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ORIGINAL PAPER

A method for modeling and analyzing the rotor dynamics of a locomotive turbocharger

Jianwu Yang · Yaju Gao · Zhifeng Liu · Chengbin Zhao · Taiti Kang · Lichao Gu · Bo Xu

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Abstract The rotor of the locomotive turbocharger is prone to malfunction of the turbocharger, and it is also the most important part of the turbocharger. The reasonable establishment of rotor dynamics model is very important in the study. It determines the accuracy of analysis. The influence of the shaft quality on the turbine and impeller was not considered in the process of analyzing the dynamic model in the passed research. But the quality of locomotive turbocharger rotor shaft is relatively large, it cannot be ignored because it will have a large bending moment in the model simplified. So the rotor dynamic equation was deduced in the case of considered the quality of the rotor axis. The rotor of the turbocharger rotor was simplified firstly, the various parts of the simplified rotor were analyzed, and the dynamic model of the rotor was established. And the dynamic model was verified by the hammer experiment. The factors considered in the dynamic model were more comprehensive, so the model would be more practical for the future research.

J. Yang (✉) · Y. Gao
Intelligent Mechanical and Electrical Equipment Laboratory, Beijing University of Technology, Beijing, China
e-mail:yangjianwu111@sina.com

Z. Liu · C. Zhao · T. Kang · L. Gu · B. Xu
Key Laboratory of Advanced Manufacturing Technology, Beijing University of Technology, Beijing, China

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1 Introduction

The fuel for diesel locomotives accounts for 60 % of oil consumption in the modern society. Meanwhile, the emissions of diesel engine exhaust gas are an important source of pollution. Although the state has introduced a new energy plan, its implementation is not yet mature. So as a kind of energy saving and environmental protection technology, the working principle of the turbocharger is the exhaust gas to drive the turbine to rotate, the turbine drives the impeller to rotate through the shaft, and the air is pressed by the impeller into the engine. So the turbocharger technology has important significance for the development of economy. Turbocharger is an important system of engine. The torque and efficiency of the engine with turbocharger will be increased by 30% $\sim$ 20%, that can reach the affect of 20% $\sim$ 10% of gasoline engine energy saving and 20% $\sim$ 10% of diesel engine energy saving. Scholars at domestic and abroad have carried out a lot of research on the turbocharger.

Li [1] established the dynamic equation of the rotor, calculated and analyzed the modal frequency and mode shapes of the free rotor. The relationship between the natural frequency of the rotor system and the stiffness of the bearing was studied, and the relationship between the two was analyzed by means of the vibration mode and the Campbell diagram. Ying [2] studied the modeling and analyzed the method of the rotor dynamics of the turbocharger under the consideration of the base excitation. The basic excitation identification of the turbocharger was carried out. The dynamic problem of the rotor under the action of the nonlinear oil film force was discussed. An effective method was used to realize the system simulation, and a method of fault diagnosis and state detection was proposed. Wan [3] researched a new type diesel engine exhaust turbocharger rotor, Compared with the experimental data, the results showed that the finite element method was more close to the engineering practice. Zhang [4] had analyzed the dynamic characteristics of the rigid rotor system by using nonlinear dynamic theory and method under some certain conditions. The results showed that the system was rich in nonlinear dynamic behavior in a certain range of parameters. The method had fast convergence rate and high accuracy, which provided a theoretical basis for the stability of the rotor-bearing system. Liu [5] used the finite element analysis method to calculate the strength and dynamic characteristics of the rotor. A finite element analysis software was used to analyze the dynamic characteristics of a turbine engine, and the critical speed of the rotor system was calculated. Yu [6] studied the dynamic characteristics of flexible rotor system, rotational speed ratio, unbalanced amount, damping ratio, viscosity as control parameters. The numerical results of the system were presented, and the long-term state of the system was predicted by the Lagrange interpolation. The numerical results of the dynamic oil film force model and the steady oil film force model were compared. The rationality of the dynamic nonlinear oil film force model was demonstrated. Wu [7] established the dynamic model of the bearing rotor system according to the actual situation. The nonlinear dynamic model of the bearing rotor system was established. The effect of the nonlinear oil film force on the rotor system was analyzed by numerical simulation. The bifurcation and chaotic motion of the system under different conditions were studied. The effects of the nonlinear responses were obtained. The complex motion forms and their evolution processes of the system were analyzed. Luo [8] considered the influence of the gravity and the unbalanced inertia force on the shaft deflection of the rotor of the high speed rotor of the turbocharger rotor. He had analyzed the overall structure of the turbocharger, improved the machining accuracy, reduced the axial vibration and improved the service life of the machine. Zhu [9] used machining process of the impeller and the impeller shaft modeling and numerical simulation. Leng [10] established the dynamic equation of 20 wm steam turbine rotor-bearing system. Xiao [11] took a discussion on the reasonable dynamic model and the corresponding efficient algorithm for the sliding bearing, gas seal and complex rotor system as the research purpose. The project with local nonlinear characteristics of rotor system of high latitude had been focused. Some innovations had been made in theory and application, and the results obtained had a positive guiding significance for the engineering application of the analysis of complex rotor system's stability. The finite element model of the flexible rotor-bearing system was established by Li [12], and the influence factors of the nonlinear factors of the oil film force support were added. The results showed that the nonlinear characteristics of the two models were significantly different, and the nonlinear characteristics of the rotor system could be obtained by using the finite element model. Ying [13] used the nonlinear oil film force and nonlinear oil film force database method to derive the tilting pad bearing. For considering the influence of inertia on tile oil film force, he analyzed the variation of the dynamic stiffness and dynamic damping coefficient with the increase in rotational speed. The linear stability index of the rotor system with different mode orders at a certain speed was calculated, and the stability of the system was obtained. Pubio [14] proposed a new method to analyze the nonlinear dynamics of a cracked rotor. The proposed method is 100 times higher than the conventional method. In addition, the use of simplified methods based on the quasi-static stiffness matrix could not be adequate. Avramov [15] considered the effect of the oil film between the asymmetry and the short bearing of the rotor, the rotor rotating torque. The oil force of the bearing was deduced, and the four nonlinear equations were derived to study the vibration of the rotor. The origination of self-sustained vibrations of rotor is studied by means of Shaw Pierre nonlinear modes. The harmonic balance method is applied to study the self-sustained vibrations with large amplitudes.

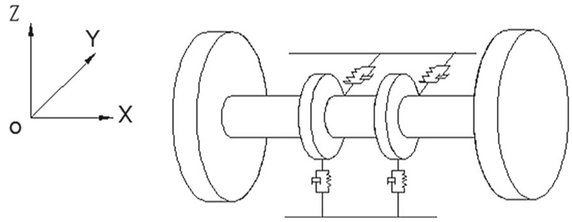

Fig. 1 Simplified model of the turbocharger rotor

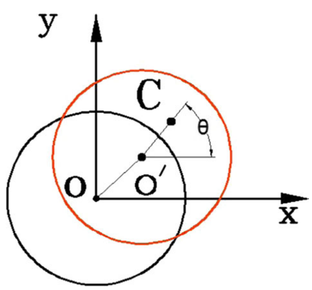

Fig. 2 The position of the axis of motion and the rest

2 The establishment and verification of the dynamic model of the rotor of the turbocharger roto

2.1 Establishment of rotor dynamic model

(1) Dynamic equation of the *I* disk

From Fig. 1, the rotor of a locomotive turbocharger rotor can be simplified to four disks and three axis segments, black circle represents the position of a shaft, and the red circle represents the position of axis rotation stability. In Fig. 2, the black circle is the static axial position and the blue circle is the center where the movement of the position. point O' is the center of the axis, point C is the center of gravity and the length of OC is e, where e is eccentricity. The angle is θ after the disk of the rotor rotated in the counterclockwise for the time of t

$\theta = \omega t$

The i disk projection displacement in the X' and Y' are

$$\begin{cases} X_i' = X_i + e_i \cos(\omega t) \\ Y_i' = Y_i + e_i \sin(\omega t) \end{cases} \tag{1}$$

X_i is the displacement of the i disk along the X axis, Y_i is the displacement of the i disk along the Y axis, e_i is the eccentricity of the i disk, ω is the angular velocity of the disk rotation, and t is the time of the rotation.

Because $\Sigma F_x = 0$, $\Sigma F_y = 0$. So by Newton's second law, it can be as follows:

$$\begin{cases} F_{xi}^{\mathrm{L}} - F_{xi}^{\mathrm{R}} - F_{bxi} = m_i \ddot{X}_i' \\ F_{yi}^{\mathrm{L}} - F_{yi}^{\mathrm{R}} - F_{byi} = m_i \ddot{Y}_i' \end{cases} \tag{2}$$

So the formula can be :

$$\begin{cases} F_{xi}^{\mathrm{L}} - F_{xi}^{\mathrm{R}} - F_{bxi} + m_i e_i \omega^2 \cos(\omega t) = m_i \ddot{X}_i \\ \\ F_{yi}^{\mathrm{L}} - F_{yi}^{\mathrm{R}} - F_{byi} + m_i e_i \omega^2 \sin(\omega t) = m_i \ddot{Y}_i \end{cases} \tag{3}$$

F_{xi}^{L} is the force on the left side of the disk in XZ plane, and F_{xi}^{R} is the force on the right side of the disk in the XZ plane. F_{yi}^{L} is the force on the left side of the YZ plane, and F_{yi}^{R} is the force on the right side of the disk in the YZ plane. X_i is the displacement of wheel along the X direction. Y_i is the displacement of wheel along the Y direction. The m_i is the quality of the i disk, and ω_r is the angular velocity of the rotor rotation. F_{bxi} is the force of bearing on the wheel along the X axis, F_{byi} is the force of bearing on the wheel along the Y axis.

(2) The derivation of the equilibrium equation of the inertia moment for the i disk.

The torque along the X axis and the torque along the Y axis are:

$$\begin{cases} M_{gx} = J_a \ddot{\phi} \\ M_{gy} = J_a \ddot{\varphi} \end{cases}$$

$$\text{so} \begin{cases} M_{yi}^{\mathrm{R}} - M_{yi}^{\mathrm{L}} - J_{ai} \ddot{\varphi}_i + \omega J_{pi} \dot{\phi}_i = 0 \\ M_{xi}^{\mathrm{R}} - M_{xi}^{\mathrm{L}} + J_{ai} \ddot{\phi}_i + \omega J_{pi} \dot{\varphi}_i = 0 \end{cases} \tag{4}$$

M_{yi}^{L} is the moment of the i wheel on the left side of the XZ plane. M_{yi}^{R} is the moment of the i wheel on the right side of the XZ plane. M_{xi}^{L} is the moment of the i wheel on the left side of the YZ plane. M_{xi}^{R} is the moment of the i wheel on the right side of the YZ plane. ϕ_i is the i wheel rotation by X, and φ_i is the i wheel rotation by Y. In summary, the kinetic equation that be obtained from (3) (4) is:

$$\begin{cases} F_{xi}^{\mathrm{L}} - F_{xi}^{\mathrm{R}} - F_{bxi} + m_i e_i \omega^2 \cos(\omega t) = m_i \ddot{X}_i \\ \quad M_{yi}^{\mathrm{R}} - M_{yi}^{\mathrm{L}} - J_{ai} \ddot{\varphi}_i + \omega J_{pi} \dot{\phi}_i = 0 \\ F_{yi}^{\mathrm{L}} - F_{yi}^{\mathrm{R}} - F_{byi} + m_i e_i \omega^2 sin(\omega t) = m_i \ddot{Y}_i \\ \quad M_{xi}^{\mathrm{R}} - M_{xi}^{\mathrm{L}} + J_{ai} \ddot{\phi}_i + \omega J_{pi} \dot{\varphi}_i = 0 \end{cases}$$
$$i = 1, 2, 3, 4$$

(3) The derivation of the mechanical equation of the i axis

The mechanical equation in the *ZX* plane is:

$$\begin{cases} X_{i+1} = X_i + L_i\varphi_i + \frac{L_i^2}{2EI_i}M_{yi}^{\mathrm{R}} + \frac{L_i^3}{6EI_i}F_{xi}^{\mathrm{R}} \\ \varphi_{i+1} = \varphi_i + \frac{L_i}{EI_i}M_{yi}^{\mathrm{R}} + \frac{L_i^2}{2EI_i}F_{xi}^{\mathrm{R}} \\ M_{y,i+1}^{\mathrm{L}} = M_{yi}^{\mathrm{R}} + F_{xi}^{\mathrm{R}}L_i \\ F_{x,i+1}^{\mathrm{L}} = F_{xi}^{\mathrm{R}} \end{cases} \tag{5}$$

The stress analysis of the shaft section in the *YZ* plane can be:

$$\begin{cases} Y_{i+1} = Y_i - L_i\phi_i + \frac{L_i^2}{2EI_i}M_{xi}^{\mathrm{R}} + \frac{L_i^3}{6EI_i}F_{yi}^{\mathrm{R}} - \frac{kL_i^4}{8EI_i} \\ \phi_{i+1} = \phi_i - \frac{L_i}{EI_i}M_{xi}^{\mathrm{R}} - \frac{L_i^2}{2EI_i}F_{yi}^{\mathrm{R}} + \frac{kL_i^3}{6EI_i} \\ M_{x,i+1}^{\mathrm{L}} = M_{xi}^{\mathrm{R}} + F_{yi}^{\mathrm{R}}L_i - \frac{1}{2}kL_i^2 \\ F_{y,i+1}^{\mathrm{L}} = F_{yi}^{\mathrm{R}} + kL_i \end{cases} \tag{6}$$

Mechanical equations of the *i* axis could be obtained from (5) and (6),

$$\begin{cases} X_{i+1} = X_i + L_i\varphi_i + \frac{L_i^2}{2EI_i}M_{yi}^{\mathrm{R}} + \frac{L_i^3}{6EI_i}F_{xi}^{\mathrm{R}} \\ \varphi_{i+1} = \varphi_i + \frac{L_i}{EI_i}M_{yi}^{\mathrm{R}} + \frac{L_i^2}{2EI_i}F_{xi}^{\mathrm{R}} \\ M_{y,i+1}^{\mathrm{L}} = M_{yi}^{\mathrm{R}} + F_{xi}^{\mathrm{R}}L_i \\ F_{x,i+1}^{\mathrm{L}} = F_{xi}^{\mathrm{R}} \\ \phi_{i+1} = \phi_i - \frac{L_i}{EI_i}M_{xi}^{\mathrm{R}} - \frac{L_i^2}{2EI_i}F_{yi}^{\mathrm{R}} + \frac{kL_i^3}{6EI_i} \\ Y_{i+1} = Y_i - L_i\phi_i + \frac{L_i^2}{2EI_i}M_{xi}^{\mathrm{R}} + \frac{L_i^3}{6EI_i}F_{yi}^{\mathrm{R}} - \frac{kL_i^4}{8EI_i} \\ M_{x,i+1}^{\mathrm{L}} = M_{xi}^{\mathrm{R}} + F_{yi}^{\mathrm{R}}L_i - \frac{1}{2}kL_i^2 \\ F_{y,i+1}^{\mathrm{L}} = F_{yi}^{\mathrm{R}} + kL_i \end{cases}$$
$$i = 1, 2, 3 \tag{7}$$

L_i is the length of the *i* axis, the E is the elastic constant, and I_i is the moment of inertia of the *I* wheel.

(4) Dynamic model of the rotor without damping could be obtained from the mechanical equation of the *i* disk and the *i* rotor. The dynamic model of the rotor is:

$$m_1\ddot{Y}_1 + \frac{12EI_1}{L_1^3}Y_1 - \frac{6EI_1}{L_1^2}\Phi_1 - \frac{12EI_1}{L_1^3}Y_2 - \frac{6EI_1}{L_1^2}\Phi_2 - \frac{kL_1}{2} = m_1e_1\omega_r^2\sin\omega_r t$$

$$J_{a1}\ddot{\Phi}_1 + \omega_r J_{p1}\dot{\Psi}_1 - \frac{6EI_1}{L_1^2}Y_1 + \frac{4EI_1}{L_1}\Phi_1 J_{a1}\ddot{\Phi}_1 + \omega_r J_{p1}\dot{\Psi}_1 - \frac{6EI_1}{L_1^2}Y_1 + \frac{4EI_1}{L_1}\Phi_1 + \frac{6EI_1}{L_1^2}Y_2 + \frac{2EI_1}{L_1}\varphi_2 + \frac{2}{3}kL_1^2 = 0$$

$$m_2\ddot{Y}_2 - \frac{12EI_1}{L_1^3}Y_1 + \frac{6EI_1}{L_1^2}\Phi_1 + \left(\frac{12EI_1}{L_1^3} + \frac{12EI_2}{L_2^3} + k_b\right)Y_2 + \left(\frac{6EI_1}{L_1^2} - \frac{6EI_2}{L_2^2}\right)\Phi_2 - \frac{12EI_2}{L_2^3}Y_3 - \frac{6EI_2}{L_2^2}\Phi_3 - \frac{kL_2}{2} = 0$$

$$J_{a2}\ddot{\Phi}_2 + \omega_r J_{p2}\dot{\varphi}_2 - \frac{6EI_1}{L_1^2}Y_1 + \frac{2EI_1}{L_1}\Phi_1 + \left(\frac{6EI_1}{L_1^2} - \frac{6EI_2}{L_2^2}\right)Y_2 + \left(\frac{4EI_1}{L_1} + \frac{4EI_2}{L_2}\right)\Phi_2 + \frac{6EI_2}{L_2^2}Y_3 + \frac{2EI_2}{L_2}\Phi_3 + \frac{2}{3}kL_2^2 - \frac{1}{6}kL_1^2 = 0$$

$$m_3\ddot{Y}_3 - \frac{12EI_2}{L_2^3}Y_2 + \frac{6EI_2}{L_2^2}\Phi_2 + \left(\frac{12EI_2}{L_2^3} + \frac{12EI_3}{L_3^3} + k_b\right)Y_3 + \left(\frac{6EI_2}{L_2^2} - \frac{6EI_3}{L_3^2}\right)\Phi_3 - \frac{12EI_3}{L_3^3}Y_4 - \frac{6EI_3}{L_3^2}\Phi_4 - \frac{kL_3}{2} = 0$$

$$J_{a3}\ddot{\Phi}_3 + \omega_r J_{z3}\dot{\varphi}_3 - \frac{6EI_2}{L_2^2}Y_2 + \frac{2EI_2}{L_2}\varphi_2 + \left(\frac{6EI_2}{L_2^2} - \frac{6EI_3}{L_3^2}\right)Y_3 + \left(\frac{4EI_2}{L_2} + \frac{4EI_3}{L_3}\right)\Phi_3 + \frac{6EI_3}{L_3^2}Y_4 + \frac{2EI_3}{L_3}\Phi_4 + \frac{2}{3}kL_3^2 - \frac{1}{6}kL_2^2 = 0$$

$$m_4\ddot{Y}_4 - \frac{12EI_3}{L_3^3}Y_3 + \frac{6EI_3}{L_3^2}\Phi_3 + \frac{12EI_3}{L_3^3}Y_4 + \frac{6EI_3}{L_3^2}\Phi_4 + \frac{kL_3}{2} = m_4e_4\omega_r^2\sin\omega_r t$$

$$J_{a4}\ddot{\Phi}_4 + \omega_r J_{p4}\dot{\varphi}_4 - \frac{6EI_3}{L_3^2}Y_3 + \frac{2EI_3}{L_3}\Phi_3 + \frac{6EI_3}{L_3^2}Y_4 + \frac{4EI_3}{L_3}\Phi_4 - \frac{1}{6}kL_3^2 = 0$$

$$m_1\ddot{X}_1 + \frac{12EI_1}{L_1^3}X_1 + \frac{6EI_1}{L_1^2}\varphi_1 - \frac{12EI_1}{L_1^3}X_2 + \frac{6EI_1}{L_1^2}\varphi_2 = m_1e_1\omega_r^2\cos\omega_r t$$

$$J_{a1}\ddot{\varphi}_1 - \omega_r J_{p1}\dot{\Phi}_1 + \frac{6EI_1}{L_1^2}X_1 + \frac{4EI_1}{L_1}\varphi_1 - \frac{6EI_1}{L_1^2}X_2 + \frac{2EI_1}{L_1}\varphi_2 = 0$$

$$m_2\ddot{X}_2-\frac{12EI_1}{L_1^3}X_1-\frac{6EI_1}{L_1^2}\Psi_1$$
$$+\left(\frac{12EI_1}{L_1^3}+\frac{12EI_2}{L_2^3}+k_b\right)X_2$$
$$+\left(-\frac{6EI_1}{L_1^2}+\frac{6EI_2}{L_2^2}\right)\varphi_2-\frac{12EI_2}{L_2^3}X_3+\frac{6EI_2}{L_2^2}\varphi_3=0$$
$$J_{a2}\ddot{\varphi}_2-\omega_r J_{p2}\dot{\Phi}_2+\frac{6EI_1}{L_1^2}X_1+\frac{2EI_1}{L_1}\varphi_1$$
$$+\left(-\frac{6EI_1}{L_1^2}+\frac{6EI_2}{L_2^2}\right)X_2+\left(\frac{4EI_1}{L_1}+\frac{4EI_2}{L_2}\right)\varphi_2$$
$$-\frac{6EI_2}{L_2^2}X_3+\frac{2EI_2}{L_2}\varphi_3=0$$
$$m_3\ddot{X}_3-\frac{12EI_2}{L_2^3}X_2-\frac{6EI_2}{L_2^2}\varphi_2$$
$$+\left(\frac{12EI_2}{L_2^3}+\frac{12EI_3}{L_3^3}+k_b\right)X_3$$
$$+\left(-\frac{6EI_2}{L_2^2}+\frac{6EI_3}{L_3^2}\right)\varphi_3-\frac{12EI_3}{L_3^3}X_4+\frac{6EI_3}{L_3^2}\varphi_4=0$$
$$J_{a3}\ddot{\varphi}_3-\omega_r J_{p3}\dot{\Phi}_3+\frac{6EI_2}{L_2^2}X_2+\frac{2EI_2}{L_2}\varphi_2$$
$$+\left(-\frac{6EI_2}{L_2^2}+\frac{6EI_3}{L_3^2}\right)X_3+\left(\frac{4EI_2}{L_2}+\frac{4EI_3}{L_3}\right)\varphi_3$$
$$-\frac{6EI_3}{L_3^2}X_4+\frac{2EI_3}{L_3}\varphi_4=0$$
$$m_4\ddot{X}_4-\frac{12EI_3}{L_3^3}X_3-\frac{6EI_3}{L_3^2}\varphi_3+\frac{12EI_3}{L_3^3}X_4$$
$$-\frac{6EI_3}{L_3^2}\varphi_4=m_4e_4\omega_r^2\cos\omega_r t$$
$$J_{a4}\dot{\varphi}_4-\omega_r J_{p4}\dot{\Phi}_4+\frac{6EI_3}{L_3^2}X_3+\frac{2EI_3}{L_3}\varphi_3$$
$$-\frac{6EI_3}{L_3^2}X_4+\frac{4EI_3}{L_3}\varphi_4=0$$

L_1, L_2, L_3 are the length of the first, second and third axis, E is the modulus of elasticity of the shaft, I_1, I_2, I_3 are the moment of inertia of the first, second and third axes. e_1 is the eccentric distance of the first disk, e_4 is the offset of fourth wheels, and ω_r is the angular velocity of the rotor. $J_{a1}, J_{a2}, J_{a3}, J_{a4}$ and $J_{p1}, J_{p2}, J_{p3}, J_{p4}$ are the first, second, third and fourth wheel of the equatorial and polar moment of inertia. k_b is bearing stiffness, and K is the gravity of the shaft section of the unit length. The rest of the variables have been defined above. The general form of the model is:

$$M_r\ddot{X}_r+w_rG_r\ddot{X}_r+K_rX_r+C=F_r$$

M is the mass matrix, G is the top matrix, K is the stiffness matrix, F_r is the excitation matrix, and C is the constant matrix.

3 Validation of rotor dynamic model

The modal calculation of the free state of the rotor is carried out in this paper. When the vibration mode of the free rotor is calculated, the supporting force of the oil film is not to be played, as the rotor is not spinning so the gyro effect is not up to the effect. $K_b=0$, $\omega_r=0$, but gravity still has a role in the rotor, so the F_r is not a zero matrix. So the type $M_r\ddot{X}_r+w_rG_r\dot{X}_r+K_rX_r+C=F_r$ is rewritten into: $M_r\ddot{X}_r+K_r\dot{X}_r+C=0$ (7).

The characteristics of rotor physical and geometrical parameters into MATLAB solution, you can get natural frequency. Parameter reference Table 1.

The rotor with a soft rope suspension under the stainless steel frame is used to simulate the free-free boundary conditions. Using the method of multi-point excitation, the rotor of the turbine impeller and the two axis of the neck were, respectively, with the hammer pulse excitation.

The layout of the sensor is shown in Fig. 3, and the sensor arranged on the impeller is corresponding to the arrangement of the sensor. Figure 4 is the sensor, and Fig. 5 is the LMS detection equipment; LMS is modal, vibration, noise test and analysis system. It is the equipment that was often used in experiments; firstly, we choose the appropriate acceleration sensors according to the selected layout. The collected signals need to be processed by the computer.

If the error of the natural frequency of the experimental results is in the range of the requirements, the establishment of the dynamic equations is reasonable.

Table 1 Parameters of the rotor parts of the locomotive turbocharger

	Modulus of elasticity (GP)	Poisson's ratio	Density (g/cm^3)
Turbine	169	0.29	7.93
Shaft	206	0.3	7.8
Impeller	87.5	0.3	2.7

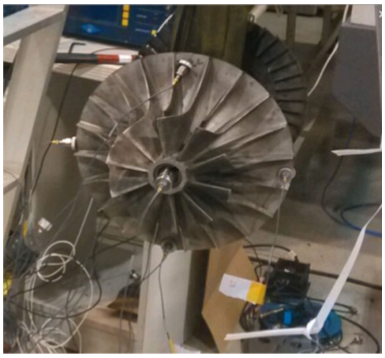

Fig. 3 Vibration sensor

Fig. 5 LMS test equipment

Fig. 4 The arrangement of the sensor on the impeller on the impeller

Take the first four order results as Fig. 6.

Usually, the error is not more than 5 %. The theoretical results and the experimental results are in error according to Table 2. Because in the process of surveying, mapping and the layout of the sensor will cause errors. But it can be seen from the above analysis that the inherent frequency of the theoretical calculation and the experimental results are within the allowable error range. So the rotor dynamics that deduced under the consideration of bending moment of the shaft is reasonable.

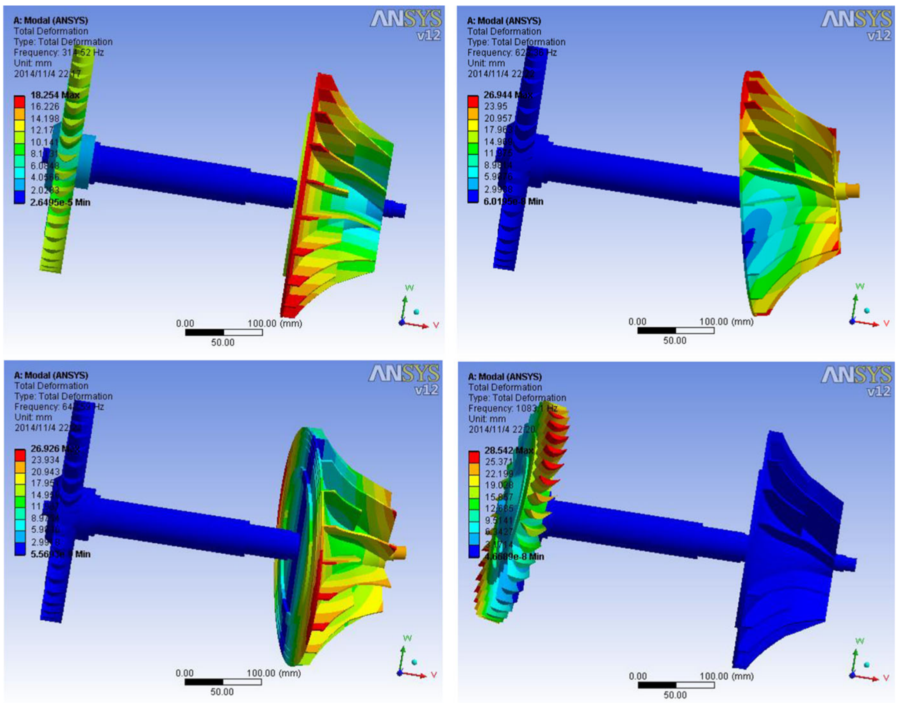

Fig. 6 From left to right for the one two three four modes

Table 2 Comparison of the results of the calculated and experimental natural frequencies

Order number	Natural frequency	Standard natural frequency	Error (%)
One	314.52	310.11	2.64
Two	623.36	643.31	3.22
Three	642.59	638.59	4.12
Four	1083.1	1064.64	4.36

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References

1. Li, G.-J.: Research on the Dynamic Characteristics and Visualization of the Turbocharger Rotor. Hebei University of Technology, Hebei (2003)
2. Ying, G.-C.: The Basic Excitation Identification and Rotor Dynamics Research of the Turbocharger. Shanghai Jiao Tong University, Shanghai (2008)
3. Wan, K.-H.: Study on the critical speed of the rotor critical speed of the turbocharger. New Technol. New process **10**, 18–19 (2009)
4. Zhang, W.-Z.: Analysis of nonlinear dynamic characteristics of the system of Jeffcott rotor—bearing. Mech. Eng. **4**, 20–22 (2011)
5. Liu, X.: Strength and Rotor Dynamics Analysis of One Type Turbocharger. Dalian Maritime University, Dalian (2013)
6. Yu, H.: The bifurcation and chaos of the flexible rotor with the dynamic oil film. Mechanics **34**(5), 827–831 (2002)
7. Wu, Q-l: A certain type of turbine rotor engine strength and rotor dynamic analysis. J. Liao Ning Shi Hua Univ. **2**, 53–57 (2008)
8. Luo, B.-B.: Machining process and dynamic balancing of the rotor parts of the turbocharger. Highw. Car **1**, 12–14 (2006)
9. Lei, Z.: Study on the dynamic behavior of the rotor system of the turbocharger rotor system considering the floating ring bearing, vol. 6, pp. 26–32. Hefei University of Technology (2013)
10. Leng, S.-X.: Linear and nonlinear oil film force model of rotor vibration stability contrast. Turbine Technol. **5**, 298–300 (2003)
11. Xiao, Z.: Rotor Bearing Seal System Dynamics Modeling and Its Characteristics Research. Fudan University, Shanghai (2006)
12. Li, Z.-F.: Two types of rotor bearing system model of nonlinear characteristics. China Mech. Eng. **2**, 218–222 (2009)
13. Ying, J.-Y.: Nonlinear Dynamic Characteristics and Stability Analysis of Tilting Pad Sliding Bearing System. Harbin Industrial University, Harbin (2009)
14. Pubio, L.: A new efficient procedure to solve the nonlinear dynamics of a cracked rotor. Nonlinear Dyn. **70**(3), 827–831 (2012)
15. Avramov, K.: Nonlinear dynamics of one disk asymmetrical rotor supported by two journal bearings. Nonlinear Dyn. **67**(2), 1201–1219 (2012)