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Modal Analysis of Scroll Compressor in Rotator System with Finite Element Method

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Abstract: In order to study the dynamic characteristics of scroll compressor crankshaft it makes the research of the crankshaft rotor of natural gas scroll compressor and analyzes the displacement and angle of inclination based on modal analysis theory and equation of rotor motion; it also calculates the dynamic coefficients of main shaft bearing and simplifies its using spring-damper system-on the basis of rotor dynamic theory; the prestressed structure modal analysis on scroll compressor crankshaft rotor is made and its vibration performance and mode under the action of the centrifugal force-using finite element analysis through the software ANSYS and at the same time it undertakes modal analyses on crankshaft under different support types and changes the support stiffness of bearing to study its effects on the entire dynamic characteristics of a rotor system. The results indicate that when rigid constraints are applied in the main shaft bearing, rotor of the crankshaft is rigid and its fundamental frequency is far less than first order natural frequency and therefore there is no resonance during working process. Maximum deformation will easily occur at the crank pin and cause substantial effects on scroll compressor's bearing. We can change the main counterweight's distance from location of the main bearing to lower crank pin's deflection. When elastic support is applied in the main shaft bearing, natural frequency of crankshaft rotor is lower compared with rigid constraints and the critical speed of rotation also will be decreased.

Key words: Scroll compressor, crankshaft rotor, vibration characteristics, modal analysis

INTRODUCTION

Scroll compressor is a new type positive displacement compressor which is widely used in refrigeration and chemical engineering due to its many advantages including low volume, light weight, high reliability and stable and sustainable gas transmission. Its excellent properties and bright prospect have gained a lot of attentions. Due to scroll compressor's unique structure and working principle, during its operation, the crankshaft system will be subject to the gas force and centrifugal inertia force generated by the eccentric rotor. Vibration of crankshaft will be transferred to the casing through the main and minor bearing and causing vibration of the casing and other parts. The damage of vibration is mainly caused by resonance because when the excitation frequency is close to a natural frequency in the system, the amplitude will be increased sharply (Liu *et al*., 1998). At present, the studies on scroll compressor's rotor are primarily focused on one single rotor but rarely on the rotor system (Du and Liu, 1999; Hao *et al*., 2011; Hu *et al*., 2007; Wang *et al*., 2012; Xu *et al*., 2012). As a matter of fact, as a main component of scroll compressor, the rotor system takes on the task of power transmission. Its steady operation is the key to the overall performance of the scroll compressor since it can affect the radial sealing clearance between the fixed scroll and orbiting scroll, the distribution of lubricant film, the eccentric wear of friction pair, the vibration of rotor and oil temperature. Therefore, when studying the subject of crankshaft dynamics it is more reasonable to consider crankshaft, orbiting scroll, main and minor counterweight and belt pulley as a whole. In this study it uses finite element analysis software ANSYS in the modal analysis on scroll compressor's rotor and analyzes rotor's critical speed of rotation under rigid support, elastic support and changing bearing's stiffness to identify its effects on the rotor's dynamic characteristics. So the research has both theoretical significance and practical value in enhancing reliability and safety of scroll compressor and extending its useful life.

BASIC STRUCTURE OF SCROLL COMPRESSOR'S ROTOR SYSTEM

The rotor system of scroll compressor is composed of crankshaft, orbiting scroll, main and minor counterweight and belt pulley. Crankshaft bearings includes main bearing (rolling bearing) and minor bearing self-aligning ball bearing. Drive bearing is the sliding bearing. The structure is as shown in Fig. 1.

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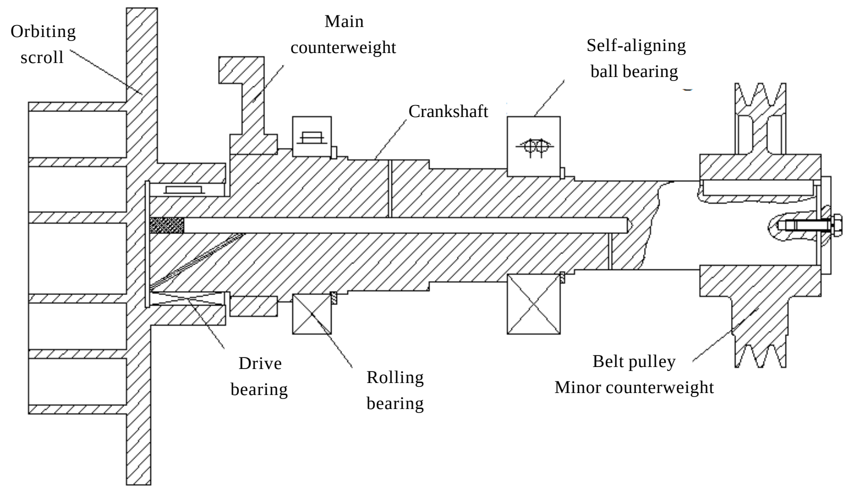


Fig. 1: Crankshaft rotor structure diagram of scroll compressor

BASIC THEORY OF PRESTRESSED MODE ANALYSIS

Based on theory of vibration mechanics, dynamic equation of the multi-degree of freedom system can be expressed as follows (Houhaun *et al*., 2012; Tiansu *et al*., 2005; Jingwen *et al*., 1988):

$$[M]\{\ddot{\delta}\}+[C]\{\dot{\delta}\}+[K]\{\delta\}=\{Q\} \tag{1}$$

where, $\{\delta\}$ is the unit node displacement, $\{\dot{\delta}\}$ is the unit node speed, $\{\ddot{\delta}\}$ is the unit node acceleration, [M] is the mass matrix, [C] is the damping matrix, [K] is the stiffness matrix, [Q] is the node load matrix.

[Q] represents the node load matrix, is usually a function of time. In a modal analysis it is used to analyze rotor's nature properties without exciting force. Therefore, $\{Q\} = 0$. Considering the effects of rotating centrifugal force from crankshaft's rotor, stiffness matrix K, representing centrifugal stress it is applied in the structure, the damping matrix is ignored and modal equation for rotor under rotating centrifugal force is obtained:

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

When the scroll compressor's rotor is rotating, radial tensile stress is released due to the centrifugal force role. Thus, when performing the modal analysis for the rotor it needs to treat this as the initial stress.

EQUATION OF ROTOR MOTION

Through discretization, the rotor shaft is divided into n segments. Unit length of the shaft is L. There is a lateral displacement v and rotation angle θ at the two ends of each shaft unit. The deflection and rotation angle of each node at the shaft unit are superposed by the four degree of freedom function of shape and position are φ_1, φ_2, φ_3, φ_4 (Wang *et al*., 2009):

$$\begin{cases} y(x) = \varphi_1 v_1 + \varphi_2 \theta_1 + \varphi_3 v_2 + \varphi_4 \theta_2 \\ \theta(x) = \varphi_1' v_1 + \varphi_2' \theta_1 + \varphi_3' v_2 + \varphi_4' \theta_2 \end{cases} \tag{3}$$

Without consideration of gyroscopic torque's effects it can replace the deflection and rotation angle of each node with degrees of freedom and use lateral displacement and speed of the two nodes at the shaft unit to indicate shaft unit's kinetic energy T and potential energy U, respectively:

$$\begin{cases} T = \frac{1}{2}\sum_i \sum_j \dot{q}_i \dot{q}_j \int_0^1 \varphi_i \varphi_j m dx = \frac{1}{2}\sum_i \sum_j m_{ij} \dot{q}_i \dot{q}_j \\ U = \frac{1}{2}\sum_i \sum_j \dot{q}_i \dot{q}_j \int_0^1 EI \varphi_i' \varphi_j' dx = \frac{1}{2}\sum_i \sum_j k_{ij} q_i q_j \end{cases} \tag{4}$$

When the shaft is divided into n segments, then there will be n+1 nodes and each node is with two degrees of freedom. Therefore, the generalized mass matrix of the system is a 2n+2 square matrix. Based on the relationship between generalized force and generalized mass it can use the superposition principle to add the generalized force,

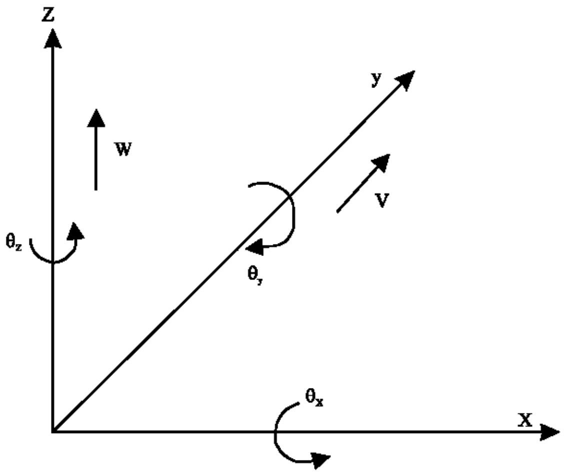


Fig. 2: Crankshaft rotor node displacement and angle coordinates of scroll compressor

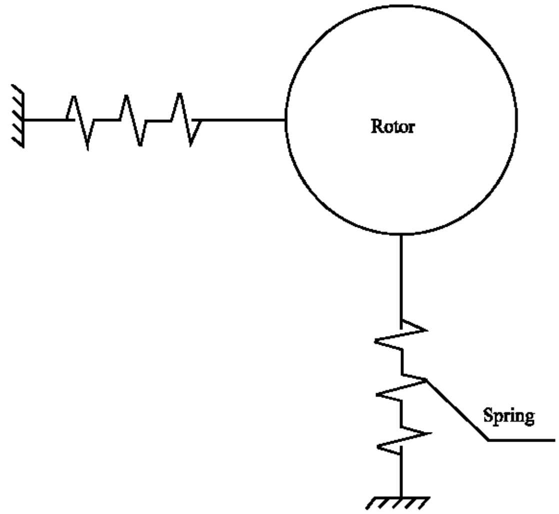


Fig. 3: Crankshaft bearing simplified diagram

corresponding to gyroscopic torque, to the Lagrange equation and obtain the equation of motion for the system:

$$(M + J)\ddot{q} + Kq = F(t) \tag{5}$$

When F(t) = 0, $(M+J)\ddot{q}+Kq = 0$. It can calculate the critical speed of rotation and the corresponding natural frequency by solving the equation.

When the scroll compressor's rotor can be divided into finite elements, the following points can serve as nodes: the point where the inner and outer diameter changes, each supporting point, external point of action and the point where the deformation occurs. There are five degrees of freedom at each node, including lateral displacement v and rotation angle θ_z in plane xoy, lateral displacement w and rotation angle θ_y in plane xoz and the deformation θ_x around axis x, just as shown in Fig. 2.

DYNAMIC COEFFICIENTS OF SCROLL COMPRESSOR'S BEARING AND THE SIMPLIFIED MODEL FOR MAIN SHAFT BEARING

The main and minor bearing of the scroll compressor are both rolling bearings which support the crankshaft, absorb the gas force and inertia force from the compression chamber and reduce frictions.

The dynamic characteristics of the bearings play a key role in rotor dynamics and yet, up to now, no systematic and complete method and data regarding rolling bearings' dynamic characteristics can be for reference. Rolling bearing is a high load hydrodynamic friction pair that not only shows elastic deformation under solid contact but also bears the influences from hydrodynamic oil film which is coupled with elastic deformation. As a result, rolling bearing dynamics is in fact a matter of liquid-solid coupling and requires the solving of the Reynolds equation and elastic equation. It can only be done under some simplified assumptions and the calculation and solving of these nonlinear equations should be through numeric iteration. In order to simplify the problem, normally in dynamic analysis of rotor it can use statistical data of rolling bearing's stiffness, between 5×10^7~5×10^8 N m^{-1}. Film damper is rather small in the rolling bearing and therefore it will be ignored during the analysis (Bangchun *et al*., 2000; Yie *et al*., 1987).

Based on the practical engineering application, the bearing support is not entirely rigid and therefore, elastic effect on the bearing should be taken into consideration. In this thesis, both horizontal and vertical spring dampers are used to stimulate bearing's constraints on crankshaft, if it makes assuming that each elastic support is composed of two spring dampers (Wang *et al*., 2010). The simplified model of the elastic support is shown in Fig. 3.

ESTABLISHMENT OF FINITE ELEMENT ANALYSIS MODEL FOR ROTOR OF SCROLL COMPRESSOR'S CRANKSHAFT

Establishment of finite element analysis model for rotor: The solid model of scroll compressor's crankshaft is built by 3D modeling software Solidworks it should be as close to the actual size as possible. In order to shorten the calculation time and reduce network flow, the belt pulley is simplified by omitting its race and ignoring small structures like chamfer, screw hole and oil hole. After the model is built in the Solidworks, an assembly is formed

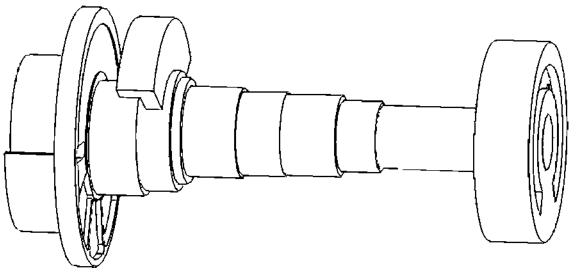


Fig. 4: Crankshaft rotor simplified physical model of scroll compressor

Table 1: Rotor parts physical parameters

Parts	Elasticity modulus (GPa)	Density ($kg\,m^{-3}$)
Crankshaft	200	7800
Counterweight	206	7000
Belt pulley	206	7000

and material attributes are given to the corresponding structures. The 3D solid model of rotor system is as shown in Fig. 4.

Physical properties and meshing of the rotor system model: The 3D model built by Solidworks will be saved as model.txt and then should be imported to ANSYS. After that, material's elastic modulus, poisson ratio, Poisson's ratio is 0.3 and density will be identified. Physical parameter of the rotor is listed in Table 1.

When scroll compressor's rotor is balanced, orbiting scroll's center of mass is moved to the center of the crank pin. When performing the finite element modal analysis, the orbiting scroll will be treated as a homogeneous disc in order to realize lumped mass and loaded to the crank pin. Meshing will be completed in software of ANSYS with 45112 nodes and 26602 elements. The simplified finite element mesh model is as shown in Fig. 5.

Boundary conditions: after the meshing is completed, loading and boundary constraints need to be added to the rotor physical model. First of all, before the modal analysis, finite element model will be given an axial rotational velocity at 293 rad sec^{-1} (scroll compressor's rotating speed is 2800 r min^{-1} when it functions normally). After that, prestressed analysis is done and the results will be used as loading and should be taken into consideration during the modal analysis. In ANSYS, the crankshaft bearing journal surface will be given cylinder constraint (Liu and Yang, 2003; Zhang *et al*., 2012), where rotation is allowable but no displacement in direction x and y (radially and tangentially). Frictionless support will be added to the shaft shoulder of the bearing cone, where

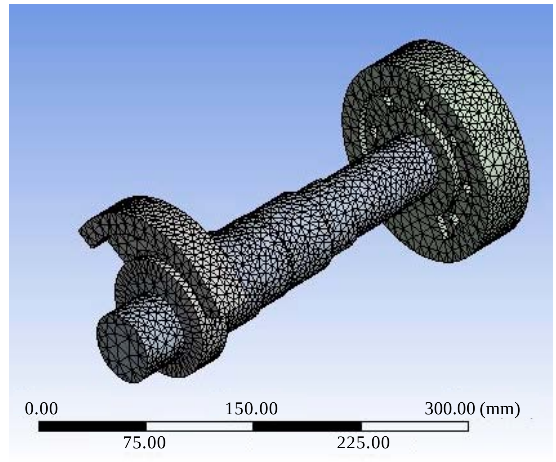


Fig. 5: Crankshaft rotor meshing model

displacement in direction z (axially) is not allowed and gravity load is added as g = 9.8 m sec^{-2}. The constraint model is shown in Fig. 6.

ROTOR MODAL ANALYSIS UNDER PRESTRESSED STRUCTURE

Since high-frequency modes will not have significant effects on the system, when performing the modal analysis, only the top five frequency modes of the crankshaft rotor will be calculated.

Modal analysis for scroll compressor's crankshaft rotor under rigid support: As shown in Fig. 7 and Table 2, the first order and second order critical speed of scroll compressor's rotor system is 6673.8 and 6710.4 r min^{-1}, respectively and the first and second mode of vibration are located on the crank outlet end. Normally, rotor should be working at a speed n, where $n<0.75n_{cr,i}$, or 1.4 $n_{cr,i}<n<0.7\, n_{cr,i+1}$ and $n_{cr,i}$ is rotor's critical speed at i order. Considering rotor's operational safety its working speed is set at more than 20% deviated from the critical speed. Therefore, the rotation speed range for scroll compressor's rotor should be lower than 5300 r min^{-1} or higher than 8000 r min^{-1}. If taking everything into account, the working speed of scroll compressor can be identified as below 5300 r min^{-1}.

Through the modal analysis under prestressed structure, the first six natural frequencies and the according modes of vibration under non-damping free vibration can be obtained for scroll compressor's rotor. Based on these natural frequencies, rotor's critical rotation speed can be calculated, as listed in Table 2.

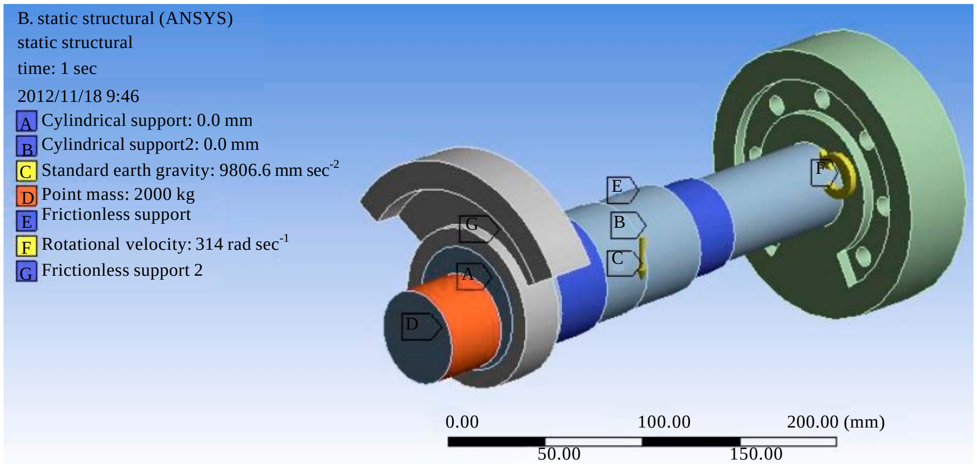
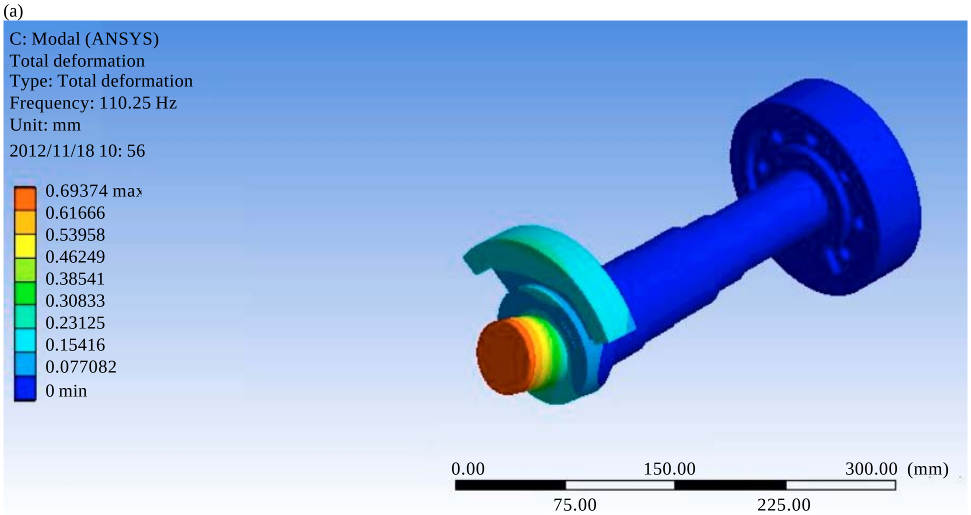


Fig. 6: Rotor boundary condition constraint model

(a)



(b)

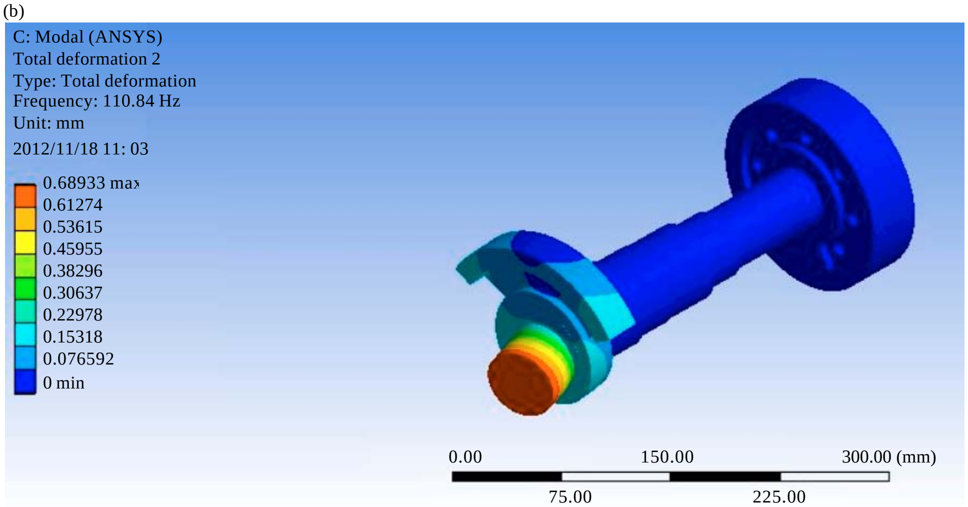


Fig. 7(a-b): Continue

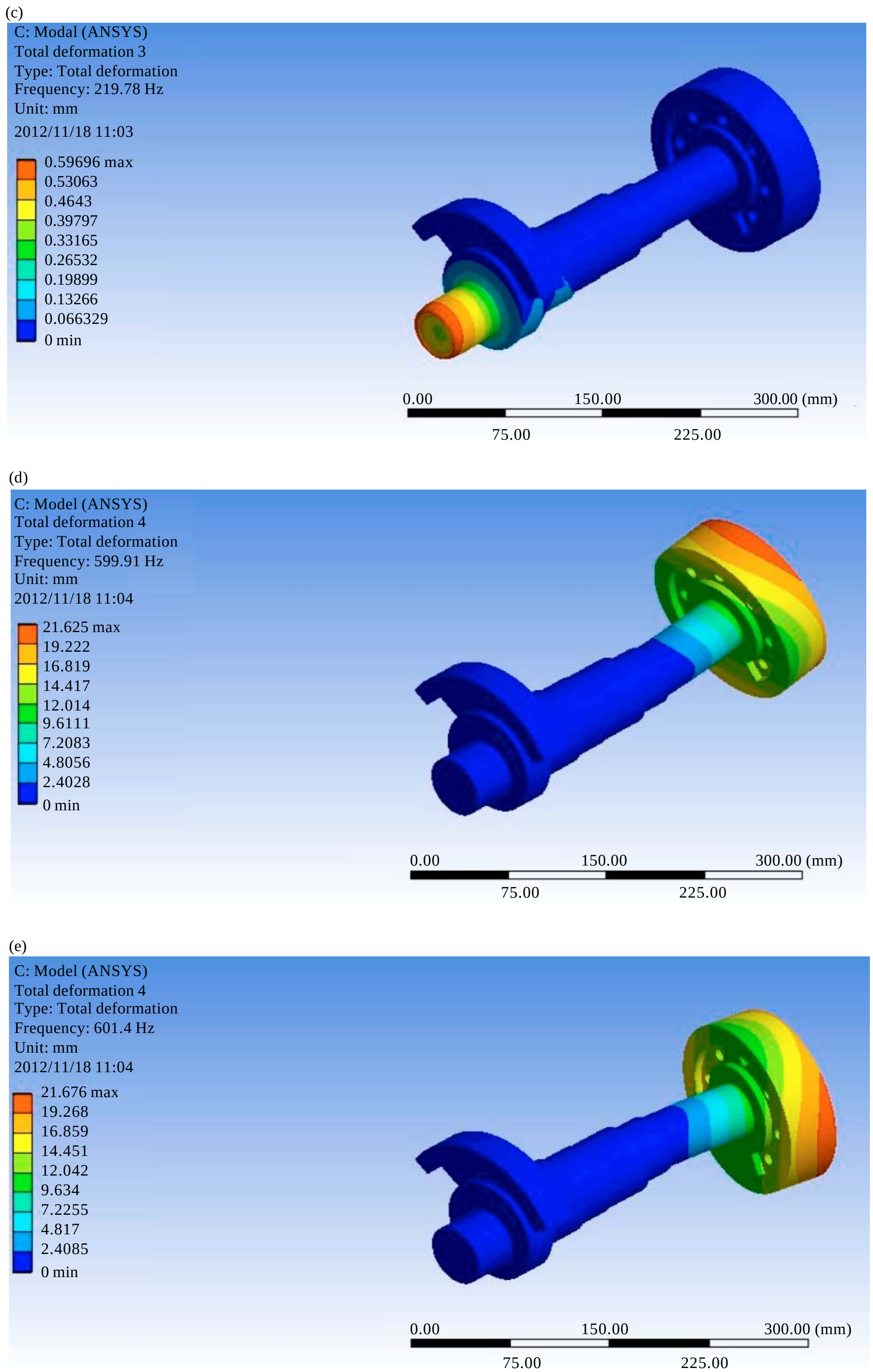


Fig. 7(a-e): All order modal vibration modes (a) First step: Bending, (b) Second step: Torsional, (c) Third step: Tensile, (d) Fourth step: Bending and (e) Fifth step: Bending

Table 2: Crankshaft rotor first five natural frequencies and critical speed under rigid support

Order	Frequency (f_0 Hz^{-1})	Critical speed (n_0 r^{-1} min^{-1})
1	111.23	6673.8
2	111.84	6710.4
3	226.38	13582.8
4	654.03	39241.8
5	655.76	39345.6

Rotor's natural frequency falls between 111.23 and 655.76 H_Z and it can be accelerated with the increasing of order. The rotor's critical rotation speed under first order natural frequency is at 6673.8 r min^{-1}, much larger than that of the scroll compressor at 2800 r min^{-1} which means that the rotor has become rigid. Rotor's fundamental frequency is 46.7 H_Z, far smaller than the natural frequency of the first two orders. Therefore, under normal working conditions, low-frequency resonance will not occur in rotor system. On the basis of the above analysis, rotor's frequency is not in the scope of the scroll compressor's resonance frequency and therefore, the crankshaft rotor meets the requirements of vibration safety.

Figure 7 shows the typical vibration mode of scroll compressor's rotor at each order. As indicated in the figure, first order and second order have a close nature frequency and the vibration mode is characterized with tangential and radial curve and twist at the end of crank pin. Vibration mode of the 3rd order shows crank pin's axial stretching. As for fourth and fifth order, the vibration of modes indicates rotor's bending and swaying, with symmetrical deformation of the belt pulley.

In the first five natural frequency and modal shape of scroll compressor's rotor, deformation occurs near the bearing constraints. Since crank pin is a cantilever beam of the crankshaft with certain deflection, orbiting scroll will be loaded to crank pin in the form of lumped mass, leaving crank pin with a more serious deformation than other parts. The deformation of crank pin has a direct effect on the orbiting scroll's drive bearing.

Bearing stiffness's effects on rotor's dynamic characteristics under elastic support: Based on the calculated damping matrix C and stiffness matrix K, elastic support is applied to the main shaft bearing. The purposes of using elastic support are (Shen *et al*., 2010; Chen, 2010): (1) Making critical rotation speed meet the design requirements by adjusting the stiffness of elastic support; (2) Reducing the dynamic amplitude by taking advantage of deformation caused by elastic support and allowing damper to absorb the vibration energy from rotor system.

Once a prestressed mode analysis is performed for the first five orders of scroll compressor's rotor, the natural frequencies and the according critical rotation speeds can be obtained as shown in Table 3. Based on the above analysis, stiffness of rolling bearing is set as 5×10^8 N m^{-1}, considering the insignificant effects of damper it will be ignored and critical rotation speed under different stiffness can be calculated. If the bearing's stiffness is altered, then its dynamic stiffness needs to be considered. Therefore, during the calculation, the dynamic stiffness will be adjusted in order to analyze its effects on the rotor modal. Table 4 shows rotor's models of the first orders with dynamic stiffness decreased and increased by 20%.

Table 3: Rigid support the crankshaft rotor first five natural frequencies and critical speed

Order	Frequency (f_0 Hz^{-1})	Critical speed (n_0 r^{-1} min^{-1})
1	89.14	5348.0
2	89.34	5361.4
3	130.95	7857.0
4	598.34	35900.4
5	600.52	36031.2

Table 4: Crankshaft rotor first five natural frequencies and critical speed under different bearing stiffness

Order	Frequency (f_0 Hz^{-1})	Critical speed (n_0 r^{-1} min^{-1})
(a) Bearing stiffness decreased 20%		
1	80.75	4845.0
2	82.14	4928.4
3	110.34	6620.4
4	499.34	29960.4
5	520.52	31231.2
(b) Bearing stiffness increased 20%		
1	99.78	5986.8
2	100.56	6033.6
3	150.76	9045.6
4	610.42	36625.2
5	628.34	37700.4

As Table 3 indicates, when elastic support is applied, under the same rotation speed, natural frequencies for the first five orders of rotor are lower compared with rigid support. With rotor's normal function, bearing's stiffness will be reduced and so is the rotor system's natural frequency. According to Table 4a, natural frequencies of fourth and fifth orders change a lot while the first and second orders show a smaller change in natural frequencies. If bearing's stiffness is increased, then the change of frequency for the first two orders are even less than that when the bearing's stiffness is reduced and the remaining 3 orders still show greater changing in natural frequencies, as indicated in Table 4b.

Under elastic support, the first order critical rotation speed of crankshaft's rotor is 5348 r min^{-1}. On the basis of the above analysis its working speed range should be below 4200 r min^{-1} or above 6400 r min^{-1}. When stiffness of elastic support is altered, the working speed range changes will be below 3800 r min^{-1} or above 5800 r min^{-1} and below 4700 r min^{-1} or above 7000 r min^{-1}.

CONCLUSION

Using finite element analysis software ANSYS, the prestressed modal for scroll compressor's rotor can be analyzed and the natural frequencies and critical rotation speeds for the first five orders are obtained. Under rigid support, rotor's working rotation speed deviates 20% from the critical rotation speed with the maximum working speed range below 5300 r min^{-1}. During normal function, rotor system's fundamental frequency is far less than the first order natural frequency and therefore no resonance will occur. Since crank pin is the cantilever beam of the crankshaft, serious deformation can be happen. By changing constraint way of main shaft bearing and using elastic support instead of rigid support, rotor's mode of vibration will be much smaller. The analysis shows that elastic support can not only enable rotor's working speed to easily avoid its bending critical speeds but also can reduce the dynamic amplitude with the deformation it caused and effectively reduce rotor's vibration at critical rotation speed. Moreover, under elastic support, rotor system's natural frequency will be changed accordingly as decreasing and increasing of bearing's stiffness, with a much smaller vibration mode than under rigid support. The software-based modal analysis of scroll compressor's rotor under prestressed structure in the study, can lay the foundation of multi-body dynamics and it will provide a guarantee for scroll compressor's stead operation.

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