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A Sound Reduction Solution of Rotary Compressor by Experimental Source Analysis

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ABSTRACT

We, Toshiba Carrier Corporation, have succeeded in reducing the sound level and improving the sound quality of outdoor unit operating sound under high-speed and high-load condition in residential inverter air conditioners by improving compressor components. In this study, we first identified the target frequency by a sensory test of the outdoor unit operating sound and analysis using a graphic equalizer. Next, we made a hypothesis that the characteristic of discharge valves in the compressor made its sound louder. To further understand the phenomenon, we observed behavior of the discharge valves with displacement sensors and measured the pressure pulsation in the compression mechanism with pressure sensors under several operation conditions. Then, we visualized the changes in the power spectrum of the pressure pulsation in compression chamber during one compressor revolution by short-time Fourier transform (STFT) of those data obtained from synchronous acquisition system. As a result, we found that the response of the discharge valve is one of the factors that worsen the pressure pulsation in the compression chambers and thus the operation sound of the outdoor unit at high-speed operation. Finally, we confirm that the sound level of the outdoor unit equipped with compressor whose discharge valve has good response is improved under high-speed and high-load conditions.

1. INTRODUCTION

Heat pump technology that uses renewable energy is attracting attention toward the realization of a decarbonized society. According to the International Energy Agency, in order to achieve carbon zero by 2050, the commercial and residential sectors have set a target of installing 1.8 billion heat pump units in the next 30 years, 10 times the current number. Among them, inverter air conditioners can achieve high efficiency and comfort at low-loads and large capacity at high-loads. However, inverter air conditioners also have several problems. For example, in contrast to conventional constant-speed type air conditioners, inverter air conditioners are operated at various speeds and tend to sound and vibration issues. With the increasing concerns for sound and sound quality particularly in many countries in Europe, operation sound is one of the major issues for inverter air conditioners. In particular, under high-load and high-speed conditions such as in cold regions or during winter heating startup, operation sound tends to increase, which may prevent the widespread use of inverter air conditioners, making a solution to this problem an urgent necessity. In this study, sensory tests were conducted on 1 to 1.5 HP outdoor unit operation sound to further clarify the issues in operation sound of residential inverter air conditioners. The 9 person participated the sensory test and responded to a questionnaire after confirming the sound quality of sounds of outdoor unit operating under several conditions. As a result, we found that the sound level and sound quality tended to deteriorate as the compressor speed increased. Therefore, we were able to clarify the mechanism of the above sound generation by experimentally measuring and analyzing the behavior of compressor components and pressure pulsation in the compression mechanism, especially at high-speed. The details are described below.

2. ANALYZING THE ISSUES

2.1 Identification of issues through sensory testing

Figure 1 shows the questionnaire results from the sensory tests of outdoor unit operation sound. Table 1 shows the employed refrigerant in the tests, and outdoor unit operating conditions complied with Japanese Industrial Standard (JIS). The sensory test participants evaluated the operating sound during compressor operation at 13.1, 38.9, 76.6, and 92.2 rps, which are actual use rotational speed. At each rotational speed, people rated the sound at three levels that "Quiet or Nothing unusual", "Nothing unusual or barely acceptable", and "Unpleasant sound". The higher the rotational speed, the more participants tended to select the worse rated item, and more than half of the participants selected "unpleasant sound" at 92.2 rps. In addition, many participants commented that the higher the rotational speed, the more the annoying and unique buzzing sound was strongly generated. Then, we used a graphic equalizer to check the operation sound at high rotational speed, and found that the buzzing sound was clearly improved when the sound around 4 and 5 kHz band was lowered. This made us predict that the frequency of the buzzing sound would be around 4 and 5 kHz.

2.2 Outdoor unit operation sound test

Figure 2 shows the power spectrum of the outdoor unit operation sound at 60 to 90 rps. The higher the rotational speed, the more sound level of 3 to 7 kHz. At high rotational speed, many prominent peaks appear in 3 to 7 kHz. Therefore, the buzzing sound is noticeable during outdoor unit operation, and it is considered to be the main cause of sound quality deterioration. As a result, we focused on the 3 to 7 kHz as a frequency that should be reduced to improve the sound quality of outdoor unit operation sound.

Table 1: Test condition

Refrigerant	Indoor temperature	Outdoor temperature
-	℃	℃
R32	20	7

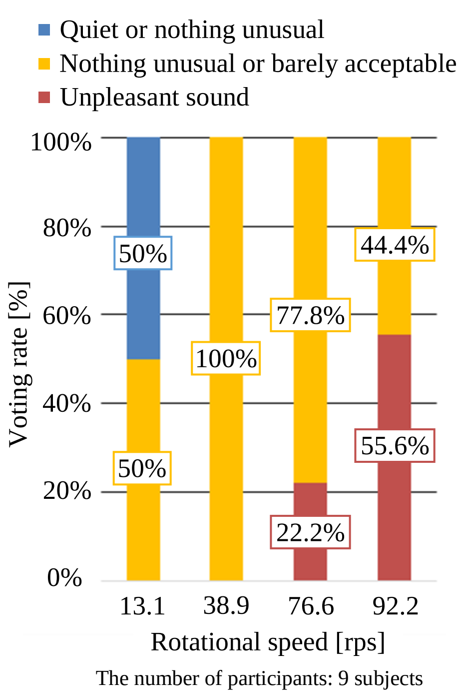


Figure 1: Result of vote

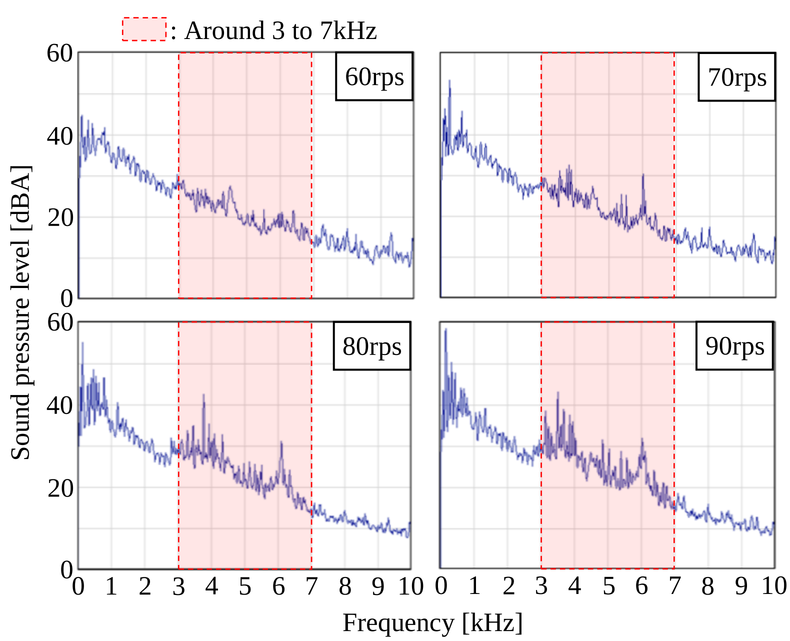


Figure 2: Power spectrum of outdoor unit operating sound

2.3 Sound source identification of outdoor unit

We conducted the sound source identification to identify the main sound radiation surface of the outdoor unit. Figure 3 shows the beamforming map of the 3.5 to 6.0 kHz sound. A red area indicating strong radiation was displayed near the base plate on the side where the compressor was installed. This shows that the base plate was the main radiation surface for the 3.5 to 6.0 kHz sound. As shown in Figure 4, the compressor in the outdoor unit is installed on the base plate of the outdoor unit through the grommets. As a result, it was considered that the vibration generated in the compressor is transmitted to the base plate of the outdoor unit through the compressor base and the grommets, and radiated as 3 to 7 kHz sound. Therefore, we evaluated the vibration of the compressor base.

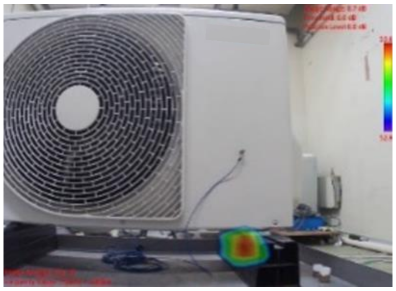


Figure 3: Beamforming map of the outdoor unit

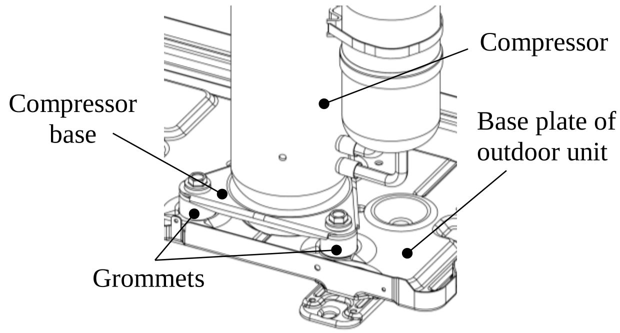


Figure 4: Installation of compressor in outdoor unit

2.4 Compressor base vibration test

We conducted the compressor base vibration test under the test conditions shown in Table 2. We used the single-axis type accelerometers with a built-in amplifier for the test, and attach them to the position shown in Figure 5. In this test, we measured the acceleration at three points near the grommets of the compressor base, and evaluated the base vibration as a three-point average. Figure 6 shows the rotational speed characteristics of the compressor base vibration in the 4 kHz and 5 kHz bands based on the 1/3-oct analysis of the data. We confirmed that the base vibration acceleration deteriorates rapidly around 70 rps, and the higher rotational speed, the worse. From these results, we estimated that the compressor is the sound source and compressor vibration transmit to the base plate of the outdoor unit through the compressor base and grommets and base plate of the outdoor unit generates the annoying sound.

Table 2: Test condition

Refrigerant	Discharge pressure	Suction pressure	Suction temperature
-	MPa_G	MPa_G	℃
R32	2.52	0.79	12.7

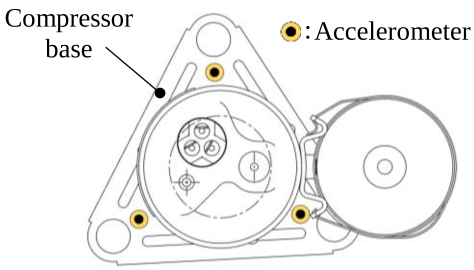


Figure 5: Measurement Position of the compressor base vibration

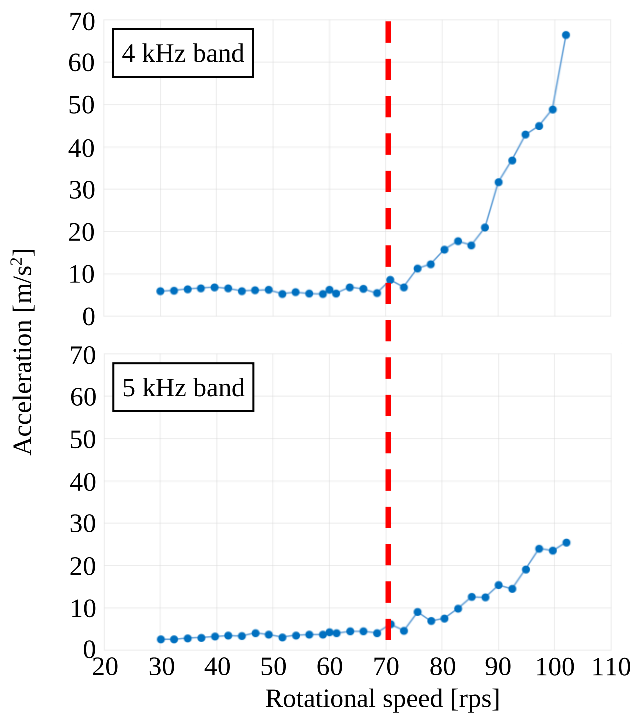


Figure 6: Results of compressor base vibration test

2.5 Structure of the compressor

The subject of this study is the twin-rotary compressor with two cylinders, which equipped in the residential inverter air conditioner. Figure 7 shows the cross-section view of the rotary compressor and Figure 8 shows the compression mechanism. Table 3 shows the specifications. Each compression chamber is formed by the cylinder, bearing, vane and partition. The DC brushless motor is mounted as the drive source. The rotor which is fixed by a shrinkage fitting to the crankshaft, is rotated by stator, and the rolling piston compresses the low-pressure refrigerant gas entering from the suction port. The rotary compressor equips a reed-type discharge valve, and the discharge port is formed in the bearing. Figure 9 shows the discharge valve and Figure 10 shows the forces acting on the discharge valve. As the refrigerant gas is compressed by the rolling piston, the gas force F_1 acting on the valve head from inside the compression chamber rises. The discharge valve opens and high-pressure refrigerant gas is discharged into the shell, when F_1 exceeds the resultant force of the gas force F_2 acting on the valve head from the shell side, and the spring force F_3 of the discharge valve. This process is performed once during one rotation in each cylinder.

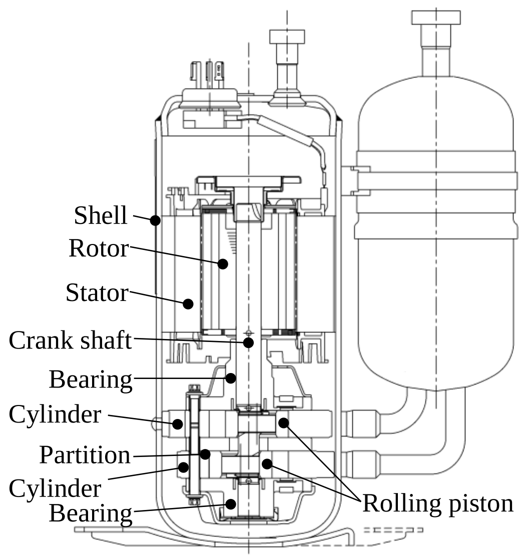


Figure 7: Cross section of rotary compressor

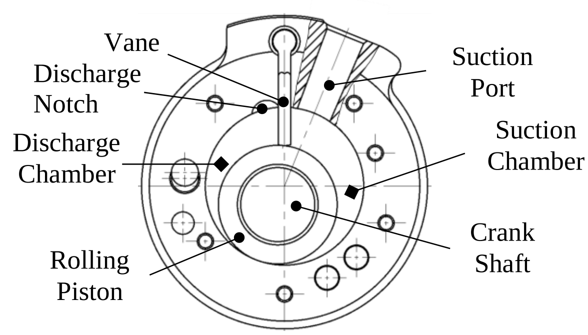


Figure 8: Compression mechanism

Table 3: Specifications

Air conditioner unit	Capacity	1 to 1.5HP
Compressor	Compression type	Rotary
	Displacement	13.6 cc

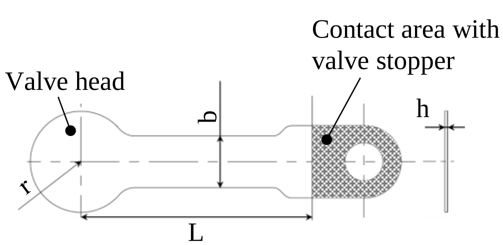


Figure 9: Discharge valve

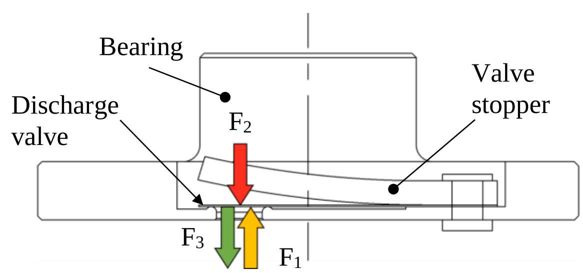


Figure 10: The force applied to the discharge valve

2.6 Dynamic characteristics of discharge valve

The inverter air conditioners control the compressor to operate in high-speed at startup in order to reach the set temperature in a short time. After reaching the set temperature, the compressor is operated at low to medium speed, the so-called intermediate rotational speed, to maintain the set temperature and to suppress power consumption. The inverter air conditioners are driven at the intermediate rotational speed on the majority of their operating time, so that they need to equip with compressors with superior performance particularly at the intermediate rotational speed to fully exert the energy-saving performance of inverter air conditioners. The optimization of discharge valve specifications is one of the countermeasures to improve performance in the intermediate rotational speed. Here, the simple formulas for discharge valve head weight, spring constant, and valve closing time, which are shown below. First, the weight of the discharge valve head is calculated by formula (1).

$$m = \rho \pi r^2 h \tag{1}$$

Where, m: valve head mass, ρ: material density, r: valve head radius, h: thickness. Second, the spring constant of the discharge valve is calculated by formula (2) which is obtained from the cantilever deflection formula.

$$k = \frac{\mathrm{E} b h^3}{4L^3} \tag{2}$$

Where, k: spring constant, E: Young's modulus, b: neck width, L: valve length.
Finally, the valve closing time is calculated by formula (3) which is obtained from the simple harmonic oscillator formula.

$$t = \pi \sqrt{\frac{m}{k}} \tag{3}$$

Where, t: valve closing time.

Table 4 shows the spesifications and the results of calculating physical property of the discharge valve which is mounted to the twin rotary compressor described previously. Calculations were made for a discharge valve with a thickness of 0.25 mm that emphasizes intermediate performance and a discharge valve with a thickness of 0.30 mm that can shorten the valve closing time at high rotational speed. With the discharge valve thickness of 0.25 mm, the spring constant k is low, so spring force F3 described in Section 2.5 are small and the valve opens easily. On the other hand, the discharge valve thickness of 0.30 mm is relatively difficult to open. The time range between completion of refrigerant discharge and the discharge valve closes is expressed by the valve closing time t. The discharge valve with thickness of 0.25mm has relatively longer valve closing time t, and tends to close later than the discharge valve thickness of 0.30 mm. The discharge valve thickness of 0.25mm is designed particularly for performance at intermediate rotational speed as easy to open at low to medium speed also. The residential inverter air conditioners employ the compressors equipped with discharge valve thickness of 0.25mm, we judged that the delay in closing the discharge valve could cause the deterioration of sound at high rotational speed.

Table 4: Specifications and physical property of the discharge valves

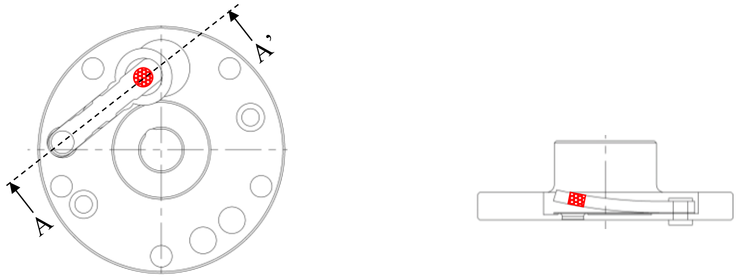
Specifications								Physical property
Thickness	Young's modulus	Density	Radius (valve head)	Length	Width	Mass (valve head)	Spring constant	Valve closing time
h	E	ρ	r	L	b	m	k	t
mm	GPa	g/mm^3	mm	mm	mm	g	N/mm	msec
0.25	208	7.86×10^{-3}	4.9	22.3	5.0	1.5×10^{-1}	3.8×10^{-1}	2.0
0.30	ditto	ditto	ditto	ditto	ditto	1.8×10^{-1}	6.6×10^{-1}	1.6

3. EXPERIMENTAL METHOD

It is necessary to obtain synchronous data on the behavior of the discharge valve and the pressure pulsation near the compression mechanism in order to reveal the phenomenon caused by the delay of closing the discharge valve during high-speed operation. Table 5 shows the test conditions, Figure 11 shows the displacement sensor installation locations. The eddy current displacement sensor is installed on the valve stopper to measure discharge valve behavior. In addition, Figure 12 shows the pressure sensor installation locations. The piezoelectric pressure sensors are mounted on the cylinder to measure pressure pulsations. The pressure sensor was installed one by one on the suction chamber side and compression chamber side. To detect the crank rotation angle of 0°, the so-called top dead center, the vane displacement is measured by the eddy current displacement sensor. The crank rotation angle was calculated based on the signal of the displacement of the vane, and we analyzed the discharge valve behavior, pressure pulsation and crank rotation angle as synchronous data.

Table 5: Test condition

Refrigerant	Discharge pressure	Suction pressure	Suction temperature	Rotational speed
-	MPa_G	MPa_G	℃	rps
R32	2.52	0.79	12.7	60, 90



(a) Top view of bearing (b) A-A’ cross section of bearing

Figure 11: Position of the displacement sensor

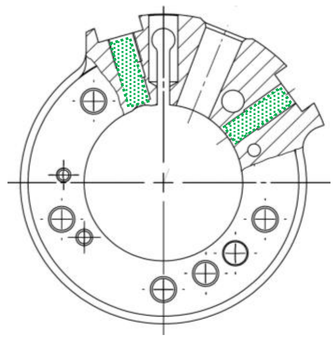
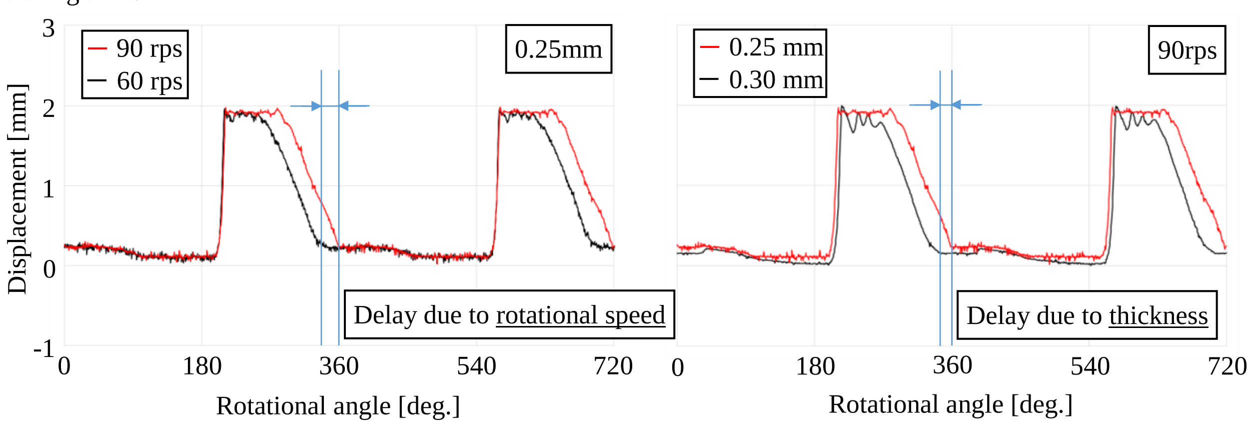


Figure 12: Position of the pressure sensors

4. TEST RESULT

4.1 Result of discharge valve displacement

Figure 13 shows the results of the discharge valve displacement measured by the displacement sensor attached to the valve stopper. Figure 13 (a) shows the comparison of compressor rotational speed at the discharge valve thickness of 0.25 mm. It was confirmed that the discharge valve closes after the crankshaft rotational angle is more progressed at higher rotational speed. Figure 13 (b) shows the comparison of the discharge valve thickness at 90 rps. It was confirmed that the discharge valve closes after the crankshaft rotational angle is more progressed when the discharge valve is thinner. In other words, higher rotational speed conditions or thinner discharge valves tend to cause delay in closing time.



(a) Comparison of rotational speed (b) Comparison of valve thickness

Figure 13: Test result of the discharge valve displacement

4.2 Pressure pulsation in compression chambers

Figure 14 shows the results of short-time Fourier transform (STFT) of pressure pulsations in the compression chambers for two revolutions acquired by the pressure sensors. On the suction chamber side, strong signals appear around 60° and 420°. This is the effect of the rolling piston passing in front of the pressure sensor. Strong signals also appear around 0° and 360°, at the top dead center, and this trend is more remarkable when the discharge valve is thin and under high rotational speed conditions. This signal appears to be caused by the re-expansion and it is thought to be related to the delay of closing of the discharge valve as described in section 4.1. Next, on the compression chamber side, the pulsation frequency grows as the crankshaft rotation progresses. At 60 rps, the signal intensity is almost the same regardless of the thickness of the discharge valve. On the other hand, at 90 rps, the strong signal is detected near the top dead center for the 0.25 mm discharge valve. For the discharge valve thickness of 0.30mm, the signal is slightly stronger, but isn't as remarkable as for the 0.25mm discharge valve.

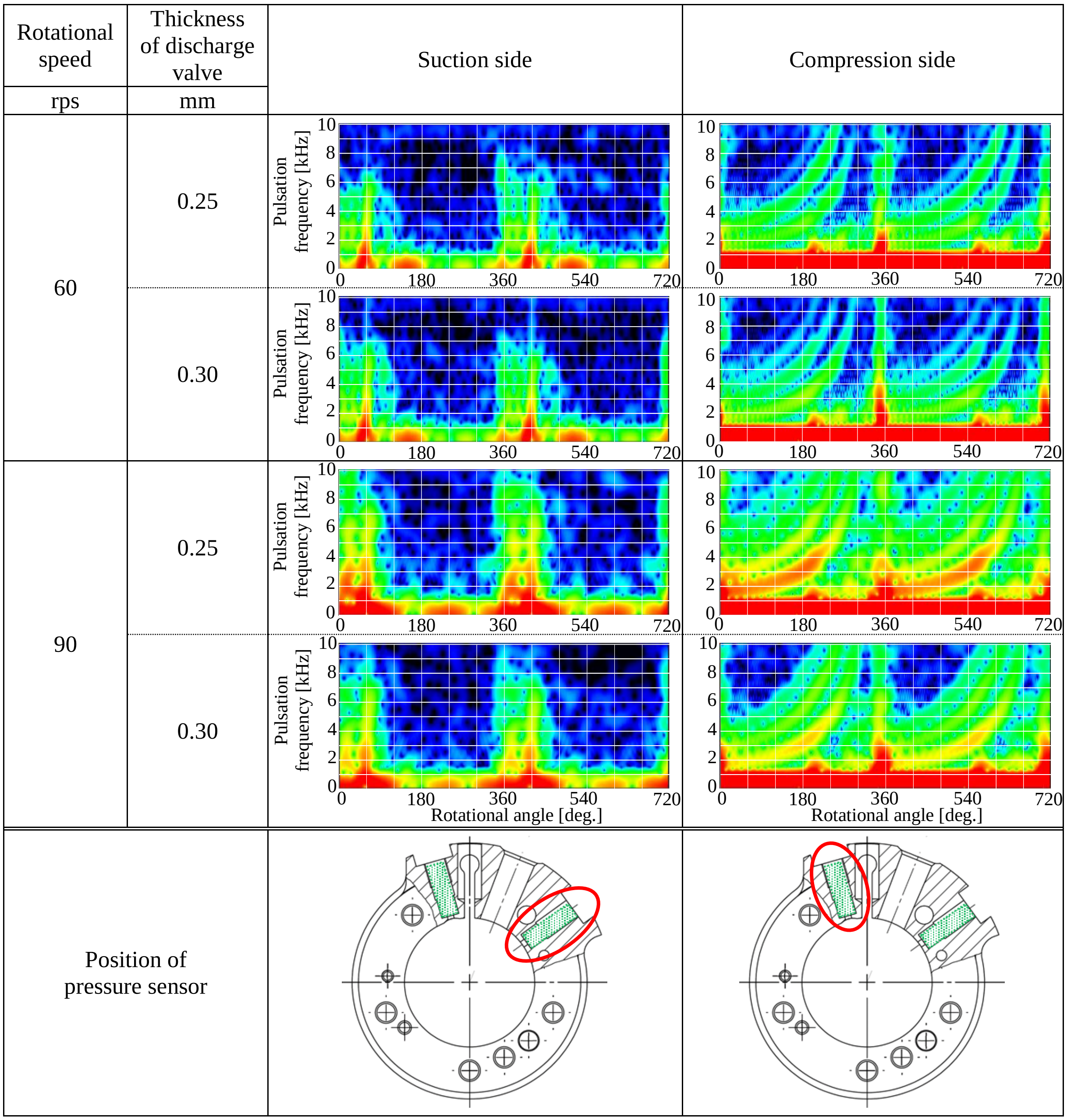


Figure 14: Test results of pressure pulsation in chambers

4.3 Pressure pulsation generation and growth mechanism

Figure 15 shows the enlarged view of the displacement of the discharge valve at the crank rotation angle when the discharge valve closes. At 60 rps, the closing timing shows almost no difference regardless of the thickness of the discharge valve. We set the crank rotation angle at this time as θ_0. At 90 rps, the discharge valve thickness of 0.30 mm close at around θ_0, but the discharge valve thickness of 0.25 mm closes long after passing through θ_0. We set the crank rotation angle at this time as θ_1. In addition, Figure 16 shows an enlarged view near the discharge port at crank rotation angles θ_0 and θ_1. The θ_0 is around the angle which the rolling piston approaches the left end of the notch in the cylinder. The crank rotation angle θ_1 is the angle which is significantly past the notch.If the discharge valve does not close around θ_0, a connecting path is formed between the main case inner space and the suction chamber of compression mechanism. The discharge valve of 0.25 mm thickness is delayed in closing from θ_0 to θ_1, the high-pressure refrigerant gas in the shell flows back into the suction chamber due to this connecting path, and the re-expansion can occur. We considered that the re-expansion changes to the strong pressure pulsation. Finally the pressure pulsation propagates as vibration to outside of the compressor through the compressor base and causes the deterioration of sound at high rotational speed.

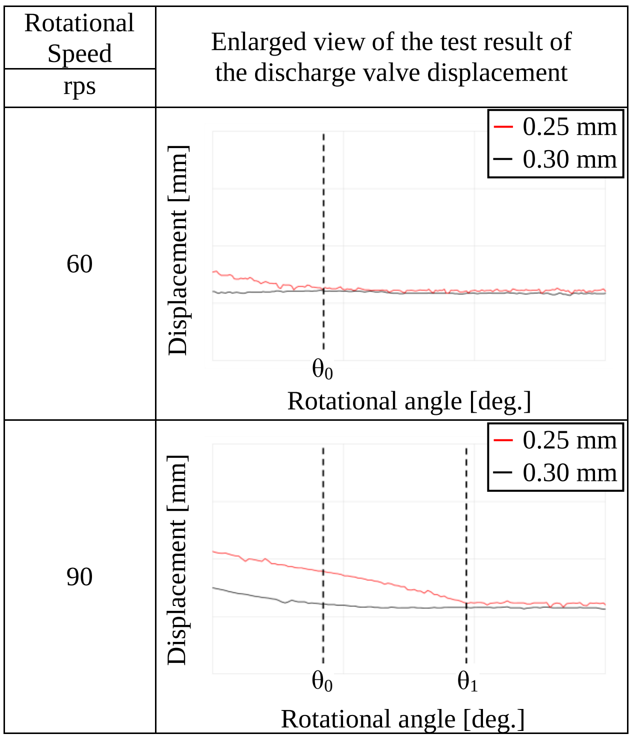


Figure 15: Enlarged view of the discharge valve displacement

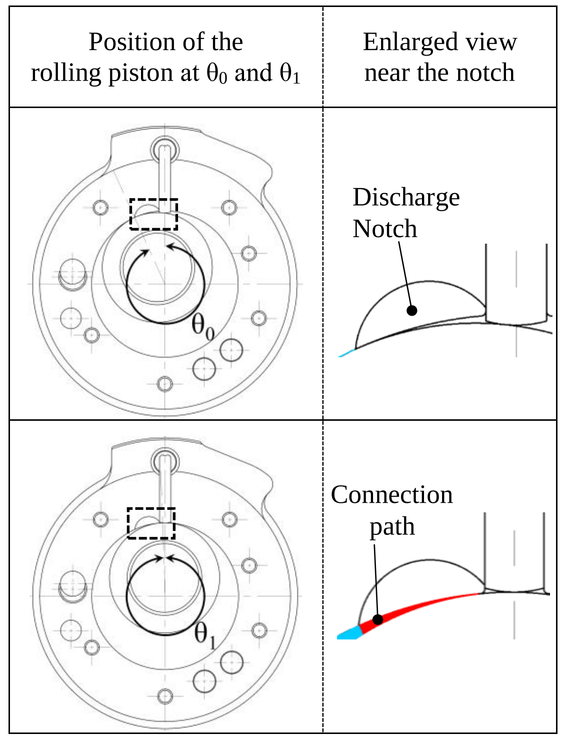


Figure 16: Connection path between the main case inner space and the suction chamber

5. EVALUATION

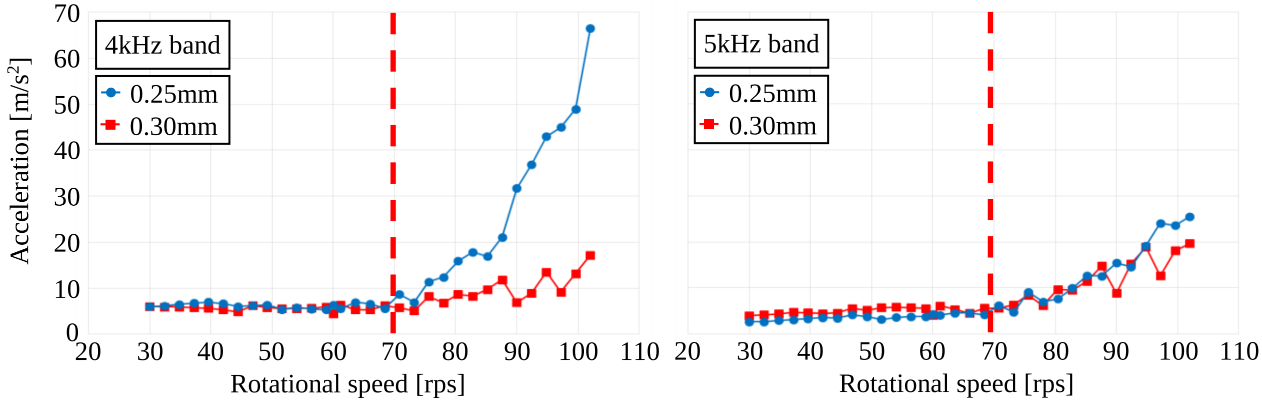
As a result of the test, we revealed that the thicker discharge valve eliminates the closing delay and suppresses the formation of the connection path and the re-expansion. In addition, to confirm the effect on outdoor unit operation sound, we conducted the compressor base vibration test and outdoor unit operation sound test using the compressor with the discharge valve changed from 0.25 mm to 0.30 mm in thickness.

5.1 Compressor base vibration test

We evaluated the vibration of the base of compressor which equipped the discharge valve of 0.30mm thickness. Table 6 shows the test conditions. Figure 17 shows the comparison of the thickness of discharge valve. In the 4 kHz band, the discharge valve thickness of 0.30mm shows obvious improvement effect and no rapid deterioration of vibration acceleration at 80rps or more, which is observed with the discharge valve thickness of 0.25mm. In the 5 kHz band, the discharge valve thickness of 0.30mm shows a slight improvement, although the effect is not as strong as in the 4 kHz band.

Table 6: Test condition

Refrigerant	Discharge pressure	Suction pressure	Suction temperature
-	MPa_G	MPa_G	℃
R32	2.52	0.79	12.7



(a) 4kHz band (b) 5kHz band

Figure 17: Results of compressor base vibration test
-Comparison of the discharge valve thickness-

5.2 Outdoor unit operation sound test

We conducted outdoor unit operation sound tests to confirm the effect of compressor vibration improvement. Table 7 shows the test conditions. Figure 18 shows the comparison of the power spectrum of the outdoor units. The sound level in 3 to7 kHz is reduced in the specifications with the discharge valve thickness of 0.30 mm, compared to the discharge valve thickness of 0.25 mm. The overall value improved by 4.7 dB, and it was also confirmed that the effect of sound reduction in outdoor unit operation under high rotational speed conditions by changing the thickness of the discharge valve.

Table 7: Test condition

Refrigerant	Indoor temperature	Outdoor temperature	Compressor rotational speed
-	℃	℃	rps
R32	20	7	90

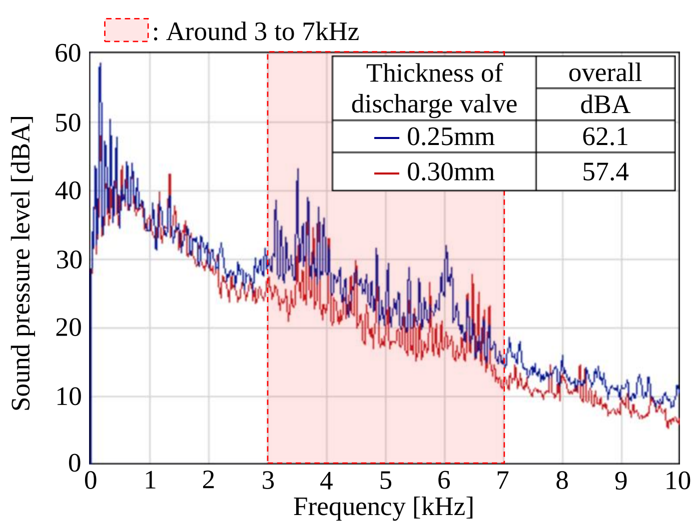


Figure 18: Power spectrum of outdoor unit operation sound
-Comparison the discharge valve thickness-

6. CONCLUSION

We obtained the following findings in the course of studying sound reduction of twin rotary compressors in residential inverter air conditioners.

・We clarified the influence of the discharge valve dynamic characteristics on the compressor base vibration and outdoor unit operating sound by acquiring displacement of the discharge valve and pressure pulsation in the compression chamber. The thin discharge valve causes delay in closing under high rotational speed conditions. Afterwards, the formation of a connecting path between the main case and the suction chamber of compression mechanism causes the backflow of compressed refrigerant to the suction chamber and the re-expansion. The pulsation frequency grows from strong pressure pulsation as the crank rotational angle advances, propagates through the mechanism, and appears as vibration outside the compressor.

・To suppress the delay in valve closing at high speeds, we manufactured compressor which equips discharge valve thickness of 0.30mm and compared the compressor with the existing 0.25mm valve. In the compressor vibration test, we confirmed the effect of the vibration reduction with the 0.30mm valve in the 4k and 5 kHz band. In addition, in the outdoor unit operation sound test, we confirmed the sound level around 3 to 7 kHz and the overall value were also improved with the 0.30mm valve.

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