Performance studies of low GWP refrigerants as environmental alternatives for R134a in low-temperature applications

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Abstract
Climate change is an important environmental issue that is causing global temperatures to rise. The primary environmental targets are to reduce carbon emissions and mitigate the impacts of climate change. The refrigeration system is a major emitter of greenhouse gases because it uses refrigerants with a high global warming potential. Due to its excellent thermophysical properties, the R134a is the most commonly used refrigerant in refrigeration systems; however, its high GWP will need to be disposed of earlier. To achieve global environmental objectives, conventional refrigerants need to be replaced with environmentally friendly and energy-efficient refrigerants. In the present work, a mathematical simulation has been carried out to check the performance of low-GWP refrigerant mixtures as environmentally friendly alternatives for R134a in a low-temperature system. In this study, a 190-L domestic refrigerator has been considered a low-temperature system. This simulation was performed using the MATLAB software, and the REFPROP database was used to obtain thermophysical properties of the refrigerants. The results showed that the COP of HFO mixtures decreased by 4–20% compared to R134a. The exergy efficiency of the R1234ze/R134a mixture improves by 4 to 16% as compared to the other mixtures and its performance is very similar to the R134a. Due to the environmentally friendly properties and flammability aspects, R1234ze/R134a (90/10) could be a good substitute for R134a in lower temperature applications and to satisfy the Montreal and Kyoto Protocol expectations.

Keywords Climate change · Low-temperature system · Exergy · Eco-friendly mixtures · MATLAB

Introduction
Climate change is a significant environmental issue and it increases global temperature. In December 2015, the Paris Climate Conference set out an international action plan to bring down global warming below 2°C. Therefore, the main environmental targets are to minimize carbon emissions and mitigate the impact of climate change (Guo et al. 2018). The refrigeration system is one of the most important emitters of greenhouse gases because it uses high-GWP refrigerants as working fluids (Aprea et al. 2017). The emission of greenhouse gases by the refrigeration system in two ways such as leakage of high GWP refrigerants is done by directly during the maintenance and charging process and indirectly through the generation of energy used to supply the system (Tassou et al. 2011, Saji Raveendran and Joseph Sekhar 2019). Moreover, the direct release of refrigerants is calculated to be responsible for 2% of the overall emissions of the equivalent CO2, while indirect releases represent about ten times the total direct releases; the total contribution to global warming by the refrigeration sector is approximately 20% (Mota-Babiloni et al. 2020).

Due to its excellent thermophysical properties, R134a is the most commonly used refrigerant in refrigeration systems; however, its high GWP will have to be phased out earlier. The European Union recently announced that alternative refrigerants will be phased out in 2036.
refrigerants would have a GWP under 150 to reduce greenhouse gas emissions (Saji Raveendran and Joseph Sekhar 2016). To achieve global environmental objectives, conventional refrigerants need to be replaced with environmentally friendly and energy-efficient refrigerants. The experimental tests carried out on refrigeration systems to replace R134a with R1234yf, the refrigeration capacity, improved slightly, making R134a an appropriate substitute for household refrigerators (Aprea et al. 2016). The R1234yf, R1234ze (E), and R600a are low-GWP refrigerants that can be used to replace R134a. The R600a is one of the best alternatives for the R134a and it has improved the energy efficiency of a refrigerating system. But, it has a very low vapor pressure at high operating temperatures and it is not suitable for low-temperature freezer applications. Moreover, the huge difference in vapor pressure between R134a and R600a had a significant impact on compressor volumetric efficiency, requiring a change in capillary tube length. As a result, R600a cannot be used to replace R134a-based refrigerators (Mohanraj 2019). Among the refrigerants, R1234yf and R1234ze may be an alternate solution for R134a due to similar efficiency and the same mass flow rate in refrigeration systems (Righetti et al. 2015).

The experimental investigation is carried out on R1234yf and R1234ze in an R134a-based refrigeration system with various evaporation and condensation temperatures. It indicates that R1234ze has a lower average refrigeration capacity and COP than R134a by around 30% and between 2 and 8%, respectively. Similarly, R1234yf has a less cooling capacity of about 9% and between 5 and 30% less COP than R134a (Navarro-Esbri et al. 2013). Moreover, the flammability of these refrigerants is a big concern. Owing to its flammability, one of Europe’s major car manufacturers declined from using R1234yf. It has been observed that adding 10% R134a to HFO refrigerants will make the refrigerant mixture non-
flammable and it is a suitable alternative for R134a due to its energy efficiency. COP, discharge temperature, and refrigeration capacity are like those of R134a (Yohan et al. 2013; Aprea et al. 2016).

An azeotropic mixture R1234yf/R134a is offered to replace R134a in various low-temperature applications. The HFO/HFC mixtures are suitable alternatives of HFC refrigerants and they have a low GWP (GWP<150). The HFO/HFC mixtures have improved performance, and their expected COP and exergy efficiency are 4 to 8.3% and 5.1 to 10.5% respectively higher than the HFO. Among the HFO mixtures, the R1234ze/R134a has enhanced energy efficiency than the R1234yf/R134a mixture (Saji Raveendran and Joseph Sekhar 2021). Moreover, R1234ze has been determined to be less flammable than R1234yf by ASHRAE, and it is classed as A2L (Kondo et al. 2012).

In the present work, a mathematical simulation has been carried out to check the energy efficiency of low-GWP refrigerant mixtures as environmentally friendly alternatives for R134a in a low-temperature system. In this study, a 190-L domestic refrigerator has been considered a low-temperature system. The refrigeration system has been modelled and the performance has been studied to find the best composition of the mixture to operate the system. This simulation was conducted using the MATLAB software, and the REFPROP 9.0 database was used to obtain the thermal-physical properties of the refrigerants. Due to the environmentally friendly properties and flammability aspects, R1234ze/R134a (90/10) could
be a good substitute for R134a in the refrigerator to satisfy the Montreal and Kyoto Protocol expectations.

**Refrigerant selection**

The essential method of identifying the suitable refrigerant is the selection of refrigerants based on its thermo-physical properties. The properties of the refrigerants obtained from REFPROP 9.0 have been plotted for the operating temperature −30 to 50°C. The vapor pressures of HFO refrigerant mixtures are lower than those of R134a as shown in Fig. 1. The vapor pressures of all the refrigerants are closer to those of R134a at low operating temperatures. The vapor pressure of R1234ze/R134a is 21.5% lower than that of R134a among the different mixtures. This can cause a reduced energy consumption. Figure 2 shows that the latent heat of R1234ze/R134a/R744 and R1234ze/R134a are 6.4% and 6.8% lower than those of R134a respectively. In the case of the R1234ze/R32/R152a mixture, the latent heat is 1.3% higher than that of R134a. The high latent heat ensures a faster rate of cooling in evaporators. The faster cooling rate will reduce the compressor running time and the energy consumption of the refrigerator.

The comparison of liquid densities of R134a and its alternatives is shown in Fig. 3. When compared to that of R134a, the liquid densities of R1234ze/R32/R152a, R1234ze/R134a/R744, and R1234ze/R134a are found to be 8.7%, 7%, and 3.8% lower, respectively. Hence, the refrigerant mass charge requirements of these refrigerants are significantly lower when compared to those of R134a. Moreover, the low liquid density ensures significant reduction in irreversibility in the condenser and capillary tube of the refrigeration system (Mohanraj 2019, Mohanraj et al. 2009).

The liquid viscosity of R134a and its alternatives is shown in Fig. 4. It is observed that the variation of liquid viscosity of R1234ze/R134a was found to be similar to that of R134a, whereas the liquid viscosity of other refrigerants such as R1234ze/R32/R152a and R1234ze/R134a/R744 was found to be lower by about 15.2% and 13.3%, respectively, when compared to R134a. The lower liquid viscosity will reduce the friction resistance in the tubes of the condenser, which affects the power consumption of the compressor and heat transfer coefficient, and it will reduce the irreversibility of all the system components. The thermodynamic properties used in this analysis are shown in Table 1.

The critical mass flux values for R134a and proposed refrigerants differ slightly, so the diameter of the capillary tube required with that of the proposed refrigerants is larger than that required with R134a, and the capillary tube length required with proposed refrigerants is shorter than that required with R134a. Moreover, due to the low liquid viscosity of the suggested refrigerants, the mass charge needs are substantially lower than those of R134a. This characteristic ensures that all system components have improved heat transfer coefficients and reduced irreversibility (Mohanraj and Andrew Pon Abraham 2020, Harish Kruthiventi and Venkatarathnam 2016).

**Analytical study**

Figure 5 shows a schematic diagram of the refrigeration system used throughout this analysis. It consists of a reciprocating compressor, an air-cooled condenser, a capillary tube, and an evaporator. The simulation of the above components was performed using the MATLAB software. To obtain the thermodynamic properties of refrigerants, REFPROP (Lemmon et al. 2007) is directly interlinked with MATLAB. A 190-L domestic refrigerator has been considered in this modeling. The dimensions of all the components are taken from the literature (Saji Raveendran and Joseph Sekhar 2021). The property variations are one dimensional, steady-state condition, constant mass flow rate, a polytropic process in
compression, the efficiency of the electrical motor is 85% and pressure drop in the heat exchanger is negligible. The above assumptions are taken into account in this analysis to minimize the complexities of the simulation.

**Compressor model**

A 100-W reciprocating type compressor has been taken for the study and it is divided into three control volumes such as compressor shell, swept volume, and discharge tube (William and Doyle Thompson 1988). The following equation is used to determine the swept volume:

\[
\dot{m}_o h_o = Q_{\text{comp}} + \frac{dW}{dt} + m_i h_i - (\Delta P)_V (1)
\]

The displacement volume is calculated at each time step using the following equation:

\[
v(t) = V_{\text{cylinder}} + \frac{\pi D^2}{8} + L_{\text{swept}} (1 - \cos(\omega t)) (2)
\]

The following equation is used to calculate the compressor work:

\[
W_{\text{comp}} = \frac{n}{n-1} P_s V \left\{ \frac{P_{\text{discharge}}}{P_{\text{suction}}} \right\}^{\frac{\gamma - 1}{\gamma}} (3)
\]

**Condenser model**

The condenser is divided into the de-superheated region, two-phase region, and sub-cooled region. The following

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**Fig. 6** T-S diagram

**Fig. 7** Flowchart for simulation model

**Table 2** Irreversibility equation of each component

| Components     | Irreversibility                                               |
|----------------|----------------------------------------------------------------|
| Compressor     | \( I_{\text{comp}} = \dot{m}_1 \psi_1 + W - \dot{m}_2 \psi_2 \) |
| Condenser      | \( I_{\text{cond}} = \dot{m}_2 \psi_2 - \dot{m}_3 \psi_3 - Q_{\text{cond}} \left( 1 - \frac{T_{\text{suction}}}{T_{\text{cond}}} \right) \) |
| Capillary tube | \( I_{\text{cap}} = \dot{m}_3 \psi_3 - \dot{m}_4 \psi_4 \) |
| Evaporator     | \( I_{\text{evap}} = \dot{m}_4 \psi_4 + Q_{\text{evap}} \left( 1 - \frac{T_{\text{suction}}}{T_{\text{evap}}} \right) - \dot{m}_1 \psi_1 \) |
correlations are used to measure the refrigerant side heat transfer coefficient in the single-phase region (Minor and Montoya 2010):

In the laminar region \((Re < 2100)\)

\[
Nu = 1.86Re^{0.33}Pr^{0.33} \left( \frac{D_l}{L} \right) \left( \frac{\mu_b}{\mu_w} \right)^{0.5}
\]  
(4)

In the turbulent region \((Re > 10,000)\)

\[
Nu = 0.023Re^{0.8}Pr^{0.33} \left( \frac{D_l}{L} \right) \left( \frac{\mu_b}{\mu_w} \right)^{0.14}
\]  
(5)

The air-side heat transfer coefficient of the condenser is calculated using the correlations below (Marques et al. 2014):

\[
Nu = 0.3 + \left( \frac{0.62Ra^{0.167}}{1 + \left( \frac{0.559}{Pr} \right)^{1.97}} \right)
\]  
(6)

**Evaporator model**

The evaporator is divided into the two-phase region and superheated region. The convective heat transfer coefficients of the single-phase region and two-phase region are computed with Equations (7), (8), and (9) (Wattelet et al. 1994; Chang et al. 2000; Cooper 1984; Navarro-Esbri et al. 2013 & Nielsen et al. 2007):

In the laminar region \((Re < 2100)\)

\[
Nu = 1.86Re^{0.33}Pr^{0.13} \left( \frac{D_l}{L} \right) \left( \frac{\mu_b}{\mu_w} \right)
\]  
(7)

In the turbulent region \((Re > 10,000)\)

\[
Nu = 0.023Re^{0.8}Pr^{0.14} \left( \frac{D_l}{L} \right) \left( \frac{\mu_b}{\mu_w} \right)
\]  
(8)

\[
h_{evap} = 55Mo^{-0.5}q^{0.67}Pr^{0.12} \left( \log Pr \right)^{0.55} + Fh_lR
\]  
(9)

**Capillary tube model**

The flow of a capillary tube can be separated into single-phase and two-phase. The capillary suction heat exchanger provision is given for \(0.3\text{–}9\text{m}\). Pressure drops in the capillary tube due to friction, momentum loss, and gravitational effects. Sudden contraction that happens by the pressure drop is calculated by following equation (Perry and Chilton 1984):

\[
P_l - P_{in} = \frac{G^2V_f}{2} \left\{ \left( \frac{1}{C_{con}} - 1 \right) + \left( 1 + \frac{A^2}{A_1^2} \right) \left( 1 + \frac{V_g}{V_f} \right) \right\}
\]  
(10)

Coefficient of contraction is calculated by the following equation:

\[
C_c = 0.55 \left( \frac{A_{in}}{A_1} \right)^3 - 0.242 \left( \frac{A_{in}}{A_1} \right)^2 + 0.111 \left( \frac{A_{in}}{A_1} \right) + 0.585
\]  
(11)

\[
COP = \frac{Q_{evap}}{W_{Comp}}
\]  
(12)
Exergy analysis

The aim of the exergy analysis is to assess the irreversibility of each component of a domestic refrigerator. Figure 6 depicts the T-S diagram and the exergy balance equations are taken from the previous studies (Bayrakci and Ozgur 2009; Saji Raveendran and Joseph Sekhar 2017a, b) and shown in Table 2.

The total exergy destruction of the system is determined by (Figure 7):

\[ I_{\text{total}} = I_{\text{comp}} + I_{\text{cond}} + I_{\text{cap}} + I_{\text{evap}} \]  

(13)

Validation

To validate the simulation model, experimental COP values have been compared with the predicted simulation results. The comparisons between the simulation and previously published experimental results with R134a for various ambient conditions are shown in Fig. 8. It shows that the difference between simulated and experimental values varies from 9 to 12%. This variation is nominal, and hence the validity of the current approach has been proven. The possible reason for this variation is the impact of various assumptions used to reduce the complexity of the mathematical modelling of the system. Since the results are close to the experimental values, this model can be used to predict the performance of the system in different operating conditions (Table 3).

Results and discussion

The energy analysis on a domestic refrigeration system through environmentally friendly refrigerant mixtures was
theoretically studied for different ambient temperatures at an evaporator temperature of −18 °C. The findings would examine the output parameters including the compressor work, COP, total exergy destruction, exergy efficiency, and TEWI.

As seen in Fig. 9, the function of the compressor work increases as the atmospheric temperature increases. This is because the input pressure of the compressor and mass flow rate increase. The R1234ze/R134a mixture has been observed to reduce compressor work from 4.3 to 8.5% than other two mixtures because of low evaporator pressure and high molecular weight of the mixture. The difference of the COP with ambient temperature was calculated and plotted using HFO mixtures in Fig. 10. It has been noted that the COP of the R1234ze/R134a mixture is higher than that of the other two mixtures by about 16.4%. This may be because the refrigerated mixture has a high latent heat.

Figure 11 indicates as the function of the total exergy destruction increases, the atmospheric temperature increases. The previous studies have found a similar pattern (Siva Reddy et al. 2012). The overall irreversibility of the R1234ze/R134a mixture is decreased from 5.2 to 12.4% than the other two mixtures that can be attributed to the high compression ratio. Since the exergy efficiency is used to determine the quality of energy used by a system, it has been calculated for HFO refrigerant mixtures and is shown in Fig. 12. The exergy efficiency of the R1234ze/R134a mixture improves from 4 to 16% as compared to the other two mixtures, and it also has a lower exergy efficiency at high ambient temperatures. Among all the HFO mixtures, the R1234ze/R134a mixture shows better performance and it is very similar to the R134a.

**Total equivalent warming impact**

A TEWI index has been defined which allows a comparison of refrigeration systems in relation to their impact on the
Table 4 TEWI for both the refrigerants

| Refrigerant          | R134a | R1234ze/R134a |
|----------------------|-------|---------------|
| Charge quantity (kg), \(m\) | 0.135 | 0.129         |
| Leakage rate per year (%), \(l\) | 6.6   | 6.6           |
| Service life (year), \(S\) | 15    | 15            |
| GWP                  | 1430  | 148           |
| Direct effect        |       |               |
| Power consumption per year (kWh), \(E\) | 701   | 756           |
| \(CO_2\) emission (kg\(CO_2\)/kWh), \(r\) | 0.89  | 0.89          |
| Indirect effect      |       |               |
| \(9358\)            | 10,093|               |
| TEWI                 | 9549  | 10,112        |

The environmental advantages of the proposed mixture have a lower GWP, reflecting their impact on global warming. The charge consumption of the proposed mixture is 4.4% lower than that of R134a in a refrigeration system because of its low liquid density, which leads to lower energy consumption. Every year, 5 million refrigerators are manufactured in India. R134a is charged for approximately 80% of these refrigerators. A significant quantity of \(CO_2\) emissions will be eliminated if refrigeration systems are charged with the R1234ze/R134a refrigerant.

Conclusion

The low-GWP refrigerants like R1234ze/R134a, R1234ze/R32/R152a, and R1234ze/R134a/R744 were theoretically studied in a low-temperature system. In this study, a 190-L domestic refrigeration system has been considered a low-temperature system. Among the mixtures, R1234ze/R134a (90/10) gives better results, with estimated COP and exergetic efficiency of 3.7 to 16.4% and 4 to 16% respectively higher than those of the other mixtures. The R1234ze/R134a has a higher TEWI than R134a by about 5.9%. This is because of less energy efficiency, but direct emissions are significantly lower than those for R134a. Even though the performance of the mixture R1234ze/R134a is slightly lower than that of R134a, it is superior to other HFO mixtures. It may also be an appropriate alternative for R134a in low-temperature applications to address environmental concerns in accordance with the Montreal and Kyoto Protocols.

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Author contribution Saji Raveendran Padmavathy handled conceptualization, methodology, and data collection. Murugan Paradesi Chockalingam was responsible for writing, reviewing, and final editing of the manuscript. Godwin Glivin was responsible for data analysis and preliminary paper writing, while Nithyanandan Kamaraj was in control of result interpretation. Venkatesh Thangaraj aided in the analysis and editing of the manuscript. The methodology and data gathering were the responsibility of Bharathiraja Moorthy. The final manuscript was read and approved by all of the authors.

Data availability The datasets generated during this work are not publicly available; however, they are available upon reasonable request to the corresponding author.

Declarations

Ethics approval and consent to participate Not applicable

Consent for publication Not applicable

Competing interests The authors declare no competing interests.

Abbreviations

- \(A\), area (m\(^2\)); \(C\), coefficient; \(COP\), coefficient of performance; \(D\), coil mean diameter (m); \(E\), Empirical constant; \(G\), mass flux (kg m\(^{-2}\) s\(^{-1}\)); \(GWP\), global warming potential; \(h\), specific enthalpy (kJ kg\(^{-1}\)); \(L\), length (m); \(m\), mass flow rate (kg s\(^{-1}\)); \(Mo\), molecular weight (g kmol\(^{-1}\)); \(n\), polytropic index; \(Nu\), Nusselt number; \(P\), pressure (kPa); \(Pr\), Prandtl number; \(Q\), heat (J); \(Ra\), Rayleigh number; \(Re\), Reynolds number; \(t\), time (s); \(T\), temperature; \(TEWI\), total equivalent warming impact; \(V\), volume (m\(^3\)); \(W\), work (W); \(X\), refrigerant quality; \(\omega\), angular velocity (rad s\(^{-1}\)); \(\mu\), dynamic viscosity (Pa s); \(\psi\), exergy (kJ/kg)
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