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INTERNATIONAL ADVANCED RESEARCHES
and
ENGINEERING JOURNAL

Journal homepage: www.dergipark.org.tr/en/pub/iarej

e-ISSN: 2618-575X

International Open Access

Volume 07
Issue 01

April, 2023

Research Article

Energy and exergy analysis in the ejector expansion refrigeration cycle under optimum conditions

Servet Giray Hacipasaoglu [a,*], ***Ilhan Tekin Ozturk*** [a]

[a]*Department of Mechanical Engineering, Faculty of Engineering, Kocaeli University, Kocaeli 41380, Turkey*

ARTICLE INFO

Article history:
Received 06 September 2022
Accepted 19 March 2023
Published 15 April 2023

Keywords:
Ejector
Ejector expansion refrigeration cycle
Exergy analysis

ABSTRACT

Refrigeration systems progress in parallel with the development of technology and the ways of saving energy in refrigeration systems are being researched. The literature suggests that incorporating ejectors in refrigeration systems can boost the coefficient of performance (COP) of the system. By utilizing ejector expansion, it is possible to improve the performance of the vapor compression refrigeration cycle (VCRC) by recapturing the expansion work that is typically lost during the expansion valve process. The present study investigation aims to contribute to the field of refrigeration by exploring the optimum pressure drop for three commonly utilized refrigerants. Specifically, the study scrutinizes the performance of an ejector based refrigeration cycle that incorporates a constant pressure mixing ejector. Utilizing the energy and exergy analyses are conducted to assess the system's performance with R134a, R600a, and R290 refrigerants across five distinct evaporator temperatures, namely 0°C, -5°C, -10°C, -20°C, and -30°C. The study further determines the optimum pressure drops in the secondary nozzle and the ejector area ratio at a specified condenser temperature, and examines the resultant total exergy destruction and exergy efficiency of the system. For R290 refrigerant; performance improvement ratio, decrease in total exergy destruction and exergy efficiency improvement ratios were found as 1.23, 54.02% and 22.97%, respectively. As a result, R290 is the most appropriate refrigerant for ejector expansion refrigeration cycle (EERC) among the refrigerants investigated as a result of the energy and exergy analyses.

1. Introduction

Refrigeration systems are preservation methods using in various applications that not only create a livable environment (commercial, residential vehicles, and buildings, etc.) but also serve to industrial sector, healthcare sectors etc. Therefore, performance improvements for the VCRC cover a wide range of applications. Reducing the throttling losses in the expansion valve is one of the methods to improve the system performance [1].

The conventional refrigeration cycle typically treats the expansion work that occurs during throttling as a form of energy loss. To compensate for this expansion work loss, it is conceivable to include a turbine instead of a throttling valve. However this is not practically possible. Instead, the ejector is preferred because of its simplicity, low cost and absence of moving parts [2]. One possible way to recover some of the kinetic energy lost during the expansion process in the VCRC is to replace the expansion valve with an ejector. By doing so, the compressor suction pressure can be increased, leading to a reduction in compression work when compared to a conventional cycle.

Theoretical studies [3-7] and experimental studies [8-12] on refrigeration systems using ejectors instead of expansion valves presented that the COP is higher than the conventional systems. Ersoy and Bilir [13] performed an ejector system exergetically and examined the effects of ejector components on system performance. During their investigation, they observed that the efficiency of the ejector improved as the performance of the system increased, but this improvement was accompanied by a decrease in the ejector's area ratio. In their study, Bilir et al. [14] conducted a theoretical analysis of the COP in an ejector expander refrigerator, focusing on the influence of the refrigerant type on COP variations. The highest COP value was obtained for isobutane (R600a) among the refrigerants studied, followed by R134a. A modified VCRC utilizing an ejector as an expansion device was evaluated using three commonly used refrigerants:

* Corresponding author. Tel.: +90-262-303-3453.
E-mail addresses: girayservet@gmail.com *(S. G. Hacipasaoglu)*, ilhan@kocaeli.edu.tr *(I. T. Ozturk)*
ORCID: 0000-0001-6698-3562 *(S. G. Hacipasaoglu)*, 0000-0003-2377-6429 *(I. T. Ozturk)*
DOI: 10.35860/iarej.1171637

R134a, R407C, and R410A. The study focused on analyzing the efficiency of the system, taking into account the condenser temperature of 35°C and evaporator temperature of 5°C. The results showed that the highest efficiency based on the second law was achieved under these conditions [15].

Gao et al. [16] studied a modified double evaporator EERC with R290. The study's findings indicate that the newly developed cycle exhibits superior energy and exergy performance compared to the conventional cycle. Specifically, the modified cycle's COP and exergy efficiency were observed to attain higher values than those of the conventional cycle under specific operating conditions.

A study conducted to compare ejector compression and VCRC found that at an evaporator temperature of 5°C and a condenser temperature of 40°C, the ejector compression system showed approximately 16% higher COP and second law efficiency compared to the VCRC system. Furthermore, the total exergy destruction observed in the VCRC was approximately 24% greater than that in the ejector expander cycle, given the same operating conditions [17].

Takleh and Zare [18] introduced a new EERC and performed a thermodynamic analysis using the first and second laws of thermodynamics with six different refrigerants: R134a, R236fa, R227ea, R500, R1234yf, and R1234ze. The authors found that, of the refrigerants examined, R1234ze exhibited 5.7% and 15.5% higher exergy efficiency compared to the conventional EERC and conventional VCRC, respectively, when operating at 40°C condenser temperature and 5°C evaporator temperature.

Cui et al. [19] proposed a new ejector-supported dual-evaporator refrigeration cycle for household refrigerator applications. According to their analysis results, the new cycle showed a 7.7% and 5.5% increase in the coefficient of performance and volumetric cooling capacity, respectively, compared to the split-ejector cycle with condenser output. Furthermore, the new cycle exhibited a 57% and 58% increase in these parameters, respectively, compared to the classical VCRC.

For household refrigerator/freezer applications, Chen et al. [20] introduced an ejector vapor compression refrigeration cycle (EVRC) that utilizes a zeotropic hydrocarbon mixture of R290/R600a. When compared to the classical VCRC, the new cycle showed significant improvements of 13.5%, 19.3%, and 13.4% in its coefficient of performance, volumetric cooling capacity, and exergy efficiency, respectively.

Bai et al. [21] conducted an experimental investigation of an ejector-automatic cascading refrigeration cycle that employed a zeotropic refrigerant mixture of R134a/R23. They conducted performance comparisons between the ejector cycle and two classical refrigeration cycles under selected operating conditions.

According to the authors, the ejector cycle demonstrated notable benefits in terms of achieving lower cooling temperatures and higher energy utilization efficiency when compared to conventional cycles. Specifically, the improvements in the COP and exergy efficiency of the ejector cycle were observed to be 9.6% and 25.1%, respectively.

Jeon et al. [22] researched the performance traits of a household refrigerator-freezer using R600a refrigerant with a condenser split ejector refrigeration cycle. To investigate the impact of entrainment ratio on pressure lift effect, mass flow rate variation, and performance coefficient improvement, the authors employed a test rig. Their findings revealed that, for comparable cooling capacity conditions, the COP of the condenser split ejector cycle could be enhanced by up to 11.4% when the entrainment ratio reached 0.18, as compared to conventional cycles.

The primary purpose of incorporating an ejector in the EERC is to minimize the level of irreversibility that typically occurs during the throttling process. When an ejector is used as an expansion valve in a refrigeration system, it is crucial to conduct an exergy analysis to determine the extent to which the irreversibility is reduced in the system and its components. By applying exergy analysis, it becomes possible to assess the irreversibilities that arise within energy systems, identify their origins, magnitudes, and distribution, and consequently, devise strategies for efficient energy utilization [23]. Despite this result, possibilities to improve performance should be investigated by using different fluids and, if any, different methods.

The purpose of this study and how it differs from previous studies in the literature is to find the optimum pressure drop for different evaporator temperatures and three different refrigerants, to perform exergy analysis for the optimum situation, and to determine the losses in each component in the system. During the analyses, optimum pressure drops were determined for each evaporator temperature, and each component's exergy destruction was examined. The originality of this study lies in determining the optimum pressure drop for R134a, R600a, and R290 refrigerants in a constant pressure mixing ejector at evaporator temperatures of 0, -5, -10, -20, and -30°C, and continuing with exergy analysis using these optimum pressure drops. In this context, while the optimum pressure drop is found; the pressure lift ratio, performance improvement ratio created by the use of the ejector system, and the ejector area ratio, which determines the ejector design parameter, based on the pressure drop in the secondary nozzle were obtained.

2. Thermodynamics Model and System Description

Figure 1 and Figure 2 demonstrate the operating schematic presentation of VCRC and EERC, respectively. As seen from these figures, the EERC is an improvement of the VCRC due to the addition of a vapor-liquid separator instead of the expansion valve, as well as an ejector to minimize throttling losses.

Figure 3 illustrates the P-h diagram obtained for the R600a refrigerant at -30°C evaporator temperature in the VCRC, and Figure 4 indicates the P-h diagram obtained for the EERC at -30°C evaporator temperature using the R600a refrigerant.

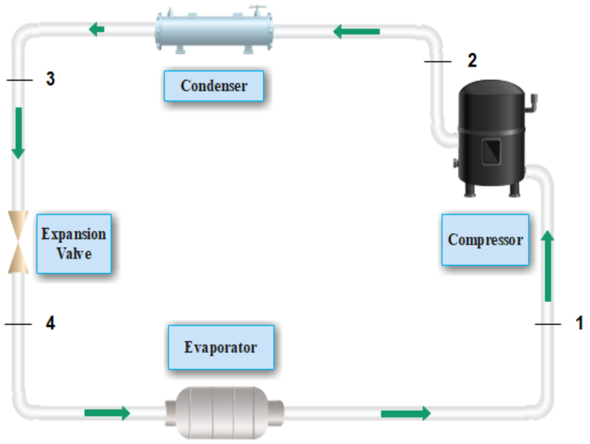

Figure 1. Schematic presentation of VCRC

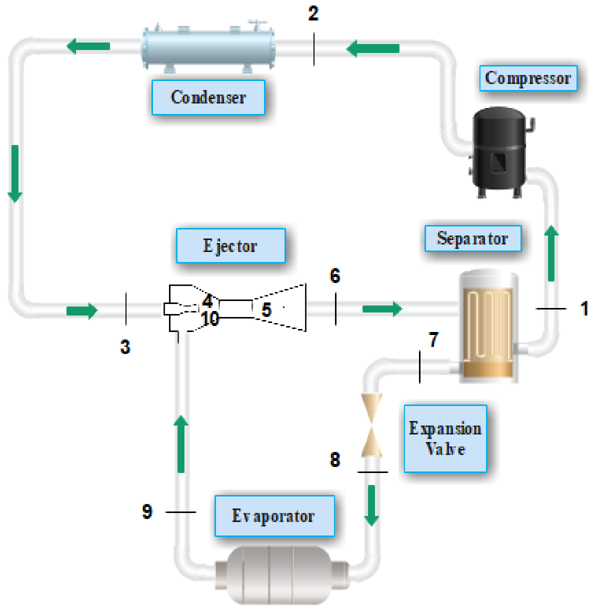

Figure 2. Schematic presentation of EERC

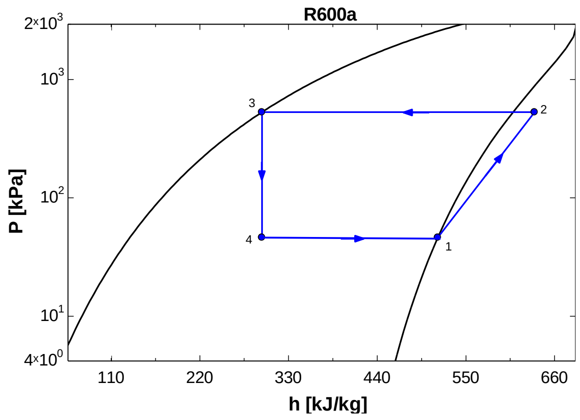

Figure 3. P-h diagram for R600a refrigerant of the VCRC.

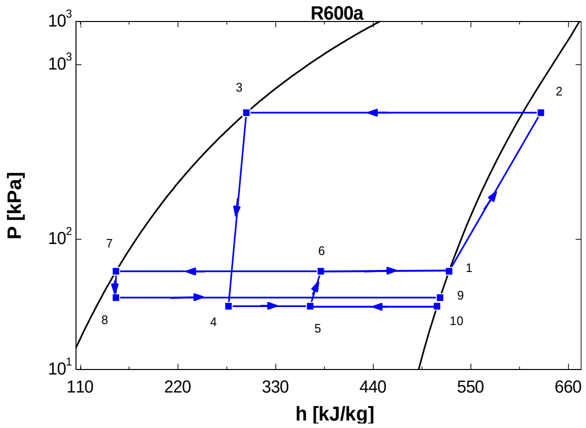

Figure 4. P-h diagram for R600a refrigerant of the EERC

This research involved the development of a thermodynamics model using a FORTRAN code specifically designed for EERC that utilizes a constant pressure ejector. The thermodynamic states of the refrigerants were obtained using REFPROP version 9.1. The model incorporates the conservation equations for mass, momentum, and energy. The thermodynamic properties of each point in the cycle were determined by considering the isentropic efficiency values for the irreversibilities in the ejector and compressor. The equations required for (constant pressure mixing) CPM modeling were used and the efficiency was assumed to be 100% for the separator (steam-liquid separator) [24]. The performance improvement ratio, ejector area ratio, and pressure lift ratio were calculated using the following methods:

Equations (1-3) are utilized to define the functioning of the motive nozzle of the ejector. Similarly, equations (4-6) are employed to represent the operations occurring in the secondary nozzle of the ejector. Moreover, equations (7-8) capture the mathematical relationships utilized to model the behavior of the ejector during constant pressure mixing. Lastly, equations (9-10) are utilized to define the functioning of the diffuser section of the ejector at the outlet. [24].

Table 1. CPM Ejector theory equations

	CPM Ejector Theory Equations
Eq. (1)	$\eta_{mn} = \frac{h_{mn,out} - h_{mn,in}}{h_{mn,out,s} - h_{mn,in}}$
Eq. (2)	$V_{mn,out} = \sqrt{2(h_{mn,in} - h_{mn,out})}$
Eq. (3)	$a_{mn} = \frac{v_{mn,out}}{V_{mn,out}(1+w)}$
Eq. (4)	$\eta_{sn} = \frac{h_{sn,out} - h_{sn,in}}{h_{sn,out,s} - h_{sn,in}}$
Eq. (5)	$V_{sn,out} = \sqrt{2(h_{sn,in} - h_{sn,out})}$
Eq. (6)	$a_{sn} = \frac{v_{sn,out}}{V_{sn,out}(1+w)}$
Eq. (7)	$V_{diff,in(mix,out)} = rV_{mn,out} + (1-r)V_{sn,out}$
Eq. (8)	$h_{diff,in(mix,out)} = rh_{mn,in} + (1-r)h_{sn,in} - \frac{V_{diff,in}^2}{2}$
Eq. (9)	$h_{diff,out} = \frac{h_{mn,in} + wh_{sn,in}}{1+w}$
Eq. (10)	$h_{diff,out} = \frac{h_{mn,in} + wh_{sn,in}}{1+w}$

With the help of the COP of VCRC and EERC, the performance of these two cycles can be compared with the help of Eq. (11).

$$R = \frac{COP_{EERC}}{COP_{VCRC}} \tag{11}$$

The area ratio given by Eq (12) is an important parameter obtained as a result of thermodynamic analysis for ejector designs.

$$A_r = \frac{a_{m,out}+a_{s,out}}{a_{m,out}} \tag{12}$$

The pressure lift ratio is determined as the ratio between the outlet pressure of the ejector and the pressure of the secondary fluid that enters the ejector, and it can be calculated by using Equation (13). The secondary flow is the flow out of the evaporator.

$$P_{lr} = \frac{P_{diff,out}}{P_{sn,in}} \tag{13}$$

Mass ratio is a parameter obtained from the entrainment ratio and is determined by Eq. (14) [24]. The primary flow is the flow out of the condenser.

$$r = \frac{m_{primary\,flow}}{m_{total}} \tag{14}$$

The reference state is denoted by a subindex of 0, an atmospheric pressure of 101.325 kPa, and 25°C ambient temperature are accepted for this study. To estimate the exergy for each component of the EERC, the mass flow rate per unit mixture in the ejector is used (Eq. 15).

$$E_i = [(h_i - h_0) - T_0(s_i - s_0)]m_i \tag{15}$$

When $i = 1, 2, 3, 4$, $m_i = r$

When $i = 5, 6$, $m_i = 1$

When $i = 7, 8, 9, 10$, $m_i = 1 - r$

The exergy losses in each component of the EERC can be obtained with the following equations (Eq. 16-20):

For compressor:

$$Ex_{comp} = (E_1 - E_2) + W_{comp} \tag{16}$$

For condenser:

$$Ex_{cond} = E_2 - E_3 \tag{17}$$

For evaporator:

$$Ex_{evap} = (E_8 - E_9) + [Q_{evap}\left(1 - \frac{T_0}{T_l}\right)] \tag{18}$$

$$T_l = T_{evap} + 5$$

For ejector:

$$Ex_{ejector} = E_3 + E_9 - E_6 \tag{19}$$

For expansion valve:

$$Ex_{exp} = E_7 - E_8 \tag{20}$$

The total exergy destruction in the system can be determined by summing up the exergy destruction of each individual component using Equation (21).

$$Ex_{total} = Ex_{comp} + Ex_{cond} + Ex_{evap} + Ex_{ejector} + Ex_{exp} \tag{21}$$

The exergy efficiency of the EERC is determined with the Eq. (22).

$$\Psi_{ej} = 1 - \frac{Ex_{tot}}{W_{comp}} \tag{22}$$

3. Operation Conditions and Model Validation

The operating conditions in the analyses are given in Table 2. Five different temperatures (0°C, -5°C, -10°C, -20°C, -30°C) were determined for the evaporator.

Table 2. Operation conditions for analyses

Evaporator temperature T_e [˚C]	0, -5, -10, -20, -30
Condenser temperature T_c [˚C]	40
Compressor isentropic efficiency (η_{comp})	0.75
Primary nozzle isentropic efficiency (η_m)	0.9
Secondary nozzle isentropic efficiency (η_s)	0.9
Diffuser isentropic efficiency (η_d)	0.8

Some general physical properties of R600a, R134a, and R290 are provided in Table 3.

Table 3. Physical properties of R600a, R134a and R290 [25]

Refrigerants	R600a	R134a	R290
Chemical name	Isobutane	1,1,1,2-Tetrafloretan	Propane
Molar mass (g/mol)	58.12	102.03	44.1
Normal boiling Point (°C)	-11.75	-26.07	-42.1
Critical temperature (°C)	134.66	101.06	96.74
Critical pressure (MPa)	3.629	4.059	4.251
Safety class	A3	A1	A3
ODP	0	0	0
GWP	4	1301	3

Figure 5 illustrates the comparison between the model created in this study and the reference model of under the same operating conditions (T_{cond} = 40°C, $\eta_{mn} = \eta_{sn}$ = 0.9, η_{comp} =0.75) using R134a as the refrigerant [26]. This comparison was performed to investigate COP with a changing evaporator temperature, with an inconsistency of 1.03%.

Figure 6 demonstrates the calculation procedure for the thermodynamic modeling of the ejector with iterative solution method.

4. Results and Discussion

Graphs of the changes in pressure lift ratio and performance improvement ratio were obtained to determine the optimum pressure drop using a constant pressure ejector for three different refrigerants while considering the basic factors affecting the performance of the ejector. Exergy analyses are performed with the pressure drops determined by these parameters.

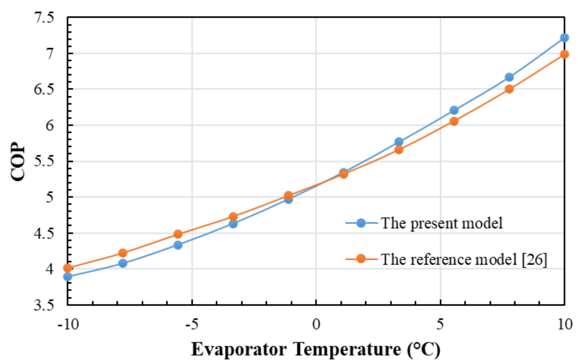

Figure 5. Comparison of the current model with (Yadav and Neeraj, 2018)

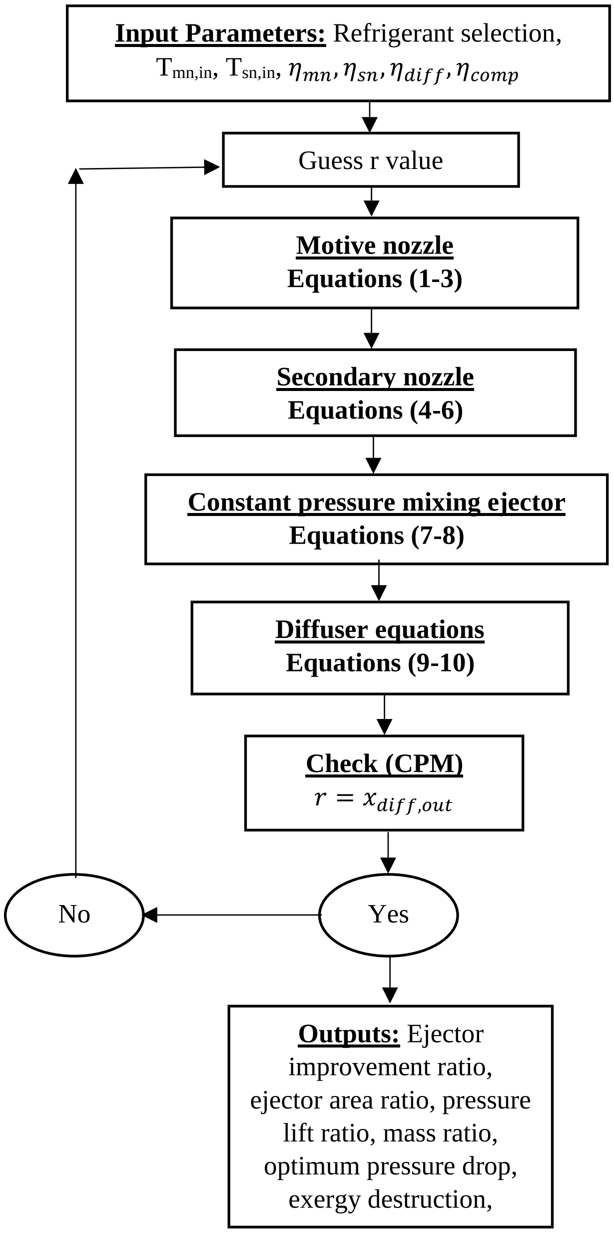

Figure 6. Flow chart of computation procedure

Figure 7-9 demonstrate the graphs obtained as a result of thermodynamic analyses using R600a refrigerant. Figure 7 displays the change in pressure lift ratio depending on the pressure drop in the secondary nozzle. The pressure lift ratio in the ejector is an important performance parameter. The higher this ratio, the higher the system performance. Because the compression work required is reduced. The aforementioned statement implies a discussion about an EERC, wherein a reduction in the compressor's work requirement could lead to an enhancement in the COP of the EERC.

In the given context, it is stated that the pressure drop at the maximum pressure lift ratio has been observed to be 5 kPa for R600a refrigerant at evaporator temperatures of 0°C, -10°C, -20°C, and -30°C. For -5°C evaporator temperature, the optimum pressure drop is approximately 9 kPa. Figure 7 and Figure 8 are actually graphs that support each other, as an increase in the pressure lift ratio means an increase in the system performance improvement ratio. The maximum value reached in the system performance improvement ratio was obtained as approximately 1.22 with a pressure drop of 5 kPa for -30°C evaporator temperature. After approximately 35 kPa pressure drop value for -30°C evaporator temperature, the performance improvement ratio decreased below 1 and has no advantage over VCRC. Looking at the same graph for -20°C evaporator temperature, it has been seen that the performance improvement ratio drops below 1 after a pressure drop of 50 kPa. This situation is also reflected in Figure 9 of the ejector area ratio graph. For evaporator temperatures of 0, -10, -20, -30°C, the optimum ejector area ratio can be found from Figure 9 by considering the optimum pressure drop amounts found in Figure 7 and Figure 8. The area ratios for each evaporator temperature are: 8.144 for 0°C, 5.85 for -5°C, 6.933 for -10°C, 5.852 for -20°C, 4.892 for -30°C.

Figure 10-12 illustrates the graphs acquired as a result of thermodynamic analyses using R134a refrigerant. For R134a refrigerant, the pressure drop with the maximum pressure lift ratio for evaporator temperatures of 0, -5 and -10°C is 15 kPa. It is approximately 14 kPa for -20°C evaporator temperature and 12 kPa for -30°C evaporator temperature.

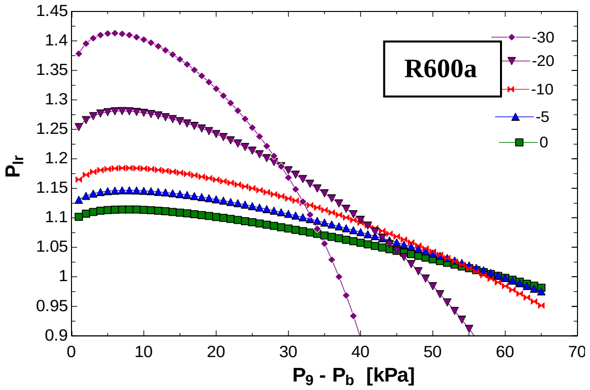

Figure 7. Variation of pressure lift ratio with pressure drop in secondary nozzle.

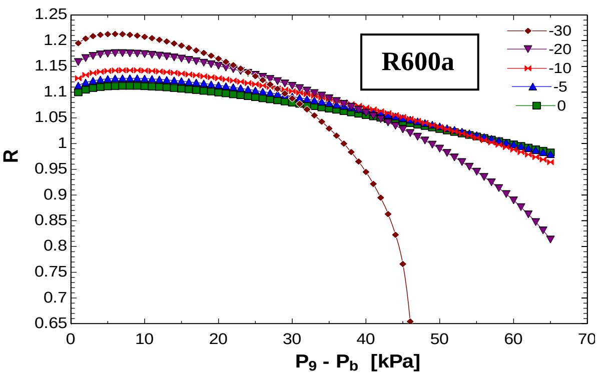

Figure 8. Variation of the performance improvement ratio with the pressure drop in the secondary nozzle.

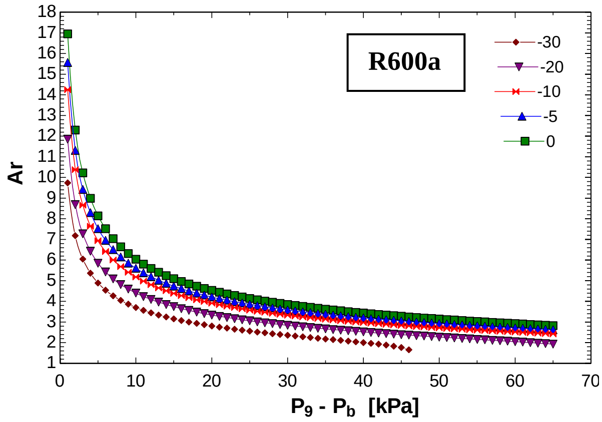

Figure 9. Variation of the ejector area ratio with the pressure drop in the secondary nozzle.

The maximum value reached in the system performance improvement ratio was obtained as approximately 1.221 with a pressure drop of 12 kPa for -30°C evaporator temperature. There is no case where the performance improvement ratio or the pressure lift ratio falls below 1 for the pressure drop values observed. The area ratios for each evaporator temperature are: 6.573 for 0°C, 6.082 for -5°C, 5.616 for -10°C, 4.892 for -20°C, 4.358 for -30°C.

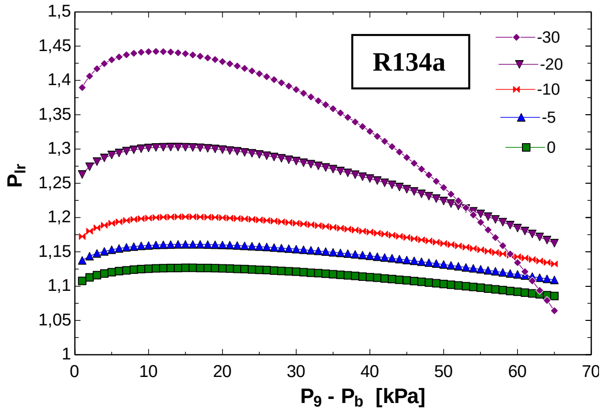

Figure 10. Variation of pressure lift ratio with pressure drop in secondary nozzle.

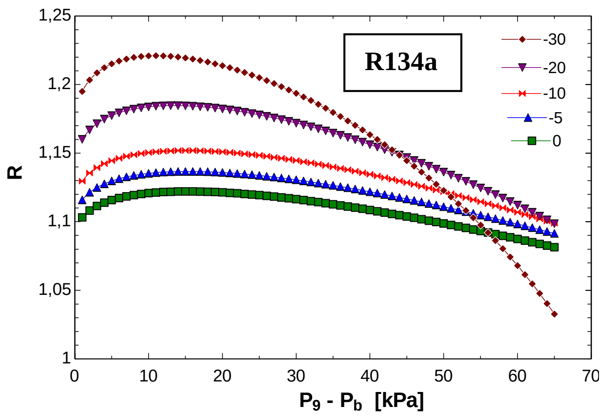

Figure 11. Variation of the performance improvement ratio with the pressure drop in the secondary nozzle.

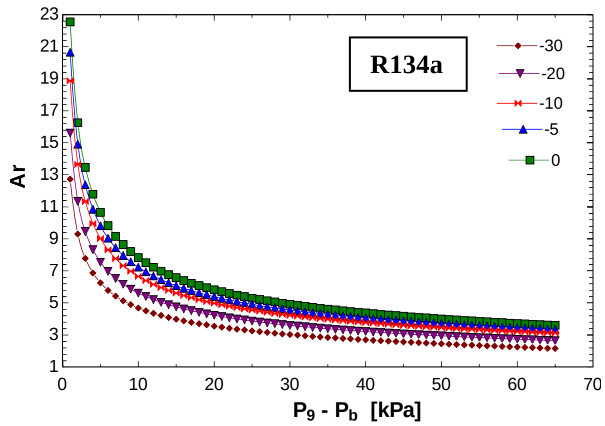

Figure 12. Variation of the ejector area ratio with the pressure drop in the secondary nozzle.

Under the conditions of a -30°C evaporator temperature and a 12 kPa pressure drop, the system performance improvement ratio for R134a refrigerant was found to be 1.221, the highest value obtained. The system performance improvement ratio for R600a refrigerant was found to be highest at approximately 1.22 under the conditions of a -30°C evaporator temperature and a 5 kPa pressure drop.

Figure 13, Figure 14, and Figure 15 indicate the graphs obtained as a result of thermodynamic analyses using R290 refrigerant. For R290 refrigerant, the pressure drop with the maximum pressure lift ratio for 0, -5, -10, -20, and -30°C evaporator temperatures is 20 kPa. The highest value obtained in the system performance improvement ratio was obtained as approximately 1.23 with a pressure drop of 20 kPa for -30°C evaporator temperature. The ejector area ratios for each evaporator temperature are: 6.705 for 0°C, 6.286 for -5°C, 5.881 for -10°C, 5.121 for -20°C, 4.424 for -30°C.

The maximum system performance improvement ratio for R134 refrigerant was achieved under the conditions of a -30 °C evaporator temperature and a 12 kPa pressure drop, with a value of 1.221.

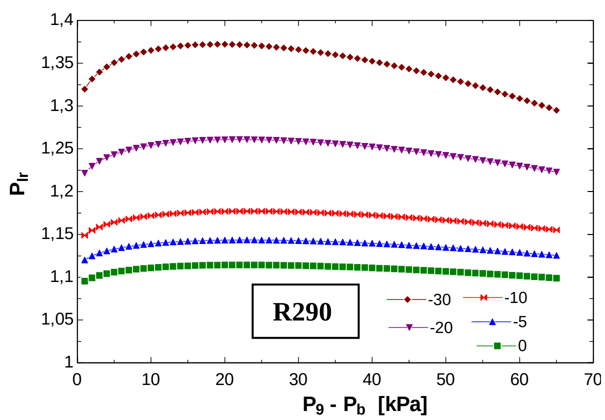

Figure 13. Variation of pressure lift ratio with pressure drop in secondary nozzle.

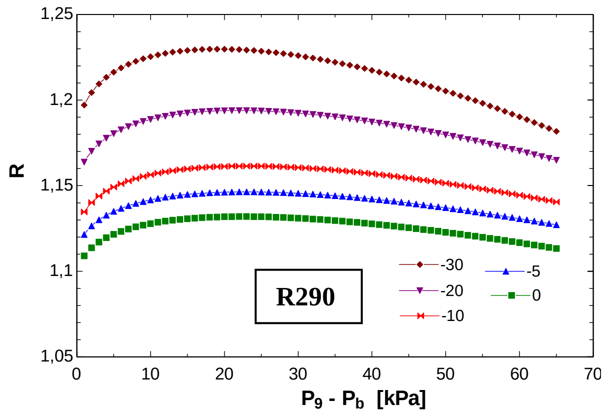

Figure 14. Variation of the performance improvement ratio with the pressure drop in the secondary nozzle.

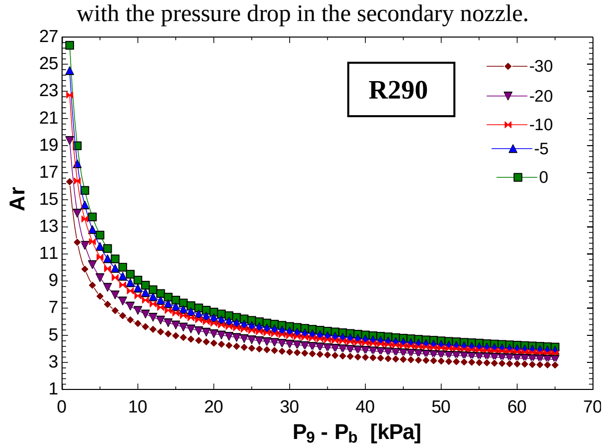

Figure 15. Variation of the ejector area ratio with the pressure drop in the secondary nozzle.

Under the conditions of a -30°C evaporator temperature and a 5 kPa pressure drop, the maximum value attained for the system performance improvement ratio of R600a refrigerant was approximately 1.22. The maximum value obtained in the system performance improvement ratio for R290 refrigerant was obtained as approximately 1.23 with a pressure drop of 20 kPa for -30°C evaporator temperature.

The primary objective of the ejector is to minimize irreversibility in the system. In the EERC and VCRC, the exergy analyses conducted in this study solely considered physical exergy, neglecting chemical, kinetic, and potential exergy.

Figure 16-21 present how the use of ejectors in the cycle will lead to a reduction in the amount of irreversibility in each component and the overall system. Exergy destruction values for each component and the overall system were computed in Figure 16 and Figure 17, utilizing the optimum pressure drop values identified earlier for R134a refrigerant at evaporator temperatures of -5°C and -30°C. For R134a, the optimum pressure drop values for evaporator temperatures of -5 and -30°C are 15 and 12 kPa, respectively. When utilizing R600a refrigerant, the optimum pressure drop values for evaporator temperatures of -5°C and -30°C are determined to be 9 kPa and 5 kPa, respectively. In the use of R290 refrigerant, the optimum pressure drop value for evaporator temperatures of -5 and -30°C is 20 kPa.

It has been calculated that the amount of exergy destruction occurring in each element of the ejector system is less than in the conventional vapor compression system. Based on the exergy analysis outcomes presented in Figure 16 for the evaporator temperature of -5°C, employing R134a refrigerant, the total exergy destruction value for the VCRC was determined to be 26.92 kJ/kg, whereas the EERC recorded a total exergy destruction value of 13.10 kJ/kg. As per the results, the total exergy destruction of the EERC reduced by 51.34% in comparison to the VCRC. The exergy destruction attributed to the expansion valve in the VCRC is calculated as 6.032 kJ/kg, whereas the EERC's expansion valve exergy destruction value is computed to be 0.0322 kJ/kg. Due to its negligible value, the expansion valve exergy destruction for the EERC was excluded from the chart.

Based on the exergy analysis outcomes presented in Figure 17 for the -30°C evaporator temperature using R134a refrigerant, the total exergy destruction values for the VCRC and EERC were determined to be 44.352 kJ/kg and 20.47 kJ/kg, respectively. The findings indicate that the total exergy destruction in the EERC was reduced by 53.84% when compared to that of the VCRC.

Upon analyzing the exergy outcomes presented in Figure 18 for the -5°C evaporator temperature with R600a refrigerant, the total exergy destruction of the VCRC was determined to be 47.22 kJ/kg, while the EERC registered a total exergy destruction value of 23.18 kJ/kg. Accordingly, the total exergy destruction of the EERC decreased by 50.91% compared to the VCRC.

In view of the exergy analysis results for -30°C evaporator temperature using R600a refrigerant in Figure 19, the total exergy destruction of the VCRC was calculated to be 75.97 kJ/kg, while the total exergy destruction of the EERC was determined to be 35.37 kJ/kg. Accordingly, the total exergy destruction of the EERC decreased by 53.44% compared to the VCRC.

Considering the exergy analysis outcomes presented in Figure 20 for the -5°C evaporator temperature utilizing R290 refrigerant, the total exergy destruction values for the VCRC and EERC were determined to be 51.91 kJ/kg and 25.04 kJ/kg, respectively. Based on the results, the total exergy destruction of the EERC decreased by 51.75% in comparison to the VCRC.

The exergy analysis results for R290 refrigerant and an evaporator temperature of -30°C showed that the total exergy destruction of the VCRC was 85.13 kJ/kg, whereas the total exergy destruction of the EERC was found to be 38.99 kJ/kg. Hence, the total exergy destruction in the EERC decreased by 54.20% compared to that of the VCRC.

In conclusion, upon comparing the exergy destruction values obtained from the analyses conducted for six different scenarios presented in Figures 16-21 for both VCRC and EERC, the most significant decrease of 54.20% was observed when using R290 refrigerant with a -30°C evaporator temperature.

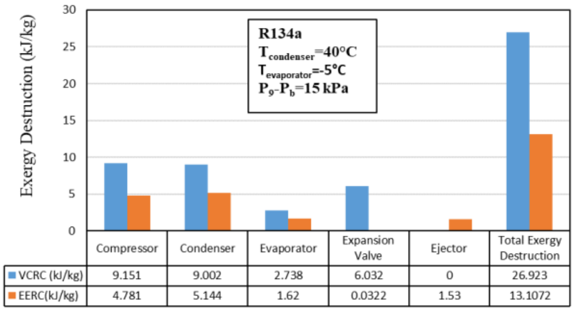

Figure 16. Comparison of EERC and VCRC exergy destruction amounts (R134a, T_{evap}=-5°C)

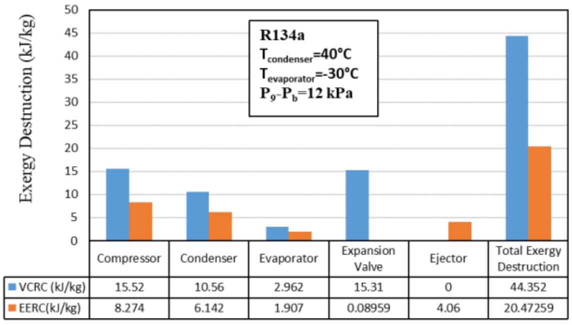

Figure 17. Comparison of EERC and VCRC exergy destruction amounts (R134a, T_{evap}=-30°C)

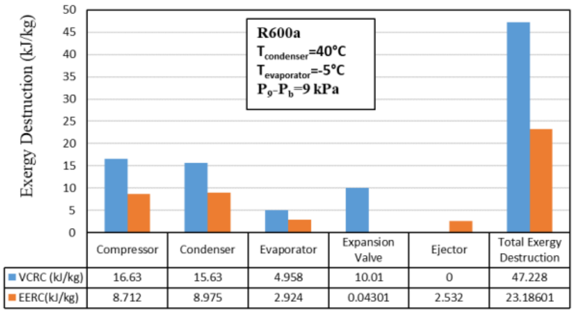

Figure 18. Comparison of EERC and VCRC exergy destruction amounts (R600a, T_{evap}=-5°C)

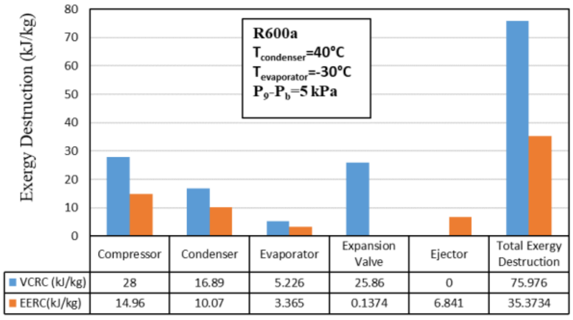

Figure 19. Comparison of EERC and VCRC exergy destruction amounts (R600a, T_{evap}=-30°C)

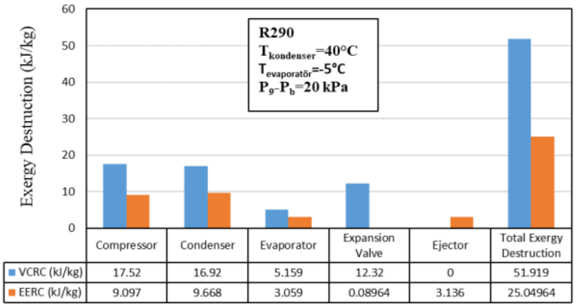

Figure 20. Comparison of EERC and VCRC exergy destruction amounts (R290, T_{evap}=-5°C)

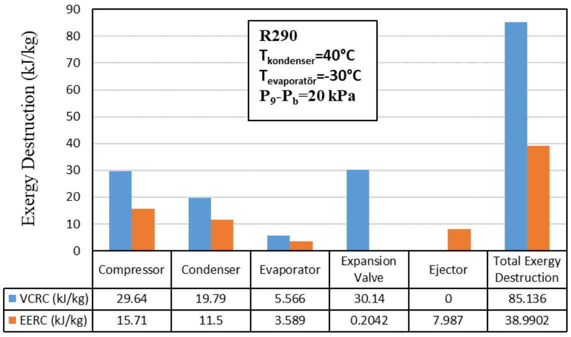

Figure 21. Comparison of EERC and VCRC exergy destruction amounts (R290, T_{evap}=-30°C)

Figures 22-23-24 exhibit how the exergy efficiency improvement ratio changes with the condenser temperature for R134a, R600a, and R290 refrigerants, respectively. As the condenser temperature increases, the rate of exergy efficiency improvement ratio also increases. Upon analyzing the situation for a single evaporator temperature, it can be noticed that the losses in the expansion valve rise as the pressure differential between the condenser and evaporator grows, which is directly proportional to the increase in the condenser temperature. Utilizing an ejector instead of the expansion valve in a refrigeration system can lead to an increase in the recycling rate of losses and thereby result in an improvement in the exergy efficiency improvement ratio. The exergy efficiency improvement ratios for R134a, R600a, and R290 refrigerants at a condenser temperature of 40°C were found to be 22.09%, 20.74%, and 22.97%, respectively. Therefore, the highest increase in terms of exergy efficiency improvement ratio was achieved with R290 refrigerant.

In Figure 25, the variation of the COP value of the EERC depending on the pressure drop in the ejector is given for R134a, R600a, R290 used in this study and R134a used in the reference model. As a result of the comparative energy and exergy analyzes for the fluids used in the study, the use of R290 refrigerant was suggested. For R290 refrigerant, the COP was found to be 6.33 at a pressure drop of 15 kPa, for R600a refrigerant at a pressure drop of 10 kPa, COP 6.243, and for R134a at a pressure drop of 25 kPa, COP was found to be 6.182. The comparison of the COP was conducted by using the graph, which takes into account the previous study in the literature that utilized R134a refrigerant. The results of this study were validated, and it was highlighted that R290 refrigerant is the preferred option among these refrigerants (T_{cond} = 40°C, T_{evap} = 5°C, $\eta_{mn} = \eta_{sn}$ = 0.9, η_{comp} =0.75).

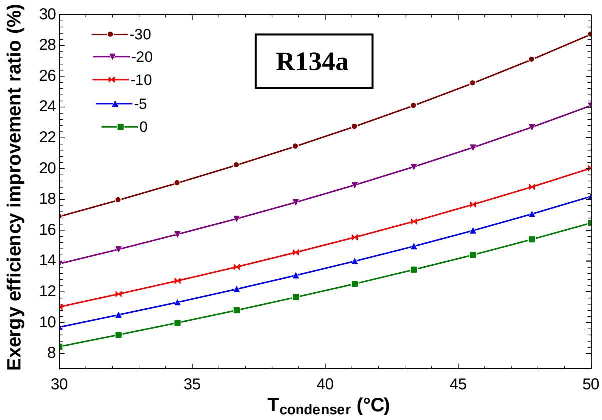

Figure 22. Alteration of the exergy efficiency improvement ratio with condenser temperature for R134a refrigerant

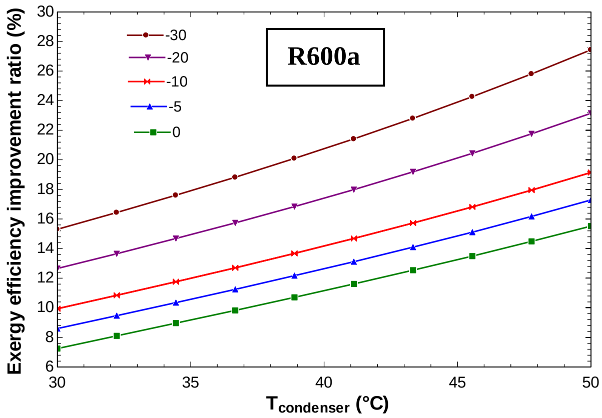

Figure 23. Alteration of exergy efficiency improvement ratio with condenser temperature for R600a refrigerant

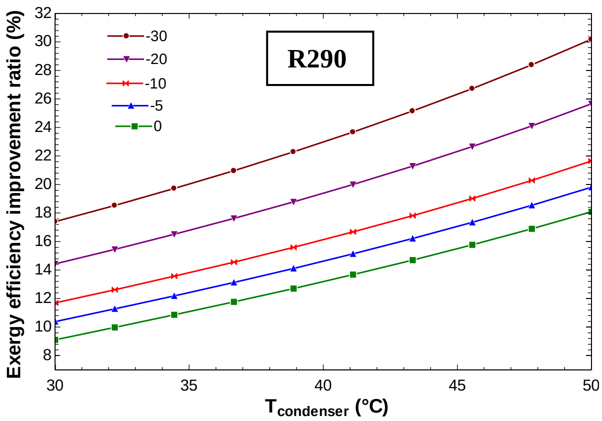

Figure 24. Alteration of exergy efficiency improvement ratio with condenser temperature for R290 refrigerant

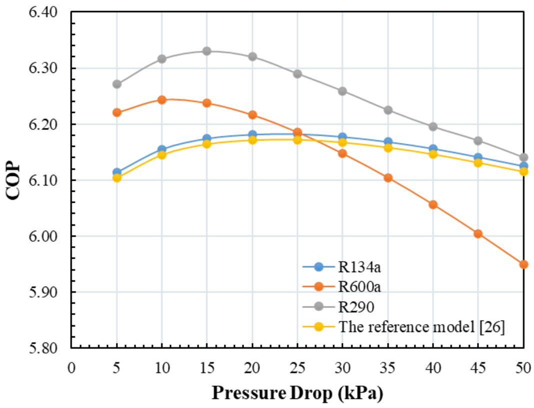

Figure 25. Alteration of pressure drop with COP

5. Conclusions

In VCRC, the use of ejectors in the system has been investigated in order to reduce the losses caused by the expansion valve and to reduce the work spent on the compressor. EERC energy and exergy analyses based on the constant pressure ejector flow model were performed. The present study focused on determining the optimum pressure drop in the secondary nozzle and corresponding ejector area ratio, total exergy destruction values, and exergy efficiency improvement ratio for a given condenser temperature. In addition, energy and exergy analyses were conducted for five different evaporator temperatures using three different refrigerants at optimum conditions. The study yielded the following findings:

- For R134a refrigerant, the pressure drop with the maximum pressure lift ratio for evaporator temperatures of 0, -5 and -10°C is 15 kPa. It is approximately 14 kPa for -20°C evaporator temperature and 12 kPa for -30°C evaporator temperature. The maximum value reached in the system performance improvement ratio was obtained as approximately 1.221 with a pressure drop of 12 kPa for -30°C evaporator temperature. It was obtained as approximately 1.13 with a pressure drop of 15 kPa for -5°C evaporator temperature.
- For R600a refrigerant, the pressure drop with the maximum pressure lift ratio for evaporator temperatures of 0, -10, -20 and -30 °C is 5 kPa. For an evaporator temperature of -5 °C, it is approximately 9 kPa. The maximum value reached in the system performance improvement ratio was obtained as approximately 1.22 with a pressure drop of 5 kPa for -30°C evaporator temperature. It was obtained as approximately 1.13 with 9 kPa pressure drop for -5°C evaporator temperature.
- For R290 refrigerant, the pressure drop with the maximum pressure lift ratio for 0, -5, -10, -20, and -30°C evaporator temperatures is 20 kPa. The maximum value reached in the system performance improvement ratio was obtained as approximately 1.23 with a pressure drop of 20 kPa for -30°C evaporator temperature.
- When comparing the exergy destruction values obtained from the exergy analyses for six different cases, it was found that the greatest decrease in total exergy destruction of the system was achieved when R290 refrigerant was used and the evaporator temperature was set to -30°C, resulting in a reduction of 54.02% for both EERC and VCRC. Also, as a result of the analyses made to determine the increase in exergy efficiency, the highest exergy efficiency improvement ratio was obtained in R290 refrigerant. As a result of the energy and exergy analyses, it can be concluded that R290 is the most suitable refrigerant for EERC among the refrigerants investigated.
- In the further studies; Thermoeconomic analysis using R290 refrigerant and performance analysis of other environmentally friendly fluids for EERC can be recommended.

Declaration

The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article. The authors also declared that this article is original, was prepared in accordance with international publication and research ethics, and ethical committee permission or any special permission is not required.

Author Contributions

İ. T. Öztürk developed the methodology. S. G. Hacıpaşaoğlu performed the analysis. İ. T. Öztürk supervised and improved the study. S. G. Hacıpaşaoğlu and İ. T. Öztürk wrote the manuscript together. İ. T. Öztürk proofread the manuscript.

Nomenclature

A_r	:Area ratio
h	:Specific Enthalpy [kJ/kg]
m	:Mass [kg]
mn	:Primary nozzle
o	:Reference situation
out	:Outlet
P	:Pressure [kPa]
P_b	:Mixing pressure [kPa]
P_{lr}	:Pressure lift ratio
r	:Mass ratio
R	:Performance improvement ratio
s	:Specific Entropy [kJ/kgK]
sn	:Secondary nozzle
T	:Temperature [°C]
tot	:Total

Greek symbols

W	:Work per unit mass [kJ/kg]
η	:Isentropic efficiency
Ψ	:Exergy efficiency improvement ratio

Abbreviations

1, 2, 3, .	:State points
comp	:Compressor
cond	:Condenser
COP	:Coefficient of Performance
CPM	:Constant Pressure Mixing
diff	:Diffuser
E	:Exergy [kJ/kg]
EERC	:Ejector Expansion Refrigeration Cycle
evap	:Evaporator
Ex	:Exergy destruction [kJ/kg]
exp	:Expansive valve
GWP	:Global Warming Potential
in	:Inlet
ODP	:Ozone Depletion Potential
VCRC	:Vapor Compression Refrigeration Cycle

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