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## Development of a Scroll Expander for the CO<sub>2</sub> Refrigeration Cycle

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### ABSTRACT

In this study, a scroll expander was developed to recover the throttling loss in CO<sub>2</sub> cycles. In cycles with an intercooler, the expander drives directly the second-stage compressor using the energy recovered from the expansion process. To carry out this function, a back-to-back scroll mechanism was applied to the expansion and sub-compression processes. A prototype of the expander/sub-compressor was designed, fabricated and tested. According to the experimental results of the prototype device, the sub-compressor increased the pressure by 1.30 MPa, while the pressure drop between the inlet and outlet of the expander was 3.19 MPa. Based on the gas state at the expander inlet, the volumetric efficiency was about 104%.

### 1. INTRODUCTION

Recently, attempts have been made to use natural refrigerants during the refrigeration cycle to reduce global warming effects. Carbon dioxide (CO<sub>2</sub>) is one of the most popular (Lorentzen and Pettersen, 1993, Lorentzen, 1994, 1995), but its practical use is still limited (Hwang and Radermacher, 1998, 1999) due to its lower coefficient of performance (COP) in some operating conditions. Many studies suggest that recovering energy from the expansion process improves a cycle's COP (Robinson and Groll, 1998, Heyl and Quack, 1999, 2000). To recover energy from the expansion process, various types of expanders were studied, such as reciprocating (NickI *et al.*, 2002, Baek *et al.*, 2002, 2005), screw (Stosic *et al.*, 2002), vane (Fukuta *et al.*, 2001, 2003), and so forth. Regarding scroll expanders, both theoretical (Westphalen and Dieckmann, 2006, Kim *et al.*, 2006) and experimental (Huff *et al.*, 2003, Kohsokabe *et al.*, 2006) research has been conducted.

However, it seems that many experimental studies are carried out with an expander converted from a compressor or oil pump. In this study, a scroll-type expander combined with a sub-compressor originally designed for a CO<sub>2</sub> refrigeration cycle with an intercooler (Baek *et al.*, 2002) is investigated to see if it efficiently utilizes the recovered energy from the expansion process of the CO<sub>2</sub> refrigeration cycle.

### 2. SCROLL EXPANDER/SUB-COMPRESSOR

Figure 1 shows a schematic of a two-stage CO<sub>2</sub> refrigerating cycle with an intercooler and an expander - the cycle configuration investigated in this study. In the cycle, only the mechanical work recovered from the expansion mechanism (EX) drives directly the sub-compression mechanism (SC), which works as the second-stage compressor after the intercooler. The design aspects of the prototype expander and its experimental results are presented in this paper.

### 3. MAJOR FEATURES OF THE EXPANDER/SUB-COMPRESSOR

The expander/sub-compressor's major features are its rotating speed and stroke volume when neither bypass nor pre-expansion is done for volumetric flow rate adjustment. Although the stroke volume ratio of the expander and the sub-compressor can be obtained from operating conditions, stroke volume per second is defined at the designed rotating speed. Performance degradation due to leakage is of concern at a low rotational speed and a large stroke volume, while the large drop in pressure at the inlet and outlet ports is of concern at a high rotational speed and a small stroke volume. For the design, it is necessary to select the best rotational speed depending on real machine tests.

The volumetric expansion ratio ( $\sigma_{V_{ex}} = V_{exi}/V_{exo}$ ) is defined as the ratio of the specific volume before expansion ( $v_{exi}$ ) and after expansion ( $v_{exo}$ ). The volumetric expander/sub-compressor ratio ( $\sigma_{V_{EC}} = V_{exi}/V_{scs}$ ) is defined as the stroke volume ratio between expansion and sub-compression. Since  $\sigma_{V_{ex}}$  and  $\sigma_{V_{EC}}$  are fixed under the design conditions, the rotational speed of the expander side ( $N_{ex} = v_{exi}/V_{exi} \times [\text{flow rate of expander}]$ ) would be different from the rotational speed of the sub-compressor side ( $N_{sc} = v_{scs}/V_{sc} \times [\text{flow rate of sub-compressor}]$ ) under off-design conditions so that the device could not be driven at synchronized speed.

To deal with such a situation, a bypass valve and a pre-expansion valve were installed. In the case of  $N_{ex} > N_{sc}$ , bypassing decreases the flow rate through the expander, and in the case of  $N_{ex} < N_{sc}$ , pre-expanding increases  $v_{exi}$ . In any case, less power is recovered because either the flow rate or the pressure difference across expansion, which is related to the recovered power, decreases.

At the design point, the bypass ratio ( $x = [\text{the bypass flow rate}] / [\text{the circulating flow rate}]$ ), and the pre-expansion ratio ( $y = [\text{pressure difference of pre-expansion}] / [\text{pressure difference between the gas cooler outlet and the evaporator inlet}]$ ) become  $\sim 0$  because the volumetric expander/sub-compressor ratio ( $\sigma_{V_{EC}}$ ) has been defined from the state of the cycle without either bypass or pre-expansion. To define the operating state at off-design points for the fixed  $\sigma_{V_{EC}}$ , one has to find a combination of intermediate pressure ( $P_m$ ),  $x$ , and  $y$ , which meets the following two requirements:

- 1) The pressure increase by the subcompression is appropriate to one that can be made by the recovered power.
- 2) The rotational speed of the expander side is synchronized to that of the sub-compressor side.

An iterative calculation is necessary to find the recovered power value meeting the two above requirements while changing  $P_m$ ,  $x$ , and  $y$ .

According to the calculation at the design point, the volumetric ratio of sub-compression ( $\sigma_{V_{sc}}$ ), which is the ratio of the specific volume at the inlet and the outlet of the subcompression, is small. The internal volume ratio should be nearly 1, and discharge valves are required.

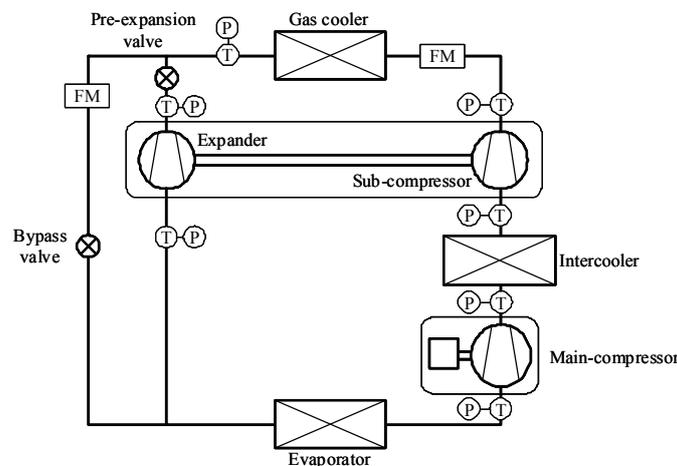


Figure 1. Schematic of a two-stage cycle with an intercooler and expander

Figure 2 shows the CO<sub>2</sub> cycle with an expander in a P-h diagram. It should be noted that the compression work needed by the main-compressor is reduced as the pressure increase of the sub-compression increases, because the slope of the isentropic compression line becomes nearly vertical when the subcompression is started in an area where the enthalpy is reduced by intercooling.

An expander stroke volume and a sub-compressor stroke volume that will satisfy the volumetric expander/sub-compressor ratio are chosen by setting the rotation speed. In addition, the most suitable expansion completion volume ( $V_{exo}$ ) can be obtained from the volumetric expansion ratio ( $\sigma_{V_{ex}}$ ), but the  $\sigma_{V_{ex}}$  may not always be satisfied due to dimension limitations.

#### 4. THE STRUCTURE OF THE PROTOTYPE

The following basic concepts are applied in the prototype:

- A dual-sided configuration of the scroll wrap (the expansion is on the bottom side, and the sub-compression is on the top side)
- An shaft penetrating the center of the orbiting scroll
- A space where the orbiting scroll moves (the space is filled with low-pressure gas after expansion)

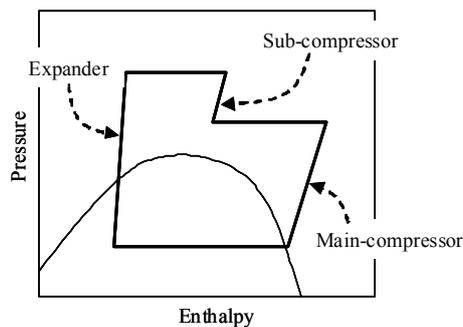


Figure 2. P-h diagram of CO<sub>2</sub> cycle with an expander

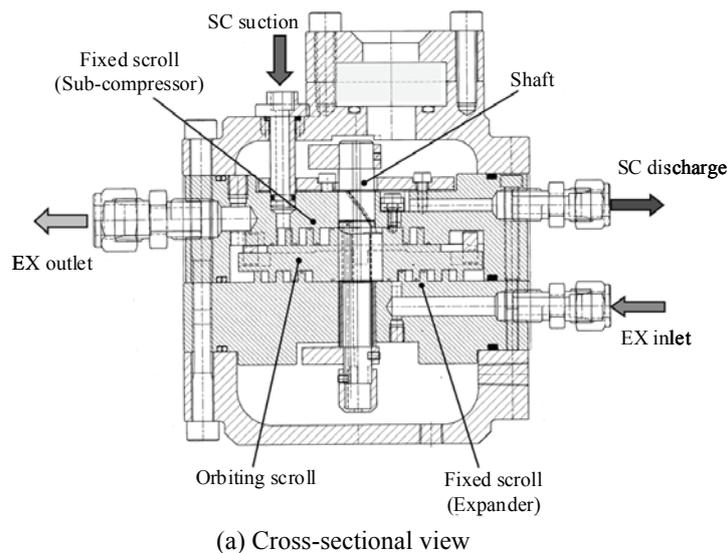


Figure 3. Structure of the prototype

The following factors are considered in applying the basic concepts described above:

- Since the scroll wraps for expansion and sub-compression are formed back-to-back upon both sides of the orbiting base plate, the orbiting radii of the expansion side and the subcompression side are the same.
- The bulb-shaped area in the large diameter is formed in the central region of the orbiting scroll, so the shaft is able to penetrate that area. This is also why the involute starts late.
- The orbiting scroll moving space (filled with evaporating pressure gas after expansion) is separated from the sub-compression side's circumference (filled with the suction pressure of sub-compression, which is equal to the intermediate pressure  $P_m$ ).

Figure 3 shows a structure of the prototype. The prototype was intended for a refrigerating cycle with an intercooler in which the sub-compressor is allocated to the second stage compression after the intercooler. The sub-compressor shares a shaft and an orbiting scroll plate with the expander. The sub-compressor is on the upper side of the orbiting scroll plate, while the expander is on the underside of the plate. Sliding loss on the orbiting scroll can be reduced because the axial gas forces acting on both sides of the plate cancel each other. The shaft penetrates the orbiting scroll and is supported by two fixed scrolls.

## 5. PERFORMANCE EVALUATION

### 5.1 The Experimental Apparatus for Performance Evaluation

Figure 1 shows a schematic diagram of the experimental apparatus used for the performance evaluation. The apparatus is equipped with an intercooler between the main-compressor driven by the motor and the sub-compressor. In addition, the intercooler and the gas cooler are water heat exchangers. By adjusting the temperature and the flow rate of water, the temperatures at the inlet of the expander and of the sub-compressor could be controlled. Additionally, bypass valve and a pre-expansion valve was installed.

The performance was evaluated by measuring the pressure, temperature, and flow rate of the CO<sub>2</sub> refrigerant. To make these measurements, pressure gauges (P), thermocouples (T) at the inlet of the expander and sub-compressor, and the mass flow meters (FM) at the inlet of the gas cooler and a bypass valve were used.

### 5.2 Definition of Efficiency

The following definitions and equations were used to evaluate the efficiency of the combined expander/sub-compressor device. The pressure ratio ( $P_r$ ) is defined as the ratio of the pressure increased by the sub-compressor ( $\Delta P_{sc}$ ) and the pressure difference between the inlet and outlet of the expander ( $\Delta P_{ex}$ ). It is shown in the following equation:

$$P_r = \frac{\Delta P_{sc}}{\Delta P_{ex}} \quad (1)$$

The volumetric efficiency of the expansion machine ( $\eta_v$ ) is defined here:

$$\eta_v = \frac{G_{ideal}}{G_r} \quad (2)$$

$$G_{ideal} = V_{exi} \cdot \rho_{exi} \cdot N_{EC} \quad (3)$$

where  $G_{ideal}$  is the ideal flow rate through the expander, which is calculated from the inlet volume of the expander ( $V_{exi}$ ), the density of refrigerant at the inlet of the expander ( $\rho_{exi}$ ), and the rotational speed ( $N_{EC}$ ).  $G_r$  is the flow rate through the expander measured by the mass flow meter. The reason why the pressure ratio ( $P_r$ ) is used is that the state of refrigerant at the expander outlet is difficult to measure.

### 5.3 Experimental Results

One of the performance evaluation results is tabulated in Table 1, and Figure 4 shows the refrigeration cycle in a P-h diagram for that case. It should be noted that the expander outlet point in the diagram is the one for the case of isentropic expansion, because its vapor quality was not measured in this experiment.

Due to circumstances related to the experimental apparatus, the experimental conditions are slightly different from those assumed when the expander was designed. In these operating conditions, the pressure drops in each heat

Table 1. Experimental result

Items		Values
Flow rate	$G_r$	329 [kg/h]
Rotational speed	$N_{EC}$	2760 [rpm]
Expander pressure difference	$\Delta P_{ex}$	3.19 [MPa]
Sub-compressor pressure increase	$\Delta P_{sc}$	1.30 [MPa]
Pressure ratio	$P_r$	0.41 [-]
Expander volumetric efficiency	$\eta_v$	104 [%]
Bypass ratio	$x$	0 [%]
Pre-expansion ratio	$y$	7.2 [%]

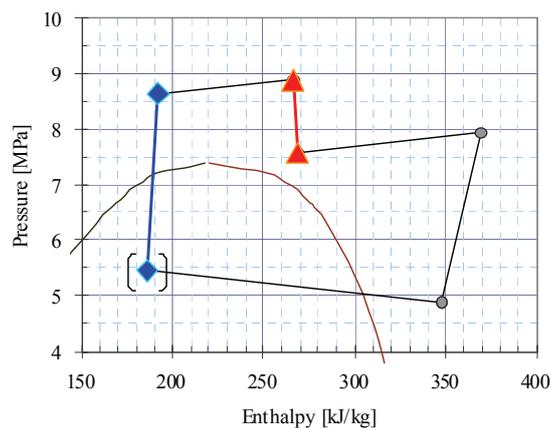


Figure 4. P-h diagram of the experimental cycle

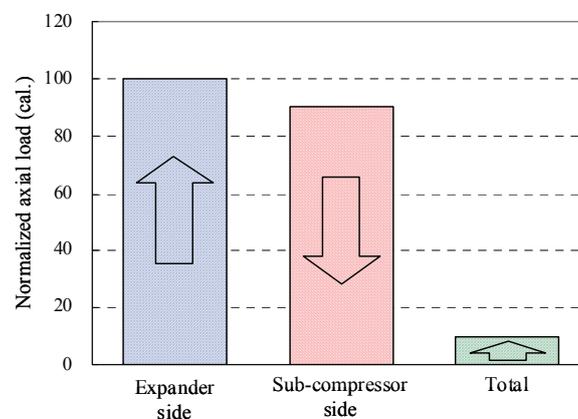


Figure 5. Normalized axial load

exchanger, and the pressure drop in the gas cooler corresponds to the pre-expansion ratio of 7.2%. In addition, enthalpy is decreased after sub-compression because heat leaks through the orbiting scroll from the sub-compressor side to the expander side. Temperature difference between the sub-compressor and the expander was about 10K or more.

The COP improvement was estimated from this P-h diagram. The case in which there is no energy recovery and no intermediate cooling is set as 100%. With intermediate cooling the cycle efficiency improvement is calculated to be 10%. With a combined intercooler and expander, the COP improvement is calculated to be about 30% on the assumption that the expansion process is isentropic. Focusing merely on the decrease of power in the main-compressor, the reduction effect by recovered energy is about 25%.

The reason why expander volumetric efficiency ( $\eta_v$ ) exceeds 100% is because of pressure loss at the inlet port and mutual, close-range interference between the fixed- and orbiting-scroll wraps (Fukuta *et al.*, 2006).

Figure 5 shows the calculated axial load acting on the dual-sided orbiting scroll under the operating conditions described above. The axial load acting on the orbiting scroll, which is calculated by subtracting the downward force of the sub-compressor side from the upward force of the expansion side, is shown after being normalized with the expansion side force as the reference value (100%). Under this condition, the axial load acting on the orbiting scroll is around one tenth of the upward force from the expansion side, because the sub-compressor side force and expander side force counteract each other. It is also thought that the sliding loss caused by the axial load is considerably reduced.

Figure 6 shows the relation between the calculated axial load acting on the orbiting scroll and Pressure ratio ( $P_r$ ) under various conditions. The Experimental results and conditions in figure 6 are  $G_r = 200\sim 300\text{kg/h}$ ,  $N_{EC} = 1800\sim 3600\text{rpm}$ ,  $\Delta P_{ex} = 1.69\sim 4.16\text{MPa}$ ,  $\Delta P_{sc} = 0.04\sim 1.62\text{MPa}$ ,  $P_r = 0.02\sim 0.42$ ,  $\eta_v = 97\sim 154\%$ ,  $x = 0\sim 39.8\%$  and  $y = 5.7\sim 34.4\%$ . Figure 6 shows that the pressure ratio increases with decreasing axial load. This is because the sliding loss increases with increasing axial load. The pressure ratio depends strongly on axial load.

## 6. CONCLUSION

In this study, a dual-sided scroll-type expander/sub-compressor combined device was designed, fabricated, and tested. Experimental results show that this design can considerably reduce sliding loss. One of the results shows the sub-compressor pressure increase ( $\Delta P_{sc}$ ) to be 1.30 MPa and the pressure ratio ( $P_r$ ) to be 0.41. Various pressure losses and heat leakage from the sub-compressor side to the expander side have been seen as factors that spoil cycle efficiency improvement through energy recovery. Therefore, it is necessary to further reduce pressure loss and heat leakage loss. However, it has been shown that the power consumed by the main-compressor can be reduced using the prototype of an expander originally designed for a CO<sub>2</sub> refrigeration cycle with an intercooler.

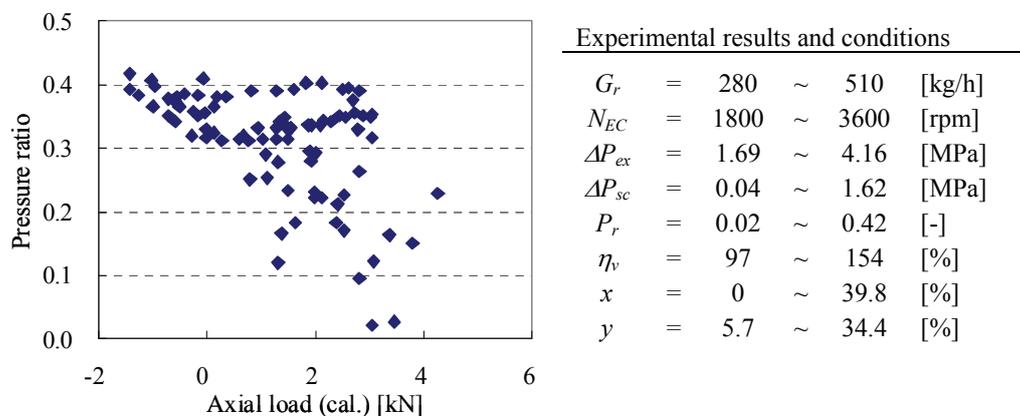


Figure 6. Experimental results under various conditions

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