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## Modal Analysis of the Rotor-Journal Bearing System of Rotary Compressor

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### ABSTRACT

In the rotary compressor, the rolling piston and the contacted rotor-journal bearing system are basically moving components. Because of the inertia forces and the periodically changing gas force acting on the rolling piston and the rotor, the journal bearing suffers very large loads. It always leads to serious vibration of the rotor-journal bearing system, and the wear of the rotor. Thus, the vibration must be reduced. In this paper, research works of vibration characteristics of the rotor-journal bearing system were carried out to provide evident of optimizing its structure to reduce the vibration. The average values of the natural frequencies and the vibration mode of the rotor-journal bearing system was obtained based on three-dimensional numerical simulations with finite element method using APDL routine. The power spectral density map of the exciting forces on rolling piston was obtained by the Fourier transformation. It could be used to judge whether resonance vibration would happen during working process of the compressor. The analysis results showed that the lateral vibration mode and the torsion vibration mode were the main vibration modes of the rotor-journal bearing system, and the first four orders of the exciting frequencies played a major role on the vibration of the system.

### 1. INTRODUCTION

The rotary type compressors are widely used in the room air conditioning systems, refrigeration and small commercial refrigeration equipments, because they have advantages such as simple structure, good dynamic equilibrium characteristics and high reliability. The rolling piston and the contacted rotor-journal bearing system are main moving parts of a rotary compressor. During working process, they withstand periodical forces, such as the gas force on the rolling piston, the inertial force on the rotor and the oil pressure at the bearing. Because these forces change over one rotor revolution, it leads to seriously vibration and large deformation of the rotor. The vibration and the deformation will change the clearance between rotor and bearing. When the vibration and the deformation is big enough, the rotor may contact the bearing. Then seriously slanting abrasion will occur between the bearing and the rotor, even a contact of the motor rotor and stator at the top of the rotor. More seriously, when sympathetic vibration occurs, the rotor would be wore out very soon. Thus, we must research the vibration mechanism of the rotor-journal bearing system, and reduce the vibration. As an effective method to investigate the characteristics of the vibration, the modal analysis of the rotor-journal bearing system becomes the focus of interest of many resent studies.

At present, many studies have been done in the calculation of the vibration characteristics and dynamic response of rotor-journal bearing system. European scholar R.Dufour (1998) carried out the modal analysis to obtain the vibration characteristics of the crankshaft of a single rotor compressor by using experimental method and the finite element analysis method. The study was one-dimensional, while the real problem is three dimensional. H. Hattori and N. Kawashima (1990 and 1991) employed a dynamical analysis of the rotor system in twin rotary compressors, under the condition that the journal bearing was lubricated well, and the pressure on the rotor at the bearing was obtained by Reynolds equations. In this study, the rotor was described by using one-dimensional beam model and the pressure gradient of the oil film was obtained based on the short bearing theory. The simplification of the bearing into infinite short bearing sometimes produces certain calculation error. XieFei et al. (2006 and 2007) built up the finite element models of the rotor-journal system by using the Timoshenko beam model. In this analysis, the dynamic response of the rotor-journal bearing system was obtained by calculating the oil pressure through the finite

bearing theory. However, structure of the rotor, such as oil paths, was simplified in the study. It would influence the deformation of the rotor. Although many scholars have done a lot of work, so far most scholars used one-dimensional beam element to build up the rotor model. And only a few studies have been carried out to calculate the vibration characteristics of the rotor-journal bearing system. Further, no work has been done by using the Fourier transformation to the exciting forces to predict whether resonance vibration happens.

In this paper, the finite element analysis model of rotor is built up by using a three-dimensional solid element to study the vibration of the rotor-journal bearing system of a rotary compressor. The dynamic characteristic parameters, such as stiffness and damping coefficient of bearing oil film, are numerically calculated according to the actual bearing structure characteristics. The modal analysis of the rotor model has been carried out by using the finite element software ANSYS, under boundary conditions of obtained equivalent stiffness and damping coefficients of oil films. Through the modal analysis of rotor-journal bearing system, the modes of the system including natural frequencies and mode shapes are obtained. The power spectral density maps are obtained by the Fourier transformation to the exciting forces, which acts on the rolling piston including the gas forces and other contact forces. These power spectral density maps can be used to judge whether resonance vibration happens during working process of the compressor and which orders of the exciting frequency plays a major role on the vibration of the system.

## 2. MODAL ANALYSIS THEORY

### 2.1 Modal Analysis Concept

Modal analysis is used to determine the vibration characteristics of a structure, namely the natural frequencies and the mode shapes of the structure. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions which determined by the inherent characteristics and the materials of a structure. They are also required if you want to do a spectrum analysis or a mode superposition harmonic or transient analysis.

By modal analysis on the rotor-journal bearing system of the rotary compressor, we can get the response of the structure withstands the different kinds of dynamic loads. On the one hand, the vibration characteristics for structure can be predicted. On the other hand, we can avoid the resonance phenomenon occurs in the design process, do the modification of the existing structure, and improve the reliability of the machine.

### 2.2 Modal analysis theory

According to the finite element method of elastic mechanics, the motion equations of the rotor-journal bearing system are firstly described as:

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = \{F(t)\} \quad (1)$$

Where  $[M]$ 、 $[C]$ 、 $[K]$  are the classical mass, damping and stiffness matrices respectively,  $\{\ddot{X}\}$ 、 $\{\dot{X}\}$ 、 $\{X\}$  are the acceleration vector, the velocity vector and the displacement vector of the node respectively, and  $\{X\} = \{x_1, x_2, \dots, x_n\}^T$ 、 $\{F(t)\}$  is the harmonic force vector, and  $\{F(t)\} = \{f_1, f_2, \dots, f_n\}^T$ .

When  $\{F(t)\} = \{0\}$ , the system is in a free vibration status. Since the effect of the damping is small in the modal analysis of the system for solving the natural frequencies and the mode shapes of the structure. The damping term in (1) can be ignored. So (1) is written as

$$[M]\{\ddot{X}\} + [K]\{X\} = \{0\} \quad (2)$$

The free vibration of the system can be regard as the superposition of a series of harmonic movement, and then take  $\{X\} = \{\Phi\}e^{i\omega t}$  as the solution of equation (2). Substituting the equation  $\{X\} = \{\Phi\}e^{i\omega t}$  to the equation (2), we can get the basic equation of modal analysis without damping as:

$$([K] - \omega_i^2 [M])\{\phi_i\} = 0 \quad (3)$$

Where  $\omega_i$  is the natural circular frequency of mode  $i$ ,  $\phi_i$  is the mode shape vector of mode  $i$ ,  $i = 1, 2, \dots, n$ .

### 3. MODAL ANALYSIS

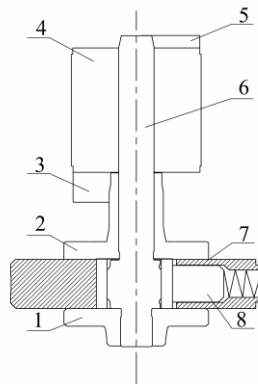
#### 3.1 Model

3.1.1 Crankshaft parameters: According to the method mentioned above, the single rotary compressor used in air conditioning system of Gree Electric Appliances has been investigated as an example in this paper. The main characteristic parameters of the rotor-journal bearing system which analyzed in the finite element modal analysis are shown in table 1.

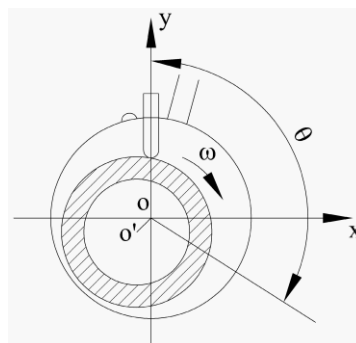
**Table 1** The main characteristic parameters

Rotational Speed (r/min)	Inlet Pressure (MPa)	Discharge Pressure (MPa)	Capacity (Nm <sup>3</sup> /min)
3600	1.0	3.363	3.5
Material	Elastic Modulus (GPa)	Poisson's Ratio	Density (kg/m <sup>3</sup> )
nodular cast iron	173	0.3	7300

3.1.2 The geometry model of the rotor system: The described modal analysis theory is used to calculate modes of the rotor system in a rotary compressor with one compression unit, as shown in Figure 1. The rolling piston is eccentrically mounted on the rotor. Hence the crankshaft drives the crank mechanism comprising mechanical parts (the eccentric crankshaft, the piston rolling in a cylinder, and the vane-spring system) which have eccentric masses. The counterweight masses ( $m_1r_1$  and  $m_2r_2$ ) are designed to reduce the effect of the mass unbalance ( $m_3r_3$ ). The compressor discharges once per rotor revolution. The coordinates  $x$ ,  $y$  and  $\theta$  of the rotor-journal bearing system are defined as shown in Figure 2, the origin and direction of the rotor rotating angle are the same as those of  $\theta$ .



**Figure 1** the rotor system in a rotary compressor with one compression unit  
1- sub bearing, 2- main bearing, 3-counterweight masses( $m_1r_1$ ), 4-motor rotor  
5- counter weight masses( $m_2r_2$ ), 6- eccentric crankshaft( $m_3r_3$ ), 7-cylinder, 8-vane



**Figure 2** The coordinates  $x$ ,  $y$  and  $\theta$

3.1.3 Element type: The unbalanced forces act on the rotor without static equilibrium due to the eccentric rotation of the rolling piston. Because of the dynamic loads, the three-dimensional solid element is needed to analysis the response of the structural with large dynamic loads in order to improve the accuracy of analysis. The 3-D 8-Node Structural Solid element(SOLID73) with Rotations which is defined by eight nodes having six degrees of freedom at each node is used for the calculation model of the rotor-bearing in this paper, as shown in Figure 3.

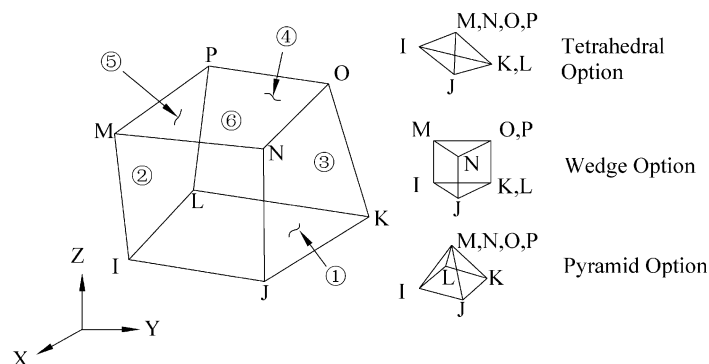


Figure 3 3-D 8-Node Structural Solid element

3.1.4 Model simplification: Before building up the modal analysis model of the rotor-journal bearing system, assumptions were made as the following:

- (1) There is no relative motion between the eccentric rotor and the rolling piston of the compressor in the whole work cycle, and the friction and forces of the oil film between the two parts is ignored.
- (2) According to the reference [3], the dynamic amplitude of the rotor center is negligible, namely the calculated oil film coefficients by using Reynolds equation can be seen as constant values.
- (3) The influence of friction and the forces of the oil film between the rolling piston and the eccentric rotor is ignored.

After doing the simplifications as above, the rotor-journal bearing system of rotary compressor is modeled by APDL routine, in which the crankshaft is meshed into SOLID73 element, the parts of the rotor fitted the bearing are meshed by structured grid, and the journals is meshed into COMBIN14 spring-damper element. The model after meshing is shown in Figure 4.

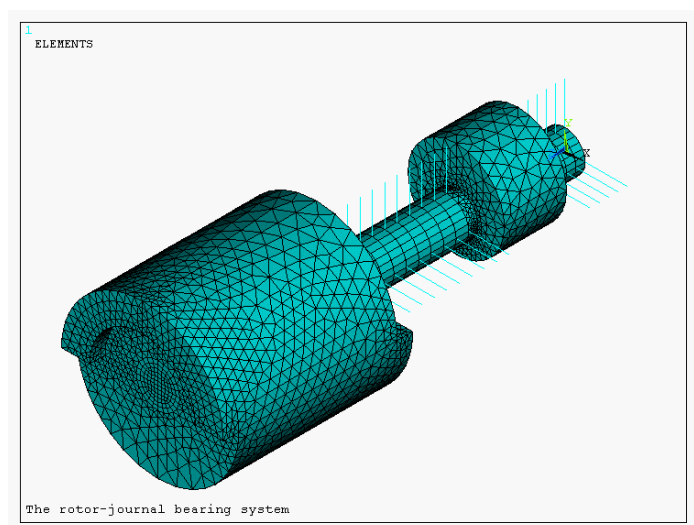


Figure 4 The meshed model for the rotor-journal bearing system

## 3.2 Boundary Conditions

3.2.1 Force boundary conditions: As outer loads acting on the rotor, the unbalanced forces due to the eccentric masses, the contact force between skateboarding and rolling piston, the oil film forces between the rolling piston and the eccentric rotor, and the gas forces in the compression unit are considered.

Due to the influence of the counterweight masses, the unbalanced forces can be ignored. The outer loads used in this analysis are shown in figure 5.

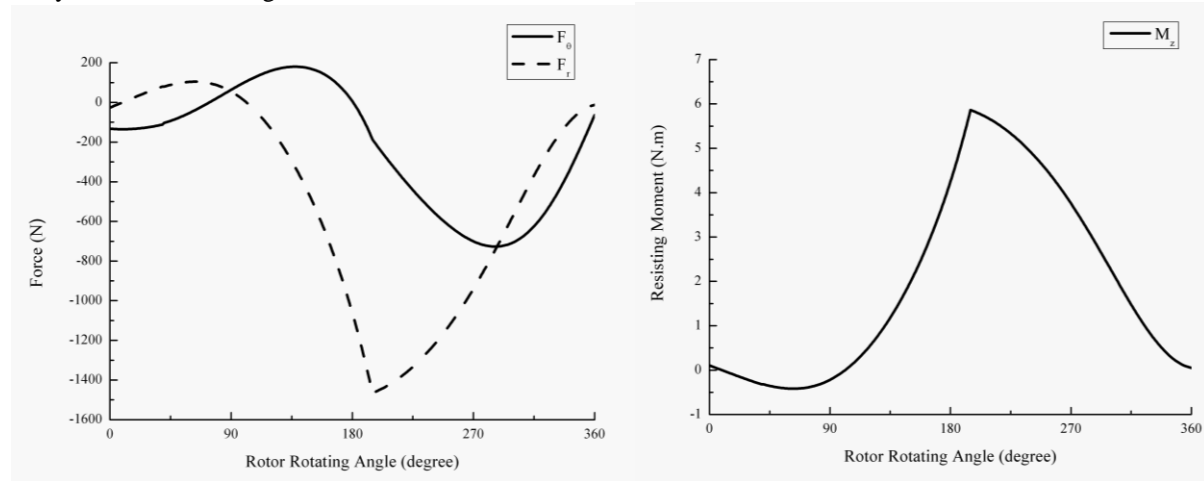


Figure 5 Outer loads profiles

3.2.2 Displacement boundary conditions: The mode of the rotor-journal bearing system is mainly influenced by its structure and the constraint of the system. As shown in fig.4, the eccentric rotor is supported by two approximated bearing. Because of the constraint of sub bearing, the z axial displacement of the whole rotor-bearing system is zero.

According to assumption of the model, the system is acted by the equivalent springs in the radial direction and peripheral direction. The stiffness and damping coefficients of the equivalent springs is the equivalent stiffness and damping of the oil film force calculated by the Reynolds equation. The calculation for the oil film coefficients is carried out for the two bearing individually.

To take account of oil film pressure of each main bearing (journal bearings), we idealized a set of linear springs and dashpots in the vertical and horizontal directions attached to the crank journal axis. The stiffness and damping coefficients of oil film in finite bearing are calculated by the following equations:

$$k_{ij} = K_{ij} \frac{\mu \Omega l}{\phi^3} \quad (4)$$

$$b_{ij} = B_{ij} \frac{\mu l}{\phi^3} \quad (5)$$

Where  $\mu$  is the viscosity of the lubricant  $l$  is the effective work length of bearing,  $\phi$  is the bearing clearance ratio,  $K_{ij}$  and  $B_{ij}$  are the stiffness and damping matrices respectively.  $\Omega$  is the instability speed for bearings, which is determined by the quality that assigned to bearings by the total quality of rotor, instability speed as follows:

$$\Omega_{st} = \frac{\mu l}{m_R \phi^3} \cdot \frac{K_{eq}}{\gamma_{st}^2} \quad (6)$$

$$K_{eq} = \frac{K_{\xi\xi} B_{\eta\eta} + K_{\eta\eta} B_{\xi\xi} - K_{\xi\eta} B_{\eta\xi} - K_{\eta\xi} B_{\xi\eta}}{B_{\xi\xi} + B_{\eta\eta}} \quad (7)$$

$$\gamma_{st}^2 = \frac{(K_{eq} - K_{\xi\xi}) + (K_{eq} - K_{\eta\eta}) - K_{\xi\eta} K_{\eta\xi}}{B_{\xi\xi} B_{\eta\eta} - B_{\eta\xi} B_{\xi\eta}} \quad (8)$$

Where  $m_R$  is the mass that the total mass of the rotor assigned to the bearing.  $\begin{bmatrix} K_{\xi\xi} & K_{\xi\eta} \\ K_{\eta\xi} & K_{\eta\eta} \end{bmatrix}$  is the stiffness matrices,

$\begin{bmatrix} B_{\xi\xi} & B_{\xi\eta} \\ B_{\eta\xi} & B_{\eta\eta} \end{bmatrix}$  is the damping matrices.

## 4. RESULTS AND DISCUSSION

The aim of modal analysis for rotor-bearing system of the rotor compressor by using the finite element software is to find out the natural frequency and its corresponding vibration mode shapes of the system. These parameters are mainly influenced by the structure types and constraint conditions without loads. In this paper the models were analyzed by using the block Lanczos method provided by ANSYS package to determine the eigen value of the Equation (1), and set the order number of modal extraction for solution.

### 4.1 Modal analysis results

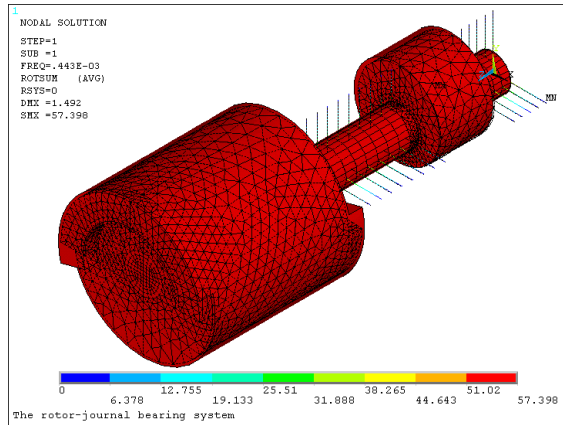
Due to the influences to the structure of the high order vibration modes are much smaller than that of the low modes, the calculated results of the first eight orders natural frequencies and the mode of vibration are listed in Table 2, the expression “L” refers to the lateral vibration and “T” refers to torsion vibration.

**Table 2** First eight calculated natural frequencies of rotor

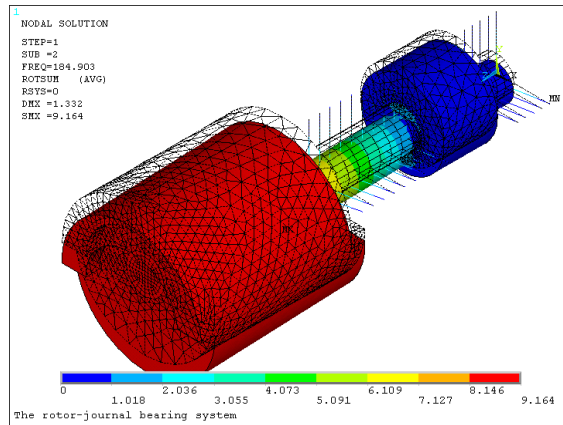
Mode	Calculated Natural Frequency ( $\omega_i$ /Hz)	Mode of Vibration
1st	0.44266E-03	T
2nd	184.90	L
3rd	195.03	L
4th	601.60	L
5th	628.59	L
6th	2803.6	L&T
7th	3141.4	L
8th	3460.6	L&T

Table 2 shows that: there are lateral vibration mode and torsion vibration in the first eight order modes. The first four order mode shapes of the rotor-bearing system were shown in Figure 6.

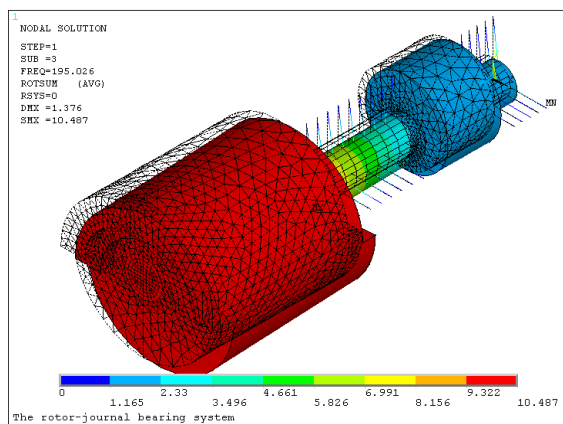
As can be seen from the figure 6: the whole of rotor-journal bearing system takes place torsion vibration along Z axis in the plane of the parallel to XY plane at the 1st mode, the whole of rotor-bearing system takes place transverse vibration respectively in parallel to the plane of YZ and XZ plane at the 2nd and 3rd mode, the rolling piston of the rotor-bearing system takes place lateral bending vibration respectively in parallel to the plane of YZ and XZ plane at the 4th mode.



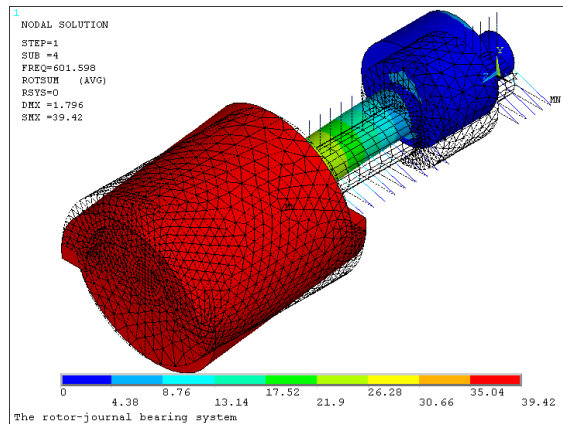
1st order mode shape



2nd order mode shape



3rd order mode shape



4th order mode shape

Figure 6 The first eight order mode shapes of the rotor-bearing system

## 4.2 Fourier transformation

The exciting forces causing the vibration of rotor-bearing system are the resultant forces acting on the rolling rotor. The whole external forces cyclical changed, and the outer load profiles per rotor revolution were shown in figure 5. The Fourier transformation was done to the resultant forces. The power spectral density maps obtained from the Fourier transformation are shown in figure 7. The figure shows that: the maximum power of harmonic frequency is at 60 Hz, 120 Hz, 180 Hz and 240 Hz. Compared with power of the first four order exciting force, power of the high frequencies are smaller. Therefore, only the influences of the first four orders of exciting forces needed to be considered.

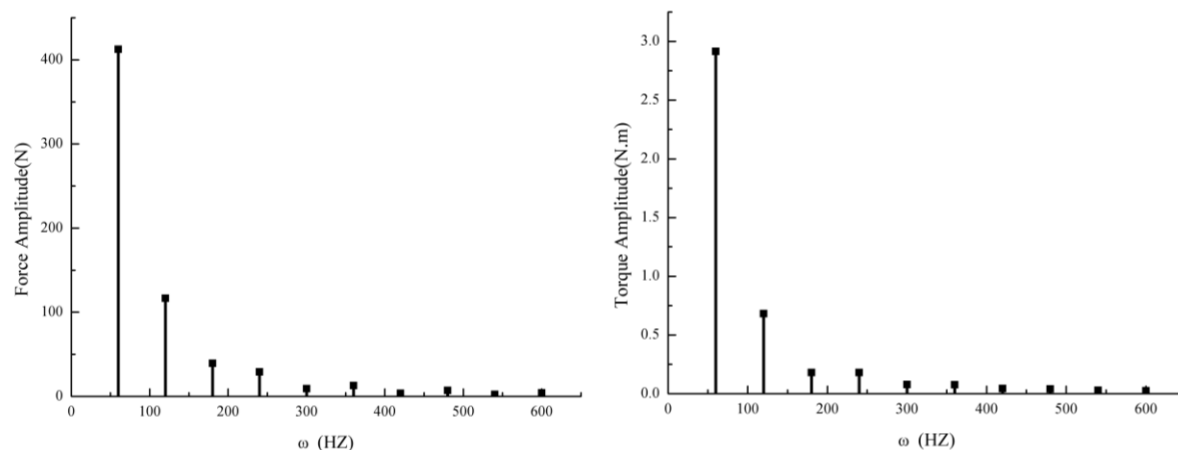
In order to avoid resonance of the structure, the exciting frequency should be outside certain limits of natural frequency as shown in following:

$$P < 0.75W \text{ and } P > 1.25W \quad (9)$$

Where  $[P]$  is the exciting frequency,  $[M]$  is the natural frequency of structure.

Substituting the exciting frequency to the equation (9), we can see: the first, the second and the fourth orders exciting frequency meet the requirement of the equation (9), so we can judge that resonance vibration would not happen to the rotor in these frequencies. But the third order the exciting frequency not match the standard that shows in equation (9), so that there would occur the resonance vibration during working process of the compressor.





**Figure 7** The power spectral density map of the force and the torque

## 5. CONCLUSIONS

(1) The low order natural frequency and vibration mode of the rotor-bearing system of a rotary compressor had been deduced in this paper. The natural frequency and vibration mode of the structure could be used in the design process to avoid resonance region and to improve the stability and reliability of the system.

(2) It was conclude that long dimensions of the length of the journal bearing led to bad stability of the rotor-journal bearing system. Long journal bearing increased the probability of deformation and instability, which would cause the collision between the motor rotor and stator. Thus, the bearing structure must be improved to improve the reliability of the system.

(3)According to the Fourier transformation to the exciting forces on the rolling piston, the maximum power of harmonic frequency were at 60 Hz,120 Hz ,180 Hz and 240 Hz. Therefore, resonance vibration would not happen to the rotor, when  $P < 0.75W$  and  $P > 1.25W$  at 60 Hz, 120 Hz, 240 Hz. However, resonance would happen at 180 Hz.

## REFERENCES

- Michael Charreyron., 1998, Dynamics Prediction of Refrigerant rotary compressor crankshaft[C]. Proceedings of International Compressor Engineering Conference at Purdue.
- H. Hattori, N. Kawashima., 1990, Dynamics Analysis of a Rotor-Journal Bearing System for Twin Rotary Compressors[C]. Proceedings of International Compressor Engineering Conference at Purdue.
- Hitoshi Hattori., 1991, Dynamics Analysis of a Rotor-Journal Bearing System with Large Dynamic Loads\*(Stiffness and Damping Coefficient Variations in Bearing Oil Films) [J]. The Japan Society of Mechanical Engineers.
- Davis K. and Henshell R D., 1972, A Timoshenko beam element [J]. Journal of Sound and Vibration, vol. 22: p. 475-487.
- Fei Xie et al., 2006, Dynamic Analysis of a Rotor-Journal Bearing System of Rotary Compressor[C]. Proceedings of International Compressor Engineering Conference at Purdue.
- Fei Xie et al., 2007, Failure Analysis of The Rotor-Journal Bearing System[C]. Proceedings of FEDSM2007 5th Joint ASME/JSME Fluids Engineering Conference July 30-August 2, San Diego, California USA.

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