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ABSTRACT

In the noise and vibration test of 8HP rotary compressor, the peak value of noise and vibration is especially high at frequency range between 600~1000Hz. The sound pressure level (SPL) exceeds the enterprise criterions and the sound quality is so tough that persons cannot bear such annoying noise. Two experiments, pressure pulsation above/below compressor motor and mechanical impacts of compressor pump-rotor unit, are particularly carried out in order to identify the noise sources. The test results confirm that the axial up-down movement of pump-rotor unit is the primary reason, causing the abnormal noise and vibration. Several measures, including partially cutting the circumferential edge of the stator and increase the height differential between rotor and stator, have been taken in order to reduce pressure pulsation above/below motor and restrain the axial movement of shaft-rotor unit. With these measures, the noise problem of 8HP rotary compressor has been solved successfully. Finally the sound pressure level decreases about 4dB and the sound quality is comfortable and good to hear.

1. INTRODUCTION

Traditionally, usage of multiple scroll compressors is widely applied for general variable refrigerant flow (VRF) system at more than 5HP capacity because of its performance and large capacity advantages. But usage of multiple scroll compressors have problems of controlling cost, which greatly limits the further expanding of commercial VRF system market. Fortunately, with the development of technology and progress of manufacture equipment, inverter rotary compressors are used not only in house-hold air conditioner market but also in commercial VRF market because of its efficiency, comfort, simple structure, low cost, etc. Recently, 5HP~12HP inverter rotary compressors, which can operate in a wide running range from low speed to high speed and from low load to high load, are developed successfully to replace similar capacity scroll compressors due to the need of cutting cost.

What is more, with the capacity getting larger and larger, more attentions should be paid to noise and vibration controlling, so how to diagnose the abnormal noise rapidly and correctly is an urgent issue for large capacity rotary compressor. In addition, the strong requirements for low noise product of customers have also enforced many compressor makers worldwide to focus on development of low noise compressor. As compressor is the main noise source of air conditioner, therefore the noise and vibration is a big concern over the course of development [1].

In this study, noise, vibration, various forces, pressure pulsation, mechanical impacts were investigated of 8HP rotary compressor. In the noise and vibration test of the compressor, the peak value was especially high at frequency range between 600~1000Hz. The sound pressure level (SPL) achieved 80.3dB with its operating speed of 60Hz, which exceeded the enterprise criterions. The sound quality was so tough that persons can't bear such annoying noise. Two experiments, pressure pulsation above/below compressor motor and mechanical impacts of compressor pump-rotor unit, were particularly carried out for the purpose of identifying the noise source. The results confirmed that the axial up-down movement of pump-rotor unit was the primary reason, causing the abnormal noise.

According to forces analysis of pump-rotor unit and two experiments, several measures had been taken, especially the height differential between rotor and stator was increased to improve the axial magnetic force dramatically, which could overcome the discharge gas force. With these measures, the noise problem of 8HP rotary compressor has been solved successfully. As a result, the peak value disappeared at frequency range between 600~1000Hz and the sound pressure level reduced approximately by 4.1dB.

2. NOISE PROBLEM DESCRIPTION

In order to diagnose the abnormal sound, noise and vibration were measured of the 8HP, large capacity compressor in hemi-anechoic room. Figure 1, Figure 2 shows the noise spectrum and vibration spectrum respectively, from which several key characteristics of the noise problem can be observed.

Firstly the high peak values mainly concentrate at frequency range between 600~1000Hz both in noise and vibration spectrum. Secondly the most serious noise problem occurs at around the operating speed of 60Hz, in which the peak value reaches to about 64.4 dB. At last, the vibration spectrum is particularly remarkable. Unlike the spectrum of X, Y direction of compressor pump, the spectrum of Z direction is extraordinary high at 600~1000Hz. Thus a conclusion can be made, that is the abnormal vibration of Z direction is the prime reason for the noise problem. This study is designed to investigate the source of the vibration issue and focus on reducing it.

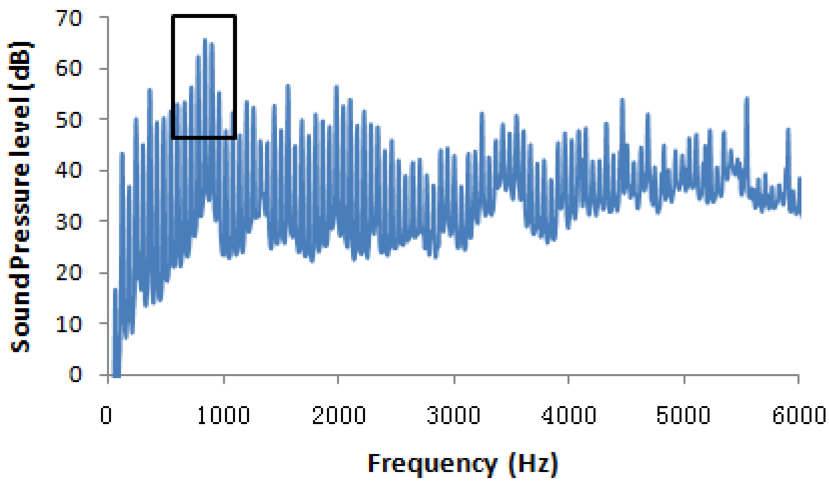

Figure 1: Noise spectrum of 8HP compressor at 60Hz running

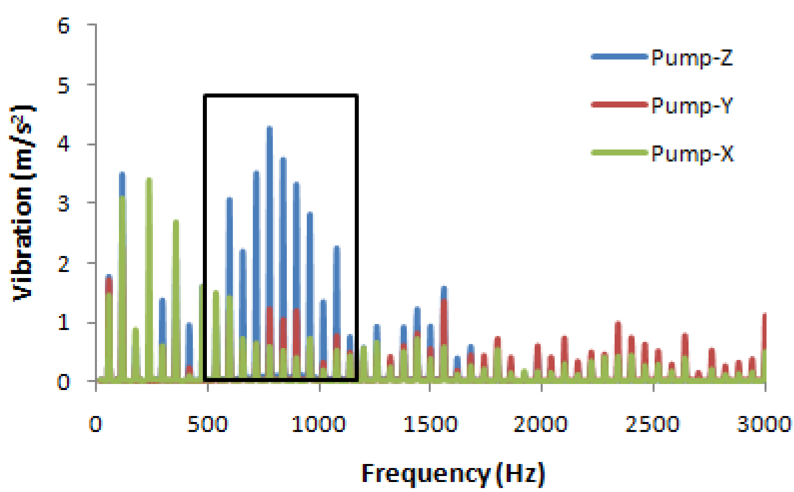

Figure 2: Vibration spectrum of 8HP compressor at 60Hz running

3. FORCE ANALYSIS AND PRESSURE PULSATION EXPERIMENT

According to the results of above noise and vibration measurement, the pump-rotor unit may move up and down in cylinder under some rather unique condition. Of all components for pump-rotor unit, the unstable movement of shaft-rotor most probably causes axial displacement up and down, which could result in serious axial impact vibration [2]. In order to investigate the mechanism and relationship between axial movement of shaft-rotor unit and the abnormal noise problem, a detailed analysis is taken on correlative forces of shaft-rotor unit, including pressure pulsation, axial magnetic force, gravity, damping effect.

3.1 Axial Force Analysis of Shaft-Rotor Unit

In discharge period of rotary compressor, the refrigerant flows into the chamber under the motor, and then reaches the chamber above the motor through motor passages. The pressure in chambers above and under the motor is different from each other, which may cause axial displacement of the shaft-rotor unit when the pressure differential force is larger than self-weight of the shaft-rotor unit and axial magnetic force of the motor. The axial motion of the shaft-rotor unit shown in Figure 3 is expressed by the following equation:

$$m\frac{d^2X}{dt^2} = \Delta PS_r - mg - C\frac{dX}{dt} - F_m \quad (1)$$

Where, m :mass of the shaft-rotor unit, X :axial displacement, ΔP :differential pressure ($\Delta P = P_2 - P_1$), S_r :sectional area of the rotor, C :damping coefficient of vibration, F_m :axial magnetic force of the motor.

The axial magnetic force is determined by the height differential Δh between rotor and stator shown in Figure 3. In addition, simulations of axial magnetic force have been completed to estimate the effect of height differential and motor torque. Fig.4 shows that the axial magnetic force increases greatly with the increasing of height differential between rotor and stator, but the increasing trend is not linear relationship. Fig.5 indicates that the axial magnetic force increases linearly with motor torque for a certain 4mm height differential.

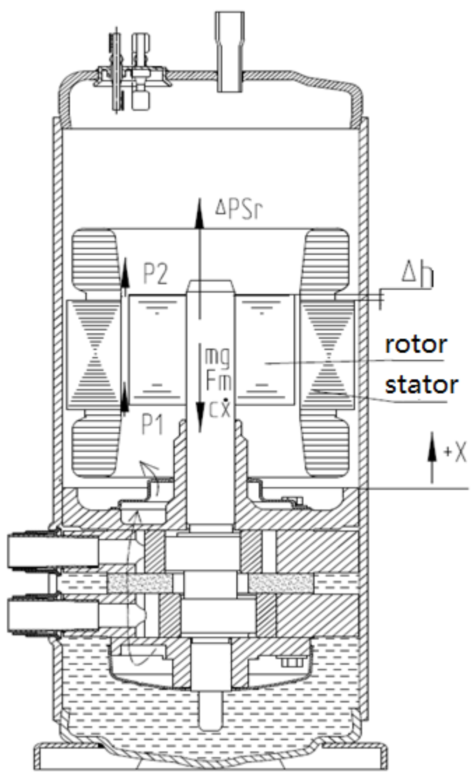

Figure 3: Axial force analysis of shaft-rotor

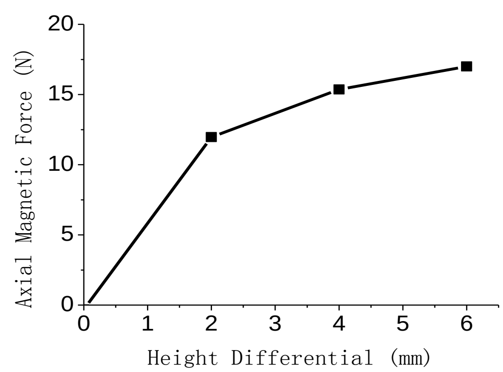

Figure 4: Axial magnetic force with height differential

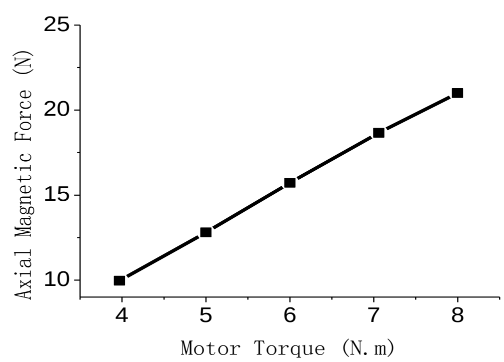

Figure 5: Axial magnetic force with motor torque

3.2 Pressure Pulsation Experiment

In recent years, with the development of high speed, large capacity compressor, the discharge pressure pulsation becomes more and more serious, which may induce various problems. What's more, due to the intermittent discharge, pressure pulsation has a great influence on the axial movement of shaft-rotor unit, especially for two-cylinder, large capacity rotary compressor. Experiment of pressure pulsation was carried out with two pressure sensors in chambers above and below the compressor motor.

Fig.6 shows the experimental results of the differential pressure ΔP between the lower chamber pressure P_1 and the upper chamber pressure P_2. As shown in Fig.6, the maximum differential pressure approaches 0.02MPa, which may introduce great impact force on pump-rotor. The axial discharge force depends on pressure pulsation in chambers above and below the motor. When the lower chamber pressure P_1 overcomes the upper chamber pressure P_2, the pressure differential force will push the shaft-rotor unit move upward, otherwise the pressure differential force will push the shaft-rotor unit move downward. However, the axial up-down displacement most probably takes place when the load condition is very tough and the pressure differential force exceeds self-weight of the shaft-rotor unit. In such situation, abnormal vibration of Z direction would be quite obvious and the noise problem emerges at the same time.

By the way, the damping force will not be discussed in detail of this paper, because the damping coefficient depends on the thickness of oil film, which is hard to provide. Although there have been some investigations on the value of the flow damping coefficient, but their results cannot been applied directly to this study. Concerning damping force of the vibration, squeeze force of oil film on the shaft thrust is superior to shearing force of oil film on the shaft journal.

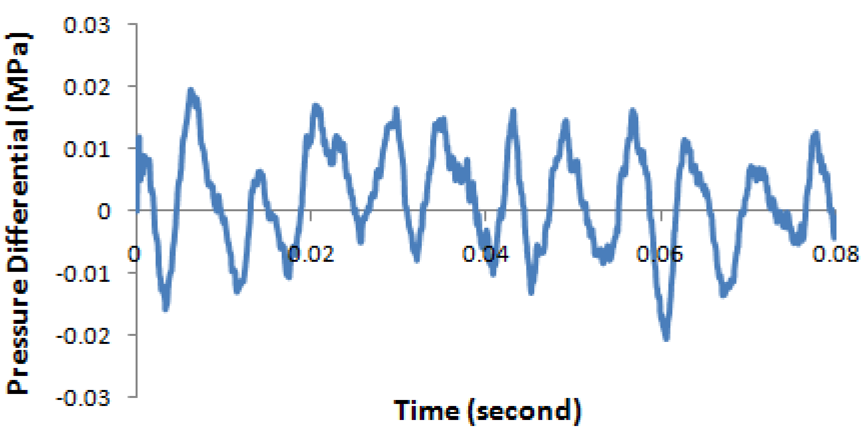

Figure 6: Pressure differential of chambers above/under motor at 60Hz running

4. MECHANICAL IMPACT EXPERIMENT

For the purpose of confirming and identifying the noise source of impact noise, experiment of mechanical impacts of compressor pump-rotor unit was particularly designed and carried out, which would help engineers comprehend the noise mechanism more clearly. As shown in Fig.7, the measurement equipments mainly consist of a pump-rotor unit, a microphone, and corresponding signal sampling & analyzer system.

In the experiment, hold the rotor with a hand and push the shaft-motor unit rotate intermittently and move up and down. This kind of movement not only brings axial up and down displacement but also operates as a normal working compressor. In this way, the unstable movement, causing abnormal vibration of Z direction and noise problem, is simulated outside the hermetic compressor. In addition, the signals of components in compressor pump impact each other are received with a microphone. Thus, the sound response of the unstable movement can be obtained easily.

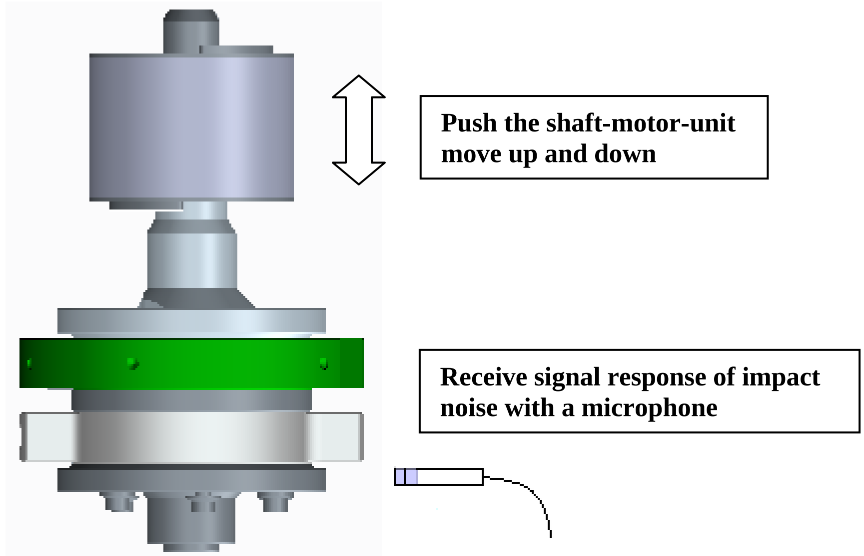

Figure 7: Experiment of mechanical impacts of pump-rotor unit

With FFT analysis, the received sound response can be transformed to frequency domain. One important thing should be mentioned, as the frequency distribution is the core interesting point of the impact response, therefore the magnitude of the response curve has been modified for easy comparison with noise and vibration spectrum. The red curve in Fig.8 and Fig.9 describes the frequency characteristics of the sound signal response for unstable movement of shaft-rotor unit. In the response curve, the maximum response mainly distributes at frequency range between 600~1100Hz, which has a strong coherence with the noise and vibration frequency distribution. And there is relatively larger response at frequency range between 1800~2100Hz, 2300~2500Hz, 3200~3500Hz, which also need more attentions.

As shown in Fig.8 and Fig.9, through comparison with the noise and vibration spectrum, it can be seen that there is a good agreement of the interested frequency range between the abnormal noise/vibration spectrum and the impact response curve. Moreover, the impact response has great effect on compressor noise at frequency range between 600~1000Hz, 1800~2100Hz, 3200~3500Hz. In addition, the shape of the noise spectrum has changed by the influence of the corresponding frequency range in Figure 8. The results confirm that mechanical impacts of shaft-rotor unit with the other components in pump are the primary reason, causing the abnormal vibration of Z direction and noise problem.

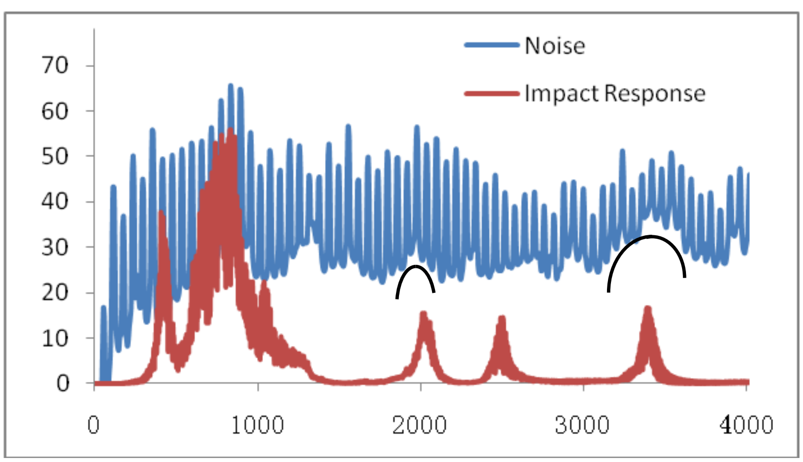

Figure 8: Comparison of noise and impact spectrum

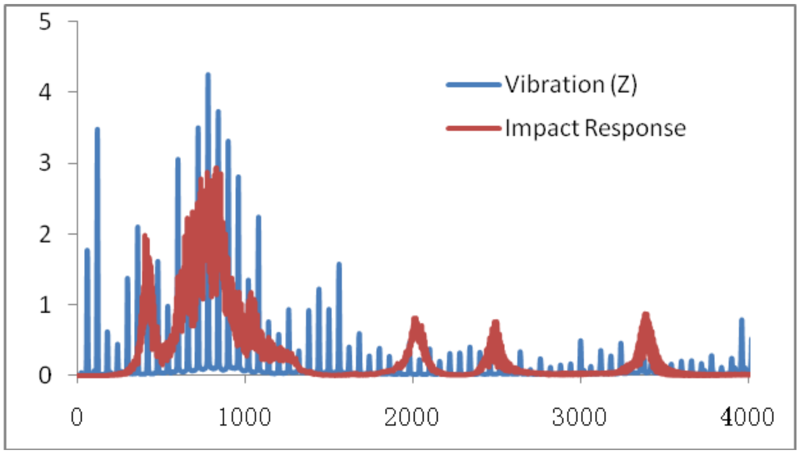

Figure 9: Comparison of vibration and impact spectrum

5. NOISE REDUCTION

From the above consideration, it is known that the shaft-rotor unit moves up and down as rigid motion, then corresponding noise and vibration are brought by being excited up-down movement of the pump. In order to solve the problem, several measures have been taken as follows:

(1) Reduction of pressure pulsation above and below motor:

A. Setting several holes passing through the rotor.

B. Partially cutting the circumferential edge of the stator.

(2) Restraint of axial movement of the shaft-rotor unit

Increase the height differential Δh between rotor and stator to improve the axial magnetic force. The direction of the magnetic force must point to the ground. Together with self-weight of the pump-rotor unit and damping force, the axial magnetic force could overcome the discharge gas force so that the shaft could be pushed down stably.

With these measures, sound pressure level of the modified compressor has been decreased from 80.3dB to 76.2dB at the operating speed of 60Hz. Compared with the original compressor, the sound pressure level is reduced approximately by 4.1dB and the peak value has been decreased approximately by 20.2dB at frequency 840Hz as shown in Table 1. The final results indicate that the measures applied take great significance for the noise problem brought by up-down movement of shaft-rotor unit.

Table 1: Comparison of sound pressure level and peak values

	Original Compressor	**Modified Compressor**	**Reduction (dB)**
Sound Pressure Level	80.3	76.2	4.1
Peak value	64.4 / 840Hz	44.2 / 840Hz	20.2

In detail, noise and vibration spectrum are compared for checking the frequency difference. As shown in Fig.10, the noise spectrum of the modified compressor has been decreased dramatically at frequency range between 600~1020Hz, 1800~2520Hz, which has a good coherence with the frequency distribution of impact response curve. In addition, the vibration spectrum of pump-Z direction has also been decreased greatly at frequency range between 300~1080Hz for the modified compressor as shown in Fig.11, which has a good agreement with the impact response curve. Fig.12 shows vibration spectrum of the modified compressor at location of pump. Obviously the vibration value of pump is low enough for an 8HP capacity compressor. As a result, the noise problem of 8HP rotary compressor has been solved successfully and the peaks disappear at frequency range between 600~1000Hz.

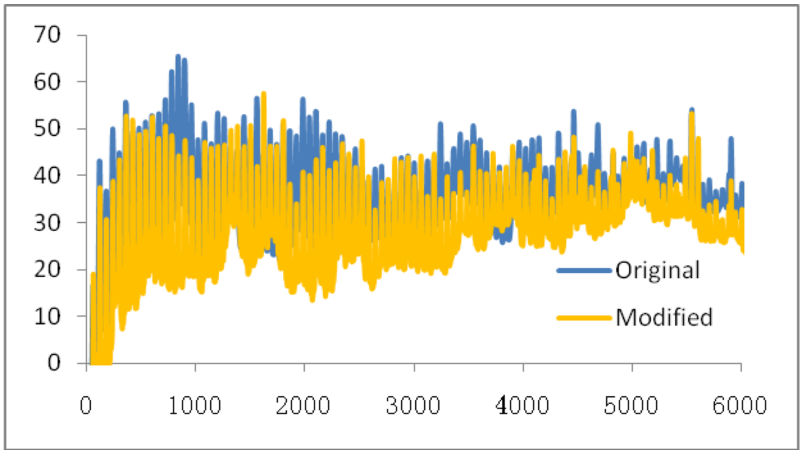

Figure 10: Noise spectrum of original and modified compressor

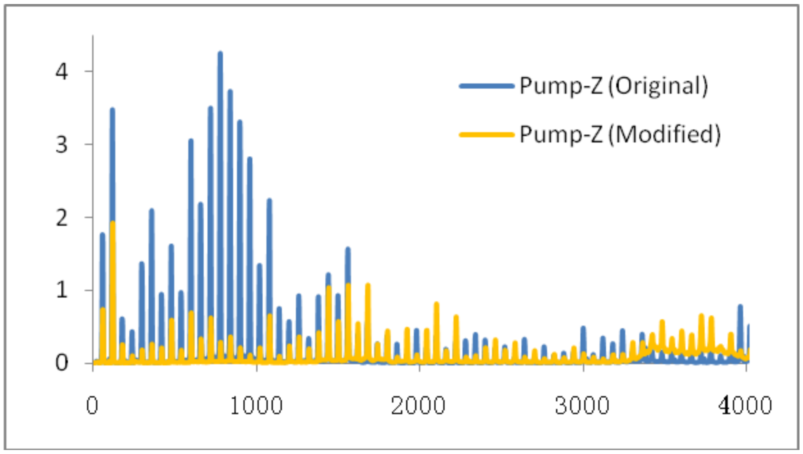

Figure 11: Vibration spectrum of original and modified compressor (Pump-Z)

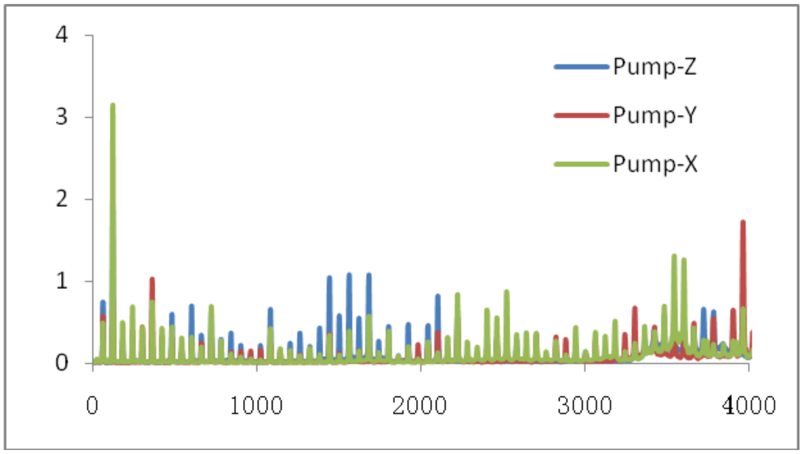

Figure 12: Vibration spectrum of modified compressor

6. CONCLUSIONS

In this paper, the noise problem of 8HP rotary compressor was investigated in order to reduce sound pressure level and improve the sound quality with large capacity compressor. According to the abnormal frequency spectrum of noise and vibration, correlative forces of shaft-rotor unit were made a careful analysis, including pressure pulsation, axial magnetic force, gravity, damping effect. Two experiments, pressure pulsation above/below compressor motor and mechanical impacts of shaft-rotor unit, were carried out for the purpose of identifying the noise source. The test results confirmed that this noise problem was originated from the axial up-down movement of the pump-rotor unit. What's more, the frequency characteristics of impact response were obtained, which had a good coherence with the frequency distribution in abnormal noise and vibration spectrum. Several measures had been taken as shown in chapter 5 to reduce pressure pulsation above/below motor and restrain the axial movement of shaft-rotor unit. As a result, the noise problem had been solved successfully and the sound pressure level decreased about 4dB and the sound quality was comfortable and good to hear.

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