Compressor Performance Comparison When Using R134 and R1234YF as Working Fluids

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COMPRESSOR PERFORMANCE COMPARISON
WHEN USING R134A AND R1234YF AS WORKING FLUIDS

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ABSTRACT

This paper examines the detailed effects on the compressor performance when using R1234yf as compared to R134a. Firstly, the effects on the compressor performance of the existing compressors which was designed for R134a when using the new refrigerant R1234yf will be presented and discussed. Secondly, the design aspects of a new compressor to be designed for R1234yf will also be presented and discussed. In the paper the rolling piston compressor was used to carry out the simulation tests. The detailed comparison of various performance parameters for the compressor are discussed and shown.

1. INTRODUCTION

In order to reduce the negative effects on the environment caused by the use of environmentally unfriendly refrigerants, worldwide efforts have been focused on replacing currently used refrigerants with one that is more environmentally friendly. One such refrigerant that has been singled out for replacement is HFC-134a (henceforth will be referred to as R134a). Most of the automotive air conditioners today use R134a as the working fluid. Legislation has been passed in Europe to dictate that by 2017, all new cars in Europe must use refrigerants that have a global warming potential (GWP) of lower than 150. R134a has a GWP of 1430. The HFO-1234yf (henceforth will be referred to as R1234yf) has been introduced specifically to replace R134a, the latter has been used in air-conditioning system in automobiles for about 20 years. R1234yf has a GWP of only 4. Extensive tests [1-5] have been carried on R1234yf to establish and determine its suitability (in all aspects including health and safety) in replacing R134a by scientists and engineers from DuPont and Honeywell. All these tests [1] show that HFO-1234yf give comparable if not better results in terms of cycle performance. In this paper the various detailed aspects of compressor performance when using R1234yf and R134a will be compared using a rolling piston compressor.

2. MATHEMATICAL MODEL

For completion, a brief account of the mathematical model is shown here, readers can refer to references [6-8] for more details. The mathematical model consists of volume, kinematics, roller dynamics, thermodynamics, valve dynamics, in-chamber heat transfer, mechanical frictional and lubrication. The volume $V(\theta)$ of the working chamber of the rolling piston compressor can be expressed in terms of the length of the compressor $l$, radii of the cylinder $R_c$ and rotor $R_r$, the rotational angle $\theta$ and the vane thickness $t_v$ as given by eqn. (1).

$$V(\theta) = f(l, R_c, R_r, \theta, t_v)$$

The variation of the properties of the working fluid in the working chamber can be obtained by applying the First Law of thermodynamics onto the working chamber of the compressor, i.e.

$$\dot{E}_{in} - \dot{E}_{out} = \frac{d(mu)_c}{dt}$$

where $E_{in}$ and $E_{out}$ and $(mu)_c$ are energy into and out of working chamber during a compressor cycle and the internal energy of the working chamber, respectively.

The real gas properties [9] of the refrigerant relate the enthalpy $h_c$ of the working fluid in the chamber to the pressure $P$ and its specific volume $v$, that is,
\( h_c = f(P, v) \) \hspace{1cm} (3)

The conservation of mass in the working chamber gives eqn. (4).

\[
\sum \frac{dm_i}{dt} - \sum \frac{dm_o}{dt} = \frac{dm_c}{dt}
\]

where \( m \) is the mass of the working fluid in the chamber and subscripts \( i, o \) and \( c \) represent in, out and chamber, respectively.

The compressor simulation model assumes that the flow through valves is a steady one-dimensional adiabatic flow [7]. Thus, the mass flow rate through valves can be expressed as

\[
\frac{dm_z}{dt} = C_d A s \sqrt{2(h_1 - h_{z1})} / \nu s
\]

where \( C_d \) indicates the combined effect of non-isentropic and flow losses, \( A \) the flow area, \( \nu \) the specific volume of the refrigerant and \( h \) the enthalpy of the working fluid. Indices 1, 2 and \( s \) indicate the upstream, downstream and isentropic conditions, respectively.

The area of the valve requires that the valve opening to be known at any instant of time. This is obtained by modelling the valve’s dynamic [7] under the pressure force during the discharge process.

The computer model was written in Fortran programming language solving simultaneously eqns. (1) to (5) using the 4th order Runge-Kutta numerical integration method. The model has been verified using R22 as the working fluid operating at operational conditions -23.3°C and 54.4°C at 2875 rev/min, by comparing its prediction with measured results, generally less than 10.0% discrepancy has been obtained, as shown in Fig.1 [6].

![Fig. 1 Comparison between measured and predicted results [6].](image)

**3. RESULTS AND DISCUSSIONS**

To compare the detailed performance of compressors when using R1234yf to that of R134a, simulation runs have been carried out using a rolling piston compressor with a displacement volume of 32 cm³ and operating at 2875 rev/min. Fig. 2 shows the basic simulation results when running the compressor at \( T_{\text{cond}} = 54.4 \, ^\circ\text{C} \) and \( T_{\text{evap}} = -10.6 \, ^\circ\text{C} \).
Fig. 2 (a) shows that the variation of the pressure-volume diagram is very similar when using R1234yf and R134a, with, R134a shows a higher final discharge pressure, with the difference in the indicated work for compressor when using both fluids to be less than 2%, as shown in Fig. 2(e). A careful check on Fig. 2 (e) reveals that the indicated work for R1234yf is higher than that of the R134a because of the higher pressure of R1234yf during the early stage of the compression, when the specific volume of the gas is large. This latter effect has resulted in a higher shaft torque required for the case of R1234yf, as shown in Fig. 2(g). Fig. 2(b) shows that the R1234yf has about 25% more mass than R134a. However this 25% more mass indicated by the R1234yf did not produces an equivalent increase in cooling capacity because the refrigerating effect of R1234yf is about 18% lower than that of the R134a, as shown in Table 1.

Fig. 2(a) and 2(c) also show that R134a has a higher final discharge pressure and temperature. As a result of marginally higher discharge pressure and lower suction pressure for R134a, the frictional losses in the R134a are marginally higher, as shown in Fig. 2(d). Fig. 2(f) and 2(h) show that the behaviour for compressor discharge valve and vane contact forces are very similar for both working fluids.

Table 1 Comparison of R134a and R1234yf properties at -10.6°C.

<table>
<thead>
<tr>
<th></th>
<th>R134a</th>
<th>R1234yf</th>
<th>[R134a-R1234yf]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sat. vapour density(kg/m³)</td>
<td>9.8164</td>
<td>12.296</td>
<td>-25.26%</td>
</tr>
<tr>
<td>Latent heat of vaporisation(kJ/kg)</td>
<td>206.40</td>
<td>169.81</td>
<td>+17.73%</td>
</tr>
<tr>
<td>Saturated pressure @-10.6°C (kPa)</td>
<td>195.90</td>
<td>216.92</td>
<td>-10.73%</td>
</tr>
<tr>
<td>Saturated pressure @54.4 °C (kPa)</td>
<td>1469.8</td>
<td>1444.5</td>
<td>+1.72%</td>
</tr>
</tbody>
</table>

In order to compare the performance of these two fluids over a wider range of operational conditions, 10 cases of simulation runs have been performed. Table 2(a) shows the cases with the a fixed condensing temperature at 54.4 °C but with varying evaporating temperatures from -40 °C to 0 °C, while Table 2(b) shows the operational conditions with a fixed evaporating temperature at -10.6 °C when condensing temperature varies from 35 °C to 55 °C.

Fig. 3 shows the results of the simulation run correspond to the operational conditions shown in Table 2(a), while Fig. 4 shows the results for cases when working under conditions given in Table 2(b).

Fig. 3(a) shows that the cooling capacity for R134a is always higher than that of the R1234yf despite the fact that R1234yf shows more mass flows through the compressor, as seen in Fig. 3(b), this is because R134a has a higher latent heat of vaporisation. This has resulted in the higher cycle COP for R134a, see fig. 3(c). The averaged torque input for compressor when using these two fluids are very similar in magnitude, as shown in Fig. 3(d).

Fig. 3(e) shows that R134a exhibits a marginally lower mechanical efficiency as it shows marginally higher frictional losses, as shown in Fig. 3(f) and 3(g). The marginally higher frictional losses exhibited by the compressor when using R134a is mainly caused by the larger pressure difference between the two chambers when using R134a as the working fluid.

Table 2 Operating conditions for cases of simulation runs

<table>
<thead>
<tr>
<th>T&lt;sub&gt;&lt;sup&gt;cond&lt;/sup&gt;&lt;/sub&gt; (°C)</th>
<th>T&lt;sub&gt;&lt;sup&gt;evap&lt;/sup&gt;&lt;/sub&gt; (°C)</th>
<th>T&lt;sub&gt;&lt;sup&gt;evap&lt;/sup&gt;&lt;/sub&gt; (°C)</th>
<th>T&lt;sub&gt;&lt;sup&gt;cond&lt;/sup&gt;&lt;/sub&gt; (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>54.4</td>
<td>-40 %55</td>
<td>-35 %55</td>
<td>-20 %55</td>
</tr>
</tbody>
</table>

Figs. 3(h) and 3(i) show that marginally higher suction and discharge losses occur in compressor using R1234yf, which is caused by its higher mass flow rate. Fig. 3(j) shows that compressor using R1234yf has a
higher volumetric efficiency. This is because the higher pressure differential between the suction and the compression chamber results in higher internal leakage in the R134a compressor, and the situation is exacerbated by a lower total mass flow of R134a.

The results show that when using R1234yf in low refrigerating temperature and high condensing temperature conditions, the performance of R1234yf is expected to be slightly lower than that of R134a, however the difference in COP and cooling capacity is expected to be less than 2%.

Fig. 4 shows the compressor performance comparison when using R1234yf and R134a and operating at lower condensing temperature situations. The results show that the compressor using R1234yf in this case is clearly a winner. The compressor gives a higher cooling capacity, higher COP and lower required torque when using R1234yf. The results also show that the differences can be as high as up to 5% better for cooling capacity and 10% better in cycle COP for compressor using R1234yf.

4. CONCLUSIONS

The results show that the performance for compressor when using R1234yf and R134a are comparable and this has reconfirmed the results available in the literature. The results also shows that the compressor working with R134a performed better than that working with R1234yf when operating under high condensing and low evaporating temperatures. However when the condensing temperature gets lower, R1234yf outperformed R134a. Over the range of operational conditions tested, the maximum difference in terms of cooling capacity is less than 5% and the COP is less than 10%.

5. REFERENCES

1. Minor, B., Spatz, M., 2008, HFO-1234yf Low GWP Refrigerant Update, 12th International Refrigeration and Air Conditioning Conference at Purdue, paper no.2349
Fig. 2 Basic simulation results when $T_{\text{comp}}=54.4$ °C and $T_{\text{evap}}=-10.6$ °C.
Fig. 3 Results when $T_{\text{cond}}=54.4^\circ\text{C}$ and $T_{\text{evap}}$ varies from -40°C to 0°C.
Fig. 4 Results when $T_{\text{evap}}=10.6 \, ^\circ\text{C}$ and $T_{\text{cond}}$ varies from 35 °C to 55 °C.