Tribology in CO₂ Scroll Compressors

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The natural refrigerant CO₂ has attracted attention as an alternative to R134a currently used in automotive air-conditioning, which has high global warming potential. A scroll compressor was developed for CO₂ automotive air-conditioning. The pressure of CO₂ is much higher than that of R134a, resulting in low CO₂ scroll compressor thrust bearing reliability and efficiency because of the large gas thrust. The static pressure assist bearing we developed was confirmed experimentally and analytically to be feasible in CO₂ scroll compressor use. We also studied elastomer, shaft-seal, and refrigerant oil for CO₂ refrigerant.

1. Introduction

The refrigerant used for automotive air-conditioning was shifted from R12 to R134a to protect the ozone layer from depletion. However, the restriction of R134a is now being considered to prevent global-warming. Consequently, carbon dioxide (CO₂), which is a natural refrigerant, has recently been attracting attention as an alternative to R134a.

Carbon dioxide is an excellent refrigerant because it is nontoxic, nonflammable, and inexpensive. On the other hand, CO₂ becomes supercritical at the high-pressure side in the refrigeration cycle and its COP (coefficient of performance) is low in a conventional simple system, so that a high-pressure control, etc. is required to improve the COP. The improvement of the system was fully described in the previous report[1].

The CO₂ compressor used for the improved system achieved an overall adiabatic compression efficiency, \( \eta_{\text{ad}} \), of 76% as a unit. However, the compressor weight was heavier than that for R134a because it was manufactured as a prototype for verifying the performance. This time, a lightweight prototype compressor with improved efficiency was produced to carry out the performance test in an experimental refrigerating system.

2. Features of CO₂ as a refrigerant

Table 1 shows the characteristics of CO₂ and R134a. While the R134a used for automotive air-conditioners has zero ozone depleting potential, it has 3100 for its GWP (Global Warming Potential, expressed as the ratio of its contribution to global-warming relative to CO₂ whose contribution is defined 1). On the other hand, CO₂ is a refrigerant low in GWP, and also non-toxic, non-flammable, and easy-to-handle material like R134a. At a glance, it would appear contradictory to use CO₂ as a refrigerant to prevent global-warming, that is to reduce CO₂ emission. However, the CO₂ is not newly produced, it is merely utilizing existent CO₂ gas, so that its actual GWP is virtually zero.

The refrigerating capacity per unit volume of CO₂ at 0 °C is about 8 times that of R134a. Accordingly, even if considering the wider operating conditions, the displacement of a CO₂ compressor is 1/8 to 1/6 that of an R134a compressor having the same capacity.

However, the saturation pressure of CO₂ is 10 times that of R134a, so that, when designing CO₂ compressors, it is necessary to solve problems caused by the higher loads of the sliding parts and the differential pressures of the sealing parts.

3. Improvement of CO₂ scroll compressor efficiency

3.1 Construction of scroll compressor for CO₂

Automotive air-conditioning scroll compressors for R134a have the excellent characteristic that the pressure drop is low especially at a high-speeds because there is no suction valve, and also another remarkable advantage is that the low discharge pulsation due to the small volumetric change rate reduces noise and vibration. These basic characteristics are kept when CO₂ is substituted for R134a as the refrigerant.

<table>
<thead>
<tr>
<th>Characteristics of CO₂</th>
<th>CO₂</th>
<th>R134a</th>
</tr>
</thead>
<tbody>
<tr>
<td>ODP*1</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>GWP 20 years*2</td>
<td>1</td>
<td>3000</td>
</tr>
<tr>
<td>Critical Temperature (°C)</td>
<td>31.1</td>
<td>101.2</td>
</tr>
<tr>
<td>Critical Pressure (MPa)</td>
<td>7.38</td>
<td>4.07</td>
</tr>
<tr>
<td>Latent heat of vaporization*3 (kJ/kg)</td>
<td>231.6</td>
<td>198.4</td>
</tr>
<tr>
<td>Saturation pressure*3 (kPa)</td>
<td>3485</td>
<td>293</td>
</tr>
<tr>
<td>Saturated vapor density*4 (kg/m³)</td>
<td>97.32</td>
<td>14.43</td>
</tr>
</tbody>
</table>

*1 ODP: Ozone Depleting Potential
*2 GWP: Global Warming Potential (in relation to CO₂ with 20 years integration time)
*3 at 0 °C
*4 at 0 °C

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Accordingly, the scroll type compressor was also chosen for use with CO₂. The outside view of the lightweight prototype compressor prepared this time is shown in Fig. 1.

Fig. 2 shows the analysis results of the three kinds of compressor losses investigated on the scroll type R134a compressor for automotive air-conditioning, the scroll type CO₂ compressor whose displacement alone is reduced (hereafter termed the pre-improvement CO₂ compressor), and the lightweight prototype compressor.

The loss by over-compression and wire-drawing of the pre-improvement CO₂ compressor is less than that of the R134a compressor, because larger density of CO₂ than R134a reduces its mass flow rate. However, the pre-improvement CO₂ compressor has a large leakage loss, because it has larger pressure difference between the high- and low-pressure sides than R134a. The mechanical loss is increased because the friction losses in the sliding surfaces (especially the thrust bearing) are increased due to the higher pressure. Therefore, the pre-improvement CO₂ compressor has a lower efficiency.

In order to improve the efficiency of a CO₂ scroll compressor, it is necessary to reduce the leakage, generated due to the higher pressure, from the high-pressure chamber and mechanical losses due to the increased load. Thus, the construction with the orbiting scroll pushed to the fixed scroll was adopted for controlling the leakage from the scroll tip surface. To reduce mechanical losses, the thrust load was reduced by applying a high-pressure to the backsides of the bearings, especially for the thrust bearing supporting the heavy load (hereafter termed a “static pressure assist bearing”).

3.2 Static pressure assist bearings

Fig. 3(a) illustrates the static pressure assist bearing. As shown in the figure, the static pressure assist bearing reduces the downward load subject to the thrust bearing by upwardly applying hydraulic pressure taken from an oil separator installed in the high-pressure side.

The static pressure assist bearing construction improves the reliability by reducing the thrust surface pressure, and also improves the mechanical efficiency. Fig. 3(b) shows the relationship between the sliding surface pressure \( P \) and the sliding part velocity \( V \) of the lightweight prototype compressor adopting the static pressure assist bearings. Fig. 3(b) also compares the surface pressures of field-proven MHI (Mitsubishi Heavy Industries, Ltd.) R134a compressor with and without static pressure assist bearings. The CO₂ compressor operates in a severe region exceeding the conventional field-proven values in sliding part surface pressure, sliding speed, and their product (PV value), which are used for the indices of reliability at a boundary lubrication region. However, by adopting the static pressure assist bearings, the PV values can be reduced to below the conventional field-proven values.

The effect of efficiency improvement by the static pressure assist bearings was verified by a unit test. Fig. 3(c) shows the results of the unit tests and indicates that the efficiency (evaluated as the overall adiabatic compression efficiency) is increased as the thrust load is reduced by the assistance of the hydraulic pressure. When the thrust load is exceedingly reduced, however, the efficiency is reduced. This indicates that there is an optimal load for peak efficiency. This efficiency reduction is caused by the increased leakage of the high-pressure side oil used for the assistance.

3.3 Verification of an actual scroll compressor

The center bar in Fig. 2 shows the results obtained by analyzing the performance of the lightweight prototype CO₂ scroll compressor with the construction.
mentioned above and the thrust bearing load optimized for efficiency. It was expected that the overall adiabatic compression efficiency would be improved from 68% in R134a to 75% in CO₂ by reducing the leakage and mechanical losses.

Using this prototype compressor, the overall adiabatic compression efficiency was determined in a running test performed at 40 km/h. Fig. 4 shows the analytic and experimental results. Although the experimental result is slightly less than the analytic result, both are nearly consistent with each other, and an efficiency of 73% is achieved at a rotation of 2400 rpm.

The figure shows that, in a swash plate compressor, a higher rpm results in increased wire-drawing loss and decreased efficiency, while in a scroll compressor, a higher rpm results in improved efficiency.

The analytic and experimental results indicate that the lightweight prototype CO₂ scroll compressor can achieve a high efficiency under wide-ranged operation.

4. Evaluation of O-ring material and refrigerant oil

In order to commercialize automotive CO₂ refrigerant air-conditioning systems, the selection of elastomer (O-rings in this case) and refrigerant oils used for the air-conditioning systems is important.

4.1 O-rings

Most O-rings in present automotive R134a refrigerant air-conditioning systems made of H-NBR (hydrogenated nitrile rubber). The leakage of CO₂ permeating through the O-ring material and rupture (blisters) have been pointed out as big problems for the O-rings used in CO₂ refrigerant. Accordingly, for the fixed sealing parts of CO₂ systems (including compressors), metal seals are used instead of O-rings. Thus, evaluation testing of O-rings for movable sealing parts, such as mechanical seals, where metal seals cannot be used, was carried out. Table 2 shows the test results. Material D was selected for the O-rings, because it has good resistance to blisters and swelling.

4.2 Refrigerant oils

Generally, the refrigerant oil used for air-conditioning requires excellent lubricity, oil-returning ability
Table 2 Result of dipping tests for O-ring

<table>
<thead>
<tr>
<th>Material</th>
<th>Volumetric change rate (%) [direct after test]</th>
<th>Volumetric change rate (%) [24 hours later]</th>
<th>Blistering</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>60.54</td>
<td>1.52</td>
<td>Yes</td>
</tr>
<tr>
<td>B</td>
<td>32.78</td>
<td>4.27</td>
<td>Yes</td>
</tr>
<tr>
<td>C</td>
<td>24.67</td>
<td>0.25</td>
<td>No</td>
</tr>
<tr>
<td>D</td>
<td>12.40</td>
<td>-2.28</td>
<td>No</td>
</tr>
</tbody>
</table>

Conditions of tests: 80°C, 15 MPa, 100 hours

Table 3 Characteristics of oils for CO₂ refrigerant

<table>
<thead>
<tr>
<th>Lubricity</th>
<th>Chemical stability</th>
<th>Oil-returning ability to compressor</th>
<th>Miscibility with CO₂</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Thermo-oxidation-stability</td>
<td>Hydrolysis-stability</td>
<td>Miscibility</td>
<td>Density</td>
</tr>
<tr>
<td>PAG</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>POE</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Alkylbenzene</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Mineral oil</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

5. Evaluation of shaft seals

A compressor for open type automotive air-conditioning has three types of sealing points: piping joints, flanges, and shaft seals. Lip seals are usually used for the shaft seals; however, they are hard to adopt to a CO₂ compressor because the inside pressure is increased to about 2 MPa. Therefore, single spring type mechanical seals were adopted because of their compact configuration.

5.1 Test equipment

Fig. 5 outlines the test equipment. The seals were installed at both ends of a pressurized container. At one end, the motor-driven rotating shaft did not pierce through the sealing part so as to facilitate the measurement of leakage from the end and observation of the seal. In addition, the leakage from the sheet surface of the mechanical seal was determined independently, being separated from the penetration leakage through O-rings.

5.2 Test method

In the leak test at a stopped state, besides the usual leakage determination, by using a sapphire glass window for the seat ring, both leakage determination and state observation were dealt with simultaneously. The refrigerant oil was used, with POE (ester oil) and PAG contained by 50% of the total volume of the tested part was filled with either POE (ester oil) or PAG used as refrigerant oil.

To determine the leakage amount, a mass flow meter was used. To cope with the cases where the low leakage level is always low, the leakage amount at the stopped state was determined by an instrument system equipped with a mass spectrometer.

5.3 Test results

Fig. 6 shows the result of the leakage test at the stopped state. In the test using PAG and that using POE, which was performed at or above the critical temperature, both leakage amounts were usually close to the detection limit on a time average, that is, large leakages were not observed, although some instantaneous values were close to the target set previously.

On the other hand, in the test using POE performed at a temperature below the critical temperature, the leakage was sharply higher.

Fig. 7 shows the states of seal surfaces at the stopped state. The seal surface circumference is in a
high-pressure region, therefore, the leakage occurs toward the inner side. In the test using PAG, the oil and CO₂ separated, and oil occupied the lower half of the vessel. In the upper half of the seal surface circumference, oil formed a meniscus, which was thought to protect the upper half from direct exposure to liquid or gas CO₂ and result in a little leakage.

On the other hand, in the test using POE, phase separation did not occur when CO₂ was liquefied and an interface was not observed at the seal surface circumference, that is, because liquefied CO₂ penetrated into clearances like oil from any position, so that the leakage seemed to be increased.

Furthermore, as the result of the rotation test, it was shown that the total leakage per hour nearly satisfies the target, taking into consideration the operating conditions of actual compressors, although the leakage was greater than when stopped.

6. Conclusions

For the CO₂ refrigerant compressor, the scroll type compressor was adopted with the orbiting scroll pushed to the fixed scroll and static pressure assist bearings were chosen to improve the efficiency. According to the analytic and experimental results, it could be confirmed that the CO₂ scroll compressor achieved an overall adiabatic compression efficiency 73%. Also, the verification tests to select the applicable elastomers and refrigerant oils for CO₂ refrigerant were carried out, and by clarifying the CO₂ leakage from the shaft seals, it could be verified that the leakage is not troublesome under ordinary operation.

In the future, the compressor will be made more compact and lighter and also the reliability for wide-ranged operation will be verified to contribute to its commercialization.

Reference