2006

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ABSTRACT

In this paper, two approaches are presented to decrease the oil discharge ratio from the CO_2 rolling piston compressor. The first approach is a numerical analysis on the oil inflow into the compression chamber. Optimizing the radial sealing line of the rolling piston and the vane groove sealing line decrease the oil discharge ratio from compression chamber to the high-pressure shell. The second approach is a numerical analysis on the mixing flow of CO_2 refrigerant and oil in the high-pressure shell. The numerical analysis revealed that there are circling flows of CO_2 refrigerant with oil mist in the high-pressure shell and that circling flows are greatly related to the oil discharge ratio from the CO_2 rolling piston compressor. According to the above-mentioned analysis, optimized dimensions of the piston and vane can be determined, and a plate to prevent circling flows of CO_2 refrigerant and a cover to separate the oil from circling flows is designed. The decrease of the oil discharge ratio from the CO_2 compressor is confirmed by the experiment. As a result, a compact highly efficient CO_2 rolling piston compressor with oil discharge ratio less than 0.1wt% has been successfully developed.

1. INTRODUCTION

In recent years, regulations against Freon-based refrigerants such as hydro-chlorofluorocarbons (HCFCs) and hydro-fluorocarbons (HFCs), which have high global-warming potential (GWP), have been tightened from the viewpoint of global warming. Consequently, much attention has been given to the development of a high-efficiency heat pump that uses CO_2, which has a low GWP and exists ubiquitously in nature. CO_2 heat pump has excellent performance in heating application. Furthermore, its coefficient of performance (COP) is 3 or more. The key device of a CO_2 heat pump is a compressor using CO_2 refrigerant. This device must be compact, efficient, and reliable.

The oil used for lubrication and cooling in the sealing and sliding portions of the compression chamber greatly influences the efficiency and reliability of the device. Consequently, it is important to maintain the oil in a stable condition in the compressor. When excessive oil is discharged from the shell with the flow of refrigerant, the oil in the compressor decreases to an unacceptably low level, which degrades the compressor's reliability. In addition, the performance of a CO_2 heat pump system generally deteriorates with an increased oil circulation ratio (OCR). This is because heat transfer is hindered when a portion of the circulating oil attaches to the wall of the tube in the heat exchanger. The heat transfer coefficient begins to decrease when OCR exceeds about 1 wt% [(1)]. Accordingly, it is important to reduce the oil discharge ratio from the compressor.

In this study, countermeasures were devised to achieve reduced the oil discharge ratio based on numerical analysis. The effect of our approach was confirmed by experimental examination. This report also describes the development of a rolling piston compressor with CO_2 refrigerant that largely reduces the oil discharge ratio.

2. Structure of Rolling Piston Compressor Using CO_2 Refrigerant and Procedure for Reducing Oil Discharge Ratio

2.1 Basic Structure of CO_2 Rolling Piston Compressor

Figure 1 shows a cross-section of the CO_2 rolling piston compressor. The motor assembly is at the center of a shell, and the compression mechanism, a one-cylinder rolling piston, is at the bottom. The major specifications of the compressor are shown in Table 1.

The refrigerant is compressed in the compression chamber and then discharged through the discharge port to the area under the rotor in a shell. After that, the refrigerant is transferred to the upper part of the shell through the gap between the rotor and stator or between the stator and shell, and then it is discharged through a discharge pipe installed in the upper part of the sell. The inside of the shell is filled with compressed high-pressure CO_2 refrigerant. There is an oil sump at the bottom, and a portion of the accumulated oil lubricates the journal bearing, flows out from the upper part of the bearing, and returns along the upper surface of the bearing to the sump. Since the inside pressure of the compression chamber during suction and compression is lower than that of the shell, oil in the sump is supplied to the compression chamber through a sliding gap, thus lubricating, sealing, and cooling the sliding portion of the compression chamber. Oil supplied to the compression chamber is mixed with CO_2 refrigerant and discharged to the interior of the shell, and then most of the oil is separated from the CO_2 refrigerant and returned to the oil sump. A part of the oil, however, is not separated but flows out along with the CO_2 refrigerant flow to the heat pump system through an discharge pipe in the upper part of the shell.

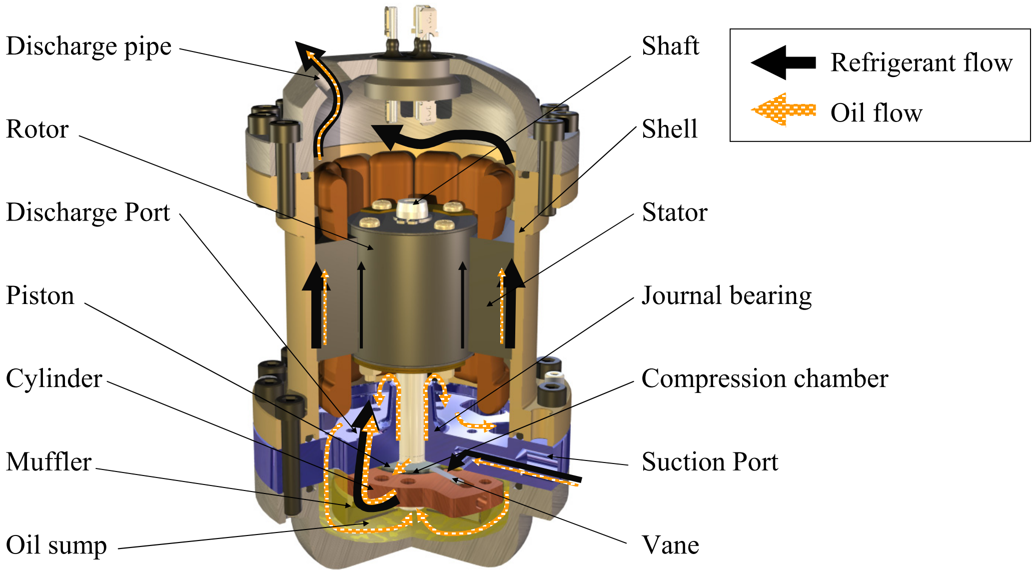

Figure 1: Cross-section of CO_2 rolling piston compressor

Table 1: Compressor major specification of CO_2 compressor

Shell diameter	Shell height	Suction volume	Motor	Refrigerant
88 mm	180 mm	1.3 cm^3	DC brushless	CO_2

2.2 Procedure for Reducing Oil Discharge Ratio

The compression performance of a prototype was evaluated, and consequently a good result was achieved with a target efficiency of 55%. However, the quantity of oil discharge was as much as 3.60 wt% for mass flow rate of refrigerant circulation. To prevent a large decrease in system performance, it is necessary to reduce the oil discharge ratio from the shell of the compressor to 1 wt% or less [(1)].

To overcome this problem, we have devised the following two countermeasures to reduce oil discharge ratio:

(1) Reduce the oil discharge ratio from the compression chamber to the shell by preventing the flow of excessive oil into the compression chamber.

(2) Reduce the oil discharge ratio from the shell to the heat pump system by separating the oil from the CO_2 refrigerant in the shell.

By taking these countermeasures, the oil discharge ratio was reduced (OCR target: 0.1 wt% or less). Details of these countermeasures are described below.

3. Reduction of Oil Inflow to Compression Chamber

In a CO_2 heat pump system, the highest pressure is 10 MPa or more and the lowest pressure is 4 MPa or less. It is assumed that this pressure difference, which is larger than that using conventional refrigerant, causes excess oil supply in the compression chamber and thus increases the oil discharge from the compression chamber to the shell.

Figure 2 shows the oil flow path to the compression chamber. The oil flows into the compression chamber mainly along the upper and lower end surfaces of the piston and the vane-sliding surface. High-pressure oil present on the inner surface of the piston and on the back surface of the vane causes the oil to flow into the compression chamber during suction and compression. Although this excess oil flow into the compression chamber can be prevented by designing each surface sufficiently wide, when the diameter of the shell is not changed the vane seal length L_V becomes shorter with an increase in the piston seal length L_P. Accordingly, the oil flow into the compression chamber was calculated with the variable of piston seal length, and the results shown in Fig. 3 were obtained. As the figure shows, although the oil flow from the upper and lower surfaces of the piston into the compression chamber decreases with the increase in L_P, the oil flow from the vane groove surface increases, since L_V decreases with the increase in L_P. From these relations, it can be concluded that the total oil inflow is the least when it proceeds from the upper and lower surfaces of piston $L_P = L_P$opt and from the vane groove surface $L_V = L_V$opt.

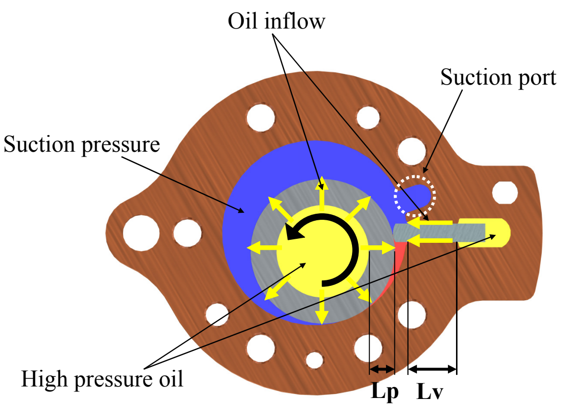

Figure 2: Oil inflow path to compression chamber

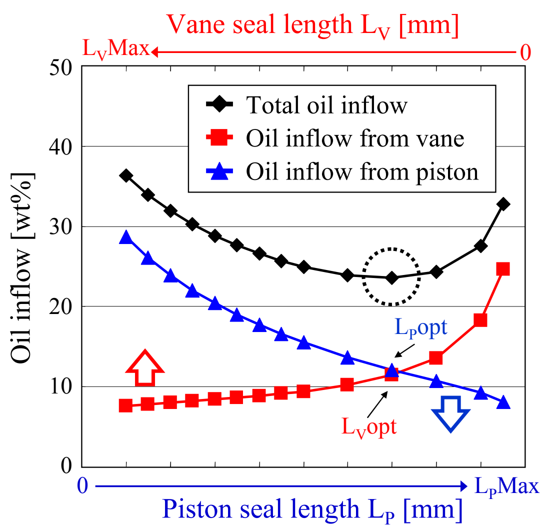

Figure 3: Relationship between each seal length and oil inflow

However, since the vane divides the compression chamber into a high-pressure space and a low-pressure space, there is always a differential pressure around the top of the vane. In the model shown in Figure 4, the differential pressure applied to the vane stress concentration at the area surrounded by the broken red line. Because this stress concentration increases when the vane groove L_V is shortened, it is undesirable to design a short vane groove L_V from the viewpoint of the groove's reliability. By considering reliability in design, it was found that the vane seal length L_V should be slightly longer than $L_V$opt. Therefore, we decided to employ this length as the optimum design value.

An experimental examination using the above optimum design showed that the oil discharge ratio was reduced by 0.4 wt%.

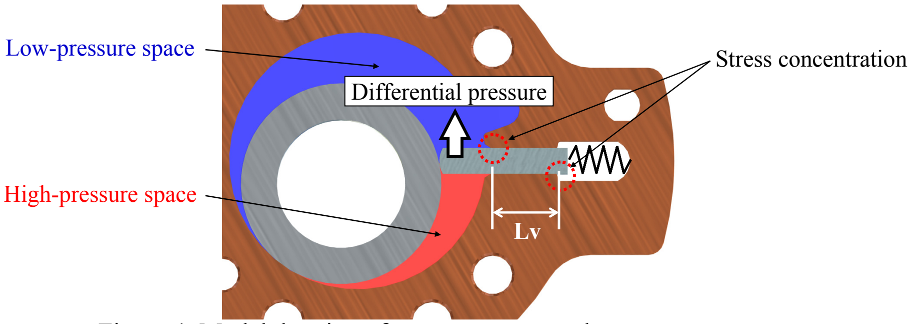

Figure 4: Model drawing of stress concentrated at vane groove

4. Acceleration of Oil Separation in Shell

4.1 Examination of Oil Separation Mechanism in Shell

The CO_2 refrigerant discharged from the compression chamber to the area under the rotor in the shell passes mainly through a gap between the stator and the shell to an upper space in the shell, from where it flows out through the discharge pipe. To separate the oil efficiently in the shell and reduce the oil discharge, it is necessary to study the flow field inside the shell. Thus, we first analyzed this for the single phase of CO_2 refrigerant by using a fluid analysis tool, FLUENT 6.0, and the κ-ε model for turbulent flow. Figure 5 and Table 2 show the calculation region and the calculation conditions, respectively.

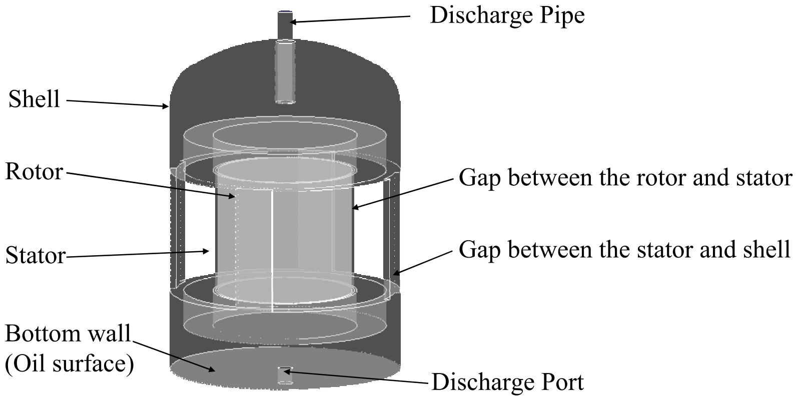

Figure 5: Calculation region

Table 2: Calculation conditions

Inlet condition					
Pressure	Temperature	Density	Viscosity	Mass flow rate	Rotor frequency
10.4 MPa	88 ℃	217.9 kg/m^3	22.85×10^{-6} Pa·s	46.0 kg/h	120 Hz

The streamline analysis results of CO_2 refrigerant inside the shell is shown in Figure 6. These results revealed that the motor rotation stirs CO_2 refrigerant, forming a circling flow in the upper and lower spaces of the rotor.

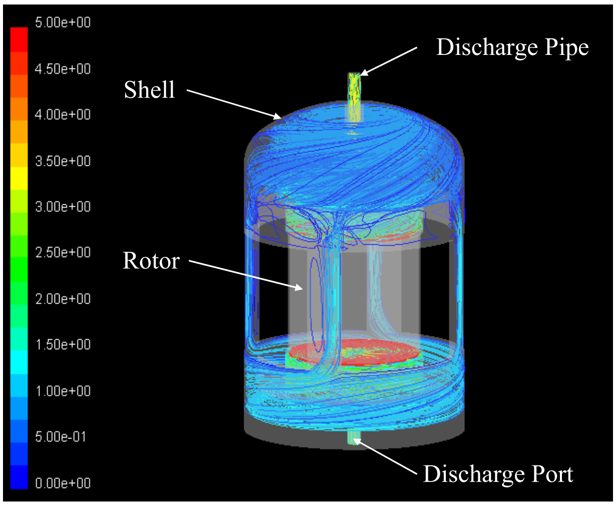

Figure 6: Analysis results of CO_2 flow in shell

The oil flow velocity distribution in the shell was analyzed next. The Dispersed Phase Model (DPM) was adopted as a multi-phase flow model so that in the calculation the oil would be regarded as a small particle mixed with CO_2 refrigerant. The calculation region is the same as shown in Figure 5, and the analysis tool is FLUENT6.0.

To examine the relationship between the circling flow velocity and the oil flow, we compared two cases: when the rotor rotates at 120 Hz and when the bottom surface of the rotor is stopped (the upper surface and the side surface rotate at 120 Hz). The results are shown in Figure 7.

When the rotor rotated at 120 Hz, the circling flow generated by the rotor was stronger and the influence of oil flow velocity was observed to be lager compared with the case when the bottom surface of the rotor was stopped. It was also observed that the oil streamlines were concentrated in the upper part of the space and moved toward the upper part of the shell. On the other hand, when the lower part of the rotor was stopped, the oil flow was slow and the oil streamlines were observed to spread over the entire space. This is because stopping the bottom surface of the rotor weakens the circling flow of the CO_2 refrigerant at the bottom of the rotor as well as the oil particle flow, and thus the oil particles tend more to move downward by the gravity spreading throughout the entire space, not limited to the upper part (the oil density is around five times larger than CO_2 refrigerant).

This finding suggests that it is possible to adjust the separation efficiency of the CO_2 refrigerant and oil in the lower space of the shell by changing the intensity of the circling flow.

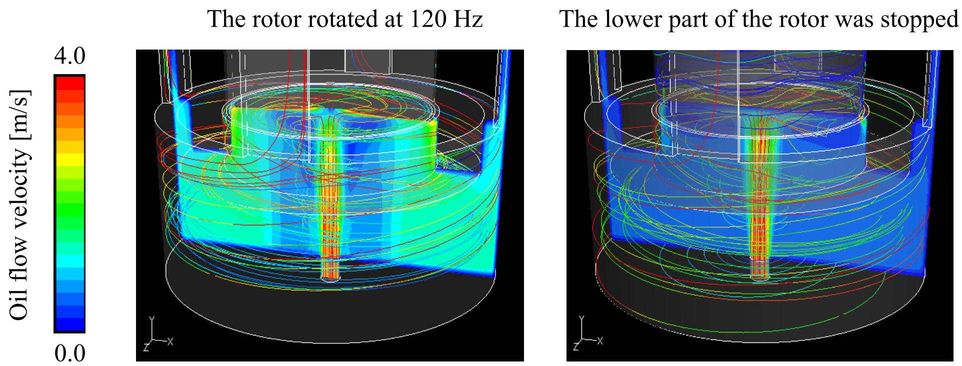

Figure 7: Oil flow rate distribution and oil streamline in lower space of shell.

4.2 Restraining Circling Flow

From the results of the above flow analysis, we devised a structure that employs a circling-flow-prevention wall at the bottom end of the rotor and examined its effect. Figure 8 shows the analysis results of oil concentration distribution when the rotor rotates at 120 Hz and when the bottom surface of the rotor is stopped (the latter case is the model in which a circling-flow-prevention wall is installed at the bottom of the motor). The calculation region and conditions are the same as those for the analysis shown in Fig. 7. The oil flow moving through the stator gap to the upper space of the shell was 74.3% when the rotor turns at 120 Hz and 42.7% when the bottom end of the rotor is stopped. Consequently, we analytically confirmed that the circling-flow-prevention wall is effective in reducing the amount of oil discharge to the upper part of the shell.

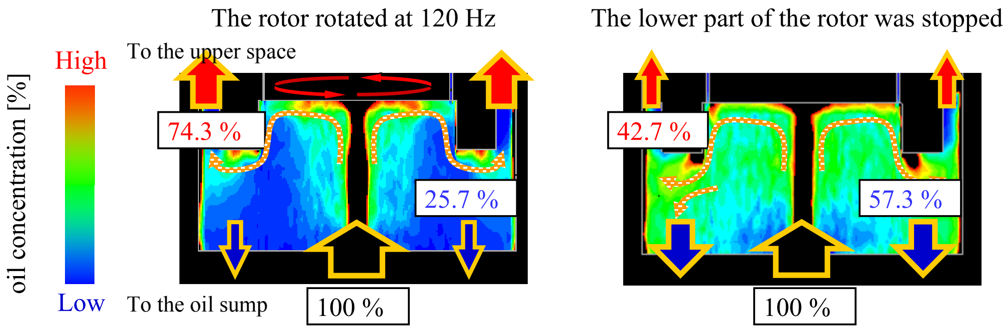

Figure 8: Influence of oil concentration distribution by circling flow

In an experiment to confirm the effect of the circling-flow-prevention wall, the oil discharge ratio was reduced to 0.12 wt%. Therefore, by separating the refrigerant and oil, the oil discharge ratio could be successfully reduced.

4.3 Journal Bearing Cover

Another method of refrigerant/oil separation is to prevent the oil around the upper surface of the journal bearing from being lifted by the circling flow.

Figure 9 illustrates the lubrication oil discharge from the journal bearing. As shown in the figure, the oil used for journal bearing lubrication flows through the gap between the bearing and shaft and then flows out from the upper part of the bearing. The oil makes contact with the flow of CO_2 refrigerant at the lower part of the rotor, being lifted by the circling flow. This can be a cause of increased oil discharge.

Therefore, a cover was installed on the journal bearing as shown in Figure 10 to prevent the oil after lubrication from making contact with the circling flow. This accelerates the oil separation. An experimental examination achieved oil discharge ratio of 0.08 wt%.

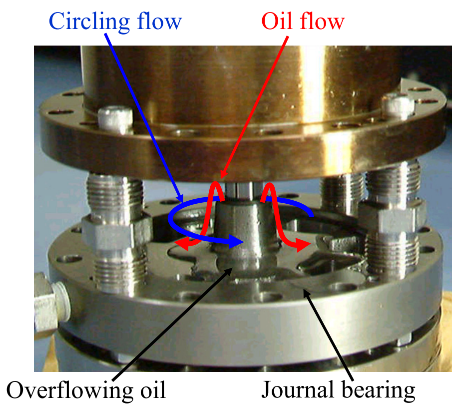

Figure 9: Flow of journal bearing lubrication oil

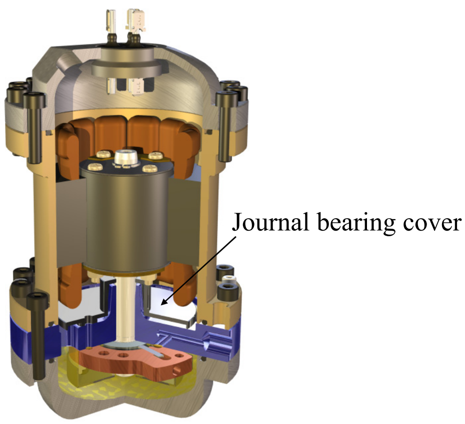

Figure 10: Cross-section of CO_2 rolling piston compressor with journal bearing cover

5. Performance Evaluation of Oil Discharge Ratio Reduction Specifications

We made a prototype compressor with optimized piston seal length and a CO_2 refrigerant/oil separation mechanism for restraining the circling flow by the rotor, and we evaluated the performance. The evaluation results are shown in Table 3. This approach not only sufficiently reduced the oil discharge ratio but also improved the compressor efficiency. The reasons for improved compressor efficiency are considered as follows: (1) reduced heat loss due to the oil by optimizing the oil quantity flowing into the compressor chamber, (2) improved volume efficiency by reducing the excessive oil in the compressor chamber, and (3) reduced oil film on the wall of the discharge port's flow path achieved by reducing excessive oil, thus reducing the pressure loss.

Table 3: Performance evaluation results

	Compression efficiency	Oil Circulation Ratio
Before measures	55.0 %	3.60 wt%
After measures	60.0 %	0.08 wt%

6. CONCLUSIONS

The oil discharged ratio from a compact rolling piston compressor using CO_2 refrigerant was reduced by taking the following two measures.

(1) Oil discharge ratio from the compressor was reduced to 0.4 wt% by targeting the reduction of oil flowing into the compression chamber and using the optimum design of the seal length on the upper and lower surfaces of the piston as well as the seal length of the vane groove.

(2) Oil discharge ratio was reduced to 0.08 wt% by installing a circling-flow-prevention plate to improve the separation of CO_2 refrigerant and oil due to the circling flow in the shell and by installing a journal bearing cover to separate the refrigerant flow and journal bearing lubrication oil.

By taking these measures, we have succeeded in reducing the discharged oil ratio from the compressor and, consequently, developed the prototype of a compact rolling piston CO_2 compressor with oil discharge ratio of 0.1 wt% or less.

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