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Identification of Noise Sources in Scroll Compressor for Air-Conditioner

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Low noise of air-conditioners is one of the most important issues because of the users' strong demand. The main source of noise in an air-conditioner is the compressor. Therefore, noise reduction in a compressor is quite significant as an element technology in the air-conditioner field. Recently, scroll compressors are widely used, because they feature low noise, due to less pulsation of gas pressure, than that of rotary compressors. For reduction of noise, the source of noise must be identified. This paper presents a detailed analysis to identify the noise source and shows the dominant factors of noise of the scroll compressor, which will make it possible to design a scroll compressor with low noise.

Key Words : Identification, Noise Source, Pressure Pulsation, Hermetic Compressor

1. Introduction

As for air-conditioners for general domestic use, a great technological progress is being made in reducing energy consumption, improving pleasantness, and making environment-friendly. Recently, noise-reduction, one of the basic elements to realize pleasant air-conditioning, has turned out to be an important subject, where researches have been concentrated. To reduce the noise of an air-conditioner, it is important to reduce the noise of the compressor. The application of scroll compressors for air-conditioners has been rapidly extended since 1990, succeeding reciprocating compressors and rotary compressors. Among our home industries, LG Electronics and Kyungwon-Century are entering into mass production of scroll compressors. The reason for scroll compressors' replacement of the existing reciprocating and rotary compressors is that they have not only advantages for vibration and noise but also high efficiency, with low torque-variation. General noise of compressors consists of the structure-borne noise coming through the resonance of the compressor's components, the air-borne noise coming through resonances of the sealed interior space of the refrigerant, and the fluid noise coming through the discharge pressure pulsation. The transmission path of the vibration and noise of the rotary compressor is shown in Fig. 1 (Iwata et al., 1990). Because compressing methods of the rotary compressor and the scroll compressor are different, their noise generation mechanisms are not the same and therfore detailed analysis of each type is required for a noise reduction. While many studies have been performed on the vibration and noise reduction of the rotary compressors, relatively less studies have been done on the vibration and noise reduction of

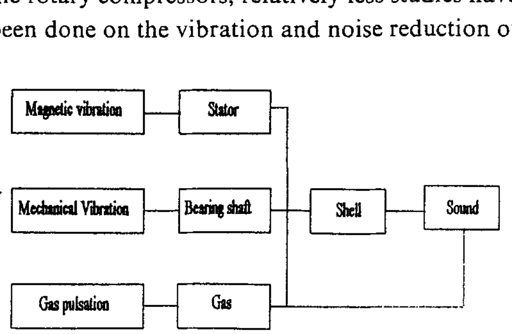

Fig. 1 Scheme of vibration and noise generation and transmission in a rotary compressor

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the scroll compressors.

Iwata et al. (1990) introduced recent movements and study results about the vibration and noise of the rotary compressors and the scroll compressors for air-conditioners. Motegi et al. (1996) and Bush et al. (1996) performed reduction of noise in a scroll compressor. It has been found that discharge gas pulsation is one of the major causes of scroll compressor noise and the top shell is a prime radiator of the acoustic energy. Sano et al. (1997) performed a systematic and synthetic examination of the noise reduction methods and analysis of the cause of noise occurrence, mainly using the scroll compressor for room air-conditioners. This examination was done in the following; 1) the noise caused by cavity resonance and gas pressure pulsation of the discharge space inside the compressor, 2) the noise caused by resonances of components, the magnetic noise of the motor. While the previous research investigated the small inverter horizontal-type scroll compressor of 1Hp., there has been no systematic study on the noise source and transmission path of large vertical-type scroll compressors.

In this paper, we identified the noise source for effective noise reduction of the vertical-type scroll compressors used for package air-conditioners of 3Hp., and analyzed, by both experiments and theories, dominating factors of the scroll compressors' noise.

2. Noise Evaluation of Scroll Compressors

The structure of a 3Hp. vertical scroll compressor is shown in Fig. 2. Its diameter and height are 146mm and about 400mm, respectively. It is driven by a motor with a rotational speed of 60Hz, using R22 refrigerant. The motor part consists of a stator that is fixed in the case and a rotor with an inserted crank shaft. And the crank shaft is supported at both ends by journal bearings of the main and sub frame. The compressing part consists of a fixed scroll and an orbiting scroll, and an oldham ring. A check valve of plate type is installed at the discharge port of the fixed

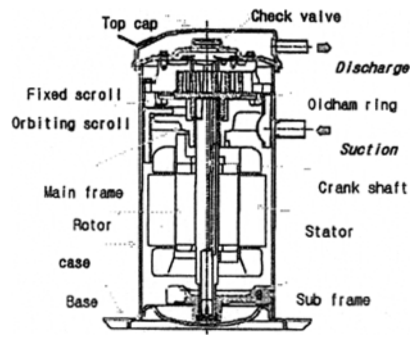


Fig. 2 Cross-sectional view of a scroll compressor

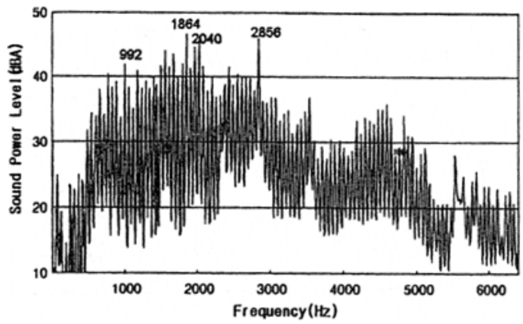


Fig. 3 Typical noise spectrum

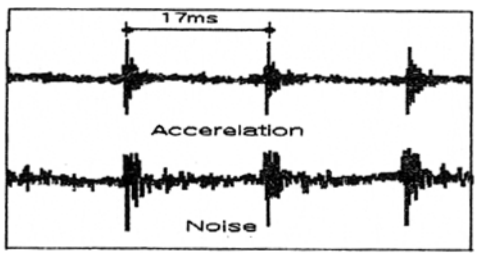


Fig. 4 Measured accelerometer and microphone time domain response

scroll to prevent the reverse rotation. The refrigerant gas is compressed in the compressing part that consists of a fixed scroll and an orbiting scroll, and the compressed gas is discharged into the space of the top cap through the discharge-port. The test conditions of the scroll compressors are the ARI standards, and the noise has been measured with the sound power of ISO 3744 standard in a semi-anechoic room.

The noise spectrum of the scroll compressor used for this research is shown in Fig. 3. We can see it shows noise characteristics of a wide band from 0.5kHz to 4.5kHz, and the main peaks exist in the bands of 0.8kHz, 1.6k～2kHz and 3kHz. Figure 4 exhibits the vibration-acceleration and the noise measured in the time domain. Note that the peaks of an acceleration of impact type pulse per rotation, and the noise peaks appear simultaneously. In this paper, we have investigated the noise source of the main peak frequencies in Fig. 3 and the exciting force of the peak in Fig. 4.

3. Measurement and Analysis

3.1 The analysis of the noise contribution by sound intensity

We measured the sound intensity to investigate the noise contribution of the components assembled inside from the direction of the noise location and frequency of the noise radiating from the scroll compressor. 145 cylindrical virtual lattice were made, and the sound intensity was measured with a sound intensity probe (B&K3520) installed 10cm away from the compressor in 3 axes directions, which was analysed through LMS CADA-X. Figure 5 shows the contribution of 1/3 octave band frequencies among the noises radiating from each part of the compressor. We can see that noise of 0.8kHz band occurs in the suction part and the motor, 1.6kHz in the compressing part, 2kHz in the motor, and 2.5kHz～3.15kHz in the top-cap. Figure 6 shows intensity maps of 1.6kHz and 2.5kHz bands. We find that the largest radiation of 1.6kHz occurs in the compressing part, and that of 2.5kHz in the top-cap.

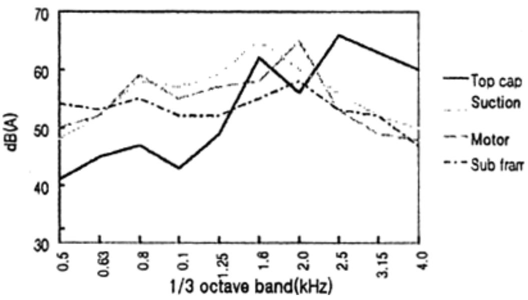
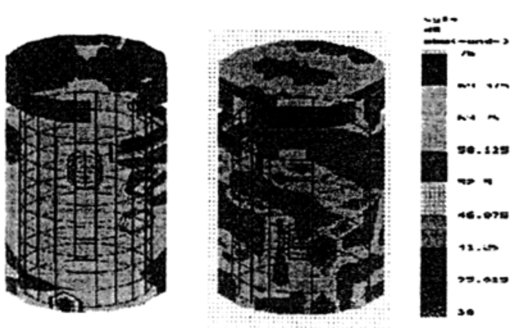


Fig. 5 Sound intensity pattern at each part



1.6 kHz band 2.5 kHz band

Fig. 6 Sound intensity map

3.2 Gas pulsation of the compression room and discharge space

Characteristics of the gas discharge pulsation and the discharge space may act as the main component of exciting force of the top-cap. In this section, these characteristics have been clarified and investigated through experiments. Pressure sensors were installed in the compression space near the discharge port to measure the gas pressure pulsation. For the gas pressure pulsation of the discharge space, a pressure sensor was installed on the top-cap. Resonance characteristics were surveyed by measuring acoustic characteristics with 4 microphones in the circumferential direction centering around the discharge pipe of the top-cap, exciting the air in the discharge space (cavity) with a speaker. The sound analysis for theoretic investigation of resonance frequencies and acoustic modes of the discharge space was done using SYSNOISE, a 3-dimensional sound analysis package. Actually, the refrigerant has different acoustic velocity from that of air, which had to be compensated. In the table 1 results of the experiment and the theoretic analysis of the resonance frequencies of the discharge area are compared, with the maximum error of 6%. The

Table 1 Resonance frequencies in discharge space

Mode	Calculation [kHz]	Experiment [kHz]	Error(%)
1	700	735	5
2	1190	1267	6
3	1490	1550	3.8

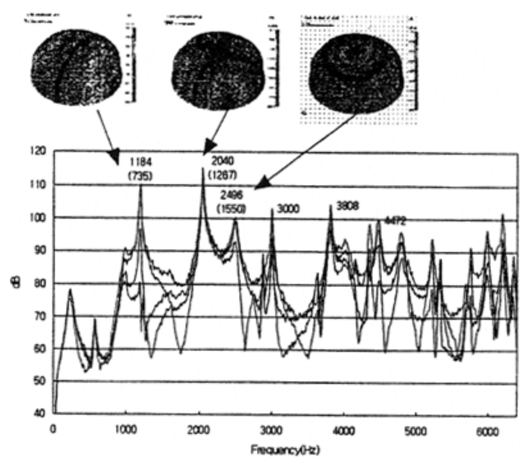


Fig. 7 Measured cavity resonances and mode shapes

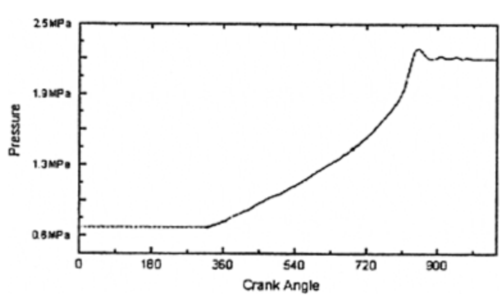
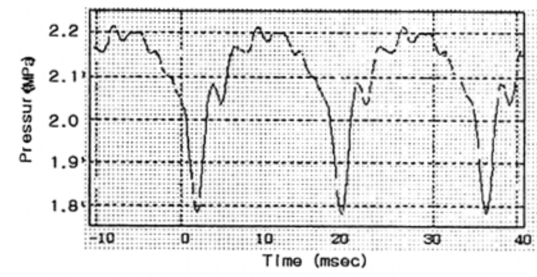


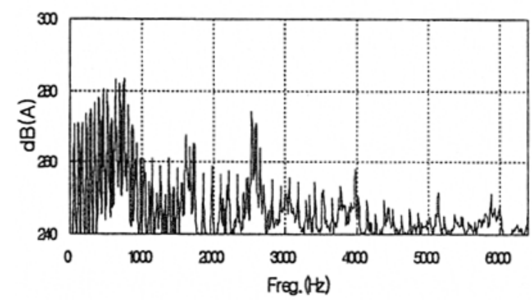
Fig. 8 Pressure pulsation in the discharge port

mode shapes, calculated with SYSNOISE are shown in Fig. 7. Note that there is the first single phase mode in 735Hz, the frequency converted into R22 in the parentheses, and the second double phase mode in 1287Hz. Figure 8 shows the pressure pulsation signal in the discharge port as a function of the crank angle. 0.7MPa~2. 2MPa of gas pressure takes place near the discharge port.

Unlike the rotary compressor, the scroll compressor does not show the pressure-pulsation of high frequency in its pressure wave in the execution of discharge port, but show that of low frequencies around 500Hz, because the check valve applied to the scroll compressor is not the reed type but the plate type. Figure 9 shows the pressure pulsation in discharge cavity as a function of time and frequency. Although there is 500Hz component in the pressure pulsation shown in Fig. 9, it does not appear quite conspicuous. The major frequency components of gas



(a) Time domain



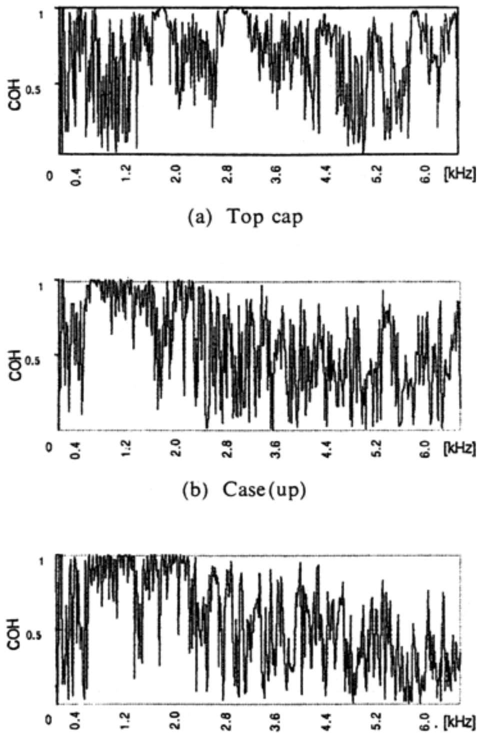
(b) Frequency domain

Fig. 9 Pressure pulsation in discharge cavity

pressure pulsation of the discharge space appears in the region lower than 800Hz, and 1.6k ~1.8kHz and 2.4~2.8kHz. These frequencies don't coincidence with the resonance frequencies of the discharge cavity, except for 700Hz.

3.3 Noise caused by resonance of the case and top cap

Vibration in the case and the top cap is one of the main noise sources caused by gas pressure pulsation of the suction and discharge cavity, or resonance of itself. The correlation of frequencies of vibration and noise of each part was examined through installing acceleration sensors at the center of the top cap and the top and bottom of the case, with microphones installed 10cm away from them, in load condition. Figure 10 exhibits the results. It can be understood that there is the correlation of vibration and noise around 1.6kHz ~1.9kHz and 2.7kHz~3.0kHz at the top cap, around 0.8kHz~3.0kHz at the upper part of the case, and 0.6kHz~1.2kHz and 1.6kHz~2.0kHz at the bottom of the case. To understand characteristics of the case and top cap of the compressor, which cause noise, we performed a modal analy-



(a) Top cap

(b) Case (up)

(c) Case (down)

Fig. 10 Coherence of vibration and noise at top cap and case

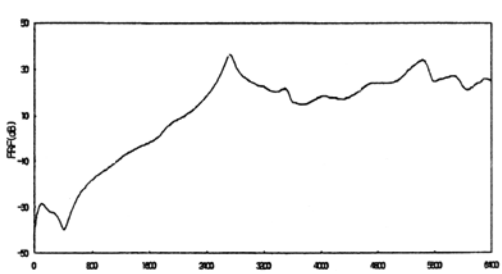


Fig. 11 FRF of top cap

sis and got FRF (frequency response function) through exciting the compressor with an impact hammer. The characteristics of frequency response of the top cap is exhibited in Fig. 11. The first resonance of the top cap appears in 2722Hz. Figure 12 shows the first mode shape (2722Hz) of the top cap and the bending mode shape (2004Hz) of the case. Most of the noise in the band of 1.6kHz~2kHz our concerned frequency range is caused by vibration of the case,

Table 2 Comparison of resonances in top cap

Mode	Calculation [Hz]	Experiment [Hz]	Error (%)
1	2946	2722	7.6
2	3946	3548	10

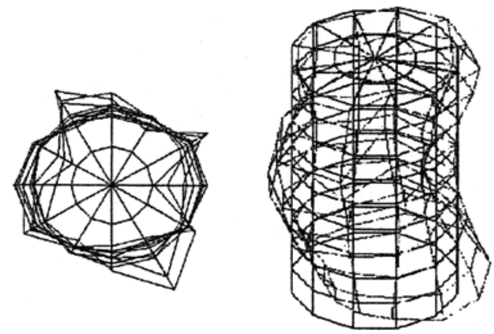
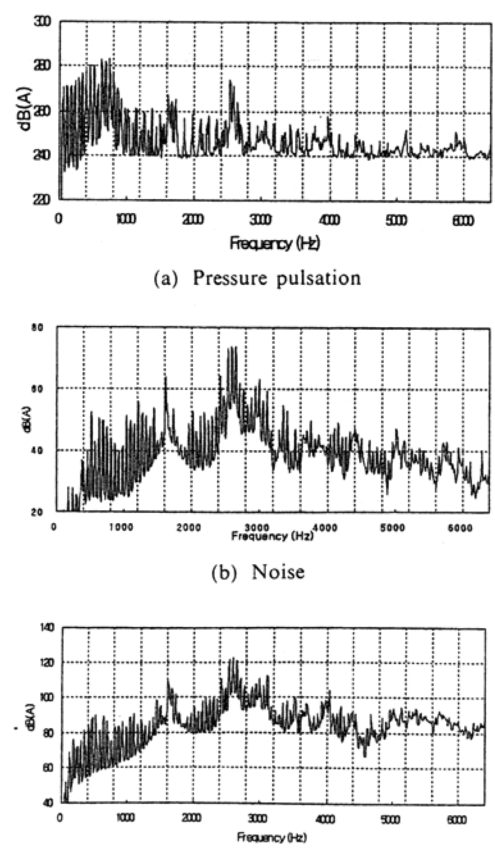


Fig. 12 Mode shape of the top cap and the case



(a) Pressure pulsation

(b) Noise

(c) Vibration

Fig. 13 Spectrums of press pulsation, noise, vibration at top cap while that of the 2.7kHz is caused by vibration of the top cap.

Experimental data of top cap, were compared to the numerical results analysed by means of ANSYS, a commercial package for structure analysis, in Table 2. Note that the first resonance of top cap coincides comparatively well, with about 7.6% of errors. In order to identify the exciting force of top cap, we measured the gas pressure pulsation of the discharge cavity, vibration of the surface of top cap and noise of a spot 10cm away from it, which are shown in Fig. 13. The resonance frequency of the top cap is around 2.7kHz, which coincides with the pressure pulsation frequency of 2.5kHz~2.7kHz, causing high-level noise.

As shown in Figs. 5 and 6 a large part of the noise radiation appears at the top cap, where the discharge part is located, because the pressure pulsation of discharged gas excites the top cap directly, and generates the noise.

3.4 Noise caused by collision of components

To identify the exciting force of the periodical acceleration and noise shown in Fig. 4, the acceleration of the fixed scroll and welded part of the case were measured in each orbiting angle, with a gap sensor installed at the suction port of the fixed scroll. The results are shown in Fig. 14. The periodical peaks of the vibration acceleration are made by collision of the scroll wraps at the starting point of suction in the scroll profile, where both of the scroll walls begin to contact, which excites the compressor and make it radiate the noise. When they are converted into frequencies, they have a great influence on the band around 2kHz.

3.5 Electromagnetic noise of the motor

The motor installed in the scroll compressor is a 2-pole single phase induction motor, whose rotor has 33 slots. A compressor removed of the compressing part (fixed scroll and orbiting scroll) was made to measure the effect of pure electromagnetic noise of the compressor. Then, vibration and noise of the surface of the case where the stator is located were measured, in non-loading condition of air-pressure, and shown in Fig. 15. The frequency of the rotor's slots in non-loading condition, related to electromagnetic exciting force, can be shown as follows, the number of the rotation being 60Hz, :

$$33 \times 60\text{Hz} \pm 120n = 1980\text{Hz} \pm 120n,\ n = 0,\ 1,\ 2, \cdots.$$

The frequency components of the electromagnetic force exciting first frequencies of motor in non-loading condition are 1860Hz 1980Hz and 2100Hz. This can be found in the spectrum of vibration and noise shown in Fig. 15. The mode shape of the case without the compressing part are shown in Fig. 16.

The resonances of the case are at 1860Hz and

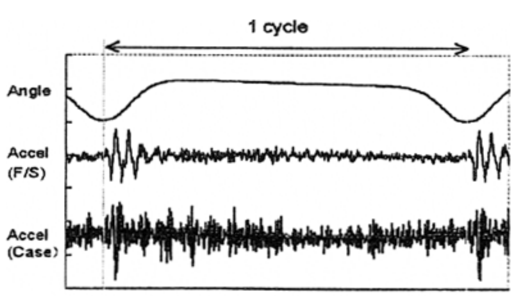
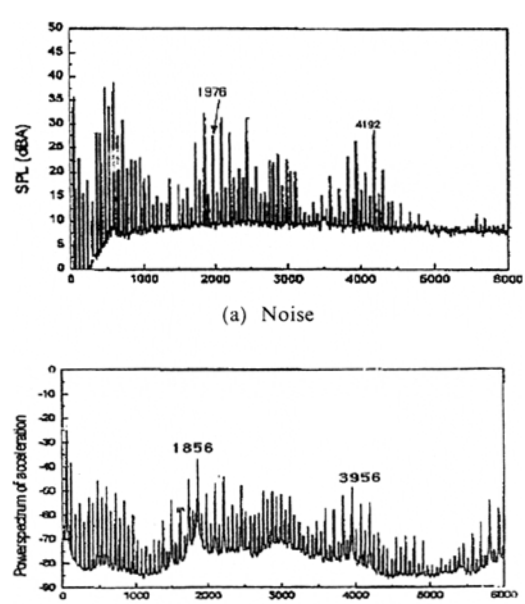


Fig. 14 Relation between angle and vibration-acceleration



(a) Noise

(b) Vibration

Fig. 15 Noise and vibration of motor

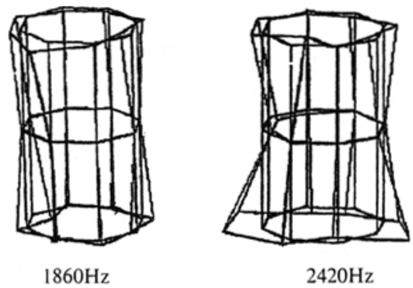


Fig. 16 Mode shape of case

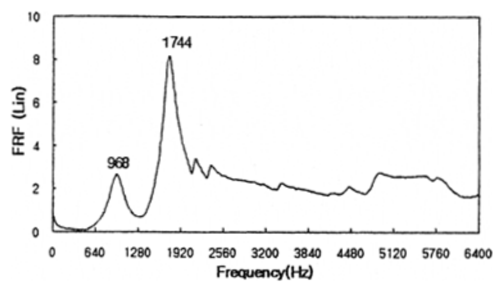


Fig. 17 FRF of rotor

2420Hz showing similar shapes to oval mode at the top and bottom. Such frequencies show a preeminent noise response excited by 1860Hz component of the electromagnetic exciting force of the motor. We measured FRFs to identify the characteristics of the stator and rotor. The stator's FRF was measured in free-free condition, while the rotor's FRF was measured being assembled with the crank shaft. The stator has its resonance frequency at 2.5kHz, the rotor at 968Hz and 1744Hz as shown in Fig. 17. The operating frequency in ARI condition being 58Hz～59Hz, the first electromagnetic frequency, in the case of 58Hz, appears at 1794Hz, 1914Hz and 2034Hz. Noise peaks are concentrated around 1750Hz ～1920Hz in the noise spectrum in ARI condition, which indicates that the electromagnetic exciting force and characteristics of the case and rotor have direct or indirect influences on it.

4. Investigation

The noise sources and transmission paths, drawn from the results of the experiments and analyses to identify the noise source, are shown in Figs. 18 and 19. Figure 18 exhibits the noise

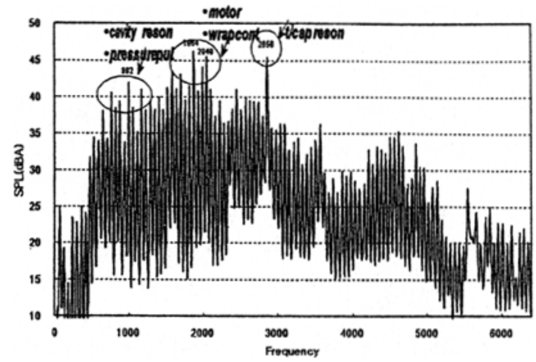


Fig. 18 Noise sources of main frequencies

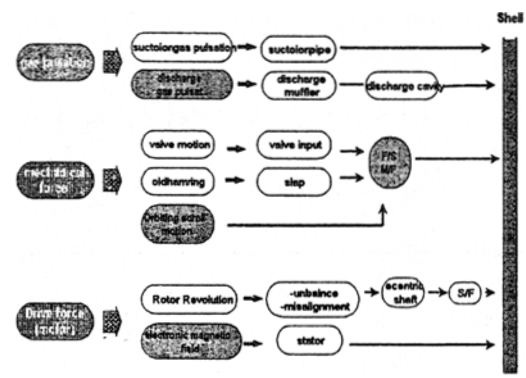


Fig. 19 Noise source and transmission path

sources of main peak frequencies.

The noise in 700Hz～1kHz is caused by the pressure pulsation of discharged gas, resonance of the discharge cavity and the resonance of the rotor. That of 1.6kHz～2.3kHz is created by resonance of the inner components and the case caused by the collision of scroll wrap and the electro-magnetic force of motor. That of 2.7kHz ～2.9kHz is caused by resonance of the top cap. Figure 19 shows the transmission paths of the main noise sources.

They are the noise generated through the top cap's getting excited by the pulsation of discharged gas, the noise generated by exciting the case through the case and the embedded main frame, by means of the collision of the wraps of the orbiting and the fixed scrolls, and the noise generated through the motor's electromagnetic force exciting the case. These noises come through a complex operation of exciting the case.

5. Conclusion

We performed analyses, experiments and the interpretation to identify the noise sources of the concerned frequencies, for the purpose of reducing the noise of scroll compressors. The followings are drawn as conclusions:

(1) The main exciting forces causing scroll compressors' noise are mainly composed of event-based noise caused by gas pulsation or mechanical collision.

(2) The pressure pulsation in discharge cavity exists in frequency region of lower than 800Hz, 1.6kHz～1.8kHz, and 2.4kHz ～2.7kHz. Its cavity resonance frequencies are around 700Hz, 1.2kHz, 1.5kHz, which does not affect the noise except for 700Hz.

(3) Resonance frequency of 2.7kHz at the top cap gets excited by the pulsation of discharge gas, causing high-level noises.

(4) Noise of 1.6kHz～2kHz band is mainly due to the effects of the motor's electro-magnetic noise and collision of the scroll wraps when suction begins, which excites the resonance of the case to cause high level noises.

(5) The waves of the collision peak seen in time domain, generated by the scroll wrap's collision at the starting point of suction in the scroll profile, are greatly affected by assemblage, and may have direct influences on the declination of noise.

It is planned to realize the noise reduction, based on the findings of this study.

References

Bush, J. W., Eyo, V. A. and Housman M. E., 1996, "Design Techniques and Resulting Structural Modifications used to Reduce Hermetic Compressor Noise," *Int. Comp. Eng. Conf. at Purdue*, pp. 967～975.

Iwata, H., Nakamura, M. and Matsusita, O., Sutou, M., 1990. "Vibration and Noise reduction of Compressors for Air-Conditioner," *Trans. of the JSRAE*, Vol. 7, No. 2, pp. 1～13.

Motegi, S. Nakashima, S., 1996. "A Study on Noise Reduction in a Scroll Compressor," *Int. Comp. Eng. Conf. at Purdue*, pp. 605～610.

Sano, K., Kawahara, S. and Akazawa, T., Ishii, N., 1997, "Experiment Study for Reduction of Noise and Vibration in Hermetic Compressor (1st Report: Reduction of Noise caused by Resonance of each Scroll Compressor Element," *Trans. of the JSRAE*, Vol. 14, No. 2, pp. 125～136.

Sano, K., Kawahara, S. and Akazawa, T., Ishii, N., 1997. "Experiment Study for Reduction of Noise and Vibrations in Hermetic Compressor (2nd Report: Noise due to Cavity Resonance in various Compressor Spaces)," *Trans. of the JSRAE*, Vol. 14, No. 2, pp. 137～148.

Sano, K., Kawahara, S. and Akazawa, T., Ishii, N., 1997. "Experiment Study for Reduction of Noise and Vibrations in Hermetic Compressor (3rd Report: Electro-magnetic Noise for Inverter Driven Compressors)," *Trans. of the JSRAE*, Vol. 14, No. 2, pp. 149～158.