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Research On Low Frequency Vibration Of Rotary Compressor

Zhiming Wen
GREE ELECTRIC APPLIANCES, INC. OF ZHUHAI, China, People's Republic of, madicus@163.com

Rongting Zhang
GREE ELECTRIC APPLIANCES, INC. OF ZHUHAI, China, People's Republic of

Yaosi Zhang
GREE ELECTRIC APPLIANCES, INC. OF ZHUHAI, China, People's Republic of

Huanhuan Gu
GREE ELECTRIC APPLIANCES, INC. OF ZHUHAI, China, People's Republic of

Yusheng Hu
GREE ELECTRIC APPLIANCES, INC. OF ZHUHAI, China, People's Republic of

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Zhiming Wen, Rongting Zhang, Yaosi Zhang, Huanhuan Gu, Yusheng Hu

Compressor and Motor Institute of Gree Electric Appliances, Inc. of Zhuhai,
Jinji West Rd., Zhuhai City, 519070, P. R. China
Phone: +86-756-8669017, Fax: +86-756-8668386,
E-mail: madicus@163.com

ABSTRACT

The abnormal noise of an outdoor domestic air-conditioner operating at low speed is experimentally analyzed. The structure-borne noise which passes through the mounting system is confirmed to be the main source of the abnormal noise due to the large low frequency vibration on compressor foot. Then the characteristic of low frequency vibration of rotary compressor including the dynamic model, exciting forces and dynamic response is researched. Based on this, mounting system including compressor foot and rubber grommet is optimized to solve this problem, more than 8dB reduced.

1. INTRODUCTION

With the popularization of the inverter-driven air-conditioner and the requirement of comfort and energy saving, the operating frequency of inverter-driven air-conditioner is lower and lower, even to 1Hz. At the same time, the concern about the noise of air-conditioner is higher and higher. So noise level is the important factor of the air-conditioner performance [1].

Generally, rotary compressor is a main noise source of air-conditioner outdoor unit, which is composed of two components, airborne noise and structure-borne noise. The airborne noise is directly radiated from the compressor shell. The structure-borne noise is caused by the vibration transmitted through mounting system, suction pipes and discharge pipes that are linked to the compressor.

To a certain domestic air-conditioner, when the compressor operates at 1440-1980 RPMs, the noise level of outdoor unit increases obviously comparing to other RPMs. The structure-borne noise caused by the mounting system of compressor is confirmed by experimental results. First, it was found from the experimental data that the low frequency vibration of compressor plays an important role in the abnormal noise. Next, the low frequency vibration of rotary compressor is analyzed. Finally, the mounting system of the compressor is optimized and a decrement of 8.9 dB in the sound pressure is achieved.

2. ABNORMAL NOISE AND IDENTIFICATION

2.1 Experiment Environment

Noise and vibration experiments are taken to the outdoor unit in semi-anechoic room. Two microphone sensors are set at two points as shown in Figure 1.

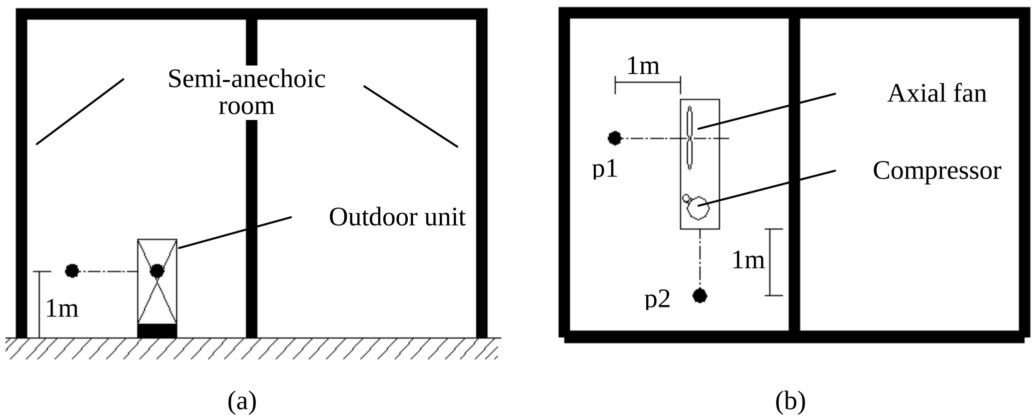


Figure 1: (a) Experiment environment. (b) Top view of (a).

2.2 Characteristic of the abnormal noise

The sound pressure level is measured from 8 to 90Hz, which is the rotation frequency of the rotary compressor. As show in Figure2, the sound pressure levels both at p1 and p2 increase sharply when the compressor operating from 24 to 33Hz (1440-1980 RPMs).

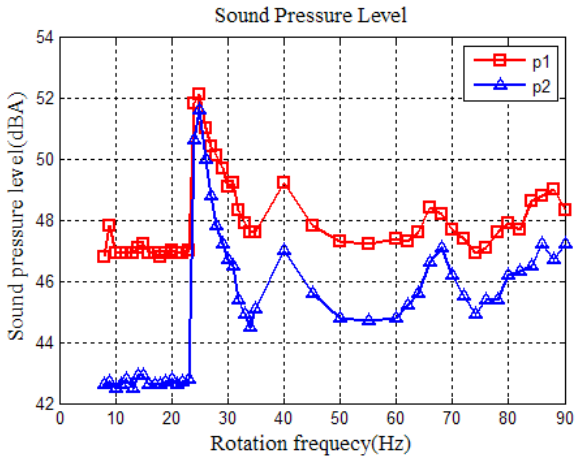
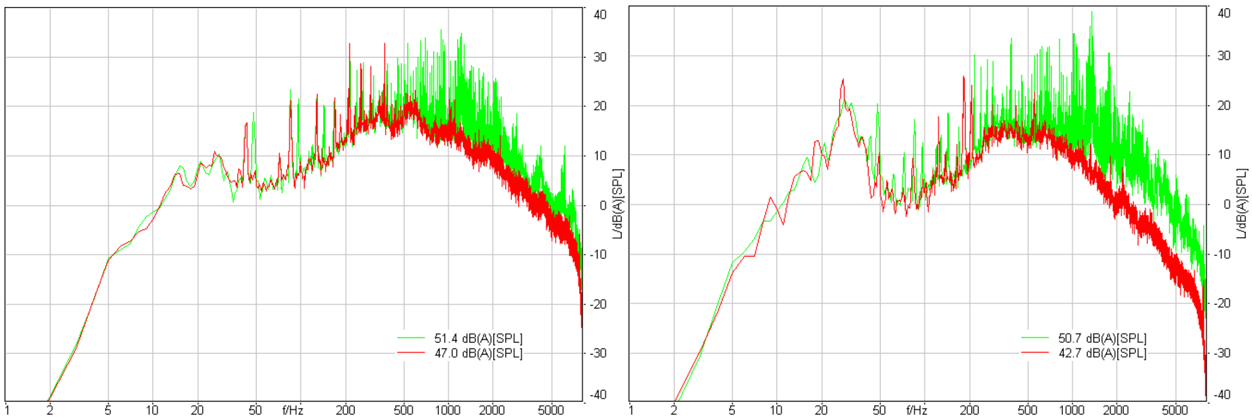


Figure2: Measured sound pressure level at p1 and p2

It can be confirmed that the abnormal noise is a problem of broad band spectrum upwards 200Hz when comparing the sound pressure spectrum at rotation frequency 23Hz with 24Hz, as shown in Figure3.



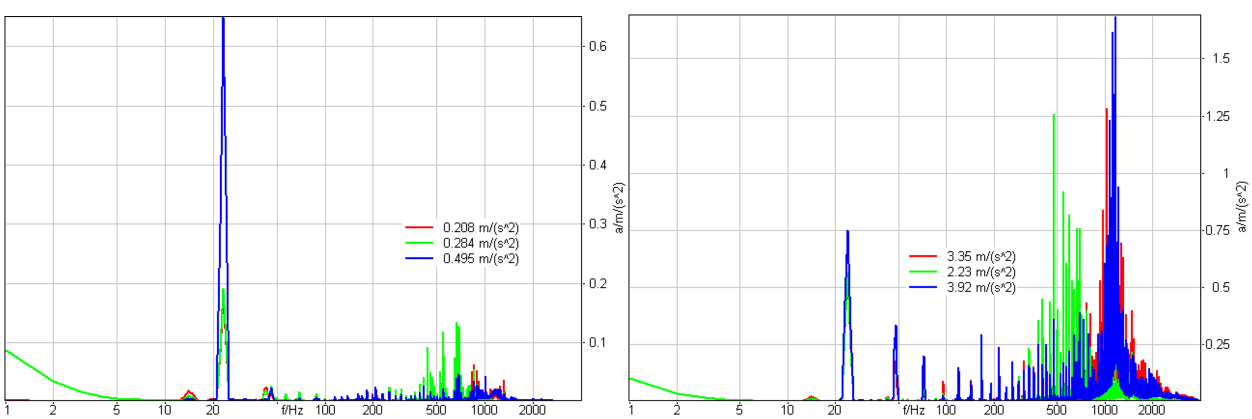
(a) p1: — 24Hz — 23Hz. (b) p2: — 24Hz — 23Hz.

Figure3: Sound pressure spectrum at different rotation frequency.

2.3 Noise source identification

Firstly, the axial fan can be eliminated as the source of the abnormal noise because the rotary speed of axial fan do not change when the rotation frequency of compressor turning from 23 to 24Hz. By the same reason, the airborne noise of compressor will not change so sharply just when the rotation frequency of compressor increases 1Hz. Thus, the structure-borne noise is the only possible reason. Finally, the structure-borne noise caused by mounting system of compressor is confirmed as the real reason by checking the vibration of pipes and unit base.

As shown in Figure 4, the acceleration at unit base increases obviously just like the characteristic of sound pressure spectrum when the rotation frequency of compressor turning from 23 to 24Hz.



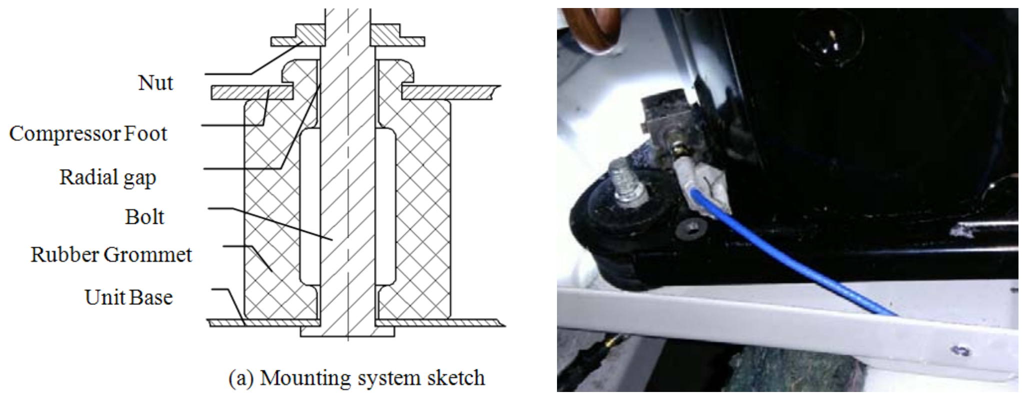
(a) 23Hz (b) 24Hz

Figure 4: Measured acceleration at unit base

(— Radial direction — Axial direction — Tangential direction)

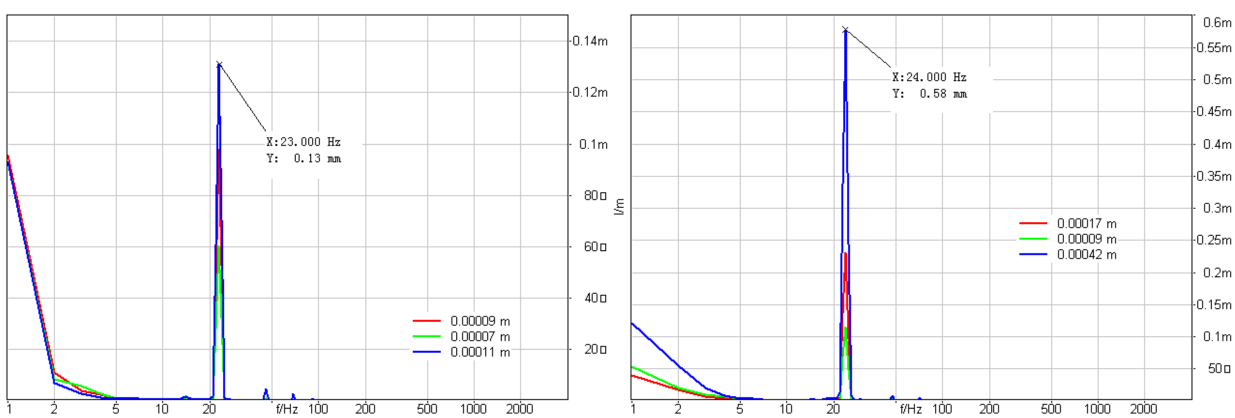
Furthermore, the 1st order tangential vibrating displacement at compressor foot increases obviously when the rotation frequency of compressor turning from 23 to 24Hz as shown in Figure 6. A rigid link between compressor foot and bolt will be formed if the low frequency tangential vibrating displacement is bigger than the actual radial gap between rubber grommet and bolt as shown in Figure 5. High frequency vibration of compressor will be directly transmitted to the unit base through the bolt while rigid link formed between compressor foot and bolt.

The low frequency vibration of compressor plays a very important role in the sound pressure level of outdoor unit.



(a) Mounting system sketch (b) Measured unit

Figure 5: Compressor mounting system



(a) 23Hz (b) 24Hz

Figure 6: Measured displacement at compressor foot

(— Radial direction — Axial direction — Tangential direction)

3. LOW FREQUENCY VIBRATION ANALYSIS

3.1 Dynamic model

To a dynamic system, the dynamic characteristic can be described by equation (1).

$$[M]\{\ddot{q}\}+[C]\{\dot{q}\}+[K]\{q\}=\{f\} \tag{1}$$

Generally, the first flexible mode of a rotary compressor is about 200Hz. So the low frequency vibration is nearly decided by the rigid modes of a compressor. As we all know, there are 6 rigid modes which can be calculated by equation (2) and (3) when ignoring the damping of a dynamic system.

$$[M]\{\ddot{q}\}+[K]\{q\}=0 \tag{2}$$

Eigen function can be expressed as follow:

$$M^{-1}K\Phi=\lambda\Phi \tag{3}$$

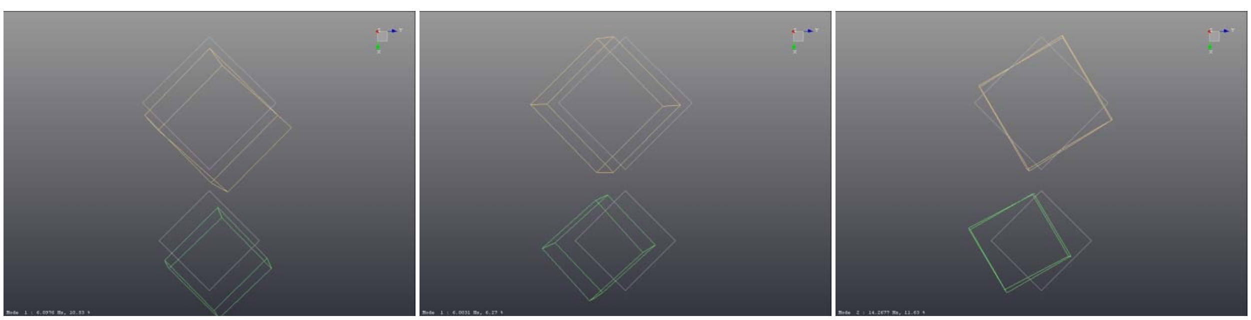
where M and K are the global stiffness and mass matrices, respectively; λ is the eigenvalues, and Φ is the eigenvectors.

Then, low frequency vibration response under a certain force $\{f\}$ can be calculated by equation (4) when considering modal damping coefficient c_i .

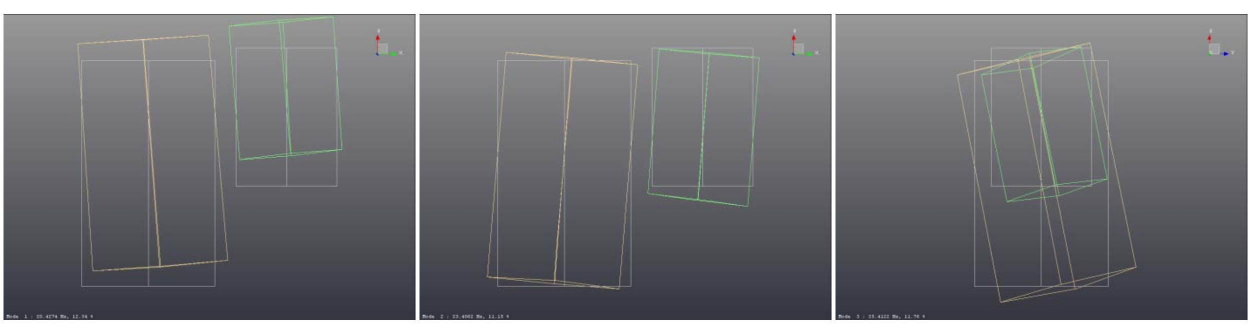
$$\{X\}=\sum_{i=1}^{6}\frac{\{\phi_i\}\{\phi_i\}^T\{f\}}{\left(k_i+j\omega c_i-\omega^2 m_i\right)} \tag{4}$$

where $\{\phi_i\}$ is the i-th eigenvector; k_i and m_i is the i-th modal stiffness and mass.

Six rigid modes are identified by impact testing of the rotary compressor as shown in Figure 7. The theoretical calculated result is accurate enough comparing to the experiment result as shown in Table 1.



(a) 1st rigid mode (b) 2nd rigid mode (c) 3rd rigid mode



(d) 4th rigid mode (e) 5th rigid mode (f) 6th rigid mode

Figure7: Measured 6 rigid modes of the rotary compressor.

Table 1: Rigid modes of a certain rotary compressor (Hz)

Rigid mode	Calculated	FEM	Measured
1	6.7	6.7	6
2	6.8	6.7	6.1
3	14.2	15.0	14.3
4	20.1	20.0	20.4
5	23.1	23.0	23.4
6	24.7	24.1	25.4

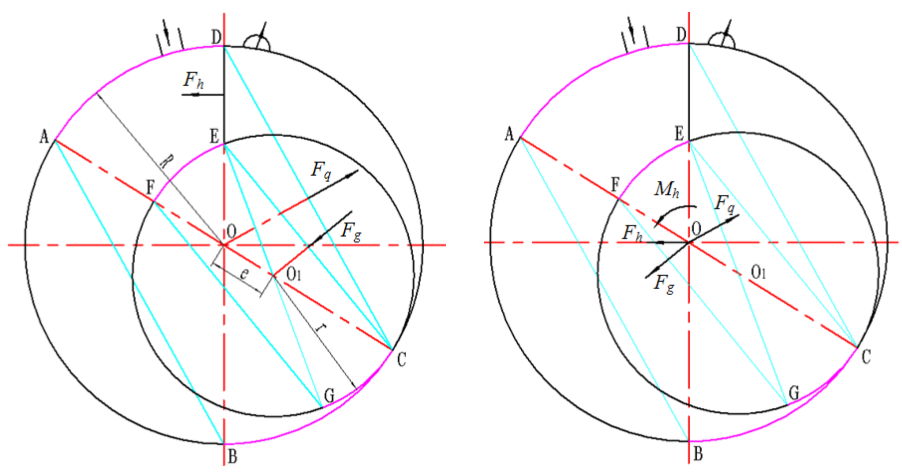
3.2 Low Frequency Forces

The low frequency forces should be discussed firstly in order to calculate the low frequency vibration. The main forces such as gas force and moment, centrifugal inertia forces, magnetic force and torque are considered based on the structure of a rotary compressor. Finally, it is found that the unbalanced axial moment is the main force deciding the low frequency vibration.

(1) Gas forces and moment

Considering the model shown in Figure 8 (a), where F_h is the gas force on blade; F_q is the gas force on cylinder; F_g is the gas force on crankshaft. As a rigid body, moving these forces to the centre of cylinder and considering the forces concentrated on cylinder as shown in figure 8 (b). The F_g do not generate an additional moment on cylinder because only radial force can be transmitted through the top and bottom bearings when taking no account of friction.

After calculation, it is found that the radial gas forces concentrated on cylinder are balanced forces and only the unbalanced moment M_h is left. Furthermore, M_h is equal to the gas moment on crankshaft with opposite direction.



(a) (b)

Figure 8: Simplified sketch of gas force

(2) Centrifugal inertia forces [2]:

Centrifugal inertia forces caused by rotational eccentric parts can be calculated by equation (5). Generally, these forces can be neglected when the rotation speed is not too high because they are designed to be a nearly balanced force.

$$F_i = m_i \omega^2 r_i \tag{5}$$

where m_i is mass of the eccentric parts (a pair of balancers, a rolling and a crankshaft eccentric part), ω is speed of rotation, r_i is the distances from the mass centre of the eccentric parts to the axis of revolution.

(3) Magnetic force and torque:

Theoretically, no dynamic concentrated radial magnetic force exists if no dynamic eccentricity existing between the axes of rotor and stator. Also the concentrated radial magnetic forces on stator and rotor are a pair of balanced forces to a rigid compressor when considering the low frequency vibration.

The magnetic counter torque on stator is a constant value and will not influence the low frequency vibration if no torque compensation algorithm is taken to the invert controller as shown in Figure 9.

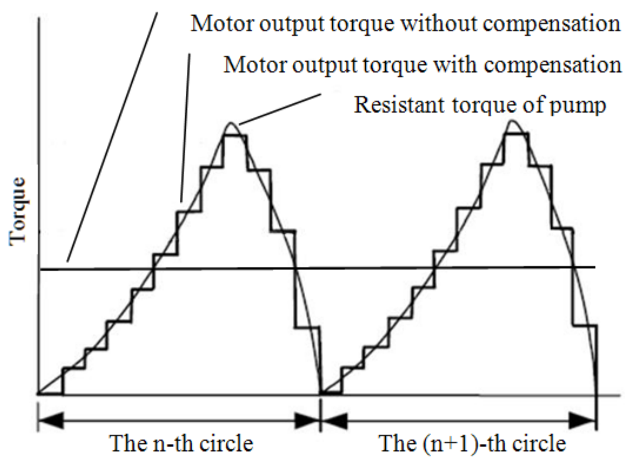
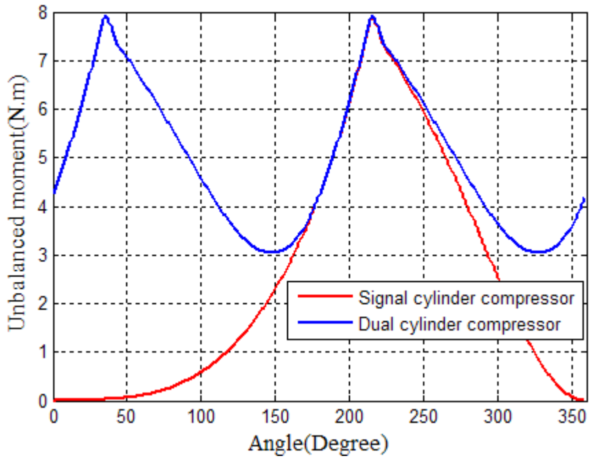


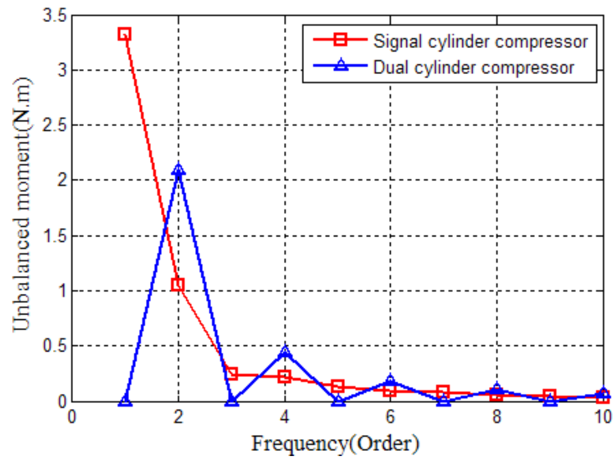
Figure 9: A sketch of torque compensation control [3]

As discussed above, the main force which influencing the low frequency vibration is the unbalanced moment. It can be calculated more accurate when using a commercial program to take account of other factors such as friction etc.

A theoretical comparison of unbalanced moment between single and dual cylinder rotary compressor is presented in Figure 10. The unbalanced moment of a signal cylinder compressor is even bigger than a dual cylinder compressor which owning double displacements and its frequency is half lower at the same rotation frequency. Single cylinder compressor is used widely in the domestic air-conditioner because of its lower cost although it may cause worse vibration problem when the unbalanced moment value is bigger and the frequency is lower.



(a) Unbalanced moment curve



(b) FFT analyzed spectrum

Figure 10: Theoretical calculated unbalanced moment

3.3 Characteristic of low frequency vibration

Firstly, the coordinate is defined as shown in Figure 11 where the *w* and *Z* axes are decided by the right hand rule.

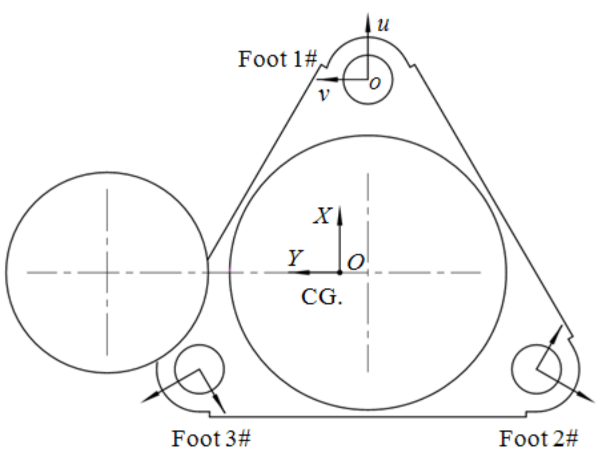
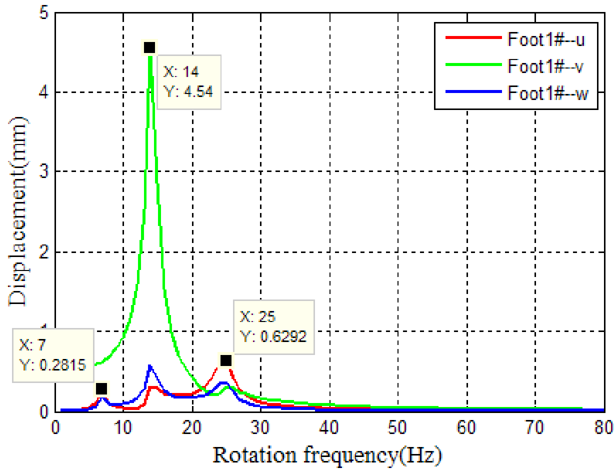


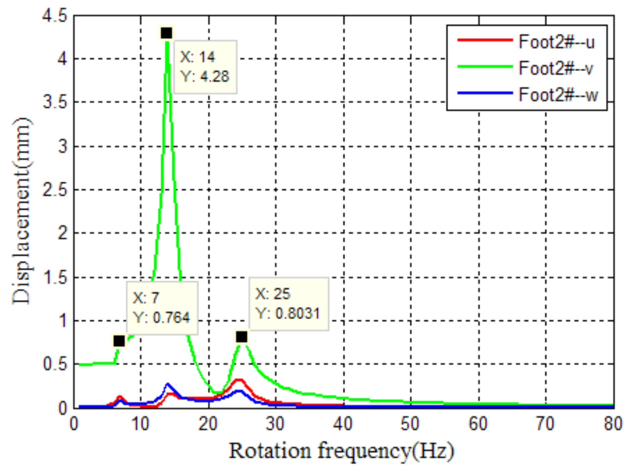
Figure 11: Coordinate definition

The vibrating displacement responses at three feet under the load of unit axial torque are presented in Figure 12(a), (b) and (c). Three response peaks appear at about 7, 14 and 25Hz which are near the natural frequency of the 2nd, 3rd and 6th rigid modes. The tangential vibrating displacement at each foot and at each frequency is nearly the biggest response against the other two directions. The maximum value of tangential vibrating displacement appears at 14Hz which is the natural frequency of the 3rd rigid mode.

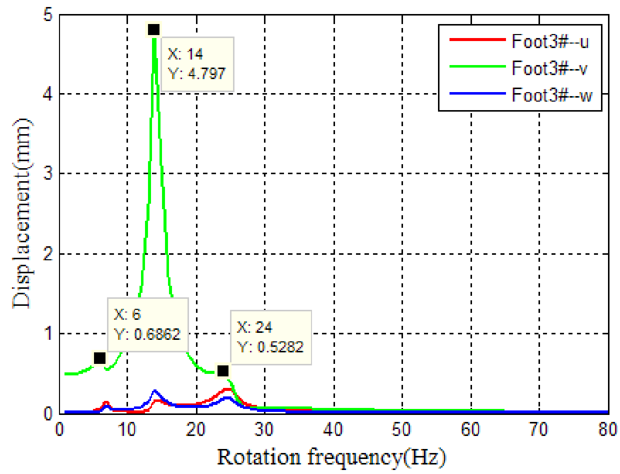
Furthermore, the characteristic of vibration under unit axial torque can be explained by discussing the data given in Table 2. The 3rd rigid mode has the largest energy distribution on the θ_Z direction which is the direction of the loaded torque. Also, the 3rd rigid mode is coupled with the 2nd and 6th rigid modes weakly under the defined coordinate. So the 3rd rigid mode will be resonated strongly when a dynamic axial torque with corresponding frequency acting on the compressor. The 2nd and 6th rigid modes will also be resonated weakly because of the weak coupling.



(a) Response at Foot 1#



(b) Response at Foot 2#



(c) Response at Foot 3#

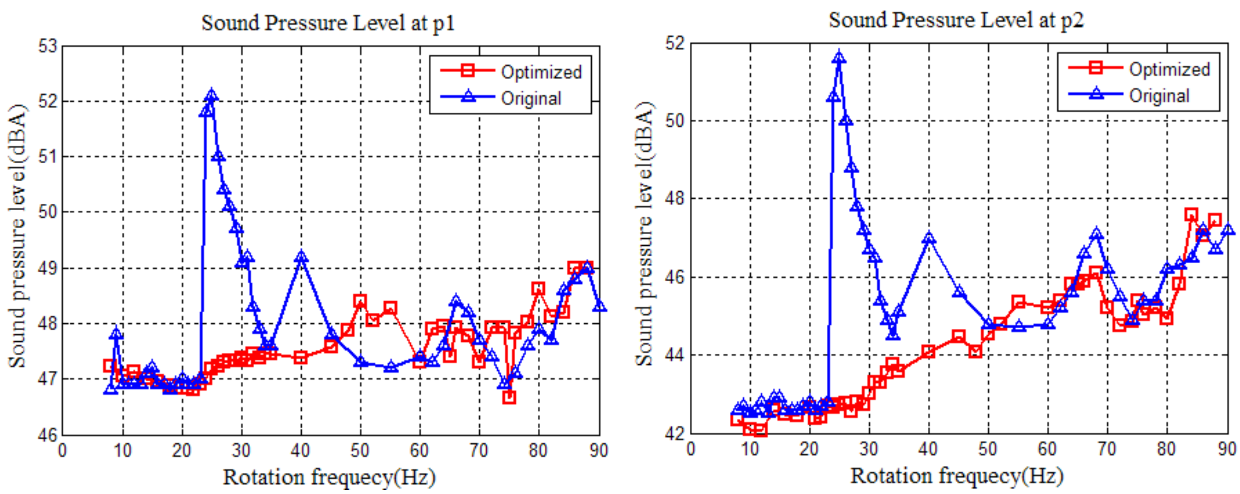
Figure 12: Vibration response under unit axial torque

Table 2: Natural frequency and energy distribution of rigid modes

Natural Frequency (Hz)		**6.7**	**6.8**	**14.2**	**20.1**	**23.1**	**24.7**
Energy Distribution (%)	X	0.04	92.11	0.23	0.00	0.00	7.61
	Y	89.52	0.04	0.00	5.41	5.03	0.00
	Z	0.27	0.00	0.00	63.15	36.56	0.01
	θ_X	10.16	0.00	0.00	31.43	58.38	0.02
	θ_Y	0.00	7.84	5.31	0.00	0.03	86.82
	θ_Z	0.00	0.00	94.46	0.00	0.00	5.54

4. IMPROVEMENT AND VALIDATION

Finally, the mounting system is optimized which including the structure of compressor foot and rubber grommet considering the abnormal noise as well as the operational stress level of suction and discharge pipes. The sound pressure levels of optimized unit is improved obviously as shown in Figure 13, 4.9 and 8.9 dB reduced at measured points p1 and p2 when the compressor operating at 25Hz.



(a) p1 (b) p2

Figure 13: Sound pressure level comparison

5. CONCLUSIONS

In this paper, the abnormal noise of a domestic air-conditioner outdoor unit when the compressor operating at certain low RPMs is experimentally analyzed. The structure-borne noise which passes through the mounting system is confirmed to be the main source of the abnormal noise due to the large low frequency vibration on compressor foot. Characteristic of low frequency vibration is researched including the dynamic model, low frequency forces and dynamic response. Based on this， mounting system including compressor foot and rubber grommet is optimized to solve this problem, 4.9 and 8.9 dB reduced at measured points p1 and p2 when the compressor operating at 25Hz.

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