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Axial and Radial Force Control for CO₂ Scroll Expander

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ABSTRACT

We made the prototype of the scroll expander for CO₂ refrigerant, which had the high-pressure vessel. We also designed the new scroll profile to make use of “over-expansion” and control the axial force on the thrust bearing. These techniques realized that the orbiting scroll never separated from the stationary scroll when the operating condition changed dramatically. As the result, the high volumetric efficiency was achieved. We also calculated the oil-film pressure of the main and eccentric bearings, considering both the dynamic behavior of the shaft and the interaction between the orbiting and stationary scroll wraps. From the calculation results, we could control the radial force between the scroll wraps, which contributed to decrease the leakage loss in expansion process.

1. INTRODUCTION

In recent years, the development and research of refrigerating and air-conditioning systems using natural refrigerants has been carried out actively instead of chlorofluorocarbon. Carbon dioxide (hereinafter CO₂), which is neither toxic nor flammable, has attracted particular attention and heat pump water heaters using CO₂ have already been commercialized in the Japanese market. Since the CO₂ cycle has large throttling loss at the expansion process, various types of expanders have been researched in order to recover this throttling loss. Under these circumstances, good characteristics are expected from the scroll type expander because it needs no control mechanism for inlet and outlet, the expansion ratio is determined by the wrap shape, and there is little torque fluctuation as multiple expansion chambers are formed simultaneously. Fukuda et al. (2006) has reported that good characteristics can be exhibited without major modifications from the CO₂ compressor. However, there are actually few cases of research on the issues and characteristics incorporating the scroll type expander into the CO₂ cycle. In this study, we have conducted analytical and experimental investigations into the particular issues for expander and show that scroll expanders have excellent characteristics.

2. PARTICULAR ISSUES FOR EXPANDER

The expansion valve serves as the control device that maintains the refrigeration cycle in the appropriate condition. If an expander is replaced with an expansion valve, the user is not only able to recover power with good efficiency, but can also demand stable performance as a control device. To put it another way, even if the expander changes the pressure difference and mass flow rate in the refrigeration cycle significantly, the refrigeration cycle still needs to be able to operate the expander with stable volumetric efficiency.

In this study, we have paid attention to the two leakage passes of the scroll expander related to volumetric efficiency – the gap in the axial direction at the tip of the wrap and the gap in the radial direction between the wraps. In regard to the gap in the axial direction, we investigated a scroll profile that makes use of “over-expansion” to ensure that the orbiting scroll never separates from the stationary scroll even when operating conditions change dramatically. In regard to the gap in the radial direction, we investigated a mechanism that uses the oil-film pressure on the shaft

bearing to force the orbiting scroll wrap onto the stationary scroll wrap. Below, we explain the technology that controls the forces in the axial and radial directions.

3. AXIAL FORCE CONTROL

3.1 Effect of over-expansion

We made the prototype of scroll expander with a seal ring between the orbiting scroll and the frame and introduced high pressure to the internal diameter side of the seal ring. Our CO₂ scroll compressor also adopted the same mechanism.

Figure 1 shows the axial force acting on the orbiting scroll comparing a compressor with an expander in the case of operation with a designed volumetric ratio and the case of operation with a lower density ratio. As shown in Fig.1, in the case of operation with the designed volumetric ratio for both the compressor and the expander, if a seal ring diameter is determined such that the resultant force of the anti-wrap side is larger than that of the wrap side, it is possible to reduce the gap in the axial direction by forcing the orbiting scroll onto the stationary scroll.

In the case of operation with a lower density ratio for the compressor, the resultant force of the wrap side is relatively larger than that of the anti-wrap side. By providing a backpressure for the relative increase of this resultant force of the wrap side, the orbiting scroll is forced onto the stationary scroll. For the expander, the resultant force of the wrap side is expected to become relatively smaller than that of the anti-wrap side because over-expansion is occurred in the expansion process.

Figure 2 shows the photo of the orbiting scrolls of the compressor and expander. We determined the volumetric ratio of the scroll expander under the rated operating condition (ambient temperature 16 deg. C., inlet water temperature 17 deg. C., and heating temperature 65 deg. C.) for our CO₂ heat pump water heater. In addition, we adjusted the wrap height so that the displacement volume was set to 0.5 cm³. We conducted an analytical examination for this scroll expander to calculate the axial force. We also mounted this prototype expander onto our CO₂ heat pump water heater to investigate the performance at start-up.

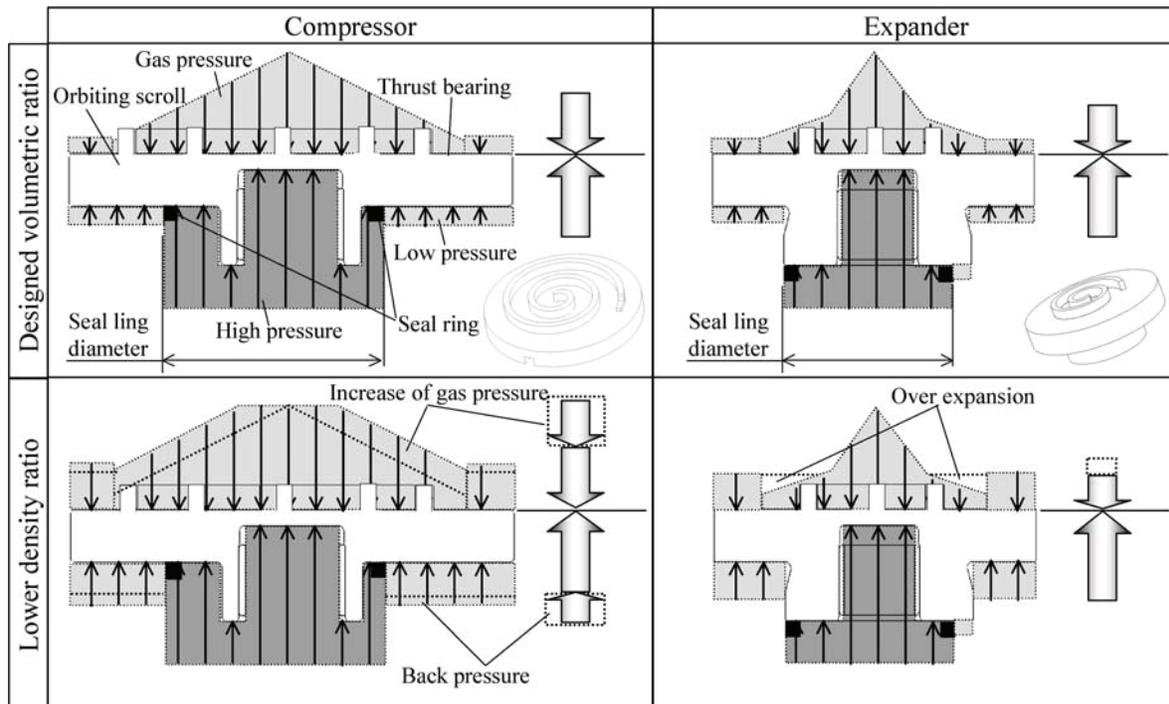


Figure 1 Axial force comparing a compressor with an expander



Figure 2 Photo of the orbiting scrolls of the compressor and expander

3.2 Calculation Results

We conducted a motion dynamics simulation of the scroll expander remodeled by that of the scroll compressor. In regard to the expansion process, we calculated the properties of pressure, density, etc., one by one for each angle of rotation using REFPROP. Figure 3 shows the axial forces acting on the orbiting scroll for each angle of rotation. If the axial force is negative, it means that the orbiting scroll separates from the stationary scroll. For the calculation conditions, we fixed the inlet pressure at 10 MPa and inlet temperature at 20 deg. C. and increased the outlet pressure from 4.0 to 7.0 MPa in 1.0 MPa steps.

As shown in Fig.3, we were able to confirm that even when the outlet pressure is increased, the resultant force of the wrap side decreases as a consequence of over-expansion and the axial force does not turn negative.

We also calculated the axial force of the scroll compressor to demonstrate the different characteristics between the compressor and expander. For the calculation conditions of the scroll compressor, we increased inlet pressure from 4.0 to 7.0 MPa in 1.0 MPa steps and fixed the super heating temperature at 10 deg. C. as the inlet temperature and outlet pressure at 10.0 MPa. Figure 4 shows the average axial force in a comparison of the compressor and expander. The horizontal axis indicates the low pressure (inlet pressure in the case of the compressor and outlet pressure in the case of the expander).

As shown in Fig.4, when the conditions are changed, the average axial force for the compressor also changes significantly. The adjusting the backpressure realized that the orbiting scroll does not separate during operation at a lower density ratio. However it caused that the average axial force becomes excessive during operation at a higher density ratio.

Meanwhile, it is found that there is little change in average axial force for the expander. Using the decline in the resultant force of the wrap side due to over-expansion prevents the separation of the orbiting scroll so it is possible to generate a stable axial force as far as operation at a lower density ratio. In other words, the scroll expander is able to use over-expansion to simultaneously keep axial force to a minimum while preventing the decrease of volumetric efficiency. We found that the scroll expander has the good characteristic to reduce the mechanical loss of thrust bearing.

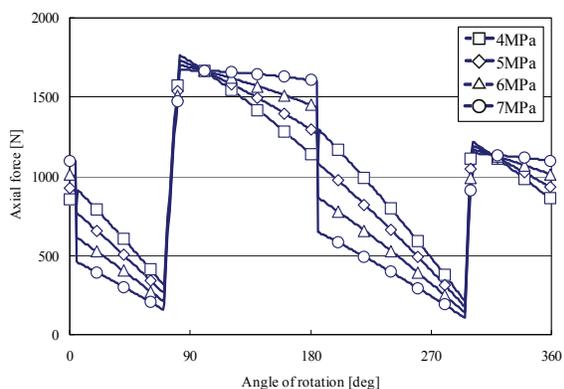


Figure 3 Axial forces for each angle of rotation

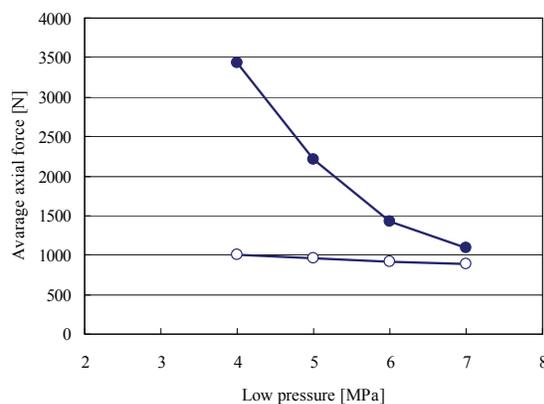
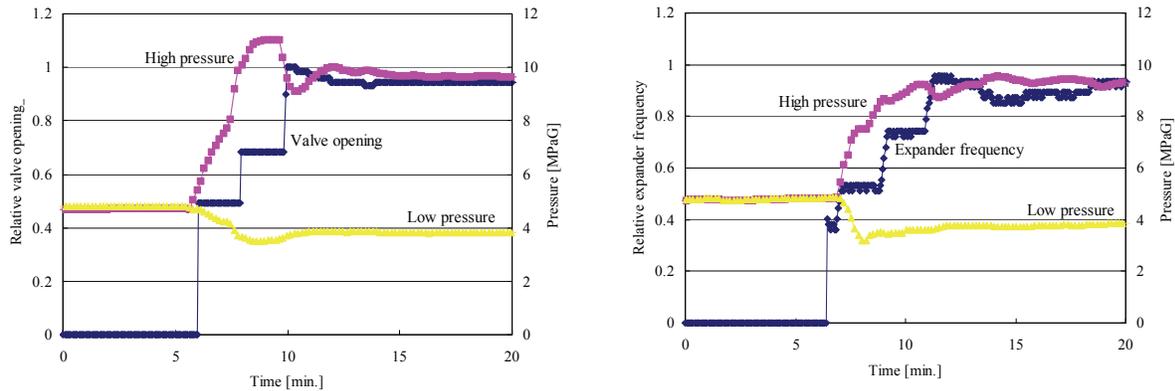


Figure 4 Average axial force

3.3 Experimental Results

Figure 5 shows the time history of high and low pressures at start-up for CO₂ heat pump water heater with the expansion valve and with the expander. The operating condition was the rated condition. As shown in Fig.5, by changing the frequency of the expander, we can control the high pressure and low pressure even if the expander replaces the expansion valve. This result indicates that the expander can be operated with stable volumetric

efficiency at start-up and there are no sudden changes of volumetric efficiency by the separation of the orbiting scroll.



(a) with the expansion valve (b) with the expander
 Figure 5 Time history of high and low pressure at start-up (a) with the expansion valve and (b) with the expander

4. RADIAL FORCE CONTROL

The shaft bearings of the scroll compressor and expander are made up of the main bearing and the eccentric bearing. Clearances exist in these bearings and a lubricant is supplied in actual operation. If these bearing clearances and the orbiting radius of the shaft are adjusted to force the orbiting scroll wrap onto the stationary scroll wrap, the gap between the wraps becomes smaller and it can be expected that there will be a reduction in leakage and an improvement of volumetric efficiency. However, excessive radial force between the wraps will invite scraping, abnormal wear and mechanical loss. Full understanding of the characteristics in the application of this radial force is required. In this section, we investigated the characteristics of the oil film pressure theoretically and confirmed the effect of the radial force control experimentally.

4.1 Oil film pressure of bearings

Figure 6 shows the definition of the forces acting on the main bearing, eccentric bearing and orbiting scroll in the case where the orbiting scroll wrap contacts stationary scroll wrap. The left of Fig.6 shows the main bearing, the middle shows the eccentric bearing and the right shows the wrap contact area. Here, the main bearing and the eccentric bearing are assembled apart from the orbiting radius r_0 . Focusing on the eccentric bearing, there are the oil film pressure forces F_{PX1} and F_{PY1} , the tangential gas pressure force F_t , the radial gas pressure force F_r , the orbiting scroll centrifugal force F_c , and the radial contact force F_w between the wraps. Focusing on the main bearing, there are the oil film reaction pressure forces R_{PX1} and R_{PY1} of the eccentric bearing, the oil film pressure forces F_{PX2} and F_{PY2} , and the centrifugal force F_{BW} of the balance weight.

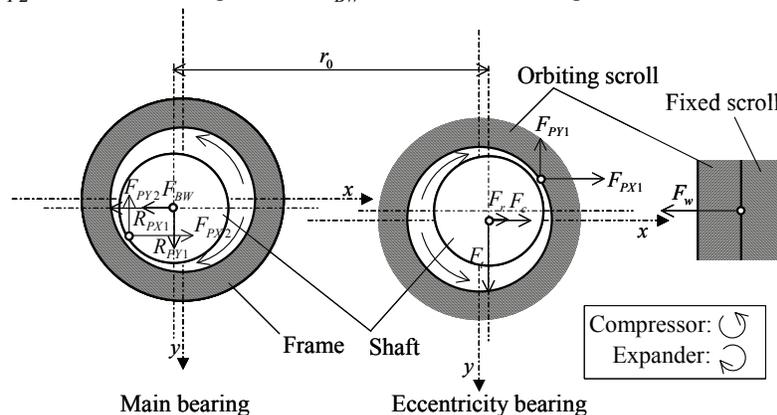


Figure 6 Definition of the forces

The rotation of the expander is clockwise and the rotation of the compressor is counter-clockwise. Only the direction of rotation differs between the compressor and expander and the definitions of the forces are the same. Figure 7 shows the oil film pressure acting on the main and eccentric bearing in detail.

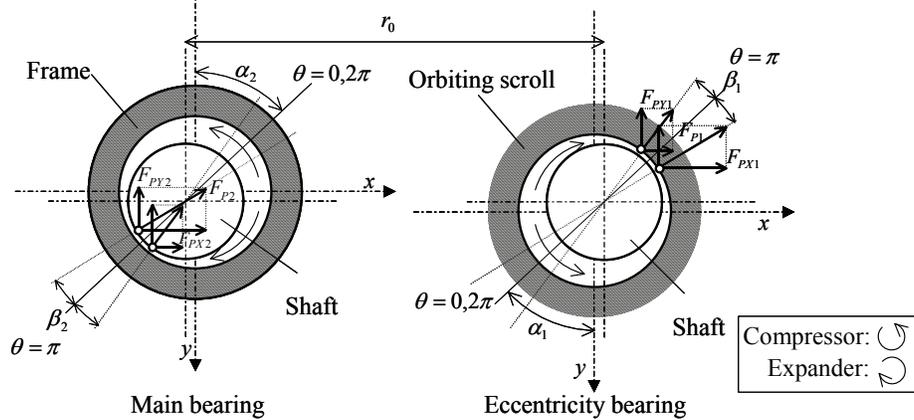


Figure 7 Oil film pressure acting on the main and eccentric bearings

Where, α_i shows the direction that will minimize the oil film thickness, represented in the following formula.

$$\alpha_i = \tan^{-1} \left(\frac{E_{Xi}}{E_{Yi}} \right) \tag{4-1}$$

Where, the suffix $i = 1$ shows the eccentric bearing, $i = 2$ shows the main bearing, E_X shows the displacement of x-axis and E_Y shows the displacement of y-axis. In addition, β_i is the angle of eccentricity and shows the angle at which the maximum oil film reaction pressure is generated. Now, assuming that the load capacity of the bearing is F_{Pi} , oil film pressure forces F_{PXi} and F_{PYi} differ for the expander and compressor. These formulas are represented as follows.

$$F_{PXi} = F_{Pi} \sin(\alpha_i + \beta_i), F_{PYi} = F_{Pi} \cos(\alpha_i + \beta_i) \tag{4-2} \text{ for the compressor}$$

$$F_{PXi} = F_{Pi} \sin(\alpha_i - \beta_i), F_{PYi} = F_{Pi} \cos(\alpha_i - \beta_i) \tag{4-3} \text{ for the expander}$$

We pay attention here to the fact that the sign for the angle of eccentricity β_i is reversed because the direction of rotation differs between the compressor and expander.

In order to calculate the distribution of oil film pressure of the bearings, we used the Sommerfeld solution based on fluid lubrication theory. For the periodic boundary condition, we adopted Gumbel's condition. In addition, we adopted Gumbel's experimental formula in regard to the effect of horizontal leakage that exists in the shaft's longitudinal direction, and considered three-dimensionality.

Lastly, we introduce the formula for the balance of the forces in the main bearing and eccentric bearing respectively. The balance of forces of the eccentric bearing is expressed in the following formulas.

$$F_c + F_r - F_w + F_{PX1} = 0, F_t - F_{PY1} = 0 \tag{4-4}$$

The balance of forces of the main bearing is expressed in the following formulas.

$$-R_{PX1} - F_{BW} + F_{PX2} = 0, R_{PY1} - F_{PY2} = 0 \tag{4-5}$$

The shaft displacement E and radial contact force F_w are numerically calculated assuming that the orbiting scroll will be constrained in the radial direction without any elastic deformation.

4.2 Calculation Results

Figure 8 shows the characteristics of the radial contact force for the expander when the bearing clearances and the orbiting radius were varied. The horizontal axis shows the average clearance of the main and eccentric bearings. The vertical axis shows the relative eccentricity that indicates the difference between the practical orbiting radius and the theoretical orbiting radius determined geometrically from the scroll profile. For the calculation condition, we assigned the rated operating condition. In addition, Figure 8 also shows the characteristics of the radial contact force for a compressor under the same conditions. As shown in Fig.8, focusing on the area that the radial contact force is larger than 0, the area of the expander is quite different from that of the compressor. In order to improve the volumetric efficiency by forcing the orbiting scroll wrap onto the fixed scroll wrap, we found that the relatively larger orbiting radius is needed for the expander.

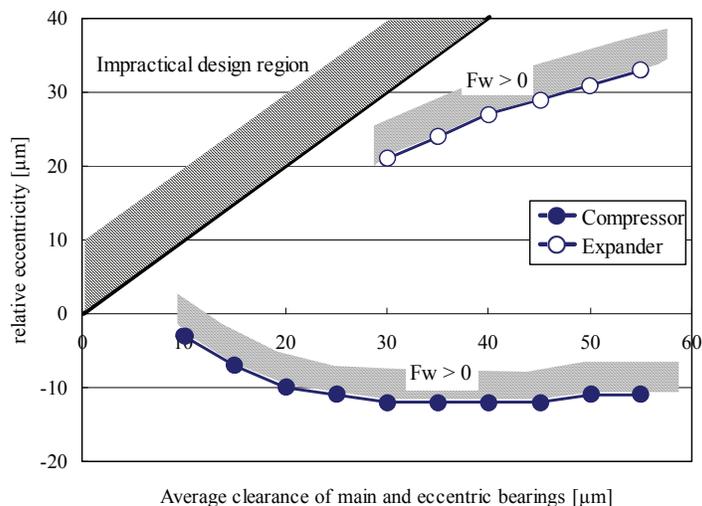


Figure 8 Characteristics of the radial contact force F_w

4.3 Experimental Results

Figure 9 shows the experimental results obtained from the measurement of volumetric efficiency for the expander when the bearing clearances and orbiting radius were varied. In addition, Figure 9 also shows the calculation results of the areas that the radial contact force is larger than 0 for the expander. As shown in Fig.9, in the vicinity of the area that the radial contact force is larger than 0, the volumetric efficiency has increased. From these results, we confirmed that the expander requires a larger orbiting radius and shaft bearing clearances in comparison to the compressor.

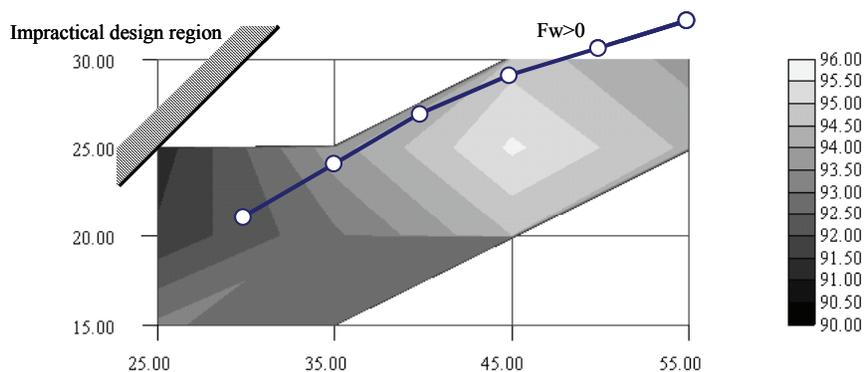


Figure 9 Volumetric efficiency when the bearing clearances and orbiting radius were varied

5. CONCLUSIONS

The authors conducted an investigation in regard to the particular issues for expanders when applying a scroll type to an expander and obtained the following conclusions:

- By using over-expansion, it is possible to prevent the separation of the orbiting scroll in a axial direction and operate the expander at a stable volumetric efficiency in a wide operating range.
- Since the axial force can be kept roughly constant even if the operating conditions change, the scroll expander has the good characteristic to reduce the mechanical loss of thrust bearing.
- As a consequence of investigating radial force considerate of the oil film pressure of the bearing, it is possible to reduce the gap in the radial direction and improve the volumetric efficiency.

The above conclusions show that the scroll type also has excellent characteristics for the use of expanders.

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