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A Study of a New Oil Injection to Compression Chambers on Scroll Compressors

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ABSTRACT

In scroll compressors, the leakage through the clearance has a large influence on performance. To improve the efficiency of scroll compressor by reducing the leakage, we carried out experiments of the oil injection in compression chambers and examined the compressor performance. As results of experiments, it became clear that the oil flow rates in which compression efficiency became highest were different between R410A scroll compressor used in air-conditioner and CO_2 scroll compressor used in heat pump water heater. Then, we newly designed an intermittent oil supply mechanism to control the oil flow rate to the compression chambers, and we made clear the oil flow characteristics of this mechanism by the theoretical and the experimental study. Using this intermittent oil supply mechanism, the performance of each compressor was improved. In addition, deriving a "leakage index" of the compressor by the dimensional analysis, there was good correlation between "leakage index" and the optimal oil injection rate on scroll compressors.

1. INTRODUCTION

Recently, environmental protection has been one of the most serious issues in the worlds. Especially, the control of global warming is the most urgent demand from the society. The Kyoto Protocol, adopted in 1997 at the third Conference of the Parties (COP3) of the United Nations Framework Convention on Climate Change, binds developed countries to reduce emissions of carbon dioxide (CO_2) and other greenhouse gasses. It requires Japan to cut 6% by 2012, compared to the figure in 1990.

One of the technologies helpful to reduce CO_2 emissions is heat pump. The development and commercialization of high efficiency heat pumps have been demanded to increase energy efficiency of air-conditioners, heaters, and hot water supply. In each home in Japan, energy used for hot water supply accounts for as high as approximately 30% of the total energy consumption. In such environment, aimed at reducing energy consumption for hot water supply, a CO_2 heat pump water heater for home use was developed and put on the market in 2001[1)]. This water heater, using CO_2 refrigerant in the heat pump cycle, is driven mostly by midnight electric power and absorbs heat from air to boil water. It contributes to reducing temporal changes of the electric power during a day and reducing CO_2 emissions.

The displacement volume of CO_2 compressor is smaller than half that of R410A compressor for air-conditioners, because CO_2 refrigerant has characteristics of high pressure and high density. In addition, pressure difference between the compression chambers becomes several times higher. As a result, the CO_2 compressor would work in low efficiency due to leakage of the refrigerant and due to the mechanical loss because of larger contact force of the sliding parts, and in worse cases like abnormal wear or damage. To overcome such problems, a large number of research and development about CO_2 rotary or scroll compressors with high efficiency and high reliability have been carried out and reported in recent years [2-4)]. While study on the leakage characteristics of CO_2 refrigerant was reported by Ishii and others [5)], however, no study has been reported about decreasing leakage by supplying the oil to the compression chambers of the CO_2 compressor.

We introduced an intermittent oil supply mechanism to supply to the compression chambers of R410A scroll compressors with an optimal amount of the oil, and reported that the compressor efficiency would increase [6]. Then, this study aims to apply that intermittent oil supply mechanism to the CO_2 scroll compressors and control the oil flow rate to the compression chambers to increase efficiency [7]. Specifically, we investigated (1) whether to apply the equation used for the oil flow rate of the intermittent oil supply mechanism on R410A scroll compressors to that of CO_2 scroll compressors, (2) the relationship between the oil flow rate to the compression chambers and the performance in a CO_2 scroll compressor, and (3) the derivation of an leakage index that closely correlates with the oil flow rate at the peak of compressor efficiency.

2. CO_2 SCROLL COMPRESOR

2.1 Structure of CO_2 scroll compressor

Specifications of the CO_2 scroll compressor used in this study are shown in Table 1. The CO_2 scroll compressor presents cooling capacity of 3,200 W at rated conditions. Fig.1 shows a cross section of the CO_2 scroll compressor. It has a high pressure shell, discharge pressure acts inside of the hermetic vessel, which contains the compression mechanism in its upper side and the motor in its lower side. The compression mechanism consists of a fixed scroll, an orbiting scroll, a crank shaft that drives the orbiting scroll by transmitting the rotary motion of the motor, a frame that supports the crank shaft, Oldham's ring that prevents the orbiting scroll from self-rotation. Between the backside of the orbiting scroll and the fame lies a seal ring. High pressure, equal to discharge pressure, acts on its inside, and medium pressure p_m between discharge pressure and suction pressure acts on its outside.

2.2 Refrigerant flow and oil path

Refrigerant gas is compressed in the compression chambers, passes through the discharge muffler, and is discharged to the hermetic vessel. Then, the high pressure refrigerant gas goes to the lower side of the vessel and cool the motor, moves back to the upper side of the vessel, and is sent to the heat pump cycle through the discharge pipe. Meanwhile, oil stored at the bottom of the vessel is pumped up by the volumetric oil pump, passes through the inside of the crank shaft, and is supplied to the compression mechanism. The oil lubricates the eccentric bearing and main bearing in turn, and moves back to the bottom of the vessel. The oil supplied to the compression mechanism is partially supplied to the outer space of the seal ring by the flow control of the intermittent oil supply mechanism (Fig. 2) , which consists of the oil passage inside the end plate of the orbiting scroll and the seal ring.

Table 1 Specifications of CO_2 scroll compressor.

Refrigerant	CO_2
Lubricants	PAG（VG100）
Shell type	High pressure vessel
Scroll wrap	Involutes
Displacement volume : V_c	4.0 cm³/rev
Outward dimensions	φ 124 × H284 mm
Motor	Blush-less DC motor
Operating speed	30 ～ 90 1/s
Cooling capacity at rated condition A	3,200 W

Cond. A : p_d =10.1 MPa , p_s = 4.1 MPa , T_s = 18 ℃

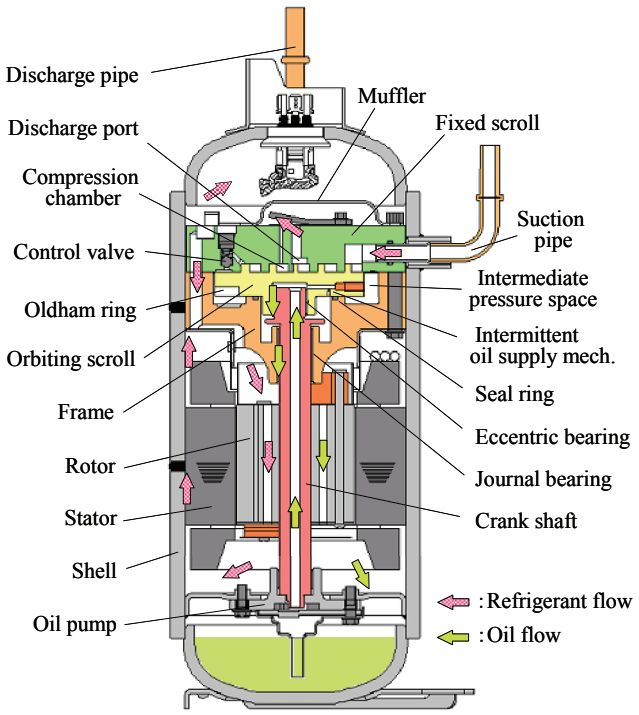


Fig.1 Cross section of CO_2 scroll compressor.

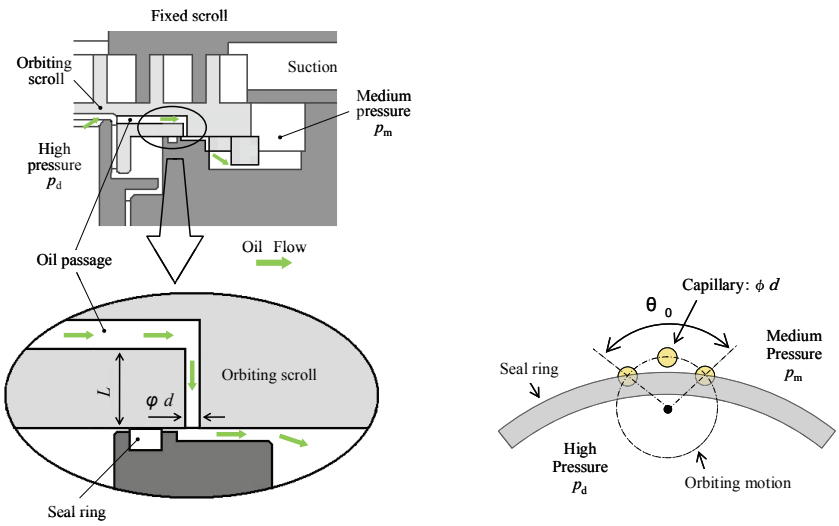
And then, the oil is supplied to the suction process in the compression chambers (hereafter "the compression chambers") through the control valve of the back pressure, lubricating and sealing the compression chamber. During this process, the pressure of the oil in the outer space of the seal ring is adjusted to medium pressure p_m by the control valve. The orbiting scroll is pushed to the fixed scroll by resultant force of the high pressure, which acts on the inside of the seal ring, and the medium pressure, which acts on the outside of the seal ring. As a result, the clearance between the scrolls when the compressor is in operation is maintained at several micrometers or shorter.

3. THE INFLUENCE OF THE OIL INJECTION ON THE COMPRESSOR PERFORMANCE

Generally, scroll compressors with small displacement volume are subjected to more significant influence of leakage trough the clearances between the wraps, thus resulting in low efficiency. Specifically, in the case of scroll compressors for CO_2 refrigerant, the volume per refrigerating capacity is approximately one-third compared to those for R410A refrigerant while pressure increases three times. Therefore, the influence of leakage on scroll compressors is significant. We have reported already that R410A scroll compressor become more efficient if the oil flow rate to the compression chambers is reduced to an extent where lubrication is properly performed, and an intermittent oil supply mechanism is effective in controlling the oil flow rate stably[7]. Then, one of the measures to reduce the leakage on CO_2 scroll compressor is optimization of the oil flow rate to the compression chambers, and an optimal oil flow rate may be different from that for R410A scroll compressor. In this respect, we studied the relationship between the oil flow rate and the compressor performance by applying the intermittent oil supply mechanism to CO_2 scroll compressor.

3.1 Intermittent oil supply mechanism

The intermittent oil supply mechanism uses orbiting motion of the orbiting scroll to limit the time for oil flowing from the high pressure (p_d) region to the medium pressure (p_m) so that the oil flow rate to the compression chambers can be controlled. Fig.2 shows the structure of the intermittent oil supply mechanism. The end plate of the orbiting scroll is equipped with an oil passage that connects the high pressure region at the center of its back side to the medium pressure region around the circumference of the back side. The capillary of the back side moves between the high pressure region and the medium pressure region that are divided by the seal ring. When the capillary reaches the medium pressure region (with its opening angle at θ_0), oil flows into the medium pressure region. We investigated how to estimate the oil flow rate in the intermittent oil supply mechanism with R410A scroll compressor, reporting that it is calculated using Equation(1), which adopts a flow that happens when the outlet valve of the passage at the deep bottom of the liquid tank is suddenly opened[8]. Then, we suppose that Equation (1) can be applied to CO_2 scroll compressor.



(a) Cross section. (b) Orbiting motion of the capillary.

Fig. 2 Intermittent oil supply mechanism

$$G_{oil} = C_{oil} \frac{\pi \rho_{oil} d^2 L \cdot f}{2} \ln\left\{\cosh\left(\frac{1}{L}\sqrt{\frac{\Delta p}{2\rho_{oil}}} \cdot \frac{\theta_o}{360} \cdot \frac{1}{f}\right)\right\} \quad (1)$$

$$\Delta p = p_d - p_m \quad (2)$$

Where, C_{oil} and Δp show a oil flow coefficient and pressure difference between inlet and outlet of the intermittent oil supply mechanism, respectively.

3.2 Verification of oil flow equation

We verified Equation (1) by calculating C_{oil} as follows, we equipped the compression mechanism with the intermittent oil supply mechanism to measure the amount of oil that flowed into the compression chambers via the intermittent oil supply mechanism, and calculated the oil flow coefficient C_{oil} of the CO_2 scroll compressor.

Fig.3 shows how to measure the oil flow rate. The CO_2 scroll compressor has two circulation passages of oil as shown in Fig.3(a). In other words, oil pumped up by the oil pump passes through Passage (1) of the crank shaft. It separates into two passages at the top end of the crank shaft. The main stream, passing through Passage (2), lubricates the eccentric and main bearings and returns to the bottom of the vessel. The other stream, passing through Passage (3), flows into the compression chambers via the intermittent oil supply mechanism, the back pressure chamber and the control valve of the back pressure. This structure makes it difficult to measure only the oil flow rate of the intermittent oil supply mechanism (Passage (3)). To make it possible, we changed the oil circulation path of the entire compressor as illustrated in Fig.3(b). Specifically, we removed the oil pump and integrated the two oil passages into one (Passage (4)), so that oil can flow from the bottom end of the crank shaft to its inside while lubricating the main and eccentric bearings, and then flow into the compression chambers via the intermittent oil supply mechanism, back pressure chamber and the control valve of the back pressure. To supply the oil, Passage (4) uses pressure difference between discharge pressure and suction pressure. We also changed the oil passage, which connected the bottom of the vessel to the bottom end of the crank shaft, to the outside of the vessel, and placed a Coriolis mass flowmeter to measure the oil flow rate in this external oil passage.

We connected the compressor, which has the new oil circulation path (Fig.3(b)), to the compressor calorimeter and measured the oil flow rate while changing operating conditions. Specifications of the intermittent oil supply mechanism and conditions used in this experiment are shown in Table 2 and 3, respectively. The oil flow rate was measured at nine conditions as follows, Condition 1, four sets of conditions (Condition 2–5) with varied discharge pressure p_d around Condition 1, and the other four sets of conditions (Condition 6–9) with varied rotational speed f.

We substituted the measured oil flow rate in the left side of Equation (1), and substituted the each value determined by the specifications of the intermittent oil supply mechanism and the test conditions in the right side of Equation (1), thus calculating the oil flow coefficient C_{oil}, which is shown in Fig. 4. In this figure, oil flow coefficients were nearly constant, not depending on the test conditions, and were 0.35 on average. Accordingly, it was confirmed that Equation (1) can be also used for estimate of the oil flow rate through the intermittent oil supply mechanism in CO_2 scroll compressor. The oil flow coefficient of R410A scroll compressor is 0.31[6], indicating that oil flow coefficients were approximately same for different refrigerants.

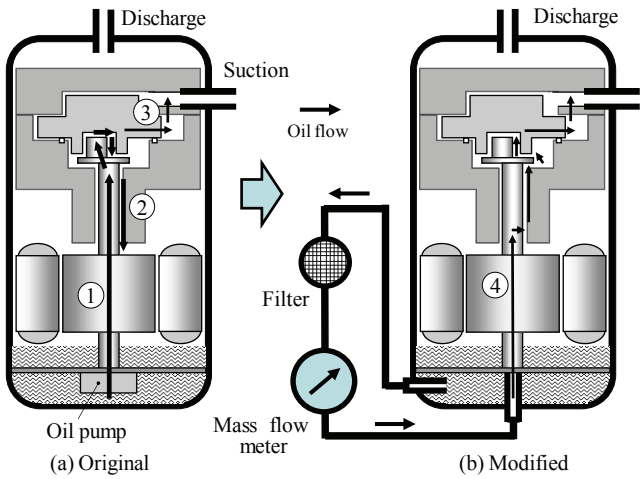


Fig.3 Measurement of oil flow rate through intermittent oil supply mechanism

Table 2 Specifications of the intermittent oil supply mechanism (1)

Diameter of capillary d [mm]	Length of capillary L [mm]	Open angle θ_0 [deg]
0.45	3.5	86

Table 3 Test conditions for CO_2 intermittent oil supply mechanism

Condition	p_d [MPa]	p_s [MPa]	Δp [MPa]	f [1/s]
1	10.1	1.6	7.6	60
2	11.1	1.6	8.7	60
3	12.1	1.6	9.7	60
4	13.1	1.6	10.6	60
5	13.7	1.6	11.2	60
6	10.1	1.6	7.6	39
7	10.1	1.6	7.6	81
8	10.2	1.6	7.6	100
9	10.1	1.6	7.6	121

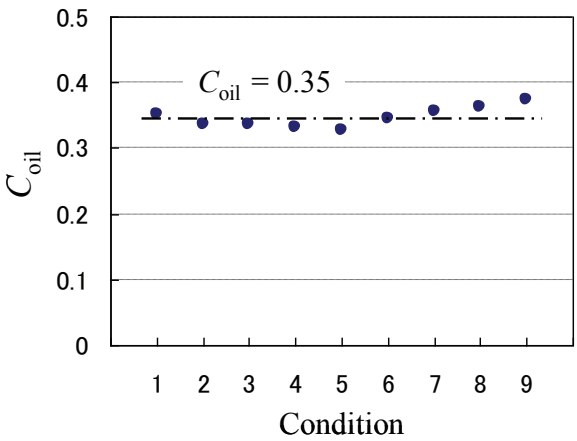


Fig.4 Oil flow coefficient on CO_2 scroll

3.3 Influence of oil flow rate on performance

3.3.1 Specifications of the intermittent oil supply mechanism and test conditions

We prepared seven types of the intermittent oil supply mechanism with different specifications to install them in the CO_2 scroll compressor (Fig.1), and investigated the relationship between the oil flow rate to the compression chambers and the compressor performance. The seven types of the intermittent oil supply mechanism, three with different capillary sizes (ϕ_d : 0.35, 0.40, 0.45 mm) and four with different opening angles (θ_0 : 56°, 72°, 86°, 98°), were used to change the oil flow rate. Their specifications and test conditions are shown in Table 4 and 5, respectively. Condition A is equivalent to the rated condition for the CO_2 heat pump water heater (ambient temperature at 6°C, inlet water temperature at 17°C, and boiling temperature at 65°C), while Condition B is equivalent to a high load condition in winter (ambient temperature at 7°C, inlet water temperature at 9°C, and boiling temperature at 90°C). The oil flow rate G_{oil} , flowing in each intermittent oil supply mechanism was calculated by using Equation (1), and the ratio of G_{oil} to the refrigerant circulating rate G_r of the compressor is shown as G_{oil}^* in Table 4.

Table 4 Specifications of intermittent oil supply mechanism (2)

No	Diameter of capillary d [mm]	Length of capillary L [mm]	Open angle θ_0 [deg]	Oil flow rate G_{oil}^* [mass %] calculation	
				Cond. A	Cond. B
a	0.35	3.5	56	2.2	3.6
b	0.40	3.5	56	2.9	4.5
c	0.40	3.5	72	3.8	6.0
d	0.40	3.5	86	4.5	7.0
e	0.40	3.5	98	5.2	8.0
f	0.45	3.5	86	5.8	8.9
g	0.45	3.5	98	6.5	10.2

Table 5 Test conditions for performance of CO_2 compressor

Condition	p_d [MPa]	p_s [MPa]	T_s [℃]	f [1/s]
A	10.1	4.1	18	66
B	11.4	3.5	9	66

3.3.2 Experimental results

The relationship between the oil flow rate to the compression chambers and the compressor performance at Conditions A and B are shown in Fig.5 and 6, respectively. The horizontal axis of the figures represents the oil flow rate expressed in both ratios of mass flow rate G_{oil}^* and volumetric flow rate Q_{oil}^* in relation to refrigerant circulating rate. The longitudinal axis represents volumetric efficiency η_v, input power W and compressor efficiency η_{comp}, all of which are expressed as the ratio to a maximum value (with *). Here, compressor efficiency η_{comp} is defined as the ratio between theoretical compression power W_{th} and input power W.

Volumetric efficiency η_v was found to have a peak as the oil flow rate increased at both Condition A and B. When the oil is supplied to the compression chambers, its sealing effect reduces leakage of refrigerant from the compression chamber to the suction space, being likely to increase volumetric efficiency. Meanwhile, high temperature oil heats the suction refrigerant and reduces volumetric efficiency, suggesting that volumetric efficiency begins to decrease when the oil flow rate exceeds a value necessary for sealing. It is suggested that the peaks appeared because of the combination of the sealing effect and heating effect of the suction refrigerant. At Condition B, where pressure difference is large, sealing effect is more necessary. Therefore, it is suggested that the peak appeared in the region where the oil flow rate of is high.

In the region where the oil flow rate was 5 mass % or lower, input power W dropped as the oil flow rate increased, thus suggesting that recompression power of the refrigerant decreased because the sealing effect of oil reduced the leakage through the clearance of the compression chambers.

Compressor efficiency is determined by the characteristics of volumetric efficiency and input power. Therefore, the peak of the compressor efficiency appeared when the oil flow rate was around 5 mass % at Condition A, and around 7 mass% at Condition B. While the R410A scroll compressor had a peak of the compressor efficiency in the range of around 2 mass %, then we found that the CO_2 scroll compressor required more oil flow rate to the compression chambers compared with R410A scroll compressor.

In the case of the CO_2 scroll compressor, the oil flow rate ranging from 5 to 7 mass % is equivalent to the range between 0.5 and 0.7 vol. %. Therefore, it was indicated that when the oil exists approximately 0.5 vol. % in the compression chambers, leakage of refrigerant reduces, thus leading to higher efficiency.

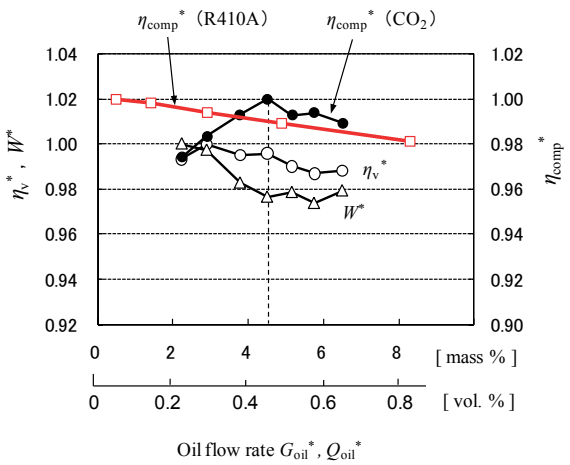


Fig.5 Relation between oil flow rate and performance (Condition A)

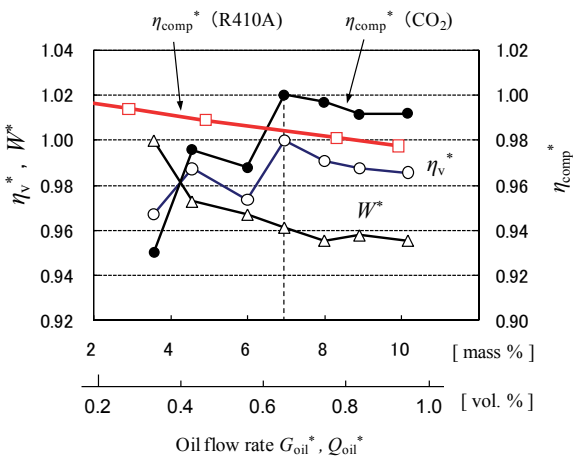


Fig.6 Relation between oil flow rate and performance (Condition B)

4. RELATION BETWEEN OPTIMAL OIL INJECTION RATE AND LEAKAGE INDEX OF THE SCROLL COMPRESSOR

We defined optimal oil flow rate ξ_{oil} as the oil flow rate to the compression chambers when the compressor efficiency has reached the peak in the scroll compressor. This optimal oil flow rate ξ_{oil} is strongly influenced by the leakage characteristics, and then we examined the relationship between optimal oil flow rate ξ_{oil} and the leakage characteristics.

First, we assumed a model (Fig.7) that refrigerant leaks from the compression chamber at high pressure p_h to the other at low pressure p_l through the clearance. Volumetric leakage flow rate V_L from the high pressure (p_h) to the low pressure (p_l) per rotation of the compressor is expressed by Equation (3).

$$V_L = S \cdot u_r \cdot \frac{1}{f} \tag{3}$$

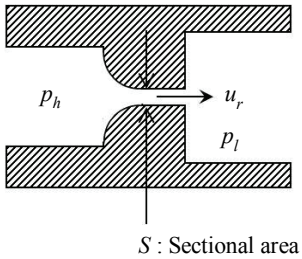


Fig.7 Leakage through the clearance

Where, S and u_r stand for the sectional area of the clearance and the leakage flow velocity, respectively. Although some equations have been proposed for estimating flow velocity u_r of a refrigerant leakage, we used Bernoulli's equation (4), assuming that the flow was an ideal fluid (with no viscosity and incompressibility).

$$\frac{p_h - p_l}{\rho_r} = \frac{u_r^{\ 2}}{2} \tag{4}$$

Volumetric leakage flow rate V_L is obtained by substituting Equation (4) in (3), and is expressed by Equation (5).

$$V_L = S \cdot \sqrt{\frac{2(p_h - p_l)}{\rho_r}} \cdot \frac{1}{f} \tag{5}$$

The influence of the leakage during compression on efficiency is estimated using the ratio of volumetric leakage flow rate V_L to displacement volume V_C, and it is expressed by Equation (6).

$$\frac{V_L}{V_C} = \frac{S}{V_C} \cdot \sqrt{\frac{2(p_h - p_l)}{\rho_r}} \cdot \frac{1}{f} \tag{6}$$

Here, let us examine sectional area S of the clearance of the scroll compressor. It is a total of area S_1 that is related to radial clearance δ_1 and area S_2 that is related to tip clearance δ_2 between the scroll wraps. Radial clearance δ_1 is present in two locations, because the scroll compressor is equipped with a pair of compression chambers. Meanwhile, tip clearance δ_2 is also present at two locations, the tips of the fixed and the orbiting scrolls. However, one of these tip clearances can be set to be very small. Therefore, we can deal with only one here. Accordingly, sectional area S of the clearance can be expressed by Equations (7), (8), and (9).

$$S = S_1 + S_2 \tag{7}$$

$$S_1 \approx 2\delta_1 \cdot H \tag{8}$$

$$S_2 \approx \pi \cdot \delta_2 \cdot D \tag{9}$$

Where, H and D refer to height and diameter of the scroll wrap, respectively. Generally, wrap height H is set to be smaller than scroll diameter D. Therefore, area S_1 of radial clearance is considered smaller than area S_2 of tip clearance, and Equation (10) can be obtained. In the case of our CO_2 scroll compressor, the ratio of S_1 / S_2 is approximately 0.05.

$$S \approx S_2 \propto \delta_2 \cdot D \tag{10}$$

Next, let us examine displacement volume V_c of the scroll compressor. As a result of dimensional analysis, displacement volume V_c is expressed by Equation (11) using wrap diameter D, orbiting radius r_0, and wrap height H.

$$V_C \propto D \cdot r_0 \cdot H \tag{11}$$

Taking Equations (10) and (11) into account, S/V_c in the right side of Equation (6) is expressed by Equation (12).

$$\frac{S}{V_C} \propto \frac{\delta_2}{r_0 \cdot H} \tag{12}$$

When substituting Equation (12) to (6), Equation (13) is obtained. Equation (13) is an expression of the influence of the leakage on the efficiency, and then the right side of Equation (13) can be defined as leakage index ζ.

$$\frac{V_L}{V_C} \propto \frac{\delta_2}{r_0 \cdot H} \cdot \sqrt{\frac{p_h - p_l}{\rho_r}} \cdot \frac{1}{f} \equiv \zeta \tag{13}$$

Equation (3) means that leakage index ζ is influenced by the refrigerant characteristics (pressure and density), and specifications (wrap height, orbiting radius, and tip clearance) and the operating condition (rotational speed) of the scroll compressor.

Therefore, we reconsidered the relationship between experimental optimal oil flow rate ξ_{oil} and leakage index ξ, illustrating it in Fig.8. In this figure, optimal oil flow rate ξ_{oil} has a width, which is related to the range of the compressor efficiency between the peak (100%) and 99.5%. On calculating leakage index ξ, we used discharge pressure p_d for higher pressure p_h and suction pressure p_s for lower pressure p_l. For the refrigerant density ρ_r , we used the density at higher pressure. We assumed that tip clearance δ_2 during operation was 5 μm. Also, Fig.8 shows experimental results of R410A scroll compressor [7], indicating that optimal oil flow rate ξ_{oil} is proportional to leakage index ξ, and their linear relationship is expressed by Equation (14).

$$\xi_{oil} = 4.4 \cdot \zeta \tag{14}$$

Consequently, we can estimate the optimal oil flow rate ξ_{oil} for a targeted scroll compressor by using the relationship expressed by Equation (14). Further, these results indicate that leakage at the tip clearance of scroll wraps is dominant in the scroll compressor.

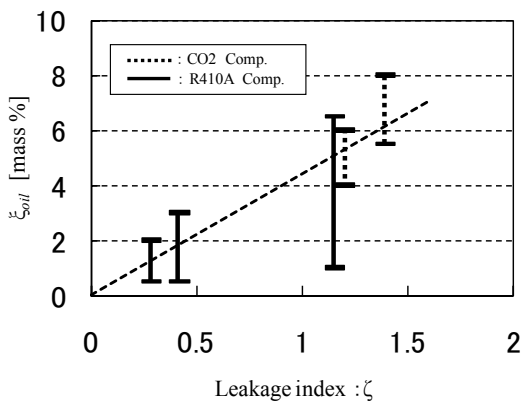


Fig.8 Relation between optimal oil flow rate and leakage index

5. CONCLUSION

Aiming to enhance the performance of CO_2 heat pump water heater, we applied the intermittent oil supply mechanism developed for the R410A scroll compressor to the CO_2 scroll compressor, and investigated the relationship between the oil flow rate to the compression chambers and the compressor performance. Accordingly, we obtained the following concusions.

(1) The oil flow rate through the intermittent oil supply mechanism in the CO_2 scroll compressor can be estimated using the same equation used for the R410A scroll compressor.

(2) The oil flow rate that increases efficiency in the CO_2 scroll compressor is in the range of 5 to 7 mass% in the ratio between the oil flow rate and the refrigerant circulating rate. It is higher than that of the R410A scroll compressor.

(3) Leakage index ζ was found to be proportional to optimal oil flow rate ξ_{oil} that increases efficiency of the scroll compressor.

NOMENCLATURE

C_{oil}	oil flow coefficient	(–)
d	diameter of the flow path	(m)
D	diameter of scroll wraps	(m)
f	rotational speed of the crank shaft	(1/s)
G	mass flow rate	(kg/s)
H	height of scroll wraps	(m)
L	length of flow path	(m)
p	pressure	(Pa)
r_0	orbiting radius	(m)
S	sectional area	(m^2)
u	velocity	(m/s)
V_C	displacement volume	(m^3/rev)
V_L	volumetric leakage rate	(m^3/rev)
W	input power	(W)
η_v	volumetric efficiency	(–)
η_{comp}	compressor efficiency	(–)
δ	clearace of scroll wraps	(m)
ρ	density	(kg/m^3)
θ_0	open angle of the capillary	(deg)
ζ	leakage index	(–)
ξ_{oil}	optimal oil flow rate	(mass %)

Subscripts

d	discharge	*h*	high pressure side
s	suction	*l*	low pressure side
m	medium	*	ratio to the base value

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