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PERFORMANCE CHARACTERISTICS OF SCROLL EXPANDER FOR CO₂ REFRIGERATION CYCLES

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ABSTRACT

To help lessen the effects of global warming, CO₂ refrigeration cycles are required in refrigeration and air-conditioning applications. Work-output expansion devices (expanders) are needed to improve the system performances, but their efficiencies are still too low to make the systems practical. The improvement in the COP of a CO₂ refrigeration cycle is highly dependent on expander efficiency. To solve this problem, we developed a prototype of a scroll expander for CO₂ refrigeration cycles and studied its performance characteristics. The experimental parameters are rotational speed, inlet pressure and temperature, exit pressure, and CO₂ mass flow rate of the scroll expander. The test results show that our scroll expander had maximum measured isentropic efficiencies greater than 70%. We confirmed that our scroll expander has a high potential for making CO₂ refrigeration systems practical.

1. INTRODUCTION

Carbon dioxide (CO₂) has been receiving attention due to global environmental problems. The coefficient of performance (COP) of CO₂ refrigeration cycles, however, is typically low compared to conventional hydrofluorocarbon (HFC) refrigeration cycles. The main reason for this low COP is the large throttling loss that results from the transcritical refrigeration cycle. Many researchers have been trying to improve the COP of transcritical CO₂ refrigeration cycles. The throttling loss can be reduced by replacing the expansion valve with a work-output expansion device (expander or ejector). Various types of expanders to recover the throttling loss of the CO₂ refrigeration cycles have been studied. A scroll-type fluid machine is an expander that has been utilized in an organic Rankine cycle. However, the amount of research on its performance for the CO₂ refrigeration cycle is relatively small.


This paper shows the experimental performance of a prototype scroll expander and presents its basic performance characteristics.

2. FEATURE OF EXPANDER EFFICIENCY

This section explains the features of the expander in comparison with the compressor. Table 1 shows the definitional expression of expander and compressor efficiency. Since the movement of the expander is opposite to the movement of the compressor, expander efficiency is defined as the reversal expression of compressor efficiency. Due to this
reversal form relation, each characteristic of the isentropic efficiency toward mechanical loss is different. Figure 1 shows the relationship between the mechanical loss ratio and the isentropic efficiency of the expander and compressor. This figure indicates that the isentropic efficiency of the expander is lower than that of the compressor at the same mechanical loss ratio. Therefore, the expander needs to be a low mechanical loss fluid machine. The decrease in mechanical loss is especially important in the expander with improvement in the volumetric and indicated efficiencies.

3. PROTOTYPE SCROLL EXPANDER

One of the features of the scroll type fluid machine is its low mechanical loss. Therefore, we selected the scroll type fluid machine as the expander. Figure 2 shows a prototype of a scroll expander, and the major specifications are listed in Table 2. This scroll expander is designed for CO₂ air-conditioning systems and has about a 700 W electrical output at the cooling rated condition. The expander inlet volume is 2.8 cm³ and the built-in volume ratio is 2.0. The scroll mechanism consists of a fixed scroll, an orbiting scroll, an Oldham coupling, a crankshaft, and a frame, and it is mounted in a housing case. The generator is connected to the scroll assembly. This scroll expander has a self-adjusting orbiting scroll support mechanism like our product’s scroll compressors (Tsubono I. et al., 1998). Behind the orbiting scroll, a back-pressure chamber is provided. The pressure in this chamber is maintained at an intermediate range between the inlet pressure and the exit pressure. The level of back-pressure sufficient to keep the orbiting scroll engaged with the fixed scroll can be found by the experimental data. This back-pressure mechanism reduces friction at the sliding portion between the orbiting and fixed scrolls and provides axial compliance for maintaining the seal at the scroll wrap tip clearance. The scroll expander lubricated by polyalkylene glycol (PAG) oil, which is stored in the bottom of the housing case. The oil is supplied to the bearings and other sliding surfaces by the pressure difference between the discharge pressure in the housing case and the intermediate pressure in the back-pressure chamber.

4. EXPERIMENT

Figure 3 shows the experimental setup. Changing the electric current or voltage of the DC electronic resistive load controls the rotational speed of the expander. The temperature and pressure of the expander (inlet and exit), the refrigerant mass flow rate, the generator power output, and the rotational speed of the expander were measured. The expander volumetric efficiency \( \eta_v \) is defined by

\[
\eta_v = \frac{G_{th}}{G}
\]

where \( G_{th} \) is the refrigerant theoretical mass flow rate, and \( G \) is the actual mass flow rate of the expander.

The expander isentropic efficiency \( \eta_{ex} \) is given as

\[
\eta_{ex} = \frac{L_g}{(G \cdot \Delta h_{ex} \cdot \eta_g)}
\]

where \( L_g \) is the generator power output, \( \Delta h_{ex} \) is the isentropic enthalpy difference of the expander, and \( \eta_g \) is the generator efficiency.

5. TEST RESULTS

5.1 Effects of Back Pressure
The control of the back-pressure was executed by adjusting the amount of gas and oil extracted from the back-pressure chamber to a low pressure line at the evaporator exit with a needle valve. In the experiment, the inlet pressure and temperature, the exit pressure, and the rotational speed of the expander were kept constant. Figure 4 illustrates the relationship between the expander efficiency and the back-pressure difference. The vertical axis represents the relative efficiency based on the maximum isentropic efficiency (1.0). The horizontal axis represents the relative pressure difference between the back-pressure and the exit pressure based on the optimum back-pressure difference (1.0). From this figure, the back-pressure difference is extremely important to improve the expander efficiency. The experiment thereafter was a result of keeping this optimum back-pressure difference.

5.2 Effects of Rotational Speed
Figure 5 shows the effects of the rotational speed of the expander. In the experiment, the inlet pressure was approximately 8.2 MPa, and the inlet temperature was 36 °C. The scroll expander performance was measured for the
exit pressure is at 4.77, 4.21, and 3.96 MPa by changing the rotational speed from 2000 to 3500 min\(^{-1}\). The optimum rotational speed of the scroll expander was in the range of 2200 to 3400 min\(^{-1}\). This optimum value does not depend on the exit pressure. The most efficient operating point is 83\%. The rotational speed, inlet pressure, and exit pressure at this point was 2655 min\(^{-1}\), 8.3 MPa, and 4.21 MPa, respectively. The operating pressure ratio and the refrigerant mass flow rate are different in Figure 5. Next, these effects are examined.

5.3 Effects of Pressure Ratio and Refrigerant Mass Flow Rate

Figure 6 shows the effects of the operating pressure ratio of the expander. This figure is a re-arrangement of Figure 5 at the rotational speed of about 2800 min\(^{-1}\). From this figure, the optimum operating pressure ratio was about 1.92. This value is a little larger than the ideal pressure ratio of 1.77 (refer to Figure 8) of the prototype scroll expander.

Figure 7 shows the effects of the operating pressure ratio at an inlet pressure of 9.2 MPa and inlet temperature between 41-43 °C. The optimum operating pressure ratio was about 1.95. This value is a little larger than the ideal pressure ratio of 1.82.

Figure 9 shows the effects of the refrigerant mass flow rate. The pressure ratio and the rotational speed are almost constant. This figure shows that the volumetric and isentropic efficiency decrease when the refrigerant mass flow rate decreases.

6. CONCLUSIONS

We have developed a prototype of a scroll expander and investigated its performance characteristics for a CO\(_2\) refrigeration cycle. As a result, the following conclusions were obtained:

- The scroll expander has a maximum measured isentropic efficiency of 83\%.
- The optimum rotational speed of the scroll expander is in the range of 2200 to 3400 min\(^{-1}\).
- The optimum operating pressure ratio is a little larger than the ideal pressure ratio of the expander.
- Our scroll expander has a high potential for making CO\(_2\) refrigeration systems practical.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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<tbody>
<tr>
<td>(G)</td>
<td>Refrigerant Mass Flow Rate</td>
<td>(kg/h)</td>
</tr>
<tr>
<td>(L_{ad})</td>
<td>Adiabatic (Isentropic) Power</td>
<td>(W)</td>
</tr>
<tr>
<td>(L_g)</td>
<td>Generator Output Power</td>
<td>(W)</td>
</tr>
<tr>
<td>(L_i)</td>
<td>Indicated Power</td>
<td>(W)</td>
</tr>
<tr>
<td>(L_s)</td>
<td>Shaft Power</td>
<td>(W)</td>
</tr>
<tr>
<td>(n)</td>
<td>Rotational Speed</td>
<td>(min(^{-1}))</td>
</tr>
<tr>
<td>(P)</td>
<td>Pressure</td>
<td>(MPa)</td>
</tr>
<tr>
<td>(P_b)</td>
<td>Back-Pressure</td>
<td>(MPa)</td>
</tr>
<tr>
<td>(T)</td>
<td>Temperature</td>
<td>(°C)</td>
</tr>
<tr>
<td>(\Delta\dot{M}_m)</td>
<td>Mechanical Loss</td>
<td>(W)</td>
</tr>
<tr>
<td>(\eta_{ex})</td>
<td>Expander Isentropic Efficiency</td>
<td>(−)</td>
</tr>
<tr>
<td>(\eta_v)</td>
<td>Volumetric Efficiency</td>
<td>(−)</td>
</tr>
<tr>
<td>(i)</td>
<td>Inlet</td>
<td></td>
</tr>
<tr>
<td>(o)</td>
<td>Exit</td>
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REFERENCES


Table 1 Definitions of Expander Efficiency

<table>
<thead>
<tr>
<th>Concept</th>
<th>Compressor</th>
<th>Expander</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chart</td>
<td>$\eta_{ad} = \frac{L_{ad}}{L_s} = \eta_v \cdot \eta_i \cdot \eta_m$</td>
<td>$\eta_{ex} = \frac{L_s}{L_{ad}} = \eta_v \cdot \eta_i \cdot \eta_m$</td>
</tr>
<tr>
<td>Isentropic Efficiency</td>
<td>$\eta_v = \frac{G}{G_{th}}$</td>
<td>$G_{th}$ : Theoretical Flow Rate</td>
</tr>
<tr>
<td>Volumetric Efficiency</td>
<td>$\eta_i = \frac{L_{th}}{L_i}$</td>
<td>$L_{th}$ : Theoretical Power</td>
</tr>
<tr>
<td>Indicated Efficiency</td>
<td>$\eta_m = \frac{L \cdot L_{th}}{L \cdot L_i}$</td>
<td>$\Delta L_m$ : Mechanical loss</td>
</tr>
<tr>
<td>Mechanical Efficiency</td>
<td>$\eta_i = \frac{L_{th}}{L_i}$</td>
<td>$L_{th}$ : Theoretical Power</td>
</tr>
</tbody>
</table>

$\Delta L_m$: mechanical loss

Figure 1 Comparison of Expander and Compressor Efficiencies
Figure 2 Prototype Scroll Expander

Table 2 Major Specifications

<table>
<thead>
<tr>
<th>Expander Type</th>
<th>Scroll</th>
</tr>
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<tbody>
<tr>
<td>Wrap Curve</td>
<td>Involute</td>
</tr>
<tr>
<td>Inlet Volume: $V_i$</td>
<td>$2.8,\text{cm}^3/\text{rev.}$</td>
</tr>
<tr>
<td>Volume Ratio: $V_r$</td>
<td>$2.0$</td>
</tr>
<tr>
<td>Oil</td>
<td>PAG (VG100)</td>
</tr>
<tr>
<td>Oil Supply</td>
<td>Pressure Difference</td>
</tr>
</tbody>
</table>

Figure 3 Experimental Setup

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Figure 4  Effects of Back-Pressure

Figure 5  Effects of Expander Rotational Speed

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Figure 6 Effects of Pressure Ratio (1)

Figure 7 Effects of Pressure Ratio (2)

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Figure 8 Expander Ideal Pressure Ratios

Figure 9 Effects of Refrigerant Mass Flow Rate