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A Study On Noise Radiation From Compressor Shell

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ABSTRACT

The noise level of refrigerating units is one of the very important factors to determine the quality of products. The acoustic radiation of the compressor installed in the household appliances can be a significant contributor to the overall noise level. A major portion of the measured noise is originated from structural vibrations of the compressor shell. This paper deals with dynamic characteristics of the compressor shell with noise radiation properties. The vibration and radiated sound were measured for various operating speeds of the compressor. Based on the results of the modal tests and the waterfall diagrams, the correlations between the vibration characteristics of the shell and its noise radiation characteristics were identified. Present results show that the vibration of the compressor shell and the noise radiated from the compressor were strongly correlated in certain frequency bands. Moreover, a new vibration absorber was proposed to reduce the vibration of the shell, resulting in the reduction of the radiated noise.

1. INTRODUCTION

Recently, noise and vibration characteristics have become important factors for choosing electric home appliances. Especially, low noise and vibration have become an essential requirement for the quality of refrigerators. In the case of compressor noise, the noise sources are diverse, and its transmission paths are quite complicated. Therefore noise reduction of home appliances can accomplished through a systematic approach considering noise sources and transmission paths.

In this study, our goal is to reduce noise and vibration of the compressor in the specific frequency range over 2k~5kHz. For this, we first determined the elastic modulus of shell material by an inverse tracking scheme based on the comparison between the experimental results by modal testing and the analytic solutions by the Euler-Bernoulli beam theory. The specimen for the modal testing is made as a cantilevered beam, which was made by the same material used in the compressor shell. After finding the elastic modulus, modal analyses and modal tests of the compressor shell were carried out. Also, we carried out acoustic analyses by using SYSNOISE, the measured frequency response function for several RPM conditions. To reduce noise and vibration of shell of the compressor, we designed a new vibration absorber to reduce the vibration in the 2.5kHz band, and verified the effect of the absorber through FRF test.

2. MODAL ANALYSIS AND MODAL TEST

2.1 Transmission Paths of Compressor Noise

The interior noise sources of a compressor are transmitted to the shell, and then radiate to the atmospheric air. The primary sources of the noise consist of pressure pulsation within cylinders, unbalanced machinery, friction noises from lubricating parts, and magnetic forces of the motor. Secondary sources are the valve systems in the suction and discharging end, muffler and cavity resonance, etc. The vibration characteristics of the compressor shell, which is the final stage of the noise transmission, is the most important factor to be carefully considered in the design of the compressors shells. .

Modal analyses need material properties. The material of the compressor shell is SHP2, and its properties such as the elastic modulus can be changed during the manufacturing process. We measured the natural frequency of a cantilever of the shell material, and the elastic modulus was determined as $1.83\times10^{8} kg/mm^{2}$.

2.2 Modal Test : Free Boundary Condition

Noises of the compressor can be divided into structure-borne noise that comes from an unbalance force, magnetic force, frictional force, etc., and air-borne noise that comes from the cavity resonance between the electro-mechanical components and the shell. High frequency peak noises generated over 2 kHz corresponds to the resonance mode of compressor shell. In order to extract the natural frequencies and mode shapes of compressor shell, we carried out FEM analyses with IDEAS, and modal test by an impact method. In the modal test, 17 measurement points are selected. In order to find the natural frequencies, we compared the FEM solutions and experimental results by modal tests of the pure shell. The present results under free and mounted boundary conditions are shown in Table 2.1 and Table. 2.2, respectively.

2.3 Constrained Modes

The shell bracket is actually constrained to be fixed to the ground. Another FEM analysis of the compressor shell with the fixed bracket was performed to decide its effects to the vibration characteristics of the shell. Table 2.2 shows the comparison between FEM analysis and modal test.

Table 2.1 Natural frequencies (Hz) : free BC

Mode	Modal testing	FEM analysis
1	2451.6	2457.4
2	2553.9	2673.4
3	2791.4	2785.3
4	2831.7	2834.8
5		3183.1
6	3241.7	3204.2
7		3302.7
8		3446.5
9	3509.1	3463.1
10	3614.6	3494.7

Table 2.2 Natural frequencies (Hz) : constrained BC

Mode	Modal testing	FEM analysis
1	2556.3	2544.5
2	2693.8	2656.9
3	2881.3	2905.7
4		2964.1
5		3245.2
6	3287.5	3301.5
7	3359.4	3305.0
8	3412.5	3464.8
9	3553.1	3507.9
10	3625.0	3706.9

2.4 Modal Testing in Operating Conditions

In order to confirm the effects of the suspension, we did modal test of the shell in operating condition. Tests are carried out in three conditions: 1) compressor with suspension, 2) compressor with suspension and lubrication oil, and 3) compressor with suspension, lubrication oil, and refrigerant. Table 2.3 shows the results of natural frequencies.

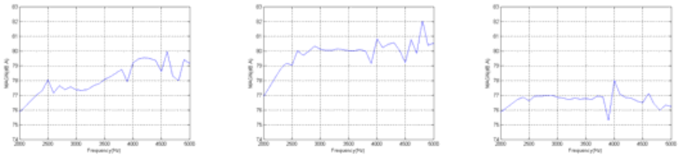
Table 2.3 Natural frequencies (Hz) : Modal test

Mode	Shell+Suspension (Free BC)	Shell+Suspension+Oil (Free BC)	Shell+Sus+Oil+Refrigerant (Constrained BC)
1	2457.0	2204.7	2262.5
2	2559.4	2521.1	2462.5
3	2793.1	2781.4	2543.8
4	2843.2	2883.2	2684.4
5	2951.8		2962.5
6	3079.3	3074.1	3137.5
7	3201.6	3214.6	3284.4
8	3412.7	3421.6	3440.6
9	3503.1	3594.2	3628.1
10	3647.2		3790.6

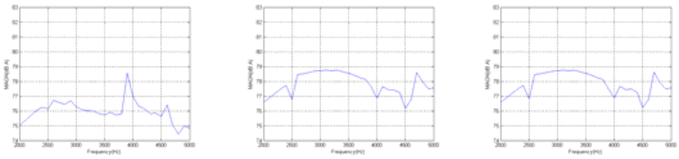
3. NOISE ANALYSIS

3.1 Noise Analysis of the Shell

We used SYSNOISE to analyze the characteristics of noise radiation from the shell. We established a field point 30cm away from the center of the compressor, and selected 6 measuring points, the front(C), top(A), back(D), bottom(B) and the left and ride sides(E, F). Fig. 3.1 shows the emitted noise at each point. In order to confirm the relation between structural resonance and radiation noise, the first mode shapes and radiation results are compared in Fig. 3.2.

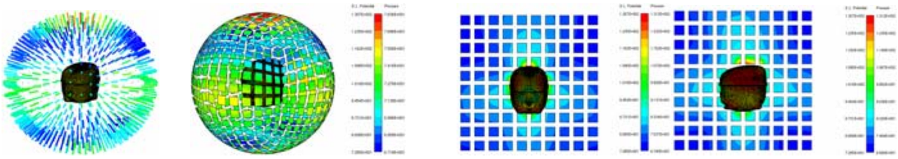


Position A (Top) Position B (Bottom) Position C (Front)



Position D (Back) Position E (Right-side) Position F (Left-side)

Fig. 3.1 Radiation noise

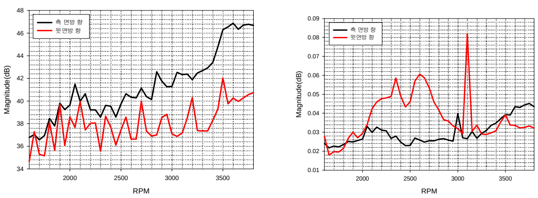


(a) A sphere with a radius of 30cm (b) y-z plane (c) x-z plane

Fig. 3.2 Mode shape & radiation noise at 1th Mode

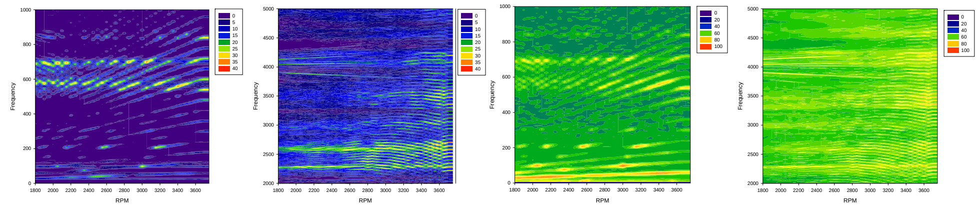
3.2 Structural Vibration and Radiated Noise

In this section, the structural vibration and radiated noise from the compressor in real operating condition is analyzed. We measured the radiation noise at 15cm away point from the shell cover, which is the half of the compressor height. Noise and vibration of the operating compressor were measured from 1600 RPM to 3800 RPM with 50 RPM interval. Fig. 3.3 shows the overall noise and vibration level at the side and top position with respect to RPMs. Fig. 3.4 shows a contour plot of the waterfall diagram. The radiation noise and vibration characteristics are shown in the low frequency range from 20Hz to 1kHz and the high frequency range from 2kHz to 5kHz. It shows the noise and vibration characteristics of the compressor. We can find global modes due to isolation rubber under 200Hz range. The flow noise due to compressor valves are found in 600~700Hz range. In the high frequency range (2000Hz~5000Hz), the structural resonances of the shell are dominant, and the peak noise at 4kHz is confirmed due to the carrier frequencies.



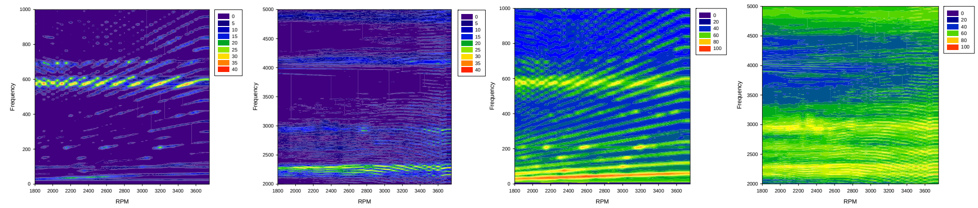
(a) Noise (b) Vibration

Fig. 3.3 Overall levels of noise and vibration at each RPM.



Frequency Range : 0~1000Hz 2000~5000Hz 0~1000Hz 2000~5000Hz

(a) Radiation Noise at the side position (b) Vibration at the side position

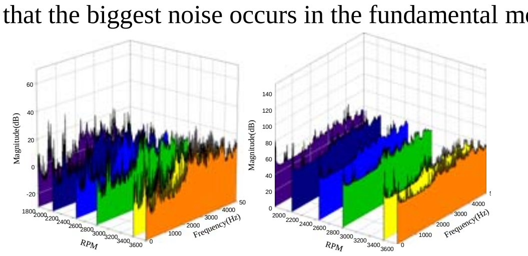


Frequency Range : 0~1000Hz 2000~5000Hz 0~1000Hz 2000~5000Hz

(c) Radiation Noise at the top position (d) Vibration at the top position

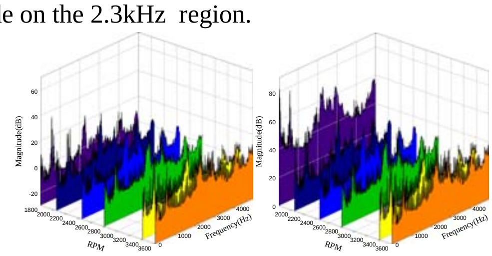
Fig. 3.4 Radiation Noise & Vibration of Compressor : Contour plot

Waterfall plots shown in Fig 3.5 and 3.6 show the emission noise and vibration level at the side and top. The noise analysis shown in Fig. 3.2 indicates that radiation noise comes largely from the top part, and from Fig. 3.6 we can confirm that the biggest noise occurs in the fundamental mode on the 2.3kHz region.



(a) noise (b) vibration

Fig.3.5 Noise and vibration of the side position



(a) noise (b) vibration

Fig.3.6 Noise and vibration of the top position

4. VIBRATION ABSORBER

Noise analyses show that the noise level from the bottom shell is dominant compared to the other part. To suppress the vibration of bottom plate, a new vibration absorber shown in Fig. 4.1 is proposed. The effect of thickness of absorber is studied to reduce the sound level in 2.5kHz band.

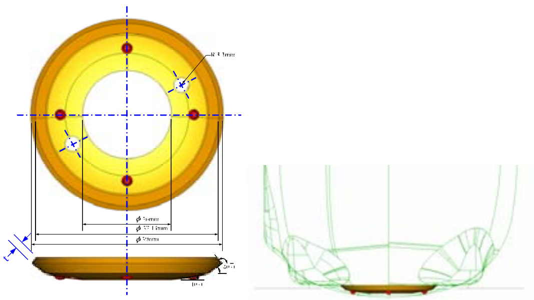
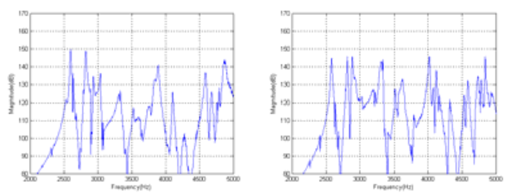


Fig.4.1 Vibration absorber



(a) FRF of shell+absorber : 2.9t (b) FRF of shell+absorber : 2.0t

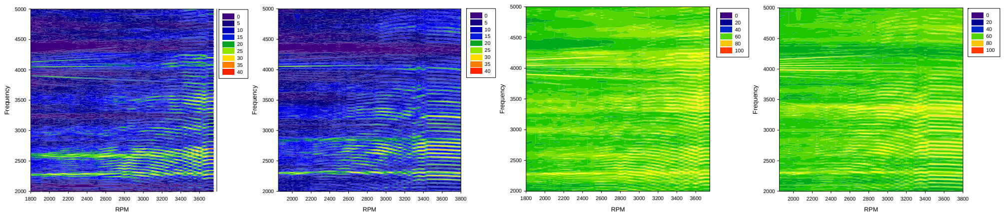
Fig. 4.2 FRF of shell+absorber

Table 4.1 is the results of modal analyses for the absorber according to the thickness variation. From these results, the optimal value of the absorber thickness is determined as 2.0mm to reduce the noise in 2500Hz band. To check the effect of absorber on vibration and noise characteristics, modal test is carried out for the compressor shell with absorber. The results are given in Fig. 4.2 as FRF curves. It shows that the vibration of the shell with absorber of thickness 2.0t is dramatically reduced in 2500Hz band , but increased in 3000Hz band, compared to the one of 2.9t which is currently used.

To check the effect of vibration and noise reduction, we measured the vibration and noise in the actual operation condition. The results are given in Fig. 4.3, as a contour plot. It shows that noise characteristics of the shell with the absorber of thickness 2.0t is better in 2500Hz band, and slightly increased in 3000Hz band, compared to that of 2.9t It is believed that absorber of 2.0t thickness can accomplish a good effect for reduction of vibration and noise level in the 2500Hz band, which is our goal.

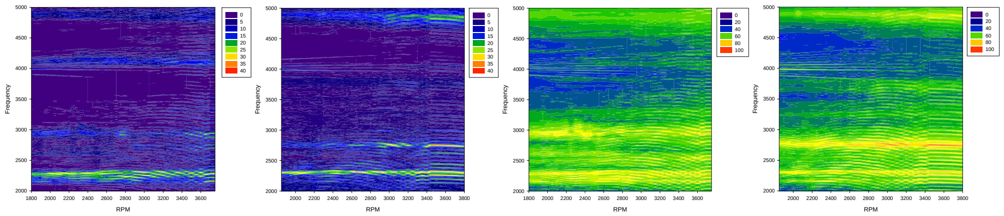
Table 4.1 Comparison of natural frequencies of absorber

	Free-free Condition		Clamped-free Condition	
Mode	2.9t	2.0t	2.9t	2.0t
1	2156.4	1670.4	3885.2	3030.3
2	2169.5	1678.3	5261.1	4228.8
3	5736.8	4477.5	5278.5	4248.8
4	5746.1	4493.3	7415.9	6189.9
5	8018.1	7442.6		8869.8



Absorber thickness : 2.9t 2.0t 2.9t 2.0t

(a) Noise Radiation at side position (b) Vibration at side position



Absorber thickness : 2.9t 2.0t 2.9t 2.0t

(c) Noise Radiation at top position (d) Vibration at top position

Fig. 4.3 Noise Radiation & Vibration of Compressor

CONCLUSION

We examined the noise and vibration characteristics of the compressor shell in order to reduce noise radiation from the compressor shell. We carried out FEM analysis and modal tests, and confirmed that the resonant frequency of the compressor shell structure appears in the frequency ranges above 2 kHz. In order to analyze the radiation noise due to the structural vibration of the compressor shell, we used SYSNOISE to do noise analysis. We measured the noise and the vibration from the compressor in operating condition, and compared with the result of noise analysis. Since the noise radiated from the bottom plate is dominant, a new vibration absorber is proposed to suppress the peak noises due to the structural resonance. The absorber is designed to tune a resonant frequency in 2500Hz band which is our goal. With this absorber, we confirmed a reduction of vibration and noise in the prescribed band.

REFERENCES

Rao, Singiresu S., 1995, *Mechanical Vibrations*, Addison Wesley, p. 526-527.

Migeot, J.L., Jones, M, 2001, *Noise Radiation by a Refrigerator Compressor*, LMS North America.

Tojo, K, Machhioa, S, and Saegusa, S, 1980, Reduction of Refrigeration Compressor, *Purdue Comp. Tech. Conf.*