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Investigation of Exergy Efficiencies of the Components of a Two-Stage Vapor Compression Refrigeration System for Different Refrigerants

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ABSTRACT

This study examines the exergy analysis of a two-stage vapor-compression refrigeration cycle (VCRC), varying the pressure ratio of compressor I in the first stage. Two-stage VCRC consists of two evaporators, one condenser, two compressors, several expansion valves, and one flash tank. Computer code was developed using Engineering Equation Solver (EES). The effects of pressure ratio on the exergy efficiencies of compressor I and II, the condenser, evaporator I, and evaporator II were investigated in a two-stage vapor compression system for three refrigerants: R1234ze, R1234yf, and R1233zd. It was observed that the exergy efficiency of compressor II increased with increasing pressure ratio, while those of compressor I, condenser, and evaporator II decreased.

KEYWORDS Two-stage refrigeration, Exergy efficiency, Pressure ratio, R1234ze, R1234yf, R1233zd

1. Introduction

Nowadays, the energy demand in the industry is increasing day by day. In addition, this demand is also due to various factors such as population growth, economic development, and technological developments. Therefore, energy efficiency has been one of the most important topics (Seyitoglu *et al.* 2022). The increase in energy demand has led to an increased focus on energy efficiency. Energy efficiency refers to the use of technology and practices that reduce the amount of energy required to achieve a specific output. In addition, increasing energy consumption also harms the environment. Refrigeration systems contribute to global warming. The effects of global warming can be reduced by using less harmful refrigerants and increasing energy efficiency. Refrigeration systems are a significant consumer of electricity (Qureshi and Zubair 2013; Seyitoglu and Kilicarslan 2018). One of the most extensive refrigeration systems is the vapor-compression refrigeration cycle (VCRC). VCRC is widely used for cooling applications such as domestic, commercial, and industrial refrigeration. In addition, many researchers have investigated cycle optimization. The main VCRC is composed of an evaporator, condenser, expansion valve, and compressor. The components of the system are connected in a closed loop. Two-stage vapor compression cycles are used in both residential and commercial buildings, where different areas have various cooling requirements.

The two-stage VCRC has been a topic of interest to researchers for several decades. The exergy examination of a multistage VCRC employing propane and ammonia as refrigerants was conducted by (Ouadha *et al.* 2005) As a result, the most exergetic losses appeared in the condenser, compressor, and expansion valve. Mancuhan (2019) analyzed the thermodynamic analysis of refrigeration systems with flash intercoolers using R152a, R290, R507, R134a, R404a, and R717 as refrigerants. As a

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result, the COP declined as the evaporator temperature decreased for all systems, while the COP declined as the condenser temperature increased for all systems. In addition, the system exergy loss for all refrigerants increased with condenser temperature.

Kılıç (2012) performed exergy analysis for a two-stage VCRC using R404a, R407c, and R507 as refrigerants. The authors found that the evaporator had the maximum irreversibility of all refrigerants at various condensing temperatures. An experimental investigation was carried out by (Yang *et al.* 2023) on a two-stage compression refrigeration system that makes use of trans-critical CO_2. The system was tested at low temperatures with and without an ejector.

Seyitoglu and Kılıçarslan (2015) performed a thermodynamic analysis of a two-stage VCRC with refrigerants R141b, R152a, R600, and R290. They demonstrated that the irreversibility of the component increased as the condenser temperature increased, and decreased as the evaporator temperature increased.

Tello-Oquendo *et al.* (2018) present a study of two-stage VCRC with steam injection for several refrigerants, taking into consideration the limited capacity of the heat sink. The key factors affecting cycle efficiency were identified, and their impact on the heating COP was examined. The most favorable conditions for the system were assessed by employing a comprehensive model that accounted for two setups: an economizer and a flash tank. After analyzing the optimization outcomes, a formula was suggested to estimate the ideal middle-area pressure, while considering the temperature lift of the secondary fluid required by the specific application. Using energy, exergy, and economic analyses, Baakeem *et al.* (2018) conducted a theoretical examination of the performance of a multistage VCRS. The findings indicated that as the sub-cooling parameter increased, the COP also increased. Among the refrigerants tested, ammonia demonstrated the highest COP and R407C had the lowest COP. Additionally, the optimization analysis revealed that R717 was the most favorable refrigerant option. Wang *et al.* (2021) established a serial and parallel two-stage evaporation technique for VCRC with independent control of humidity and temperature in their study. The findings indicated that R600 outperformed R134a with respect to its refrigeration capabilities, as it exhibited a higher specific refrigeration capacity and coefficient of performance. On average, the specific refrigeration capacity of R600 is 102.72% greater than that of R134a.

The literature review has shown that the exergy efficiencies of the components of a two-stage vapor compression haven't been studied. Refrigerants such as R1234yf, R1234ze, and R1233zd (E), which have zero ozone depletion potential and very low global warming potential, are also used. The second law approach of the two-stage VCRC was performed using EES software in this study. The effects of the pressure ratio on the components' exergy efficiencies, including evaporator I-II, condenser, and compressors I and II, were observed.

2. System Description

Fig. 1 depicts a schematic of the two-stage vapor-compression refrigeration system. This study includes two compressors, two evaporators, a condenser, four expansion valves, and an intercooler. Compressor I receives the refrigerant in state 1 as saturated vapor. The refrigerant enters the intercooler after leaving compressor I at state 2. The refrigerant exits the intercooler at state 3 as a saturated vapor. Then, the refrigerant enters compressor II, where it is compressed to condenser pressure. It moves into the system's condenser. At this stage, the refrigerant rejects heat into the surrounding air before leaving as a saturated liquid at state 5. At this point, the refrigerant is divided into two ways. The refrigerant of the first way enters evaporator I at state 11. The refrigerant in the other path enters the intercooler at state 6. The refrigerant exits the intercooler at state 7 and enters evaporator II at state 8. Refrigerant from evaporators I and II combine in the mixing chamber. The cycle is completed as the refrigerant leaves the mixing chamber and re-enters the compressor.

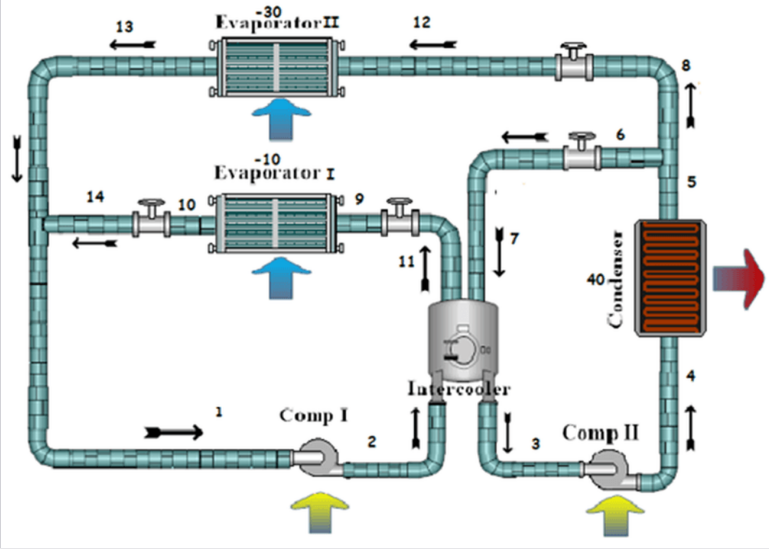

Figure 1. Schematic diagram of the two-stage refrigeration cycle.

Refrigerants with a high heat capacity and a low boiling point are preferred, as they can absorb large amounts of heat and transfer it efficiently. Additionally, refrigerants should be non-toxic, non-flammable, and have a low environmental impact. The refrigerants used in this study have zero ozone-depleting potential (ODP) but some global warming potential (GWP). R1234ze, R1234yf, and R1233zd are members of the hydrofluoroolefin (HFO) family. The refrigerant properties are illustrated in Table 1 (Wikipedia no date).

Table 1. Thermophysical properties of the selected refrigerants.

Refrigerants	Ozone Dep. Pot. (ODP)	Global Warm. Pot. (GWP) (100 yr)	Boiling Point Temp. (°C)	Crt. Temp. (°C)	Safety Group
R1234yf	0	4	−29	94.7	A2L
R1234ze	0	6	−19	109.36	A2L
R1233zd (E)	0	1	19	165.5	A1

The T-s and P-h diagrams of the cycle for each refrigerant used in this study are shown in Fig. 2.

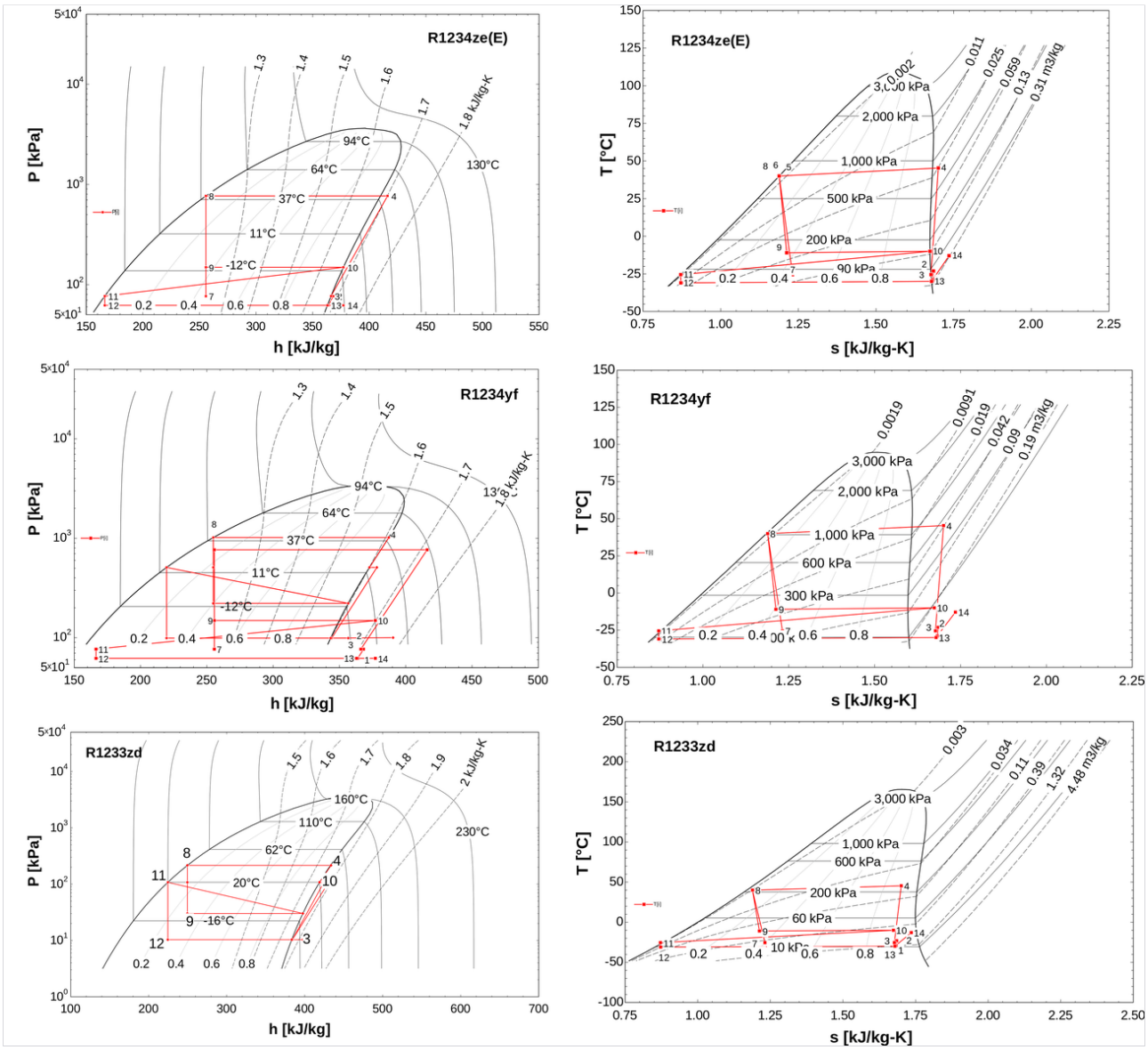

Figure 2. The refrigeration system's P-h and T-s diagrams for each refrigerant.

3. Thermodynamic Approach

To assess the exergetic performance of the two-stage VCRC, a thermodynamic study of the proposed refrigeration cycle is conducted. The assumptions made for the thermodynamic analysis of the refrigeration cycle in this study are as follows:

- The refrigeration cycle components operate in a steady-state, steady-flow process.
- Kinetic and potential energy changes during the analysis are negligible.
- Compressors and expansion valves are adiabatic.
- The pressure loss during the flow of the refrigerant through the pipes, condenser, and evaporator is negligible.
- The ambient has a temperature $T_0 = 25$ °C and pressure $P_0 = 1$ atm

Exergy can be defined as the quality and potential work measure of different energy types depending on the environmental conditions. Exergy analysis provides information on losses across all components of a system. Thanks to this analysis, the significance level of those losses within the system can be understood, and the best working conditions can be determined. To calculate the

exergy productivity of all elements of a compression-type theoretical refrigeration system, the exergy that enters and exits each element should be calculated. With respect to the second law of thermodynamics, the exergy balance can be shown (Dincer and Kanoglu 2010).

$$\dot{E}_Q - \dot{E}_W + \dot{E}_{mass,in} - \dot{E}_{mass,out} = \Delta\dot{E} = \dot{I} \tag{1}$$

Where:

- $\dot{E}_Q$ = exergy transfer accompanied by heat transfer (kW)
- $\dot{E}_W$ = exergy transfer accompanied by work (kW)
- $\dot{E}_{mass,\,in}$ = exergy rate of mass entering the control volume (kW)
- $\dot{E}_{mass,\,out}$ = exergy rate of mass exiting the control volume (kW)
- $\dot{I}$ = irreversibility rate (exergy destruction) (kW)

$$\dot{E}_Q = \dot{Q}(1 - \frac{T_0}{T}) \tag{2}$$

$$\dot{E}_w = \dot{W} \tag{3}$$

$$\dot{E}_{mass} = \dot{m}_r \psi \tag{4}$$

Where:

- $\dot{Q}(1 - \frac{T_0}{T})$ = the exergy transfer rate that is accomplished by the heat transfer rate $\dot{Q}$
- T_o = environment temperature (dead state temperature) (K)
- T = boundary temperature (K)
- $\dot{m}_r$ = mass flow rate of the refrigerant (kg/s)
- ψ = specific flow exergy (kJ/kg)

$$\psi = (h - h_o) - T_o(s - s_o) \tag{5}$$

Where:

- h = specific enthalpy (kJ/kg)
- h_o = environment enthalpy (dead state specific enthalpy) (kJ/kg)
- s = specific entropy (kJ/(kg·K))
- s_o = environment entropy (dead state specific entropy) (kJ/(kg·K))

The following formulas can be used to determine the irreversibility of each system component:

$$\dot{I}_{comp1} = \dot{m}_1 T_0 (s_2 - s_1) \tag{6}$$

$$\dot{I}_{comp2} = \dot{m}_2 T_0 (s_4 - s_3) \tag{7}$$

$$\dot{I}_{cond} = \dot{m}_4 T_0 \left[(s_5 - s_4) + \frac{(h_4 - h_5)}{T_0} \right] \tag{8}$$

$$\dot{I}_{eva\,I} = \dot{m}_9 T_0 \left[(s_{10} - s_9) + \frac{(h_9 - h_{10})}{T_{rs}} \right] \tag{9}$$

$$\dot{I}_{eva\,II} = \dot{m}_{11} T_0 \left[(s_{13} - s_{12}) + \frac{(h_{12} - h_{13})}{T_{rs}} \right] \tag{10}$$

Where:

- T_{rs} = reference state temperature (K)
- $1, 2, 3, 4, 5, 9, 10, 11, 12\ and\ 13$ = state numbers in Fig. 1

The exergy efficiency of the components is defined by taking into account all hypotheses and equations as follows:

$$\varepsilon_{comp1} = 1 - \frac{\dot{I}_{comp1}}{\dot{W}_{comp1}} \tag{11}$$

$$\varepsilon_{comp2} = 1 - \frac{\dot{I}_{comp2}}{\dot{W}_{comp2}} \tag{12}$$

$$\varepsilon_{cond} = 1 - \frac{\dot{I}_{cond}}{\dot{m}_4(\psi_4 - \psi_5)} \tag{13}$$

$$\varepsilon_{eva\,I} = 1 - \frac{\dot{I}_{eva\,I}}{\dot{m}_9(\psi_{10} - \psi_9)} \tag{14}$$

$$\varepsilon_{eva\,II} = 1 - \frac{\dot{I}_{eva\,II}}{\dot{m}_{11}(\psi_{13} - \psi_{12})} \tag{15}$$

The most preferred software for thermodynamic analysis, EES (Klein 2020) was used in this study. Where:

- ε = exergy efficiency

4. Results and Discussion

R1234ze, R1234yf, and R1233zd are used as refrigerants in a two-stage vapor-compression refrigeration system. Several parameters, such as compressor isentropic efficiencies, condenser temperature, evaporator I temperature, and evaporator II temperature, are kept constant while generating the program code. The effect of the pressure ratio of compressor II on the exergy efficiencies of the system components, such as compressors I and II, the condenser, and evaporators I and II, is presented and discussed in the following paragraphs.

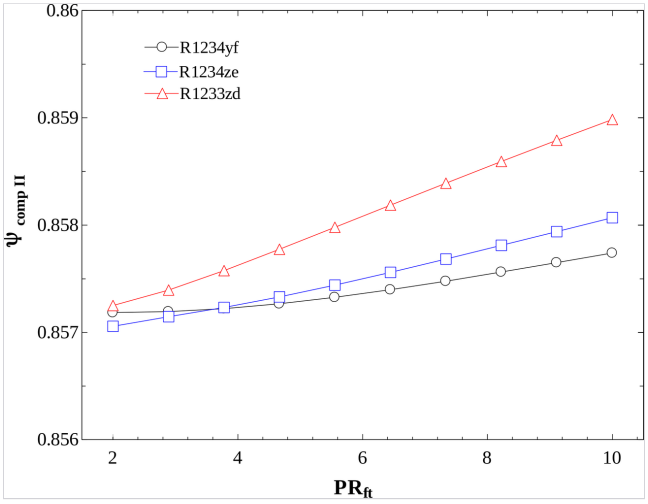

Figure 3. Variation of the exergy efficiency of compressor II with the pressure ratio of compressor II.

Fig. 3 depicts the change of exergy efficiency of compressor II with respect to the pressure ratio of compressor II. As the pressure ratio of compressor II increases, both the power consumed by compressor II and its exergy destruction increase. The ratio of the increase in the exergy destruction in compressor II is a little bit less than the power consumed by compressor II. As a result of this, the exergy efficiency of compressor II increases for refrigerants R1234ze, R1234yf and R1233zd as shown in Fig. 3. At the lowest pressure ratio (2), the exergy efficiencies obtained using refrigerants R1234ze, R1234yf and R1233zd are almost the same and as the pressure ratio increases, the difference between the exergy efficiencies for the refrigerants increases as depicted in Fig. 3. As the pressure ratio is varied between 2 and 10, the exergy efficiency of compressor II increases at very little amount for the refrigerants in the cycle. The highest exergy efficiency of compressor II is obtained with R1233zd as refrigerant in a two-stage cycle, while the minimum is obtained with R1234yf.

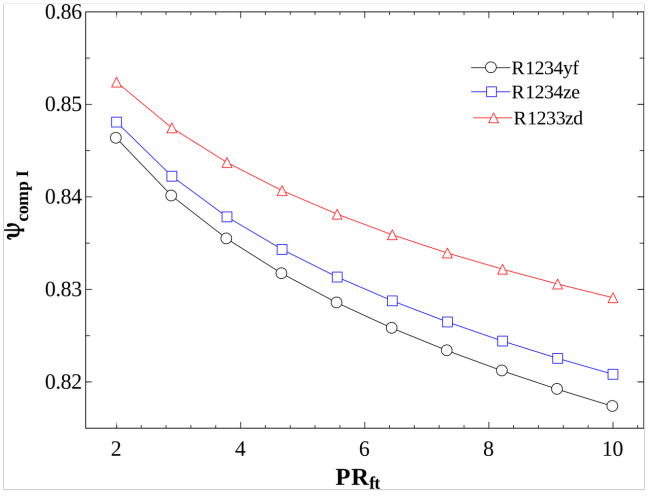

Figure 4. Variation of the exergy efficiency of compressor I with the pressure ratio of compressor II.

Fig. 4 depicts the effect of the pressure ratio of compressor II on the exergy efficiency of Compressor I. In the computer code developed, the pressure ratio of compressor II is changed by lowering its suction pressure while holding its discharge pressure constant. The suction pressure of compressor II and the discharge pressure of compressor I are equal to each other. The suction pressure of compressor I is the saturation pressure corresponding to evaporator I temperature. Evaporator I temperature is held constant. An increase in the pressure ratio of compressor II decreases the pressure ratio of compressor I. Unlike compressor II, as the pressure ratio of

compressor II increases, both the power consumed by compressor I and the exergy destruction in compressor I increase. The decrease in the exergy destruction of compressor I is less than the power consumed by compressor I. Therefore, the exergy efficiency of compressor I increases for the refrigerants including R1234ze, R1234yf, and R1233zd, as shown in Fig. 4. As seen in Fig. 4, the highest exergy efficiency of compressor I is obtained using R1233zd as the refrigerant in the cycle. Conversely, the lowest exergy efficiency of compressor I is obtained with R1234ze. When using R1233zd as the refrigerant, the pressure ratio increased by 5 times, while the efficiency decreased by about 1.02 times.

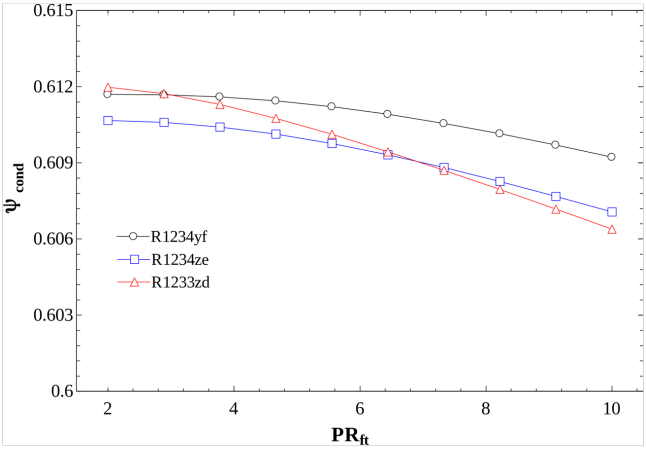

Figure 5. Change of the exergy efficiency of condenser with the pressure ratio of compressor II.

As the pressure ratio increases, the exergy efficiency of the condenser decreases for the refrigerants R1234ze, R1234yf, and R1233zd used in a two-stage vapor-compression refrigeration cycle. As shown in Fig. 5, the pressure ratio negatively affects the condenser exergy efficiency over the range of 2 to 10. The condenser's exergy efficiency declines as the pressure ratio rises. R1234yf yields the most efficient condenser exergy compared to other refrigerants, whereas R1233zd results in the least efficient condenser exergy among them.

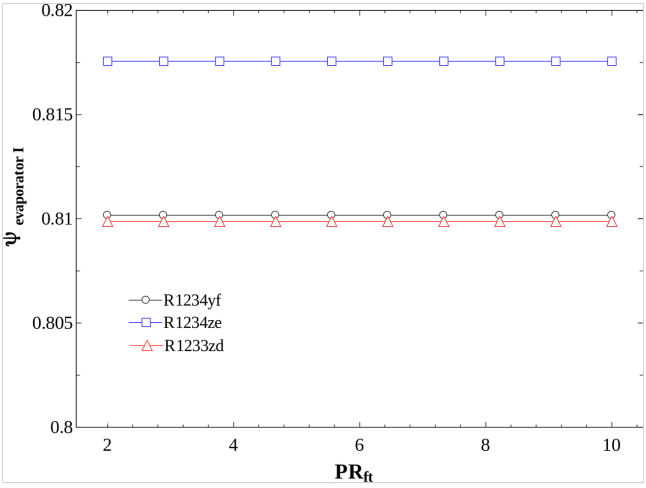

Figure 6. Variation of evaporator I exergy efficiency with pressure ratio.

Fig. 6 illustrates how the exergy efficiency of evaporator I varies with the pressure ratio. The pressure ratio is changed from 2 to 10. The temperature of evaporator I is kept constant at −10 °C. As seen in Fig. 6, the exergy efficiency of evaporator I remains constant for all refrigerants tested, even as the pressure ratio is increased. R1234ze yields the highest possible evaporator I exergy efficiency, whereas R1233zd results in the lowest possible evaporator I exergy efficiency.

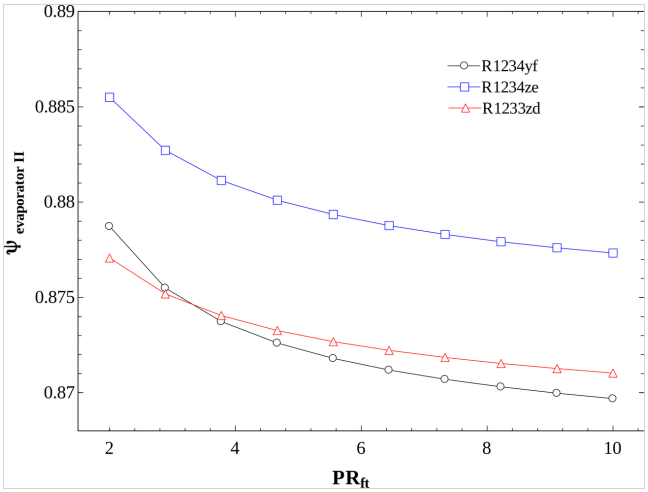

Figure 7. Variation of evaporator II exergy efficiency with pressure ratio.

Fig. 7 depicts the change of the exergy efficiency of evaporator II with respect to the pressure ratio. As the pressure ratio is increased from 2 to 10, the exergy efficiency of evaporator II decreases for all refrigerants studied. Figure 7 shows that R1234ze obtains the maximum evaporator II exergy efficiency, while the minimum evaporator II exergy efficiency is obtained by using R1234yf as a refrigerant.

In this section, the effect of pressure ratio on the refrigeration cycle, including two evaporators, two compressors, and a condenser, was discussed. When all results are examined, the exergy efficiency of components generally decreases while the pressure ratio increases. However, only one component, the compressor, has its exergy efficiency increasing.

5. Conclusion

The exergy efficiency of the system components was examined in relation to the operational parameter, namely the pressure ratio. The remarkable results obtained as a result of the analysis are presented below:

- The exergy efficiency of each component within the system was analyzed. It has been found that the condenser is the component with the lowest exergy efficiency.
- Among the system components, the maximum exergy efficiency is generally provided through the refrigerant R1234ze.
- When the pressure ratio increased, the compressor's exergy efficiency decreased. However, the compressor's exergy efficiency decreased while the condenser temperature increased.
- When the general results were examined, the compressor II exergy efficiency increased with changes in the pressure ratio.
- The pressure ratio had a little impact on the condenser's exergy efficiency.
- The exergy efficiency of evaporator I remains constant with the increase of pressure ratio.
- The highest exergy efficiency of evaporator I and II was obtained with R1234ze as a refrigerant.

Declarations

Author contributions. Ali Kilicarslan: Conceptualization, Methodology, Formal analysis, Supervision, Investigation, Writing (review and editing). Sertaç Samed Seyitoglu: Investigation, Formal analysis, Writing (original draft).

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Data availability. Not applicable. This study did not generate or analyse any datasets.

Ethics. None

Generative AI. None

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