New Structure Design and Rigidity Analysis of A Dynamic Balance Machine

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Abstract: Rotating machinery is important in the national economy and the people's livelihood, it is widely used in the industrial production, aerospace, hydroelectric power, transportation and other fields. With the development of production, the requirement of rotating mechanical properties and rotor manufacturing precision is higher, it is urgent to adopt advanced technology in dynamic balance test.Combined the research of domestic and foreign scholars, the theory and technology of rotor balancing principles have been summarized. The thesis has designed the mechanical structure of hard bearing dynamic balance machine, described the balancing method of influence coefficient and in-depth analyzed the two key technologies in the vibration analysis and research on test system of dynamic balance machine on the basis of the theoretical research.

Keywords: Supporting frame; Unbalance value; Modal analysis

1 Introduction

The design and research of dynamic balance machine can be traced back to the 1870s. Martinson designed the early dynamic balance machine model. Later, in the late 19th century and the early 20th century, Stodola of Switzerland and Akimoff of the United States conducted in-depth research on the design of dynamic balance machine, which made the design of dynamic balance machine get fast development. The mechanical structure of the dynamic balance machine is the platform for the development of the test system and the experimental analysis, and it is the key to affect the dynamic characteristics of the system[1-5].

After the 20th century, with the research and development of balancing technology by researchers, the dynamic balance machine improved by Heyman was applied to industrial production. The dynamic balancing technology is also more and more mature. The products of each dynamic balance machine manufacturer are more and more, the functions are more and more perfect, and the test accuracy is also constantly

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improved[6]. Schenck is the most representative company in foreign countries, and Beijing Qingyun and Shanghai Jianping are the most representative companies in China. These companies produce a wide range of dynamic balance machine products, including low-speed and high-speed, fixed and mobile, special and general, vertical and horizontal.

Starting from the design of mechatronics research project, this paper mainly completed the mechanical structure design of horizontal hard support dynamic balance machine, and completed the design and calculation of driving system, transmission system and support system according to the design requirements of low-speed dynamic balance machine.

2 Overall mechanical design

The mechanical structure of the horizontal hard support dynamic balance machine is shown in Fig. 1. The mechanical system is mainly composed of three parts: motor drive system, belt drive system and support frame system. (1) The drive system mainly completes the power drive from the motor to the rotor to ensure that the rotor rotates stably at the balanced speed; (2) the transmission system mainly transmits the power of the motor to the rotor through the belt to ensure the accuracy and stability of the rotor in the test process, so the general requirements for the dynamic balance machine are that the drive system is stable, the structure is simple and practical; (3) In addition to stably supporting the rotor, the supporting frame system is mainly used to measure the vibration information of the unbalance. The design of the supporting swing is the key to the design of the dynamic balance machine.

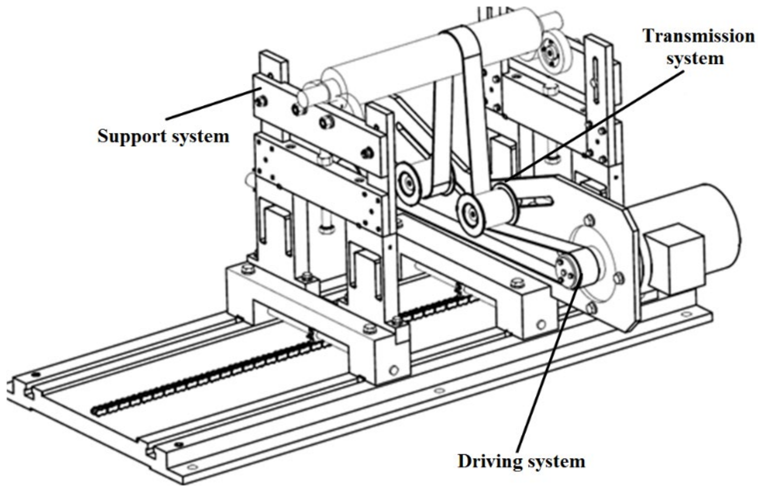

Fig.1 The model of dynamic balance machine

3 Mechanical characteristics of dynamic balance machine

3.1 Dynamic design requirements

The key technology of the hard support balancing machine system is to make the natural frequency of the system within the specified design range. During the design process of the supporting system, the natural frequency of the rotor supporting system must be far higher than the working frequency of the rotor, so as to ensure that the vibration of the dynamic balance machine will not lead to the resonance of the mechanism, the vibration of the supporting frame is low, ensuring that the vibration amplitude of the frame is proportional to the centrifugal force generated by the rotor unbalance, The magnitude and phase of the unbalance can be obtained by measuring the horizontal vibration of the frame. The frequency ratio between the working frequency and the natural frequency of the system is between 0.1-0.3 [7,8].

3.2 Vibration model of support system

Before analyzing and discussing the vibration characteristics of the dynamic balance machine, we must have a certain understanding of the vibration theory of the dynamic balance machine, and provide the theoretical basis for the finite element modal analysis and the dynamic balance machine experiment. Using the analytical method , the vibration model of the rotor on the mechanical support frame system is established first, as shown in Fig. 2.

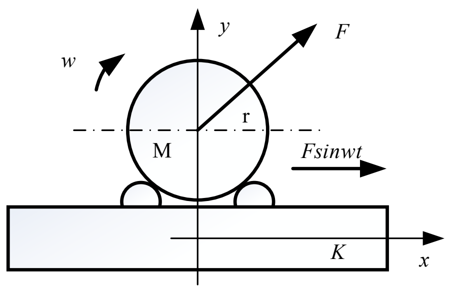

Fig.2 The force decomposition of rotor system

The rotor is clamped on the dynamic balance machine, the supporting frame supports the rotor stably by the rollers at both ends, and drives the rotor to rotate stably under the balanced speed through the motor driving belt. With the rotor rotating, the centrifugal inertia force will be generated due to the existence of the unbalance, which will cause the vibration of the supporting frame system. The Fig. shows the force state of the rotor on the vibration system of the hard supporting frame, and the vibration equation of the rotor support system in the X direction can be listed:

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$$M\frac{d^2x}{dt^2}+C\frac{dx}{dt}+Kx=Fx_0\sin\omega t \tag{1}$$

Where, F —Centrifugal force generated with rotor rotation unbalance, M —Total mass of rotor support system in vibration, ω —Rotating angular speed of rotor, K — Total stiffness of the rotor support system of the balancing machine in X direction, $x_0\sin\omega t$ —The displacement in X direction when the support frame vibrates. C — Damping of rotor support system. Ignore the damping of the supporting pendulum, and let $\omega_0^2=K/M$, It can be seen from the formula analysis, Based on the relationship between ω and ω_0 , i.e. the frequency ratio between the working frequency and the natural frequency of the rotor system, and $\omega/\omega_0<<1$ in the hard supported dynamic balance machine, Equation (1) can be reduced to

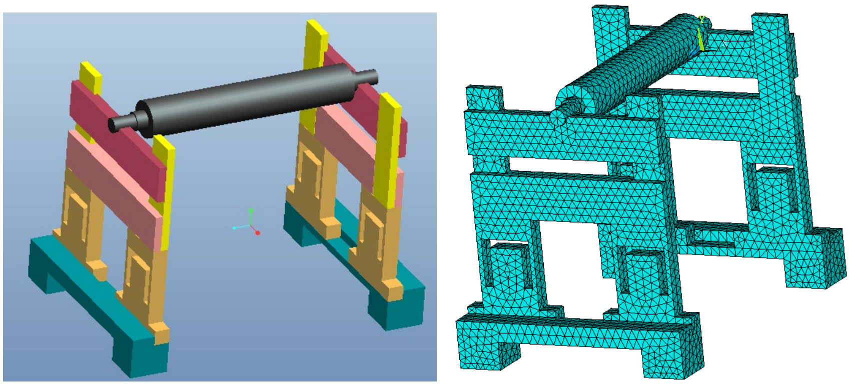
$$x_0\approx\frac{mr\omega^2}{K}=F/K \tag{2}$$

It can be seen that when the rigidity of the hard support frame is large enough, the vibration displacement of the frame is directly proportional to the centrifugal force produced by the unbalance of the rotor.

4 Dynamic calculation of balance machine

4.1 Finite element modal analysis of dynamic balance machine

The analysis of dynamic characteristics is the key of system design[9,10]. The model of dynamic balance machine designed by three-dimensional software is imported into the finite element software for modal analysis. Due to the complex structure of the dynamic balance machine model, it is necessary to simplify the model, ignoring the base, driving plate, belt and motor which are not related to vibration, so that the model is simplified as the rotor and the supporting frame supported at both ends. Fig. 3 (a) is the simplified model. During the finite element analysis, 45 steel is selected according to the design requirements, with a density of 7890kg / m3, an elastic modulus of 206gpa and a Poisson's ratio of 0.269. The finite element analysis of the dynamic balance machine is carried out by using solid186 element. The X, y and Z direction translational constraints are applied at the base of the supporting frame. The finite element model after mesh generation is shown in Fig. 3 (b).

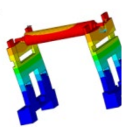
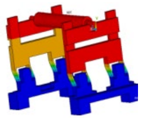
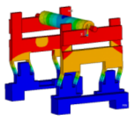
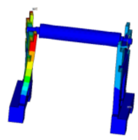


(a) After simplification (b) Mesh generation

Fig.3 The modal of dynamic balance machine

Through the finite element modal analysis of the designed horizontal hard support dynamic balance machine, the first four frequencies and natural mode shapes of the dynamic balance machine are obtained as shown in Table 1.

Table 1 The results of modal analysis

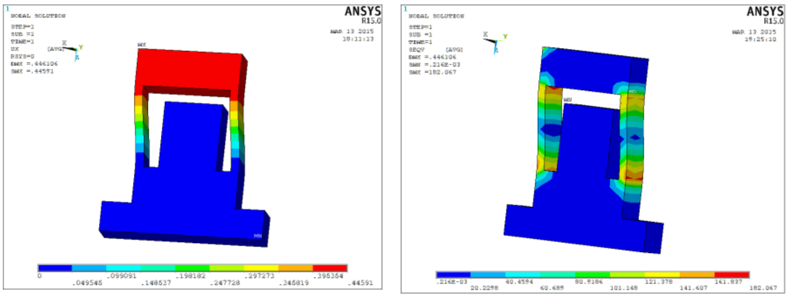
Order	1	2	3	4
Frequency	58.93Hz	104.39Hz	127.18Hz	247.69Hz
Mode shape				

4.2 Analysis and calculation of the rigidity of the supporting swing frame

(1) Stiffness calculation and frequency ratio analysis

The model of dynamic balance machine is simplified, and the supporting plate model of 3D modeling design is imported into ANSYS, and the material parameter setting is the same as above. Constrain all degrees of freedom of the lower surface of the vertical plate, observe the deformation of the vertical plate model under the horizontal force $f = 5000N$, as shown in Fig. 4 is the deformation and stress diagram of the vertical plate. Under the working condition, the horizontal displacement is 0.446106mm, so the single vertical plate stiffness $k = 1.12 \times 10^7 N/m$.

After determining the mass of the rotor system, the natural frequency ω_0 of the rotor system can be obtained by the formula $k = M * \omega_0^2 / 4$, so the total horizontal stiffness of the support system can be obtained as $K = 4.48 \times 10^7 N / m$.



（a）Deformation （b）Stress

Fig.4 Deformation and stress under horizontal force 5000N

According to the relationship between the stiffness and the natural frequency, the natural frequency of the rotor system can be calculated, $\omega_0 = \sqrt{K/M}$, in which the calculation of the natural frequency of the system also requires a variable, that is M, the mass of the rotor system. During the design of dynamic balance machine, according to the design requirements and working conditions of dynamic balance machine, the bearing range of dynamic balance machine is 10kg-200kg.

In the design process of the hard supported dynamic balancer, it is required that the working frequency ω of the balancer should be far less than its natural frequency, i.e. $0.1\omega_0 \le \omega \le 0.3\omega_0$ is required. Therefore, after calculating the natural frequency of the rotor system, the range of the working frequency of the rotor can be calculated which required by the change of the load of the balancer, as shown in Table 2 below.

Table 2 Frequency analysis under different mass

M(Kg)	10	50	80	110	140	170	200
Natural Frequency	337.0	150.7	119.0	101.6	90.1	81.8	75.4
ω_{min} (Hz)	33.70	15.07	11.90	10.16	9.01	8.18	7.54
ω_{max} (Hz)	101.10	45.21	35.70	30.48	27.03	24.54	22.62

Based on the above analysis and discussion, when balancing the rotors with different masses, it is necessary to select the appropriate working speed. It can be seen that in order to ensure the vibration stability and accuracy of the supporting frame of the dynamic balance machine with hard support, the working speed of the rotor should be correspondingly reduced under the condition of the rotor with large balance mass. And with the increase of the tested rotor mass, the selection range of the working speed becomes smaller.

(2) Parameter calculation of balancing machine

According to the finite element analysis, the total stiffness of the system is $K = 4.48 \times 10^7 N/m$. During the test, the diameter of the unbalanced rotor is 100mm. According to the calculation, the total mass of the rotor system is $150kg$. If the required working speed of the designed dynamic balancer is less than $900r/\min$, then the working frequency $\omega = 900/60 = 15Hz$. Therefore, only if the natural frequency $\omega_0 \geq 15/0.3 = 50Hz$ of the rotor system is met, can the design of the hard supporting system $\omega \leq 0.3\omega_0$ be guaranteed, which meets the design requirements.

$$\omega_0 = \sqrt{\frac{K}{M}} = 546.5rad/s \tag{3}$$

The natural frequency of the corresponding system is 87Hz, which is greater than 50Hz, so the design of the supporting frame of the balancing machine is reasonable.

$$\frac{\omega}{\omega_0} = \frac{900*pi/30}{546.5} = 0.17 \leq 0.3 \tag{4}$$

In the design process of the hard supporting frame of the dynamic balance machine, the ratio of the inertial force of the supporting frame vibration and the centrifugal force generated by the unbalance of the rotor must not be higher than a given error rate. According to the calculated frequency ratio of 0.17, it can be seen from the frequency ratio error table (Table 3) that the error rate ξ = 0.03 of the designed dynamic balance machine, which can meet the design requirements.

Table 3 Frequency ratio with error correspondence

ξ	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
ω/ω_0	0.1	0.14	0.17	0.196	0.22	0.24	0.26	0.27

(3) Load and speed range of balancing machine

By substituting the conversion formula $\omega = \frac{2\pi n}{60}$ and $K = M\omega_0^2$ of angular frequency into $\frac{\omega}{\omega_0} \leq \sqrt{\frac{\xi}{1+\xi}}$, we can know that

$$Mn^2 \leq \frac{900K\xi}{\pi^2(1+\xi)},\ M \leq \frac{900K\xi}{\pi^2 n^2(1+\xi)},\ n^2 \leq \frac{900K\xi}{\pi^2 M(1+\xi)} \tag{5}$$

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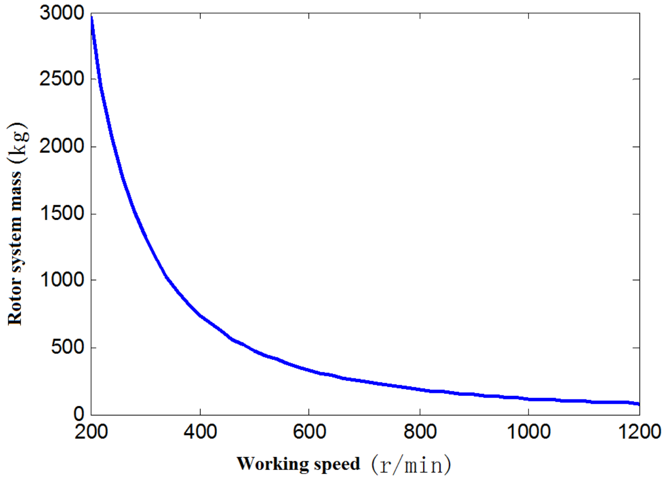


Fig.5 Relations between speed and mass

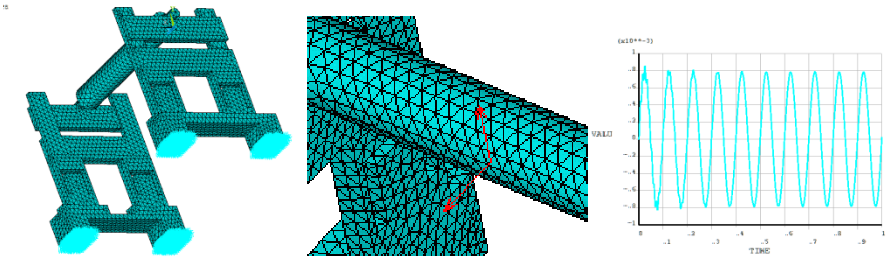
4.3 Vibration response analysis of dynamic balance machine system

The vibration response analysis of mechanical structure or system is to solve the dynamic problem of mechanical system considering the existence of external excitation. Based on the dynamic model of the system, the dynamic equation is listed to solve the displacement, velocity or acceleration response as a function of time. For the designed dynamic balance machine system, when there is unbalance in the rotor, the centrifugal force will be generated along with the unbalance of the rotor. In the working process, the centrifugal force is regarded as an external excitation. The vibration response analysis of the dynamic balance machine is to solve the vibration problem of the dynamic balance machine system caused by the unbalanced centrifugal force.

According to the vibration model of the support system as shown in Fig. 2, it can be seen from formula (2) that in the design of the hard support dynamic balance machine, the displacement response $x_0 = \frac{mr\omega^2}{K}$, where the rigidity $\boldsymbol{K}$ of the support swing system is known, $K = 4.48 \times 10^7 N/m$,assuming that there is unbalance $10000 g \cdot mm$ in the rotor, when the balancing speed $600 r/\min$, the displacement response of the dynamic balancing system with time can be obtained by theoretical calculation. Substitute each parameter into

$$x_0 = \frac{10000 \times 10^{-6} \times (2\pi f)^2}{4.48e^7} = \frac{10^{-2} \times 62.8^2}{4.48 \times 10^7} = 8.7 \times 10^{-7} m \tag{6}$$

Based on the finite element model of dynamic balance machine, the vibration response of dynamic balance machine system is considered. The centrifugal force generated by the unbalance of the rotor is equivalent to the two component forces along the X and Y directions applied at a point of the rotor, the magnitude of which varies with time, $1e^{-2}\omega^2\sin(\omega t)$ and $1e^{-2}\omega^2\cos(\omega t)$ respectively. Here, the working speed of the dynamic balance machine corresponding to the theoretical calculation is 600 r/min, so the angular speed is $\omega = 62.8rad/s$. as shown in Fig. 6, the base is fully constrained, and the unbalanced centrifugal force is applied to the rotor.



(a) Constraint (b) loading (c) vibration displacement response

Fig.6 Vibration response analysis

As shown in Fig. 6 (c), the vibration displacement response of the dynamic balance machine system is obtained through the vibration response analysis of the modal superposition method. It can be seen that the vibration of the supporting frame caused by the unbalance mass is a regular sine wave with the vibration amplitude of $8.5\times10^{-7}m$, which is little different from the theoretical calculation result. It can be seen that the finite element vibration response analysis can characterize the vibration information of the rotor unbalance, and the theoretical calculation results also verify the correctness of the modal analysis of the dynamic balance machine.

5 Experimental modal analysis

In order to obtain relatively accurate data of percussion experiment, the method of single point excitation and multi-point vibration pickup is adopted in the experiment. Four acceleration sensors are installed on one side of the supporting frame of the dynamic balance machine, which are named as 1, 2, 3 and 4 vibration pickup points from top to bottom. The force hammer is used to knock around the four vibration pickup points in turn, and the sensor layout is shown in Fig.7. The frequency response curve of the percussion test data obtained by LMS acquisition system is shown in Fig. 8.

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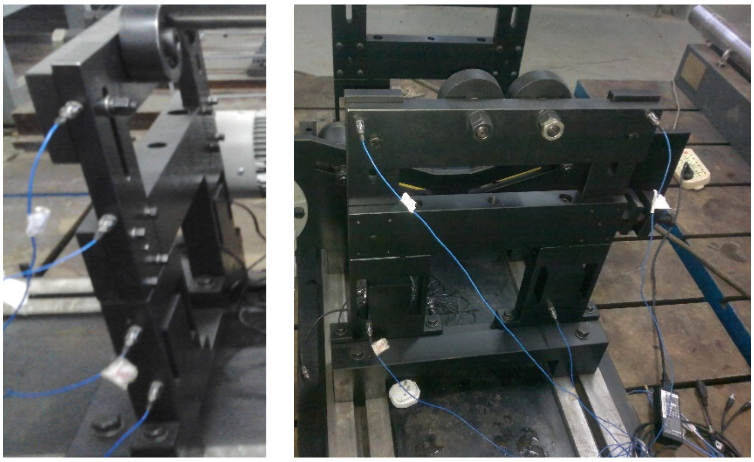

Fig.7 The sensor arrangement of hammering test

It can be seen from the Fig. that 89.84Hz has a vibration peak, while 50Hz and 100Hz peak are electric current interference, so the natural frequency of the rotor support system obtained by the experimental modal analysis is about 89.84Hz. The vibration signal data picked up near the measuring point 2 is as shown in Fig. 8, and the natural frequency is 88.2Hhz. Therefore, it can be seen that the data measured in the experiment is stable. The natural frequency of the system is 89.06Hz which taken as the average of two experimental results. The natural frequency of the rotor support system of the dynamic balance machine is much higher than the working frequency of the rotor, so the design of the dynamic balance machine is reasonable.

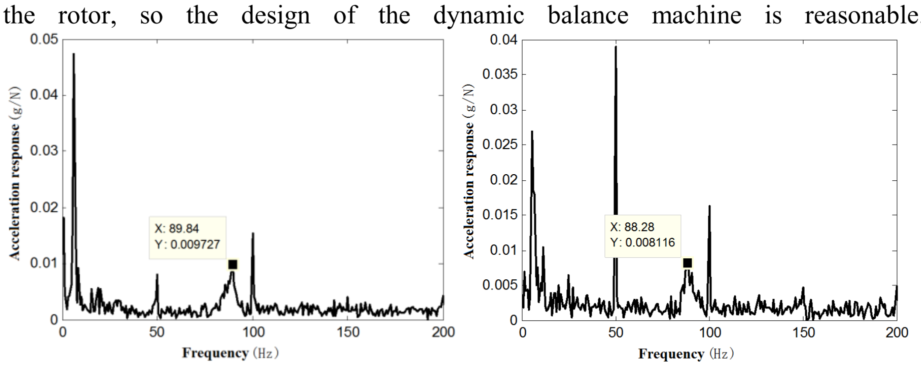

Fig.8 The frequency response of hammering point 1

The comparison between the results of finite element analysis method and experimental test is shown in Table 4. The natural frequency value obtained from the modal analysis of the whole dynamic balance machine by using the finite element method is larger than the result of the experimental modal measurement. This is because the connection stiffness between the parts of the finite element model is larger than the actual situation, resulting in the rigidity of the whole system in the simulation results becoming larger, and the natural frequency of the finite element method larger than the experimental result.

Table 4 The results contrast of modal analysis

	Modal analysis of the whole machine	**Experimental modal analysis**
Natural frequency	104.39	89.06
Deviation	15.33	
Error rate	17.2%	

6 Conclusions

Firstly, the structure of the dynamic balance machine is designed, the theoretical model of the rotor support system is built, and the dynamic characteristics of the dynamic balance machine system are analyzed based on the modal analysis theory. Secondly, the theoretical analysis results are verified by modal test, which ensures the structure rationality of the dynamic balance machine.

（1）Based on the vibration model of the support system, the working principle of the hard support dynamic balance machine is deduced. When the rigidity of the hard support frame is large enough, the vibration displacement of the frame is proportional to the centrifugal force produced by the rotor unbalance.

(2) The dynamic characteristics of the dynamic balance machine system are analyzed. The natural frequency of the rotor support system is obtained by using the finite element modal analysis and the modal test of the dynamic balancing machine. Based on the design theory of the hard support balancing machine, the natural frequencies of the balancing machine system obtained by two kinds of finite element analysis methods are far greater than the working frequencies of the rotor, which meet the design requirements of the hard support balancing machine $\omega \leq 0.3\omega_0$. The second and third modes of the dynamic balance machine are the left and right swing modes of the supporting frame, which are the modes concerned in the dynamic balancing test. In the case of a certain system stiffness, with the increase of the load of the balancing machine, the working frequency of the rotor decreases.

(3) The difference between the vibration displacement response of the dynamic balance machine system simulated by the finite element software and the theoretical calculation result is very small, which verifies the correctness of the finite element modeling analysis.

(4) The simulation results are verified by the experimental modal analysis method, and the experimental values are obtained by the method of single point excitation and multi-point vibration pickup. The results obtained by the finite element analysis method are close to the experimental values, which verifies the rationality of the structural design of the dynamic balance machine.

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