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BASIC OPERATING CHARACTERISTICS OF CO\textsubscript{2} REFRIGERATION CYCLES WITH EXPANDER-COMPRESSOR UNIT

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ABSTRACT

A transcritical CO\textsubscript{2} refrigeration cycle with an expander was studied to improve the coefficient of performance (COP) of the cycle of a simple expansion valve by recovering the throttling loss. We developed an expander-compressor unit. A scroll type expander and a rolling-piston type rotary sub-compressor were adopted and connected them with a shaft. With this unit, we made a CO\textsubscript{2} refrigeration cycle for a small capacity air-cooled chiller and investigated the basic operating characteristics of the cycle. In experiments, the expander-compressor unit was shown to be stable and to improve the COP of the CO\textsubscript{2} chiller cycle under both cooling and heating conditions. The test results indicated that the COP improvement of the cycle was more than 30\% while the total efficiency of the expander-compressor unit was 57\%. Expander-compressor unit are a key technology for use in CO\textsubscript{2} refrigeration systems.

1. INTRODUCTION

Recently, natural refrigerants have been receiving attention due to global environmental problems. Carbon dioxide (CO\textsubscript{2}), a natural refrigerant, is a potential substitute for hydrofluorocarbons (HFCs) because of its incombustibility, non-toxicity and low global warming potential (GWP). The coefficient of performance (COP) of CO\textsubscript{2} refrigeration cycles, however, is typically low compared to the HFC refrigeration cycles. The main reason for this low COP is large throttling loss that results from a transcritical refrigeration cycle. Many researchers have been trying to improve the COP of transcritical CO\textsubscript{2} 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 have been studied to recover the throttling loss of the CO\textsubscript{2} refrigeration cycles. For example, Nickl et al. (2005) developed a third generation CO\textsubscript{2} reciprocating expander, which has three independent expansion stages. Hays and Brasz (2004) analyzed a transcritical turbine-compressor for CO\textsubscript{2} refrigeration and heat pump systems. Westphalen and Dieckmann (2004) analyzed a scroll expander for CO\textsubscript{2} refrigerant-cooling systems. Similar research is going on in many other companies and universities with different types of expander designs. However, the amount of research for operating characteristics of the CO\textsubscript{2} refrigeration cycle with expanders is relatively small. Fukuta et al. (2001) studied the theoretical performance of the CO\textsubscript{2} cycle with a vane compressor-expander combination. Taking account of the loss due to miss-matching of the operating condition, the theoretical COP of the cycle with the expander was found to be approximately 1.5 times larger than the cycle without the expander. Hiwata et al. (2003) investigated the theoretical performance of the cycle with the expander for both cooling and heating operations. A new refrigeration cycle (change over cycle) was proposed as a cycle with an expander for CO\textsubscript{2} air-conditioning systems.

This paper shows the experimental performance of a transcritical CO\textsubscript{2} refrigeration cycle with an expander-compressor unit and presents its operating characteristics for both cooling and heating conditions.
2. CO₂ REFRIGERATION CYCLE WITH EXPANDER-COMPRESSOR UNIT

A CO₂ refrigeration cycle with an expander-compressor unit is shown in Figure 1. This cycle is for an air-cooled heat pump chiller. The specifications of the cycle components are shown in Table 1. The refrigeration cycle consists of a main compressor (MC), an oil separator, a four-way valve, an air-side heat exchanger, an expander-compressor unit, a water/refrigerant heat exchanger, check valves, and etc. The expander-compressor unit includes an expander (EX) and a sub-compressor (SC).

In Figure 1, a solid arrow shows the flow of the refrigerant during cooling operation and a broken arrow shows the flow during heating operation. The four-way valve changes these two operating modes. In both cooling and heating operation, check valves fix the flow direction of the refrigerant to the expander so that the expander always recovers the energy. This recovered energy directly drives the sub-compressor. Here, the sub-compressor acts as a boosted compressor and reduces the power of the main compressor.

Figure 2 shows an ideal P-h diagram of the CO₂ cycle with the expander-compressor unit. An expansion process and a compression process are assumed to be isentropic. The line from point A to B indicates the compression process of the sub-compressor, and the line from point B to C indicates the compression process of the main-compressor. The line from point D to E indicates the expansion process of the expander. The total efficiency of the expander-compressor unit \( \eta_t \) is defined by

\[
\eta_t = \frac{\Delta h_{sc}}{\Delta h_{ex}}
\]

where \( \Delta h_{sc} \) is the isentropic enthalpy difference of the sub-compressor, and \( \Delta h_{ex} \) is the isentropic enthalpy difference of the expander. In Figure 2, the evaporating temperature \( T_e \) is 0.8°C, the suction super-heat degree is 0°C, the heat rejection pressure \( P_h \) is 9.0MPa and the expander inlet temperature \( T_i \) is 36°C. The theoretical COP improvement of the cycle with the expander-compressor unit is shown in Figure 3. It can be seen that for the given conditions, the cycle with the expander-compressor unit has a 30% improvement in COP over the basic transcritical CO₂ cycle, while the total efficiency of the expander-compressor unit \( \eta_t \) is 50%.

3. EXPANDER-COMPRESSOR UNIT

Figure 4 shows a sectional view of the expander-compressor unit (ECU). We used a scroll type expander and a rolling-piston type rotary sub-compressor. Both elements were connected with a crankshaft, so both rotate at the same speed. The ECU design condition is a cooling rated condition. The ECU is lubricated by polyalkylene glycol (PAG) oil, which is stored in the bottom of the casing. The end of the crankshaft is under the oil. The oil pick up supplies the oil to the axial bore of the crankshaft and then feeds it to bearings and other sliding surfaces. The oil separator supplies oil to the casing oil sump (see Figure 1).

4. EXPERIMENT

The test conditions are shown in Table 2. The experiments were carried out at different refrigerant charges, ambient temperatures, and water inlet temperatures for both cooling and heating conditions. The performance of the CO₂ cycle with the ECU was compared with the performance of the CO₂ cycle without the ECU (with expansion valve).

In one testing cycle, the temperature and pressure at each component, the refrigerant mass flow rate, the compressor power input, and the rotating speed of the ECU were measured. The system capacity calculated to the amount of heat exchange from the water side. The cooling or heating capacity \( Q \) is given as

\[
Q = \rho_w \cdot C_w \cdot V_w \cdot \Delta T
\]

where \( \rho_w \) is the water density, \( C_w \) is the heat capacity of water, \( V_w \) is the volume flow rate of water, and \( \Delta T \) is the average temperature difference between the inlet and outlet water of the water/refrigerant heat exchanger.

The COP is the ratio of the cooling or heating capacity to the compressor power input, which is defined by

\[
\text{COP} = \frac{Q}{W_i}
\]

where \( W_i \) is the compressor power input.

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In experiments, the ECU was stable under both cooling and heating conditions.

5. TEST RESULTS

5.1 Effects of refrigerant charge
Figure 5 shows the effects of the refrigerant charge on system performance. Figure 5(a) is a cooling rated condition and Figure 5(b) is a heating rated condition. The relative COP, the system capacity, and the compressor power input were charted. In Figure 5(a), overall, when the refrigerant charge increases, the cooling capacity increases, but its rate of increase is gradually saturates. On the other hand, the compressor power input slightly increases, so the COP has a maximum point at a specific refrigerant charge. The optimum refrigerant charge is 6.4 kg with the ECU and 6.7 kg without the ECU. In Figure 5(b), the results of the heating rated condition are similar to but smaller than the refrigerant charge of the cooling rated condition. The optimum refrigerant charge is 3.35 kg with the ECU and 3.6 kg without the ECU. In Figure 5, the CO₂ refrigeration cycle with the ECU improved the COP by more than 30%.

5.2 Effects of inlet water temperature
Figure 6-7 shows the effects of inlet water temperature. The refrigerant charge remained at the optimum value. Figure 6 shows the relative COP at the cooling condition. Figure 7 shows the relative COP at the heating condition. The COP with the ECU is above the COP without the ECU. These results indicate that though the ECU was designed for the cooling rated condition, it improves the COP of the cycle under the different operating conditions. Figure 8 shows a P-h diagram of the CO₂ cycle with the ECU at cooling rated condition. In this figure, the line from point A to B indicates the compression process of the ECU, and the line from point D to E indicates the expansion process of the ECU. A two-dot chain line means a constant entropy S. Based on this figure, the results from Equation (1), the total efficiency γ₁ is 57%.

6. CONCLUSIONS

We have developed an expander-compressor unit and investigated the basic operating characteristics of the CO₂ refrigeration cycle. As a result, the following conclusions were obtained:
- The expander-compressor unit improves the COP of the CO₂ chiller cycle under both cooling and heating conditions.
- The COP improvement of the cycle with the expander-compressor unit was more than 30%, while the total efficiency of the expander-compressor unit was 57%.
- Expander-compressor unit are a key technology for use in CO₂ refrigeration systems.

REFERENCES

Figure 1: CO₂ air-cooled chiller cycle with expander-compressor unit

Table 1: Specifications of CO₂ cycle components

<table>
<thead>
<tr>
<th>Component</th>
<th>Type</th>
<th>Scroll</th>
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<tbody>
<tr>
<td>Main Compressor</td>
<td>Type</td>
<td>AC, 200V, 3.75kW</td>
</tr>
<tr>
<td>Expander-Compressor Unit</td>
<td>Type</td>
<td>Scroll</td>
</tr>
<tr>
<td>Expander</td>
<td>Type</td>
<td>Rolling-Piston Rotary</td>
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<tr>
<td>Sub-compressor</td>
<td>Type</td>
<td>Cross-Fin Tube</td>
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<tr>
<td>Air-side Heat Exchanger</td>
<td>Type</td>
<td>Plate and Tube</td>
</tr>
<tr>
<td>Water/Refrigerant Heat Exchanger</td>
<td>Type</td>
<td></td>
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</tbody>
</table>
Figure 2: P-h diagram of CO₂ cycle with expander-compressor unit

Figure 3: COP improvement of CO₂ cycle with expander-compressor unit
Figure 4: Cross-sectional view of expander-compressor unit

Table 2: Test conditions

<table>
<thead>
<tr>
<th>Expansion Device</th>
<th>Operating Mode</th>
<th>Charge of CO₂ [kg]</th>
<th>Ambient Temperature: $T_a [^{\circ}C]$</th>
<th>Water Inlet Temperature: $T_{wi} [^{\circ}C]$</th>
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</thead>
<tbody>
<tr>
<td>Expansion Valve</td>
<td>Cooling</td>
<td>4.2~7.0</td>
<td>35</td>
<td>12</td>
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<td></td>
<td></td>
<td>6.7</td>
<td>35</td>
<td>9.7~15.9</td>
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<tr>
<td></td>
<td>Heating</td>
<td>3.2~4.1</td>
<td>7</td>
<td>40</td>
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<tr>
<td></td>
<td></td>
<td>3.6</td>
<td>7</td>
<td>35~45</td>
</tr>
<tr>
<td>Expander-Compressor Unit</td>
<td>Cooling</td>
<td>4.5~6.4</td>
<td>35</td>
<td>12</td>
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<tr>
<td></td>
<td></td>
<td>6.4</td>
<td>35</td>
<td>9.7~15.5</td>
</tr>
<tr>
<td></td>
<td>Heating</td>
<td>2.8~3.45</td>
<td>7</td>
<td>40</td>
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<tr>
<td></td>
<td></td>
<td>3.35</td>
<td>7</td>
<td>35~45</td>
</tr>
</tbody>
</table>
Figure 5: Effect of refrigerant charge on performance
Figure 6: Effect of water inlet temperature on COP (cooling mode)

Figure 7: Effect of water inlet temperature on COP (heating mode)

Figure 8: P-h diagram of cooling rated condition