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Clearance Control of Scroll Compressor for CO₂ Refrigerant

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ABSTRACT

This study presents a higher compression efficiency scroll compressor for CO₂ heat pump water heaters, focusing on reduction of leakage loss. Gas leakage through radial and axial clearances is caused by two predominant factors; “uneven thermal expansion on scroll members due to temperature difference between suction and discharge” and “difference of thermal expansion coefficient between fixed and orbiting scroll.” First, we measured temperature distributions on fixed and orbiting scrolls in operation, second, designed several new scroll profiles to reduce leakage loss in consideration of the uneven thermal expansion, and achieved a higher efficiency CO₂ scroll compressor.

1. INTRODUCTION

For preventing the global warming, world attention is focusing on natural refrigerants for heat pump applications, where CO₂ refrigerant is one of the most practical natural gases because of its non-toxic and non-flammable properties. Moreover, the heat pump cycle with CO₂ refrigerant works efficiently in heating for large temperature difference such as CO₂ heat pump water heaters, which have already been commercialized in Japan. On the other hand, pressure difference in the CO₂ cycle is larger than in other refrigerants' cycle. Therefore, a scroll compressor, which has a plurality of isolated compression chambers and smaller pressure differences between adjacent chambers, is suitable for CO₂ refrigerant. We have commercialized an accumulator-less CO₂ scroll compressor in 2004.

The efficiency of our CO₂ scroll compressor should be improved further for energy conservation, where we have to reduce leakage loss and friction loss in particular.

For reduction of the leakage loss, it's ideal to minimize clearances on seal points between the compression chambers at every crank angle. Actually, the leakage loss exists because of uneven clearances, which appear due to temperature gradients on the compression mechanism. Moreover, unevenness of the clearances is enlarged due to the difference in thermal expansion coefficients between aluminum alloy orbiting scroll and cast iron fixed scroll, which we adopt for realizing high reliability and high efficiency.

In this study, we focused especially on the uneven radial clearances of scroll compressor in operation. First, we measured temperature distributions on the compression mechanism in operation. Second, based on these results, we designed several new scroll profiles to minimize and uniform the radial clearances on seal points of scroll wrap walls. Thus, we improved the performance of CO₂ scroll compressor, which we report on below.

2. BASIC STRUCTURE OF CO₂ SCROLL COMPRESSOR

The cross-sectional view of our conventional type of CO₂ scroll compressor is shown in Figure 1. The cast iron fixed scroll and the aluminum alloy orbiting scroll are mated to form a plurality of compression chambers. As the orbiting scroll moves, the compression chambers move towards the center, reducing the volume to compress the refrigerant. The orbiting scroll is linked to the Oldham's ring and the crankshaft driven by the motor, thus performing the orbiting motion as the crankshaft rotate.

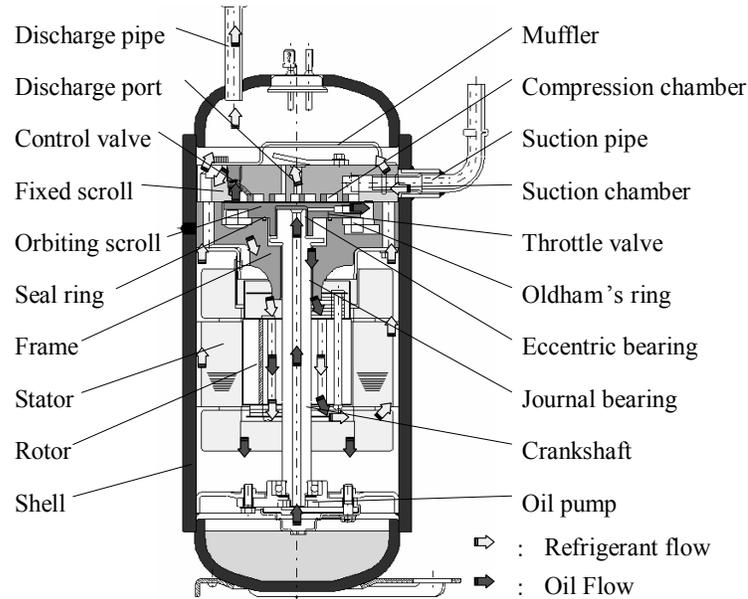


Figure 1: Cross-sectional view of conventional CO₂ scroll compressor

2.1 Axial clearance

As shown in Figure 2, a seal ring is disposed between the orbiting scroll and the frame, to separate a high-pressure space inside the seal ring and an intermediate pressure space outside the seal ring, which is kept at a specified intermediate pressure by a control valve attached in the fixed scroll. Thereby, the orbiting scroll is pressed against the fixed scroll during operation not to disengage (overturn) from the fixed scroll. Thus, the bottom of the orbiting scroll wrap contacts on the tip of the fixed scroll wrap closely.

On the other hand, there is some clearance between the tip of the orbiting scroll wrap and the bottom of the fixed scroll wrap, so-called "tip clearance", considering the difference in axial thermal expansion of the fixed and orbiting scroll wraps.

Hiwata et al. (2006) found that optimum slopes exist for the tip and bottom surface on the orbiting scroll wrap to minimize the axial clearance with thermal expansion and welding deformation in operation.

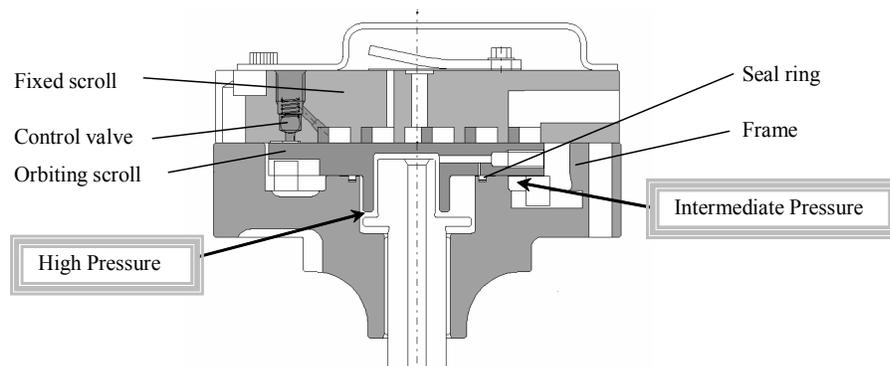


Figure 2: Enlarged view of conventional compression mechanism

2.2 Radial clearance

The compression process in our scroll compressor is shown in Figure 3. Compression chamber A, formed by the outer wall of the orbiting scroll wrap and the inner wall of the fixed scroll wrap, and Compression chamber B, formed by the inner wall of the orbiting scroll wrap and the outer wall of the fixed scroll wrap, have separate compression processes. The compressed refrigerants in both compression chambers are mixed in the scroll center to discharge out of the compression mechanism through a discharge port on the bottom of the fixed scroll wrap.

Compression chamber A starts compressing at the crank angle in Figure 3a, and Compression chamber B starts at the crank angle in Figure 3c, where it has progressed 180 degrees from the crank angle in Figure 3a.

A plurality of seal points on the walls of orbiting and fixed scroll wraps suppress refrigerant leakage between compression chambers with higher and lower pressures. For example, at the crank angle shown in Figure 3b, there are five seal points; A1 to A3 and B1 to B2.

Under general operating conditions, the orbiting scroll is pressed against the fixed scroll in the radial direction while compressing refrigerant due to three forces on the orbiting scroll; the oil film force on the eccentric bearing, the centrifugal force and the tangential gas force. Therefore, the compression chambers don't have any radial clearances on the seal points at any crank angle theoretically by contacts with wraps on every seal point.

However, in the actual operation, difference in the radial thermal expansion of the orbiting and fixed scrolls decreases the contact points, thereby, the radial clearances appear on the seal points without contacts. In addition, due to temperature gradients from the scroll center to the outside, caused by temperature difference between suction and discharge gases, the radial clearances become more uneven and complicated.

Therefore, it's important for the reduction of leakage loss to minimize and uniform the radial clearances on every seal point of scroll wrap walls at any crank angle, by grasping the temperature difference between the orbiting and fixed scrolls and the temperature distributions on the scrolls in operation.

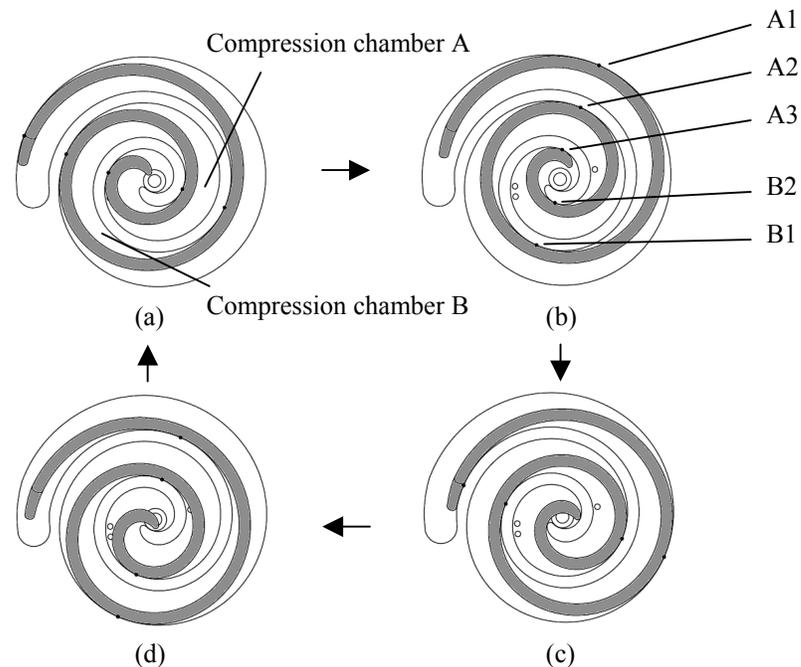


Figure 3: Scroll compression process

3. PERFORMANCE EVALUATION CONDITIONS

For the purpose to measure the compressor performance on 4.5kW CO₂ heat pump water heater, which is the dominant capacity class in Japan market, the compressor unit was tested under the rated operation mode (Condition 1) and the winter operation mode (Condition 2) as shown in Table 1.

Using a calorimeter, capacity and input of the compressor were measured to evaluate COP.

Table 1: Performance evaluation conditions

		Temperature [degree C.]			Compression ratio
		Output water	Input water	Outdoor	
Condition 1	Rated operation mode	65	17	16	2.5
Condition 2	Winter operation mode	90	9	7	3.3

4. TEMPERATURE DISTRIBUTIONS ON COMPRESSION MECHANISM

At first, we measured temperatures on the compression mechanism, the basic data for optimization of the radial clearances in operation.

Figure 4 shows the temperature distribution results with a plurality of thermocouples in the orbiting and fixed scrolls under Condition 1, which is the most frequent and important operation mode. The temperatures were measured on a line joined by the center and the suction. The horizontal axis R_r represents a radius ratio of the measurement points relative to the radius of the end of the fixed scroll wrap, with plus sign on the suction side, and the vertical axis T_r represents a factor of the thermocouple temperature t , in which the suction temperature T_s is defined as 0 and the discharge temperature T_d is 1, where $T_r = (t - T_s) / (T_d - T_s)$.

From these results, we confirmed the temperatures of the orbiting and fixed scrolls were different and the temperature of the fixed scroll was higher than the orbiting scroll. During normal operation modes, such as the rated condition (excluding overload operation modes with high compression ratio), the temperature of the discharge gas in the space between the back of the fixed scroll and the muffler is generally higher than the temperatures of the orbiting and fixed scrolls. On the contrast, the back space of the orbiting scroll is filled with oil cooled down to a certain extent. Thereupon, one may consider that the fixed scroll temperature is higher than the orbiting scroll temperature due to such backgrounds.

In addition, we confirmed temperature gradients, to the suction side in particular, existed on the orbiting and fixed scrolls, and that of the fixed scroll was larger than the orbiting scroll. One may consider that the difference in these gradients are caused by the difference in the thermal conductivities between the orbiting and fixed scrolls.

As a result, one may conclude that in order to minimize and uniform the radial clearances in operation, a special attention should be paid to considering both of the temperature difference between the orbiting and fixed scrolls and the temperature gradients on each scroll.

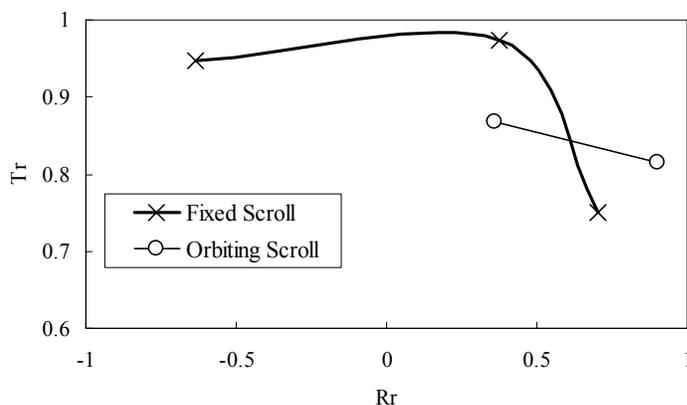


Figure 4: Temperature distribution results (Cond. 1)

5. OPTIMIZATION OF RADIAL CLEARANCES IN OPERATION

Based on the results as described above, we obtained the radial clearances in operation on all seal points analytically. As shown in Table 2, two temperature conditions were assumed to confirm the predominant factor, “temperature difference” between the orbiting and fixed scrolls or “temperature gradients” on the scrolls.

Table 2: Temperature conditions (T_r) in analysis of radial clearances in operation

	Orbiting scroll		Fixed scroll	
	Center	Outside	Center	Outside
Temperature 1	0.85		1.05	
Temperature 2	0.90	0.80	1.00	0.75

Only “temperature difference” factor was assumed for Temperature 1, which were general values without “temperature gradient”, obtained by overall results in this and past studies. Temperature 2 assumed both “temperature difference” and “temperature gradient” factors, which were the values in Figure 4.

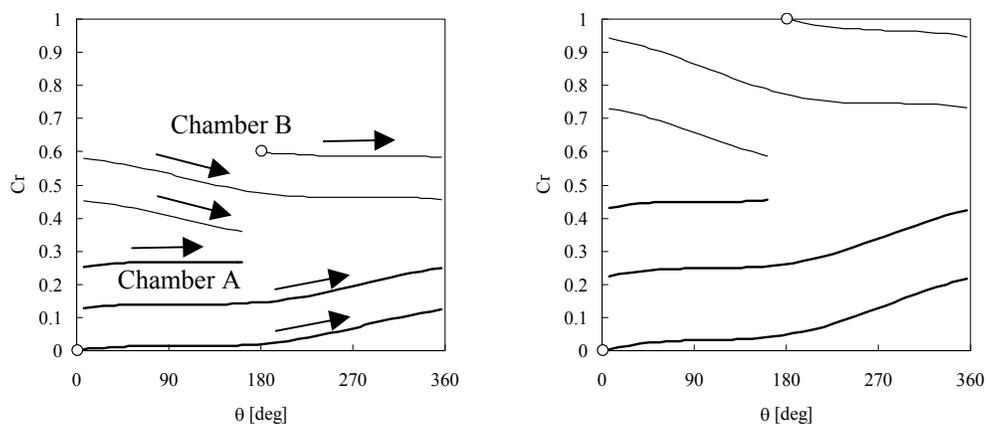
Figure 5 shows the radial clearances in operation for the conventional scroll compressor. The radial clearances in Temperature 1, without “temperature gradient”, are shown in Figure 5a, and those in Temperature 2, with “temperature gradient”, are shown in Figure 5b, where the temperature gradients were assumed to be constant from the center to the outside.

The vertical axis Cr (hereinafter called “coefficient of radial clearance”) represents the radial clearance ratio relative to the range of the radial clearances in Temperature 2, and the horizontal axis θ represents the crank angle, referred to the crank angle when Compression chamber A starts compressing suction gas. The radial clearances of Compression chamber A are indicated with thick lines, and Compression chamber B, thin lines. The radial clearances start changing continuously from the open circles, which are the enclosing points of the compression chambers, in the direction of the arrows.

As shown in Figure 5, the calculated results for the conventional compressor, designed without considering thermal expansion, indicate that the radial clearances for the 360 degrees section from the enclosure in Compression chamber A, that is the radial clearances on the outer wall of the orbiting scroll for one turn from the end of the involute, were the smallest values and the gas leakages were not suppressed sufficiently on other seal points, which had large radial clearances.

In addition, the radial clearances in Temperature 2 were even larger than Temperature 1, thereby, one may consider that the radial clearances in operation are very uneven due to both of the “temperature difference” and the “temperature gradient”.

Our conventional radial clearance curves take wave shapes because the centers of involute base circles don't lie in the centers of the end plates of the orbiting and fixed scrolls.



(a) Temperature 1

(b) Temperature 2

Figure 5: Conventional radial clearances in operation

In order to reduce leakage loss due to the uneven radial clearances in operation, we carried out designs and evaluations of new scroll profiles with two means to minimize and uniform the radial clearances at any crank angle. One is a profile with smaller involute base circle radius of the orbiting scroll than the fixed scroll, and the other is a profile with partly offset involutes relative to the base involutes. The outline of the involutes with offsets for the orbiting scroll is shown in Figure 6. The involute ending side (outside) of the outer wall and the involute starting

side (center side) are offset relative to the base involutes.

We designed four new scroll profiles for Temperature 1 and 2 with these means, namely “involute with reduced base circle radius” and “involute with offset”. Figure 7 and 8 show the calculated radial clearances in operation for the new scroll profiles. We designed them for the minimum value in radial clearances equivalent to the minimum value in the conventional radial clearances in order to prevent abnormal abrasions on the wrap walls.

As shown in the figures, we confirmed that it was possible to uniform most of the radial clearances in operation on every seal point easily by the “involute with reduced base circle radius” and the “involute with offset”.

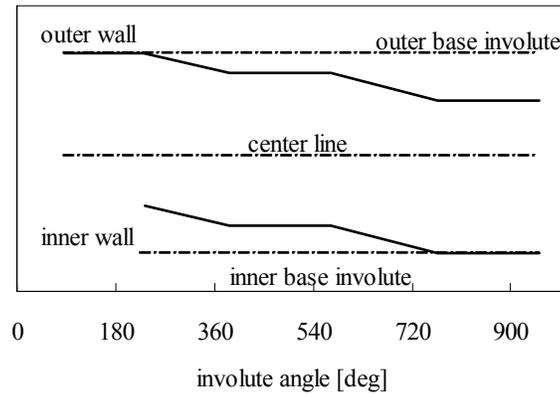
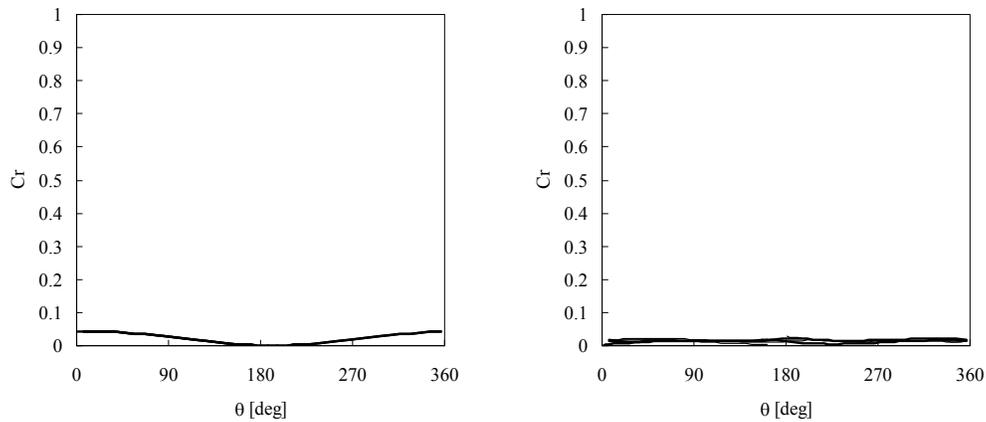


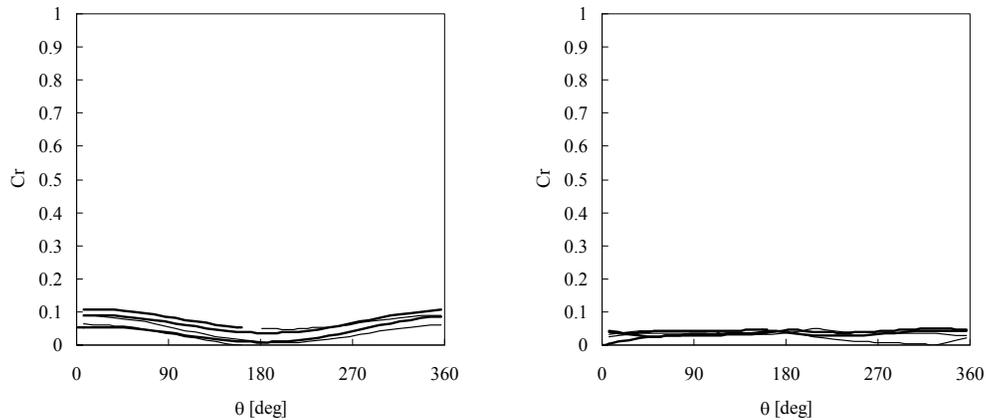
Figure 6: Outline of involute with offset



(a) Involute with reduced base circle radius

(b) Involute with offset

Figure 7: Radial clearances of new scroll profiles designed for Temperature 1



(a) Involute with reduced base circle radius

(b) Involute with offset

Figure 8: Radial clearances of new scroll profiles designed for Temperature 2

6. PERFORMANCE EVALUATION RESULTS

The performance evaluation results for the new scroll profiles are shown in Figure 9. Profile A1 and A2 are the “involute with reduced base circle radius” profiles, designed to optimize the radial clearances in Temperature 1 and 2, and Profile B1 and B2 are the “involute with offset” profiles. The vertical axis represents COP ratio relative to the conventional scroll profile.

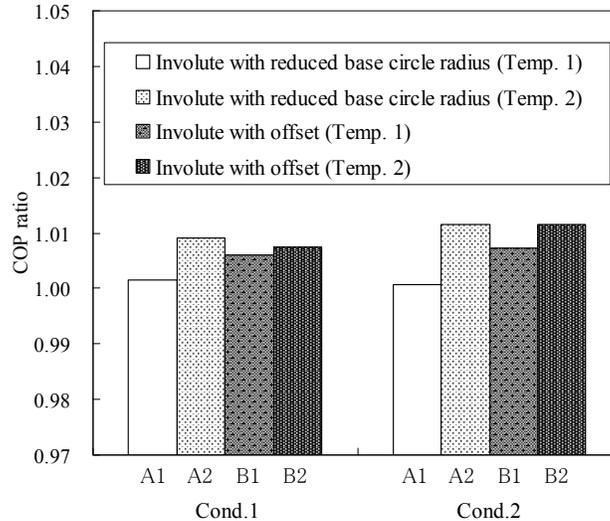
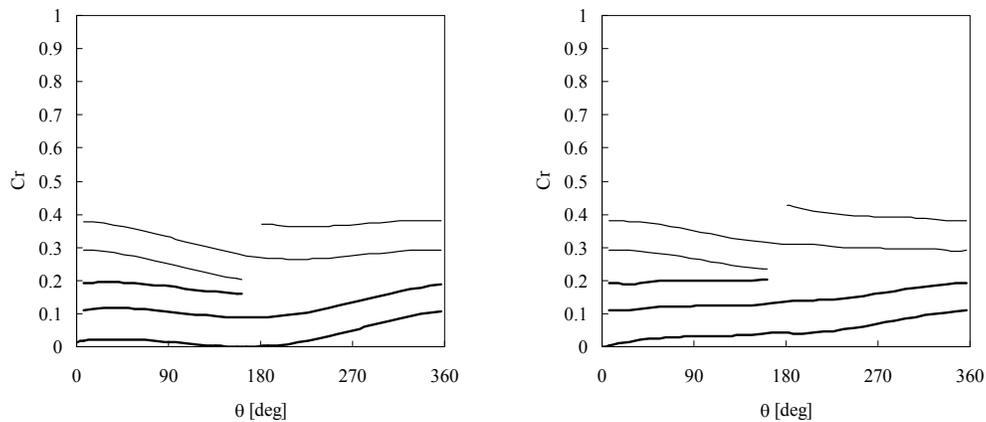


Figure 9: Performance evaluation results

These results suggest the following three points:

- 1) The performances (COP) of new scroll profiles were improved, whereby we confirmed the possibility of high efficiency by means of minimization and uniform of the radial clearances considering thermal expansion.
- 2) Under Condition 2, which has 25 K higher output water temperature than Condition 1 and is considered to have slightly different temperature distributions on the compression mechanism from Condition 1, the performances were improved. Unfortunately, we couldn't measure temperature distributions under Condition 2 to be compared to the distributions under Condition 1. Nevertheless, the improvements under Condition 2 suggest that a slight difference in assumed temperatures doesn't concern the radial clearances considerably.
- 3) The performances of new scroll profiles designed for Temperature 2, which considered the temperature gradients, were higher than the profiles for Temperature 1. One may conclude that it's reasonable to calculate the radial clearances considering both of the “temperature difference” and the “temperature gradient”, and that's a key point for improvement of efficiency.



(a) Involute with reduced base circle radius (Profile A1) (b) Involute with offset (Profile B1)

Figure 10: Radial clearances of new scroll profiles designed for Temperature 1 in Temperature 2

Figure 10 shows the radial clearances of Profile A1 and B1, designed for Temperature 1 and calculated in Temperature 2.

The fluctuations of the coefficient of radial clearance C_r for Profile A1 and B1 were about 0.4, and those for Profile A2 and B2 were from 0.05 to 0.1, as shown in Figure 8. They were smaller than 1.0, which was the fluctuation of C_r for the conventional model. Thereby, improvement of the performance by reduction of leakage loss was expected and the actual performances were improved as expected.

Profile A1 and B1 were also expected to have almost the same amount of leakage loss. While, the actual performance of B1 was higher than A1. It suggests that Temperature 2, which is considered to be a reasonable temperature distribution for calculation of the radial clearances in operation, is not applied to the actual temperature distributions yet. As shown in the temperature distribution results, Figure 4, the temperature gradients were not symmetric with respect to the centers of the scrolls and the actual temperature distributions are considered to be rather complicated. Further studies are required to establish the method of exact radial clearance analysis by grasping the temperature distributions on the compression mechanism in detail.

7. ESTABLISHMENT OF HIGH EFFICIENCY CO₂ SCROLL COMPRESSOR

As described above, the performance of the new CO₂ scroll compressor was improved under Condition 1 and 2, as shown in Table 3, due to the “involute with reduced base circle radius” profile and the “involute with offset” profile, which were designed for Temperature 2, considering both of the temperature difference and the temperature gradients, to minimize and uniform the radial clearances in operation. The resultant performances were improved by about 1%, compared with the conventional scroll compressor. Further improvement in performance are expected by grasping the temperature distributions on the compression mechanism in detail.

In addition, the temperature difference and the temperature gradients on the orbiting and fixed scrolls suggest that the optimization method for the radial clearances in operation, conducted in this study, will contribute to an improvement in performance of a scroll compressor with the same thermal expansion coefficients in the orbiting and fixed scrolls.

Table 3: Improvement of performance with optimization method in radial clearances

New scroll profile	Improvement of performance	
	Cond. 1	Cond. 2
Involute with reduced base circle radius	0.9 %	1.1 %
Involute with offset	0.7 %	1.2 %

8. CONCLUSIONS

We investigated the scroll compressor for CO₂ heat pump water heater and obtained the following conclusions:

- We carried out measurement of temperature distributions on the orbiting and fixed scrolls, thereby the temperature difference between the orbiting and fixed scrolls and the temperature gradients on the scrolls were confirmed.
- Higher efficiency can be achieved by appropriate scroll profiles with minimized and uniformed radial clearances in operation, considering temperature distributions and difference in thermal expansion coefficients of the scrolls.
- The resultant performance of scroll compressor adopting “involute with reduced base circle radius” or “involute with offset” profiles was improved by about 1%, compared with the conventional scroll compressor.

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