineers of Monastir, I am about to Develop my doctoral thesis at the Tunis National School of engineers in the unit of energy and Environment research

Lakhdar kairaoueni: Professor at the National School of Tunis engineers and director of the Energy and Environmental Research unit.