Exergetic performance analysis of low GWP alternative refrigerants for R404A in a refrigeration system

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Abstract
According to the regulation of European Union laws in 2014, it was inevitable to switch to low global warming potential (GWP) fluids in the refrigeration systems where the R404A working fluid is currently used. The GWP of R404A is very high, and the potential for ozone depletion is zero. In this study, energetic and exergetic performance assessment of a theoretical refrigeration system was carried out for R404 refrigerant and its alternatives, comparatively. The analyses were made for R448A, R449A, R452A and R404A. The results of the analysis were presented separately in the tables and graphs. According to the results, the cooling system working with R448A exhibited the best performance with a coefficient of performance (COP) value of 2.467 within the alternatives of R404A followed by R449A and R452A, where the COP values were calculated as 2.419 and 2.313, respectively. In addition, the exergy efficiencies of the system were calculated as 20.62%, 20.22% and 19.33% for R448A, R449A and R452A, respectively. For the base calculations made for R404A, the COP of the system was estimated as 2.477, where the exergy efficiency was 20.71%. Under the same operating conditions, the total exergy destruction rates for R404A, R448A, R449A and R452A working fluids were found to be 3.201 kW, 3.217 kW, 3.298 kW and 3.488 kW, respectively. Furthermore, parametric analyses were carried out in order to investigate the effects of different system parameters such as evaporator and condenser temperature.

Keywords: alternative refrigerant; R448A; R449A; R452A; COP; exergy

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1. INTRODUCTION

In today’s world, refrigeration systems have a significant role in our daily lives owing to the rapid development and growth of the economy and technology during the past century. However, halocarbon refrigerants used in these refrigeration systems have also increased environmental problems [1]. In recent years, significant regulations and policies have been implemented in order to reduce emissions that cause the greenhouse effect and deplete the ozone layer. The main objectives according to these implementations are to lower the impact of climate change and to reduce carbon emissions. These impacts are quantified by well-known parameters, which are the global warming potential (GWP) and ozone depletion potential (ODP). One of the prevention approaches is to lower the production and utilization of refrigerants, which have a higher impact on the environment [2,3,4].

R404A is a commonly used commercial refrigerant, particularly in supermarket refrigeration systems [5]. It is a mixture of R125, R143A and R134A with 44%, 52% and 4% mass percentages, respectively [6]. It has 0 ODP value; however, its GWP value is 3800 [7,8], which is relatively high. In the choice of a refrigerant to be utilized in a cooling system, not only the ODP value but also the GWP value is required. Besides the lower environmental effects, it is desirable to use refrigerants with better energy performance [9]. As alternatives of R404A, R448A (N40) [10], R449A (XP40) [11] and R452A (XP 44) [12] can be used in commercial refrigeration systems. R448A is a blend of HFO, which is non-flammable [13]. It is used for the replacement of R404A and R22 in the cooling systems. It has a low GWP value and less harm for the atmosphere [14]. R449A is an HFO blend [15] and is used for the replacement of R404A and R507 in the cooling systems. It has a low GWP value of 1397 [16]. R452A is also an HFO refrigerant blend and it has a low GWP value,
which is 2141. It is used as a replacement for R404A and R507 in low-temperature transport refrigeration applications [17,18].

Over time, an extensive literature has developed an analysis of refrigeration systems working with R404A and comparing it with low-GWP refrigerants. Llopis et al. [19] investigated R407H as an alternative cooling fluid for R404A. Their analysis reported a reduction of ~7.7% in the energy consumption rate at low temperatures. The authors also stated that energy performance could be negatively affected over time because of the R407H compressor discharge temperature. Cardoso et al. [20] investigated the R407H and R454A refrigerants instead of the R404A. The authors found that both refrigerants’ coefficient of performance (COP) values were nearly 15% and 9% higher than that of the R404A, respectively. Also, it was pointed out that the tested alternative refrigerants could be utilized over a long period. In another study, Saengsikhao et al. [21] used R463A as an alternative working fluid for R404A. Their results showed that the COP of the alternative refrigerant was higher than that of the R404A by 10% for low-temperature applications. The life cycle climate performance of a transport refrigeration system was investigated by Li [22]. In the study, a detailed comparison of the system for R404A and R452A refrigerants were made. It was reported that R452A cooling gas has a lower emission reduction (5%–15%) for the food transportation systems. The researcher also made comparisons of pressure-enthalpy and temperature-entropy for the R404A and R452A working fluids. According to this study, R452A was reported as the most suitable refrigerant for the replacement of R404A. Mota-Babiloni et al. [23] conducted an experimental comparison of a refrigeration system for R448A working fluid instead of R404A. According to their study, R448A had a smaller cooling capacity, with a higher COP value and lower energy consumption rate. Based on their analysis, R448A with lower GWP was reported as an innovative refrigerant, which was an energy-efficient alternative to the R404A working fluid. Makhnatch et al. [15] performed an experimental analysis for indirect supermarket refrigeration systems. They stated that R449A was the most suitable alternative for R404A because of its thermodynamic properties. They made different measurements for both refrigerants in the supermarket indirect refrigeration system. In their results, the R449A working fluid showed a lower GWP and also a lower cooling capacity. Vaitkus and Dagilis [24] analyzed alternatives refrigerants of high GWP working fluids for eutectic refrigerating systems. In their study, the analyzed cooling fluid alternatives were the CO₂, HFC/HFO mixtures and hydrocarbons. According to the results of their study, the R448A, R449A, R407F and R407A refrigerants showed a 12%–16% reduction in refrigerating capacity. Mendoza-Miranda et al. [25] made an evaluation of the refrigeration system for R404A and R134A working fluids instead of R448A and R450A in the modified micro-fin evaporator model. They made experiments for different operating conditions. According to their results, the R450A and R404A refrigerants were found to be more suitable in the proposed model, but R448A and R134a predictions were also accurate. Sethi et al. [26] made an experimental study regarding R404A replacements for commercial refrigeration. They stated that R448A and R455A were the most suitable refrigerators for the replacements of R404A. R448A matches the capacity with 4%–8% higher efficiency compared to R404A. R448A resulted 9%–20% energy savings for the supermarket refrigeration systems. Devecioglu and Oruc [27] made a comparison of new generation low GWP value refrigerants at the trial stage. In their study, R448A was used instead of R404A; R450A, R513A, R1234yf and R1234ze(E) gases were used instead of R134a, DR-33, L40, DR-7 and DR-5; R447A was used instead of R410A; and finally, N₂O and R444B refrigerants were used as alternatives to R22. They presented their results with detailed graphics and tables.

Although there are many studies in the literature, a detailed comparative investigation of refrigerants R448A, R449A and R452A, which are alternatives for R404A, have never been addressed. Since the R404A refrigerant is widely used in low and medium temperature applications, especially for commercial refrigeration applications, the search for alternatives with lower GWP to replace R404A has gained importance. Therefore, in this study, a comprehensive thermodynamic performance analysis of a refrigeration system is carried out for alternative refrigerants R448A, R449A and R452 in comparison to R404A. These fluids are selected due to their thermodynamic properties and lower GWP values, which are 64%, 67% and 50% lower than that of the R404A, respectively. For the performance investigation, the first energy analysis is conducted in order to determine the COP value. After, exergy analysis is carried out for determining the exergy destruction rates of the whole system comparatively under the same operating conditions. In addition, a parametrical study is performed to examine the effects of different system parameters on the system performances.

2. SYSTEM DESCRIPTION

The vapor compression refrigeration cycle is the most widely used cycle in refrigeration machines, air conditioning systems and heat pumps [28]. The schematic diagram for the vapor compression refrigeration system is given in Figure 1. As illustrated in the figure, the vapor system mainly consists of four components—evaporator, compressor, condenser and expansion device. The refrigerant entering the compressor as the saturated steam is compressed up to the condenser pressure. In the superheated state, the refrigerant enters the condenser, where it is cooled and condensed. The fluid exits the condenser as a saturated liquid by discharging its heat to the environment. After that, it passes through the expansion valve and enters the evaporator by expanding to low pressure. At the evaporator pressure, the refrigerant is below ambient temperature, and it absorbs heat from the refrigerated space. Finally, the fluid exits the evaporator in a saturated vapor state and enters the compressor to complete the cycle.

The performance assessment of the vapor compression refrigeration cycle is conducted for three different working fluids, which may be alternatives to R404A. The main characteristics of these refrigerants are given in Table 1. As can be seen from the table, the properties of the refrigerants are close to each other; however,
Table 1. The main characteristics of the tested refrigerants [30].

| Refrigerant | R404A | R448A | R449A | R452A |
|-------------|-------|-------|-------|-------|
| Molar mass (kg/kmol) | 97.6  | 86.3  | 87.2  | 103.5 |
| Boiling point (°C) | −46.45 | −45.9 | −46   | −47   |
| Critical temperature (°C) | 72.12  | 82.68 | 81.05 | 75.05 |
| Critical pressure (kPa) | 3735   | 4595  | 4500  | 4014  |
| ODP | 0    | 0    | 1282  | 1945  |
| GWP | 3859 | 1387 | 1282  | 1945  |
| Safety class | A1   | A1   | A1    | A1    |

Figure 1. Schematic for the vapor-compression refrigeration cycle (modified from [29]).

For the performance assessment of the refrigeration cycle for different refrigerants, the governing mass balance (MB) and energy balance equations for a steady-state and steady flow system are given below [28,32]:

\[
\sum m_{in} = \sum m_{out} \quad (1)
\]

\[
\dot{Q} + \sum \dot{m}_{in}h_{in} = \dot{W} + \sum \dot{m}_{out}h_{out} \quad (2)
\]

where \( \dot{m} \) is the mass flow rate, \( \dot{Q} \) is the heat transfer, \( \dot{W} \) is the work and \( h \) is the specific enthalpy. In addition, the subscripts in and out represent inlet and outlet flows, respectively.

The general exergy balance equation for a control volume is defined as:

\[
\dot{E}_{XQ} - \dot{E}_{XW} = \sum \dot{m}_{out}e_{out} - \sum \dot{m}_{in}e_{in} + \dot{E}_{Xdest} \quad (3)
\]

Here, \( \dot{E}_{XQ} \) and \( \dot{E}_{XW} \) are the exergy of heat and work, respectively, \( e \) is the specific flow exergy and \( \dot{E}_{Xdest} \) is the exergy destruction rate. The exergy transfer associated with heat and work transfer, the flow exergy and exergy destruction rate can also be written as follows:

\[
\dot{E}_{XQ} = \dot{Q} \left( \frac{T - T_0}{T} \right) \quad (4)
\]

\[
\dot{E}_{XW} = \dot{W} \quad (5)
\]

\[
e = (h - h_0) - T_0 (s - s_0) \quad (6)
\]

\[
\dot{E}_{Xdest} = T_0 \dot{S}_{gen} \quad (7)
\]

In above equations, \( T \) is the temperature, \( s \) represents entropy, subscripts 0, dest and gen stand for reference state, destruction and generation.

After applying the principles of conservation of mass, first and second laws of thermodynamics, the models for mass balance (MB), energy balance (EnB), entropy balance (EtB), and exergy balance (ExB) equations for each system components can be described as follows:

**Evaporator:**

MB: \( \dot{m}_4 = \dot{m}_1 = \dot{m} \quad (8) \)

EnB: \( \dot{m}_4h_4 + \dot{Q}_E = \dot{m}_1h_1 \quad (9) \)

ExB: \( \dot{m}_4e_4 + \dot{Q}_{Eva} \left( 1 - T_0/T_E \right) = \dot{m}_1e_1 + \dot{E}_{Xdest,E} \quad (10) \)

EtB: \( \dot{m}_4s_4 + \dot{Q}_E/T_E + \dot{S}_{gen,E} = \dot{m}_1s_1 \quad (11) \)

3. THERMODYNAMICS ANALYSES

A comprehensive thermodynamic evolution is carried out to analyze traditional vapor compression refrigerant cycle performance for alternative refrigerants. The engineering equation solver (EES) software program [31] has been utilized for the analysis. During the calculations, the following assumptions are made:

- All the operations are in steady-state conditions.
- The changes in potential and kinetic energies are neglected.
- Pressure drops in all elements of the system are neglected.
- Compressor operation is assumed to be adiabatic.
- Heat losses to/from the cycle are neglected.
- The dead state temperature and pressure are taken as 101 kPa and 25°C.

For the performance assessment of the refrigeration cycle for different refrigerants, the governing mass balance (MB) and energy balance equations for a steady-state and steady flow system are given below [28,32]:
Figure 2. Pressure-enthalpy diagram for the refrigerants.

Figure 3. COP values for the cooling system according to using various working fluids.

Figure 4. Exergy destruction rates for each component of the cooling system according to various working fluids.

Compressor:

MB: \( \dot{m}_1 = \dot{m}_2 = \dot{m} \) \hspace{1cm} (12)

ExB: \( \dot{m}_1 \dot{e}_1 + \dot{W}_{\text{Comp}} = \dot{m}_2 \dot{e}_2 + \dot{E}_{\text{dest,Comp}} \) \hspace{1cm} (14)

EnB: \( \dot{m}_1 h_1 + \dot{W}_{\text{Comp}} = \dot{m}_2 h_2 \) \hspace{1cm} (13)

EtB: \( \dot{m}_1 s_1 + \dot{S}_{\text{gen,Comp}} = \dot{m}_2 s_2 \) \hspace{1cm} (15)
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Condenser:

\[ \dot{m}_2 = \dot{m}_3 = \dot{m} \]  
\[ \dot{h}_2 = \dot{h}_3 + \dot{Q}_C \]  
\[ \dot{m}_2 \dot{h}_2 = \dot{m}_3 \dot{h}_3 + \dot{Q}_C \left(1 - \frac{T_0}{T_C}\right) + \dot{E}_{dest,C} \]  
\[ \dot{m}_2 s_2 + \dot{S}_{gen,C} = \frac{\dot{Q}_C}{T_C} + \dot{m}_3 s_3 \]  

Expansion valve:

\[ \dot{m}_3 = \dot{m}_4 = \dot{m} \]  
\[ \dot{m}_3 h_3 = \dot{m}_4 h_4 \]  
\[ \dot{m}_3 e_3 = \dot{m}_4 e_4 + \dot{E}_{dest,ExpV} \]  
\[ \dot{m}_3 s_3 + \dot{S}_{gen,ExpV} = \dot{m}_4 s_4 \]  

The energetic and exergetic efficiency equations for the proposed cooling cycle are written as follows:

\[ \text{COP} = \frac{\dot{Q}_E}{W_{Comp}} \]  
\[ \psi = \frac{\dot{E}_{QE}}{W_{Comp}} \]  

4. RESULTS AND DISCUSSION

Based on the assumptions explained in the previous section, the energy and exergy performance of the refrigeration cycle was analyzed for R404A and its alternatives. The main input parameters of the cycle are given in Table 2. During the calculations, the operating conditions for R404A and other alternative refrigerants were supposed to be the same. According to the calculations using the data provided in Table 2, the P-h diagrams were plotted for all refrigerants (Figure 2).

In Figure 3, COP values of the refrigeration system for each working fluid are presented. As can be seen from the figure, the highest COP value was obtained for R404A fluid with a value of 2.477. Within the alternative refrigerants, the closest performance was obtained for R448A with a COP value of 2.467 and followed by R449A and R452A.

In Figure 4, the exergy destruction rates were given for each system component. According to the figure, the highest exergy destruction occurs in the condenser and compressor for all refrigerants. This is due to the pressure increase and friction losses in the compressor. In addition to that, as seen in the figure, the highest exergy destruction based on the compressor obtained for R404A working fluid. This is followed by three other fluids with a closer value. Exergy destruction in the condenser is caused by the heat transfer due to the finite temperature difference. Considering the overall system, the highest total exergy destruction is estimated for R452A refrigerant. According to the results, the total exergy destruction values of R404A, R448A, R449A and R452A working fluids were calculated as 3.201 kW, 3.217 kW, 3.298 kW and 3.488 kW, respectively. It is clear from the figure that the closest performance to R404A is estimated for R448A in terms of the second law.

Figure 5 shows the exergy efficiency of the refrigeration system for all refrigerants. As mentioned previously, the higher exergy destruction ratio results in lower exergy efficiency. According to the results, exergy efficiencies for R404A, R448A, R449A and R452A were found to be 20.71%, 20.62%, 20.22% and 19.33%, respectively. In previous studies on R452A refrigerant, a similar exergy efficiency was found, which was 30.9% [33]. As can be seen from the figure, the R404A fluid with the lowest exergy destruction reached the highest exergy efficiency. In the same manner, the refrigeration system working with R448A showed a similar performance to that of R404A. The lowest exergy efficiency was calculated for R449A and R452A.

Table 2. Input parameter values assumed in the simulation models.

| Parameters                      | Value |
|--------------------------------|-------|
| \(\dot{Q}_E \) (kW)            | 10    |
| \(T_C \) (°C)                  | 45    |
| \(T_E \) (°C)                  | -10   |
| Isentropic efficiency          | 0.75  |
| Superheating temperature (°C)  | 5     |
| Subcooling temperature (°C)    | 5     |
| \(T_0 \) (°C)                  | 25    |
| \(P_0 \) (kPa)                 | 101   |

Figure 5. Exergy efficiency values for the cooling system according to using various working fluids.
In Figure 6, the relative exergy destruction rates were given for each system component, for all refrigerants. As seen from the figure, the destruction rate distribution differs for the refrigerants for each system component. As an example, the highest destruction occurs in the condenser; however, for R404A, it is 30% of the total destruction while 36% for R448A, 34% for R449A and 30% for R452A. Another interesting result is that, as declared above, the best exergetic performance of the system was obtained using R404A; however, in terms of the compressor, the relative destruction was found to be the highest among the other components. Thus, when using R404A, extra attention must be paid to the compressor during the system design.

Figure 7 shows the exergy destruction of each system component for all refrigerants, with the variation of the evaporator temperature. As seen from the figure, with the increasing evaporator temperature, the exergy destruction was decreased in the compressor and expansion valve for R404A refrigerant fluid. The reason for the reduction of exergy destruction in the compressor is that the compressor load decreases due to the increase in the evaporator temperature. As the thermal load from the condenser decreases, the exergy destruction in the condenser decreases. However, as a result of the decrease in the evaporator load, the decrease in the cooling load, the exergy destruction in the evaporator decreases significantly. As seen from the graphs, with the variation of evaporator temperature, the refrigeration system shows similar performance for R448A and R449A. On the other hand, R452A shows similar characteristics with R404A working fluid.

The effect of the evaporator temperature change between $-10^\circ$C and $0^\circ$C of R448A refrigerant on the exergy destruction on system equipment is also shown in Figure 7. From the figure, it is calculated that the temperature change between $-10^\circ$C and $0^\circ$C decreases in exergy destruction by 0.53 kW—0.17 kW for the evaporator. The reason for this decrease in exergy destruction is due to the increasing evaporator temperature. In addition, it has been observed that there is a decrease in exergy destruction in other equipment that make up this temperature increase system. In addition, for R449A and R452A, similar results were obtained with the variation of evaporator temperature.

For all refrigerants, the effect of condenser temperature on COP and exergy efficiency are given in Figures 8 and 9. As can be seen from Figure 8, with the increasing condenser temperature from 25°C to 45°C, the COP of the refrigeration system decreases. In the same manner, the exergy efficiency also decreases with the increase of condenser temperature for all refrigerants (Figure 9).

In Figure 10, the effect of condenser temperature on exergy destruction is given for all refrigerants. Increasing the condenser temperature for all fluids increases the total exergy destruction and decrease the exergy efficiency. In Figure 10, the total exergy destructions of the system operating temperature values between 25°C and 45°C are shown for each fluid. As can be seen from the graph, the lowest total exergy destruction is seen in the R404A fluid. The fluid with the closest value to the R404A fluid was again R448A. Another interesting result that, increasing the temperature in the condenser from 25°C to 45°C, resulted in an increase of $\sim$100% in the total exergy destruction.
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Figure 7. Variation of exergy destruction rates for the individual system components with the evaporator temperature.

Figure 8. Variation of COP with condenser temperature.

The effect of increasing evaporator temperature on COP is given in Figure 11, comparatively. As can be seen from the figure, increasing the evaporator temperature from $-10^\circ C$ to $0^\circ C$ caused an increase in COP values by 30% for all refrigerants. This is due to the fact that, with the increase of the temperature, the corresponding saturation pressure also increases, resulting in a smaller pressure ratio. Thus, the power consumption of the compressor decreases with the pressure ratio. In Figure 12, the variation of evaporator temperature was plotted against exergy efficiency. As for the COP, again, with the increase of the temperature, exergy efficiency of the refrigeration cycle also increases for all working fluids.
The effect of evaporator temperature on the total exergy destruction is given in Figure 13. The figure is plotted by varying the evaporator temperature between $-10^\circ$C and 0$^\circ$C for all refrigerants. In contrary to exergy efficiency, the exergy destruction rate decreases with the temperature increase for all refrigerants. Again, R404A fluid appears to have the lowest total exergy destruction, followed by R448A, R449A, and R452A. According to the calculations, with the increase of evaporator temperature from $-10^\circ$C to 0$^\circ$C, the total exergy destruction of the cycle for the R404A decreases from 3.11 kW to 2.18 kW.

5. CONCLUSIONS

In this study, the energy and exergy analyses of a refrigeration cycle for R448A, R449A and R452A, which are alternatives of R404A, were presented comparatively. From the analysis, the highest COP value was obtained as 2.477 for the R448A within the alternative refrigerants. The lowest COP value was calculated for R452A as 2.313. In terms of exergy efficiencies, the highest efficiency was calculated for R448A fluid with 20.62%, while the lowest exergy efficiency was found to be 19.33% for R452A. According to the exergy destruction results for the individual system components, the highest exergy destruction occurred in the condenser with a rate of around 30%–36% for alternative refrigerants. The second highest destruction rate was calculated for compressor, followed by expansion valves and evaporator. Under the same operating conditions, the total exergy destruction rates of the refrigeration cycle were found to be 3.201 kW, 3.217 kW, 3.298 kW and 3.488 kW, for R404A, R448A, R449A and R452A, respectively. According to the results, R448A and R449A can be preferred both environmentally and exergetically as an alternative to R404A at ideal evaporator temperatures. In addition, R452A working fluid has shown that it can be environmentally beneficial, especially for cooling systems operating at relatively high evaporator temperatures. In light of all these data, it is concluded that R448A can be a good alternative for R404A due to its better performance in terms of energy and exergy efficiencies. Furthermore, the economic savings based on choosing cooling working fluid are also possible. In the continuation of this
study, it is planned to evaluate the system in terms of cost analysis and optimization. Therefore, it will be aimed to determine the ideal operating conditions of R448A with a thermoeconomic optimization study.

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