Performance Evaluation of Centrifugal Refrigeration Compressor Using R1234yf and R1234ze(E) as Drop-In Replacements for R134a Refrigerant

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Abstract: With the increasing global requirements for environmental protection, refrigerants with ODP of 0 and low GWP are widely concerned and applied. In this paper, the CFD numerical method simulates the R134a centrifugal compressor directly replaced by R1234yf and R1234ze(E). The results show that at the same compressor rotational speed, using R1234yf to replace R134a directly can obtain a higher cooling capacity, but it reduces COP by about 12.5%; using R1234ze(E) to replace R134a directly reduces the cooling capacity under partial working conditions, the COP is reduced by about 7.0%. When the evaporation temperature, condensation temperature, and cooling capacity are the same, compared with the R134a unit, the COP of the R1234ze(E) unit is reduced by about 5.14%, and it is reduced by about 8.93% for the R1234yf unit. For the R134a centrifugal chiller, the drop-in replacement of R134a with R1234ze(E) can obtain better system performance compared with R1234yf.

Keywords: centrifugal compressor; chiller; refrigerant; R1234yf; R1234ze(E); R134a

1. Introduction

R134a is an HFC (hydrofluorocarbon) refrigerant that is widely used in automotive air conditioning and water chillers with centrifugal compressors. However, its value of global warming potential (GWP) is 1300, which contributes to global warming [1,2]. With increasing global environmental requirements, the usage amount of R134a has been requested to reduce [3,4]. One kind of replacement is HFO (Hydrofluoroolefin) refrigerants. This kind of refrigerant has attracted extensive attention and is applied in some applications due to its 0 ozone depletion potential (ODP) and low GWP. Additionally, the representatives of HFOs include R1234yf, R1234ze(E), and R1233zd(E) [5,6]. R1234yf is mainly used to replace R134a in automobile air conditioners, while R1234ze(E) or R1233zd(E) is commonly used to replace R134a in centrifugal chillers [7,8]. At present, some companies are developing centrifugal chillers using R1234ze(E) and R1233zd(E) as refrigerants, such as Danfoss, Klima-Therm Company, Carrier, Mitsubishi Heavy Industries, Trane, and Gree [9,10].

Many scholars studied the replacement of R134a refrigerant. Aral et al. [11] developed an automotive air conditioning (AAC) and automotive heat pump (AHP) system employing refrigerants R134a and R1234yf. They tested them in cooling and heating operation modes under broad ranges of compressor speed and air inlet temperature conditions. Additionally, the results revealed that R1234yf performs better in heating relative to cooling mode. It can be used as a replacement for R134a in not only AAC but also AHP systems at the expense of slightly lower energy effectiveness. Qi [12] performed thermodynamic analysis for the R1234yf mobile air-condition (MAC) system associated with the R134a system under three typical vehicle operating conditions. The performance improvement potentials
by superheating, subcooling, and compressor performance were mainly focused on and
discussed. It was concluded that adding an internal heat exchanger and improving com-
pressor efficiencies would be good options for future R1234yf MAC system enhancement.
Mota-Babiloni et al. [13] presented an energy performance evaluation of two low-GWP
refrigerants, R1234yf and R1234ze(E), as drop-in replacements for R134a in a vapor com-
pression system using a reciprocating compressor. Additionally, the volumetric efficiency,
cooling capacity, and coefficient of performance (COP) were analyzed. Results showed that
the COP values were about 7% lower for R1234yf and 6% lower for R1234ze than those
obtained using R134a. Additionally, the use of an internal heat exchanger reduced the
COP differences for both replacements. Chen et al. [14] investigated the performance of
R1234yf and R134a in an oil-free vapor compression refrigeration (VCR) system. The results
showed that R1234yf is similar to R134a in terms of operating pressure and temperature,
but R1234yf has an 11% and 16% deterioration in cooling capacity and COP, respectively.

Many scholars compared the boiling and condensation heat transfer coefficients of
R1234yf, R1234ze(E), and R1233zd(E) with those of R134a. Most scholars showed that the
boiling and condensation heat transfer coefficients of R1234yf, R1234ze(E), and R1233zd(E)
are lower than those of R134a. For example, Yang et al. [15,16] provided an experimental
analysis of flow boiling and condensation heat transfer of R1234yf and R134a in a small
circular tube. The test results showed that the boiling and condensation heat transfer
coefficients of R1234yf are lower than those of R134a. Nagata et al. [17] comparatively
assessed the free convective condensation and pool boiling heat transfer coefficients (HTCs)
of 1234ze(E) on a smooth horizontal tube made of copper with an outer diameter of
19.12 mm. Results showed that the condensation HTC and pool boiling HTC of R1234ze(E)
are slightly lower than that of R134a. Ubara et al. [18] experimentally evaluated the falling
film evaporation and pool boiling heat transfer coefficients of R1233zd(E) on a single
horizontal tube. The experimental results for R1233zd(E) were compared with those for
R134a. The results showed that the pool boiling and falling film evaporation heat transfer
coefficients were lower for R1233zd(E) than for R134a. Therefore, in the water chiller, it
is necessary to consider increasing the heat transfer area of the evaporator and condenser
when R1234yf, R1234ze(E) and R1233zd(E) are used to replace R134 as the refrigerant.

In spite of a number of studies mentioned above, research on the drop-in refrigerant
replacement of R134a centrifugal chiller is rare at present. HFOs are considered new
generation refrigerants with the most potential to replace R134a, with broad application
prospects. At present, there are still R134a centrifugal chillers in use in the market. With
the gradual prohibition of R134a, it is of great significance to study the refrigerant direct
replacement technology for the existing R134a centrifugal chillers. Moreover, due to the
difference in working principles from the positive displacement (screw, scroll, reciprocating,
etc.) refrigeration compressor, the centrifugal refrigeration compressor is more sensitive to
the physical properties of the refrigerant.

Among HFO refrigerants, R1234yf, R1234ze(E), and R1233zd(E) are the suitable refrig-
erants that are researched and applied in centrifugal water chillers. However, R1233zd(E)
cannot be used as a drop-in replacement refrigerant for R134a through the analysis in the
next section. Therefore, this paper took the centrifugal refrigeration compressor as the
research object and used the CFD numerical method to simulate and analyze the compres-
sor using R134a, R1234yf, and R1234ze(E), respectively. Parameters such as the pressure
ratio, power, and isentropic efficiency of the centrifugal compressor were calculated. The
R134a centrifugal chiller was selected, the refrigerant was directly replaced by R1234yf and
R1234ze(E), and the refrigeration performance (including cooling capacity and COP) was
compared and analyzed.

2. Comparative Analysis of Refrigerant Physical Properties

In consideration of the thermophysical properties of refrigerants, R1234yf, R1234ze(E),
and R1233zd(E) can be considered the drop-in replacement refrigerants of R134a [19,20].
Table 1 compare the main physical property parameters of R134a with the refrigerants
R1234yf, R1234ze(E), and R1233zd(E) [21–23]. It can be seen that R1234yf, R1234ze(E), and R1233zd(E) have extremely low GWP compared with R134a.

Table 1. Main physical property parameters of refrigerants.

<table>
<thead>
<tr>
<th></th>
<th>R134a</th>
<th>R1234yf</th>
<th>R1234ze(E)</th>
<th>R1233zd(E)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molecular formula</td>
<td>CH2FCF3</td>
<td>C3H2F4</td>
<td>C3H2F4</td>
<td>C3H2CF3</td>
</tr>
<tr>
<td>Molecular weight</td>
<td>102.0</td>
<td>114.04</td>
<td>114.04</td>
<td>130.5</td>
</tr>
<tr>
<td>Critical pressure (MPa)</td>
<td>4.059</td>
<td>3.382</td>
<td>3.636</td>
<td>3.624</td>
</tr>
<tr>
<td>Critical temperature (°C)</td>
<td>101.6</td>
<td>94.7</td>
<td>109.4</td>
<td>166.5</td>
</tr>
<tr>
<td>Normal boiling point (°C)</td>
<td>-26.07</td>
<td>-29.45</td>
<td>-18.97</td>
<td>18.3</td>
</tr>
<tr>
<td>GWP</td>
<td>1430</td>
<td>4</td>
<td>6</td>
<td>1</td>
</tr>
<tr>
<td>ODP</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Security level</td>
<td>A1</td>
<td>A2L</td>
<td>A1</td>
<td>A1</td>
</tr>
</tbody>
</table>

The saturation lines of four refrigerants on the pressure-specific enthalpy diagram are shown as thin lines in Figure 1a. It can be seen that the saturation lines of R1234ze(E) and R1234yf are similar to that of R134a, and the saturation line of R1234ze(E) has the smallest deviation from that of R134a. The saturation line of R1233zd(E) is quite different from that of R134a. Under the same pressure, it has a relatively higher specific enthalpy value than R134a.

![Figure 1a](image1a.png)  
(a)  

![Figure 1b](image1b.png)  
(b)  

Figure 1. Thermodynamic figure comparison of 4 refrigerants: (a) Pressure-enthalpy diagram, (b) Pressure-temperature diagram.

The working conditions are set as evaporation temperature 5.5 °C, condensation temperature 37 °C, superheating temperature 3 °C, and subcooling temperature 4 °C. The theoretical cycles of the four refrigerants are represented on the pressure-enthalpy diagram (shown as the thick lines in Figure 1a). Pressure is the operating pressure of the system and will affect the working process of the compressor. Specific enthalpy directly affects system performance. It can be seen that the pressure of R1234yf is the closest to that of R134a, but the specific enthalpy value of R1234yf in the gas phase region is smaller than that of R134a; the specific enthalpy value of R1234ze(E) is the closest to that of R134a, but its pressure is lower than that of R134a; the difference between R1233zd(E) and R134a is large, the pressure of R1233zd(E) is lower than that of R134a.

Figure 1b show the saturation pressure of four refrigerants at different temperatures. The curve of R1234yf and R134a is the closest, followed by R1234ze(E). At the same temperature, the saturation pressure of R1233zd(E) is lower than that of R134a. Additionally, with the increase in temperature, the difference between them increases gradually.
Due to the low pressure of R1233zd(E), the volume flow rate under the same cooling capacity is quite different from that of other refrigerants, leading to a significant reduction in the rotational speed of the compressor impeller and a large difference in impeller design size [24]. Because the physical properties of R1233zd(E) and R134a are quite different, it is difficult to directly replace R1233zd(E) in the original centrifugal chillers using R134a. Therefore, this paper does not consider the drop-in replacement of R1233zd(E) in the R134a centrifugal chiller. The working fluids of R1234yf and R1234ze(E) have similar physical properties to R134a, and they can potentially be used as drop-in refrigerants.

For centrifugal chillers, if we want to replace the refrigerant without changing the compressor, the cooling capacity per unit volume of the replaced refrigerant should be similar to that of the original refrigerant. Under the mentioned working conditions above, the cooling capacity per unit volume is calculated using the following Equation (1):

\[ q_v = q_0 \times \rho_1 \]  

(1)

where \( q_0 \) (kJ·kg\(^{-1}\)) is the unit cooling capacity; \( \rho_1 \) (kg·m\(^{-3}\)) is the density of the compressor inlet.

The cooling capacity per unit volume of the four refrigerant units is shown in Table 2. It can be seen that the cooling capacity per unit volume of R1234yf and R1234ze(E) is not much different from that of the R134a unit, the cooling capacity per unit volume of R1234yf is 0.95 times that of R134a, and the cooling capacity per unit volume of R1234ze(E) is 0.75 times that of R134a. At the same time, the cooling capacity per unit volume of R1233zd(E) is quite different from that of R134a, which is 0.22 times that of R134a. Therefore, it may be feasible to replace R134a with R1234yf or R1234ze(E) directly.

**Table 2. Cooling capacity per unit volume of 4 refrigerant units.**

<table>
<thead>
<tr>
<th></th>
<th>R134a</th>
<th>R1234yf</th>
<th>R1234ze(E)</th>
<th>R1233zd(E)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporation temperature (°C)</td>
<td>5.5</td>
<td>5.5</td>
<td>5.5</td>
<td>5.5</td>
</tr>
<tr>
<td>Condensation temperature (°C)</td>
<td>37</td>
<td>37</td>
<td>37</td>
<td>37</td>
</tr>
<tr>
<td>Superheating temperature (°C)</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Subcooling temperature (°C)</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Cooling capacity (kJ·m(^{-3}))</td>
<td>2717.4</td>
<td>2590.8</td>
<td>2033.0</td>
<td>593.1</td>
</tr>
</tbody>
</table>

**3. Refrigeration Cycle System**

The working circuit of the two-stage compression refrigeration cycle system used in this paper is shown in Figure 2a: the low-pressure refrigerant gas from the evaporator is sucked in by the first stage impeller of the compressor, compressed to the intermediate pressure, and mixed with the refrigerant vapor from the intercooler. Additionally, the gas is sucked in by the second stage impeller of the compressor, compressed to the condensation pressure, enters the condenser to be condensed into the refrigerant liquid, and then enters the liquid reservoir. The liquid from the liquid reservoir is divided into two routes: A portion of liquid enters the intercooler to reduce the temperature and becomes a subcooled liquid. After the pressure is reduced by the throttle valve, it is evaporated in the evaporator. Another portion of liquid flows into the intercooler for evaporation after depressurization by the throttle valve. The pressure-specific enthalpy diagram corresponding to this circuit is shown in Figure 2b.
3. Refrigeration Cycle System

The working circuit of the two-stage compression refrigeration cycle system used in this paper are set, and they are shown in Table 3.

Table 3. Working conditions of centrifugal chiller system.

<table>
<thead>
<tr>
<th></th>
<th>R134a</th>
<th>R1234yf</th>
<th>R1234ze(E)</th>
<th>R1233zd(E)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporation temperature (°C)</td>
<td></td>
<td>5.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporation pressure (kPa)</td>
<td>355.78</td>
<td>379.04</td>
<td>263.96</td>
<td>60.707</td>
</tr>
<tr>
<td>Condensation pressure (kPa)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Superheating temperature (°C)</td>
<td></td>
<td></td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>Subcooling temperature (°C)</td>
<td></td>
<td></td>
<td>4</td>
<td></td>
</tr>
</tbody>
</table>

4. Numerical Calculation Methods

4.1. Basic Governing Equations

To simulate the internal flow of centrifugal compressors, the CFD method is used in this paper. Additionally, the details of the governing equations which are used during the simulation process are as follows:

1. Mass conservation equation

   The mass conservation equation can be expressed as:

   \[ \frac{\partial \rho}{\partial t} + \text{div}(\rho u) = 0 \]  \hspace{1cm} (2)

   where \( t \) is the time, \( \rho \) is the density, and \( u \) is the velocity vector.

2. Momentum conservation equation

   The momentum conservation equation can be expressed as:

   \[
   \begin{align*}
   &\frac{\partial (\rho u_x)}{\partial x} + \text{div}(\rho u_x u) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + F_x \\
   &\frac{\partial (\rho u_y)}{\partial y} + \text{div}(\rho u_y u) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{yx}}{\partial x} + \frac{\partial \tau_{zy}}{\partial z} + F_y \\
   &\frac{\partial (\rho u_z)}{\partial z} + \text{div}(\rho u_z u) = -\frac{\partial p}{\partial z} + \frac{\partial \tau_{zz}}{\partial z} + \frac{\partial \tau_{zx}}{\partial x} + \frac{\partial \tau_{zy}}{\partial y} + F_z
   \end{align*}
   \]  \hspace{1cm} (3)

   where \( u_x, u_y, u_z \) is the component of the velocity vector in the \( x, y, z \) direction, \( p \) is the pressure on the micro-element body, \( \tau_{ij} \) (\( i = x, y, z, j = x, y, z \)) is the deviatoric stress component, \( F_x, F_y, F_z \) is the force on the micro-element body.
(3) Energy conservation equation

The energy conservation equation can be expressed as:

\[
\frac{\partial}{\partial t}\left[\rho (e + \frac{V^2}{2})\right] + \text{div}\left[\rho (e + \frac{V^2}{2})V\right] = \rho q + \text{div}(k\nabla T) - \text{div}(pV) + \text{div}(\tau \times V) + F \times V \tag{4}
\]

where \( e \) is the internal energy per unit mass of fluid, \( V^2/2 \) is the kinetic energy per unit mass of fluid, \( q \) is the heat source per unit volume, \( \text{div}(k\nabla T) \) is the imported heat, \( -\text{div}(pV) \) + \( \text{div}(\tau \times V) \) is the work conducted by surface force, \( F \times V \) is the work conducted by mass force.

4.2. Geometrical Model

The fluid domain of the centrifugal compressor is extracted as the physical model, which mainly includes the impeller, diffuser, and volute (as shown in Figure 3). The main design parameters of impellers are shown in Table 4.

![Figure 3. Model of impeller, diffuser, and volute.](image)

Table 4. Main design parameters of centrifugal compressor.

<table>
<thead>
<tr>
<th>Design Parameters</th>
<th>First Stage Impeller</th>
<th>Second Stage Impeller</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impeller hub diameter (mm)</td>
<td>25.0</td>
<td>37.0</td>
</tr>
<tr>
<td>Impeller inlet diameter (mm)</td>
<td>66.7</td>
<td>64.5</td>
</tr>
<tr>
<td>Impeller outlet diameter (mm)</td>
<td>127.2</td>
<td>114.3</td>
</tr>
<tr>
<td>Impeller outlet width (mm)</td>
<td>5.7</td>
<td>4.9</td>
</tr>
<tr>
<td>Number of blades</td>
<td>16</td>
<td>16</td>
</tr>
<tr>
<td>Diffuser inlet radius (mm)</td>
<td>66.0</td>
<td>59.5</td>
</tr>
<tr>
<td>Diffuser outlet radius (mm)</td>
<td>94.0</td>
<td>85.4</td>
</tr>
<tr>
<td>Diffuser width (mm)</td>
<td>6.7</td>
<td>5.8</td>
</tr>
<tr>
<td>Radius at 0° of volute profile (mm)</td>
<td>94.0</td>
<td>85.4</td>
</tr>
<tr>
<td>Radius at 360° of volute profile (mm)</td>
<td>138.7</td>
<td>131.4</td>
</tr>
</tbody>
</table>

4.3. Meshing

The hexahedral structured mesh was divided for the single channel of impeller and diffuser, and the unstructured tetrahedral mesh was divided for the volute. The mesh independence verification of the first stage compressor is shown in Figure 4. The outlet pressure when the grid number is 988,413 is only 0.04kPa lower than that when the grid number is 946,364, so the total number of meshes for the first-stage impeller, diffuser, and volute was 946,364. The same method is adopted for the mesh independence verification of the second stage compressor. The total number of meshes for the second-stage impeller, diffuser, and volute was 872,912. The value of \( y^+ \) is 25. There are eight boundary layers; the thickness of the first boundary layer is 0.036 mm. The mesh is shown in Figure 5.
The efficiency is an important parameter of centrifugal compressors, especially in refrigeration applications. The total inlet pressure of the three fluids is set to

\[ P_1 \]

where

\[ h_1 \]

is the specific enthalpy value of the working fluid at the compressor inlet; and

\[ h_2s \]

is the specific enthalpy value of the working fluid at the outlet of the isentropic compression process.

The impeller flow channel is set as the rotating calculation domain, and the diffuser and volute flow channel are set as the static calculation domain. Both the impeller and the diffuser use the periodic interface. Additionally, the interface between the impeller and diffuser and the interface between the diffuser and volute both use the frozen rotor model. The surface of impellers, diffuser, and volute are all set as non-slip, smooth, and adiabatic wall surfaces. The maximum number of iteration steps is 500, an automatic timescale is used, and the timescale factor is 10. Convergence is considered when the residuals converge.

Steady-state simulation is used. The heat transfer model is set to the Total Energy. The turbulence model uses the Shear Stress Transport (SST) model [26]. Due to the analysis found that R1233zd(E) cannot be used as a drop-in replacement refrigerant for R134a, the fluids are set to R134a, R1234yf, and R1234ze(E), respectively. Among them, R1234ze(E)
uses the NIST physical property software [27]. The refrigerant physical property data is converted into a physical property table (RGP file), which is imported into the solver for the solution. The physical property table takes pressure and temperature as independent variables. The boundary conditions of the total temperature and total pressure inlet and flow rate outlet are chosen. The total inlet pressure of the three fluids is set to 355.78 kPa, 379.04 kPa, and 263.96 kPa (their saturation pressures at the evaporation temperature of 5.5 °C), respectively. The total inlet temperature is 8.5 °C (superheating temperature is 3 °C), and the outlet flow rate is set according to the working conditions.

4.5. Performance Parameters

The working condition parameters of the refrigeration system which are used in the calculation of the system performance are shown in Table 3. The cooling capacity and the coefficient of performance (COP) are calculated based on two kinds of data. One is the simulation result of the compressors, the other is from the parameters in Table 3 and the thermophysical properties of the refrigerants.

The isentropic efficiency is an important parameter of centrifugal compressors, which is defined as the ratio of the isentropic power of the compressor to the actual compression power, which can also be calculated by the following equation [28]:

$$\eta_s = \frac{h_2 - h_1}{h_2^s - h_1}$$  \hspace{1cm} (5)

where $h_1$ (kJ·kg$^{-1}$) is the specific enthalpy value of the working fluid at the compressor inlet; $h_2$ (kJ·kg$^{-1}$) is the specific enthalpy value of the working fluid at the outlet of the actual compression process; $h_2^s$ (kJ·kg$^{-1}$) is the specific enthalpy value of the working fluid at the outlet of the isentropic compression process.

The cooling capacity of the unit is calculated by the following equation [29]:

$$Q_0 = q_m\eta_0 = q_m(h_1 - h_8)$$ \hspace{1cm} (6)

where $q_m$ (kg·s$^{-1}$) is the mass flow rate of the refrigerant; $\eta_0$ (kJ·kg$^{-1}$) is the unit cooling capacity; $h_1$ (kJ·kg$^{-1}$) is the specific enthalpy value of the compressor inlet; $h_8$ (kJ·kg$^{-1}$) is the specific enthalpy value of the evaporator inlet.

The COP of the system is calculated by the following equation:

$$\text{COP} = \frac{Q_0}{P_{ed} + P_{eg}}$$ \hspace{1cm} (7)

where $P_{ed}$ (kW) is the input power of the low-pressure compressor; $P_{eg}$ (kW) is the input power of the high-pressure compressor. They are calculated by the CFD simulation model.

5. Results and Discussions

5.1. Centrifugal Compressor Performance

In order to reflect the performance of the compressor under different working conditions, the relationship between the various speeds, inlet flow rate, and outlet pressure (or pressure ratio) of the compressor is usually expressed in the form of curves, which are called the flow characteristic curve or performance curve of the compressor [30].

In this paper, the CFD numerical method was used to calculate the performance curves of the centrifugal refrigeration compressors of the three refrigerants R134a, R1234yf, and R1234ze(E) at a rotational speed of 21,000 r·min$^{-1}$. Due to their different physical property parameters, their curves are different. The results are shown in Figure 6. Three figures show the relationship of pressure, compressor power, and efficiency with the mass flow rate. The mass flow rate represented by the $x$-axis is the inlet mass flow rate of the first stage impeller.
Figure 6a show the relationship between the mass flow rate and pressure ratio of the centrifugal compressor. The pressure ratio is defined as the ratio of the outlet pressor of the second stage of the compressor to the inlet pressure of the first stage, which is the total pressure ratio of the compressor. The intermediate injection pressure of gas from the economizer is set the same as the outlet pressure of the first stage compressor. The mass flow rate range of the R134a compressor under the stable working condition is 1.17 kg·s\(^{-1}\) to 2.64 kg·s\(^{-1}\); the mass flow rate range of the R1234ze(E) compressor under the stable working condition is 1.0 kg·s\(^{-1}\) to 2.2 kg·s\(^{-1}\), which is moved to the left and smaller than that of R134a compressor, and the range is smaller by 18.4%; the mass flow rate range of the R1234yf compressor under the stable working condition is 1.58 kg·s\(^{-1}\) to 3.2 kg·s\(^{-1}\), which is moved to the right and larger than that of R134a compressor, and the range is larger by 10.2%. The pressure ratio range of the R134a compressor is 1.9 to 2.8, that of the R1234ze(E) compressor is 2.1 to 3.2, and that of the R1234yf compressor is 2.2 to 3.3. The pressure ratio range of the R1234ze(E) and R1234yf compressor is about 22.2% larger than that of the 134a compressor. Additionally, the pressure ratio curve of R1234ze(E) and R1234yf is higher than that of R134a, which is because the molecular weights of R1234ze(E) and R1234yf are greater than that of 134a. As the molecular weight of gas increases, the pressure ratio curve moves up [31].

![Pressure ratio vs Mass flow rate](image1)

(a) Pressure ratio vs Mass flow rate, kg·s\(^{-1}\)

![Power vs Mass flow rate](image2)

(b) Power vs Mass flow rate, kg·s\(^{-1}\)

Figure 6. Cont.
When the mass flow rate is less than 2.0 kg·s\(^{-1}\), the pressure ratio of the R1234ze(E) compressor is greater than that of the R134a compressor, and when the flow rate is greater than 2.0 kg·s\(^{-1}\), the pressure ratio of the R134a compressor is greater than that of R1234ze(E) compressor; when the flow rate is the same, the pressure ratio of the R1234yf compressor is greater than that of the R134a compressor and the R1234ze(E) compressor. Compared with R1234yf, the mass flow rate-pressure ratio performance curves of R1234zd(E) and R134a are closer.

Figure 6b show the mass flow rate-power performance curve of the centrifugal compressor. At the same mass flow rate, the power of the R1234ze(E) compressor is less than that of the R134a compressor, and the difference between them increases gradually with the increase of the mass flow rate, the minimum is about 3.1% less, and the maximum is about 10.2% less; the power of the R1234yf compressor is greater than that of the R134a compressor; the minimum is about 9.9% greater, and the maximum is about 23.2% greater.

The mass flow rate–efficiency performance curve of the centrifugal compressor calculated by CFD is shown in Figure 6c. The isentropic efficiency increases first and then decreases with the increase of the flow rate. The maximum isentropic efficiency of the R1234ze(E), R134a, and R1234yf compressors are all 86–87%, which indicates that the flow conditions in the compressor are still maintained in a good state after the refrigerant R134a is drop-in replaced by R1234ze(E) or R1234yf.

5.2. Cooling Capacity

The cooling capacity at different condensation temperatures is shown in Figure 7. It can be seen that when the condensation temperature is less than 310 K, the cooling capacity of the R1234ze(E) unit is less than that of the R134a unit; when the condensation temperature is greater than 310 K, the cooling capacity of the R1234ze(E) unit is greater than that of the R134a unit. When the condensation temperature is the same, the cooling capacity of the R1234yf unit is greater than that of the R134a unit and the R1234ze(E) unit. Therefore, in the R134a centrifugal chiller, compared to R1234ze(E), directly replacing R134a with R1234yf obtains a higher cooling capacity.
5.3. Coefficient of Performance (COP)

The COP at different condensation temperatures is shown in Figure 8. It can be seen that when the condensation temperature is less than 312 K, the COP of the R1234ze(E) unit is less than that of the R134a unit, which is about 7.0% smaller. When the condensation temperature is the same, the COP of the R1234yf unit is less than that of the R134a unit, which is about 12.5% smaller. Therefore, compared with R1234yf, the reduction of the COP is less when R1234ze(E) is used to replace R134a directly in the R134a centrifugal chiller.

![Figure 7. Cooling capacity at different condensation temperatures.](image)

5.4. COP Comparison under Same Conditions

The rotational speeds of the R134a and R1234ze(E) compressors are set as 18,000 r·min⁻¹, 21,000 r·min⁻¹, and 24,000 r·min⁻¹, respectively. When the rotational speed of the R1234yf compressor is the same as that of the R134a compressor, it cannot achieve the same cooling capacity at the same evaporation temperature and condensation temperature, so the rotational speed of the R1234yf compressor is reduced, set to 17,000 r·min⁻¹, 20,000 r·min⁻¹ and 23,000 r·min⁻¹, respectively. The mass flow rate-pressure ratio performance curve of three refrigerants R134a, R1234ze(E), and R1234yf is shown in Figure 9a, the mass flow rate-power performance curve is shown in Figure 9b, and the mass flow rate-efficiency performance curve is shown in Figure 9c.
Figure 9. Performance curve of centrifugal compressor: (a) Mass flow rate-pressure ratio performance curve; (b) Mass flow rate-power performance curve; (c) Mass flow rate-efficiency performance curve.
Figure 10 show the variation of cooling capacity with condensation temperature for R134a, R1234ze(E), and R1234yf units at the same evaporation temperature. When the evaporation temperature, condensation temperature, and cooling capacity of the R134a unit and the R1234ze(E) unit are the same, that is, at the intersection of the black line and the red line in Figure 10, the COP comparison of the two units is shown in Table 5. The COP of the R1234ze(E) unit is reduced by about 5.14% on average compared to that of the R134a unit. When the evaporation temperature, condensation temperature, and cooling capacity of the R134a unit and the R1234yf unit are the same, that is, at the intersection of the black line and the blue line in Figure 10, the COP comparison of the two units is shown in Table 6. The COP of the R1234yf unit is reduced by about 8.93% on average compared to that of the R134a unit.

![Figure 10](image)

**Figure 10.** Diagram of the change of cooling capacity with condensation temperatures.

**Table 5.** COP comparison of R1234ze(E) unit and R134a unit.

<table>
<thead>
<tr>
<th>Condensation Temperature (K)</th>
<th>COP of R134a</th>
<th>COP of R1234ze(E)</th>
<th>Percentage of Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>300.8</td>
<td>10.360</td>
<td>9.822</td>
<td>5.19%</td>
</tr>
<tr>
<td>310.5</td>
<td>6.852</td>
<td>6.477</td>
<td>5.47%</td>
</tr>
<tr>
<td>322.8</td>
<td>4.435</td>
<td>4.224</td>
<td>4.76%</td>
</tr>
</tbody>
</table>

**Table 6.** COP comparison of R1234yf unit and R134a unit.

<table>
<thead>
<tr>
<th>Condensation Temperature (K)</th>
<th>COP of R134a</th>
<th>COP of R1234yf</th>
<th>Percentage of Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>296.0</td>
<td>12.318</td>
<td>11.462</td>
<td>6.95%</td>
</tr>
<tr>
<td>305.2</td>
<td>8.163</td>
<td>7.416</td>
<td>9.15%</td>
</tr>
<tr>
<td>319.4</td>
<td>4.990</td>
<td>4.457</td>
<td>10.7%</td>
</tr>
</tbody>
</table>

5.5. Results Verification

This model can run stably in the experiment of the R134a centrifugal water chiller. At the rotating speed of 21,000 r·min\(^{-1}\), the comparisons between the compressor pressure ratio and power calculated by the CFD method and the system cooling capacity and COP calculated by the thermodynamic analysis method and the experimental values are shown in Figure 11. It can be seen that they are relatively close, indicating that the results in this paper are reliable. The reason for the difference is that there will be some flow losses in the experiment, which are not considered in the CFD calculation.
Figure 11. Cont.
In this paper, the CFD numerical method is used to simulate the internal flow of the centrifugal refrigeration compressor, which uses R134a, R1234yf, and R1234ze(E) as refrigerants, respectively. For the R134a centrifugal chiller, the refrigerant is drop-in replaced by R1234yf and R1234ze(E), and the refrigeration performance is compared and analyzed by thermodynamic analysis method under the set working conditions. The main conclusions are as follows: the mass flow rate range of the R1234ze(E) compressor is moved to the left and is smaller than that of the R134a compressor; that of the R1234yf compressor is moved to the right and is larger than that of the R134a compressor. The pressure ratio curve of R1234ze(E) and R1234yf is higher than that of R134a.

At the same mass flow rate, the power of the R1234ze(E) compressor is 3.5% lower than that of the R134a compressor. Additionally, the power of the R1234yf compressor is 13.7% larger than that of the R134a compressor.

At the same rotational speed, in the R134a centrifugal chiller, although using R1234yf to directly replace R134a can obtain higher cooling capacity, the COP is reduced by about 12.5%; using R1234ze(E) to replace R134a can reduce the cooling capacity under certain conditions, the COP is reduced by about 7.0%.

When the evaporation temperature, condensation temperature, and cooling capacity of the R134a unit and the R1234ze(E) unit are the same, the COP of the R1234ze(E) unit is reduced by about 5.14% on average than that of the R134a unit. Additionally, the COP of the R1234yf unit is reduced by about 8.93% on average compared to that of the R134a unit. Therefore, in the R134a centrifugal chiller, compared with R1234yf, directly replacing R134a with R1234ze(E) can obtain better refrigeration performance.

The model can run stably in the experiment of the R134a centrifugal water chiller. The experimental results are relatively close to the calculated results in this paper, which verifies that the results in this paper are reliable.

In this paper, only the drop-in replacement of centrifugal compressors in the R134a centrifugal chiller is studied, and the replacement of heat exchangers and other parts in the system is not considered. Hence, research on the whole system of the drop-in replacement of R134a should be carried out in the future. In addition, relevant experimental research should be carried out in the future.
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Conflicts of Interest: The authors declare no conflict of interest.

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