Modal analysis of AC quadrupole magnet system for CSNS/RCS

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Abstract: The quadrupole magnet of the China Spallation Neutron Source (CSNS) Rapid-cycling Synchrotron (RCS) is operated at a 25 Hz sinusoidal alternating current which causes severe vibration. The vibration will influence the long-term safety and reliable operation of the quadrupole magnet. By taking the quadrupole magnet and girder as specific model system, a method for analyzing and studying the dynamic characteristic of the system is put forward by combining theoretical calculation with experimental testing. The theoretical modal analysis results coincide with the experimental testing results. It shows that the dynamic characteristic parameters of the structure can be obtained by modal analysis which will provide a theoretical basis for the further study and the magnet girder optimal design of CSNS/RCS.

Key words: quadrupole magnet, girder, modal analysis, testing modal

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1 Introduction

The CSNS accelerators consist of an 80 MeV H^- linac and a rapid cycling synchrotron of 1.6 GeV [1, 2]. The RCS ring is a four-folded symmetrical topological structure. There are 48 sets of quadrupole magnets distributed in the whole RCS ring, and the magnets will be operated at a 25 Hz rate sinusoidal alternating current. The magnetic core and coils made severe vibration especially at the frequency 25 Hz through the vibration testing of the CSNS quadrupole magnet at the current of 915DC add 652AC，and the testing project and results are shown in Fig. 1. The maximum amplitude is 4.107 μm at the vertical direction (y). The main amplitude is at 25 Hz，and the frequency doubling of 50 Hz and 75 Hz is much little than the exciting frequency. At the same time the vibration influences other equipment through the magnetic measurement girder.

Quadrupole magnet girder system with complex structure and high-precision adjustment is one of the most important equipment of the CSNS/RCS. Because of the self-excited vibration, the comprehensive technical index of requirement is different from other accelerators which vibration was caused by the ground vibration. So

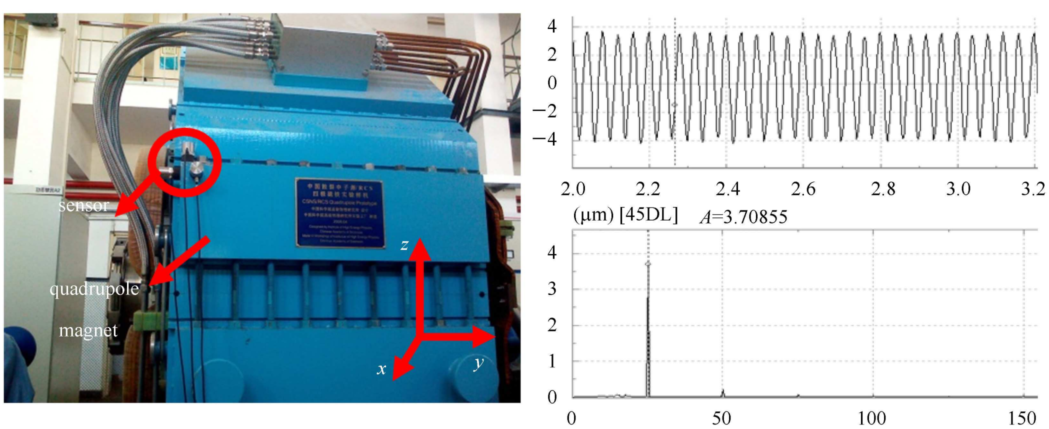

Fig. 1. Quadrupole magnet vibration test of CSNS/RCS.

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it is necessary to study the dynamic characteristic and reduce the vibration of the system [3, 4]. The theoretical modal analysis and testing modal analysis are the main research methods. The theoretical modal analysis is based on the liner vibration theory and finite element method to research the relationship among the excitation, system and response. Domestic and foreign scholars obtained many achievements by the theoretical modal analysis. The testing modal analysis uses the input and response parameters to obtain the modal parameters (frequency, damping ratio and vibration mode) [5]. The dynamic characteristic of the girder is very important. This paper adopts the quadrupole magnet & magnetic measurement girder system as the research object. The theoretical and testing methods are used to study the dynamic characteristic of the system.

2 Theoretical modal analysis

2.1 Theoretical modal theory

The system suffers from the vibrating force which comes from the magnet. The whole structure is a multi-degree-of-freedom system, and the vibration differential equation can be expressed in the formula below,

$$\boldsymbol{M}\boldsymbol{X}''+\boldsymbol{C}\boldsymbol{X}'+\boldsymbol{K}\boldsymbol{X}=\boldsymbol{F}(t), \tag{1}$$

where $\boldsymbol{M}$ is the system mass matrix, $\boldsymbol{C}$ and $\boldsymbol{K}$ are the damping and stiffness matrix, $\boldsymbol{X}''$ is the system acceleration matrix, $\boldsymbol{X}'$ and $\boldsymbol{X}$ are the velocity and displacement matrix. $\boldsymbol{F}(t)$ is the vibrating force matrix of quadrupole magnet.

The response of the whole system can be regard as the superposition of the natural frequency and vibration mode parameters in the state of non-damping free vibration. The nonzero solution condition of the constant-coefficient-linear-homogeneous differential of Eq. (1) is

$$|\boldsymbol{K}-\omega^2\boldsymbol{M}|=0. \tag{2}$$

The natural frequency ω_i^2 and main vibration mode $\{\varphi_i\}$ can be obtained from Eq. (2), where i=1, 2,$\cdots$, n.

2.2 Finite element modal analysis (FEMA)

The finite element structure (FE) consists of quadrupole magnet, magnetic measurement girder and one-layer rubber plate. The quadrupole magnet is composed of silicon steel sheets, steel plate and coil; the girder is composed of steel plate and adjusting mechanism (AM). Considering the FE accuracy and computational cost, some measures are used to simplify the FE. The structure physical properties are listed in Table 1.

Table 1. Parameters of material.

name	Young's modulus/Pa	poisson's rate	density/ (kg/mm^3)
A3 steel	2.09E11	0.269	7890
silicon steel sheet	1.97E11	0.26	7650
rubber sheet	2E7	0.35	2000

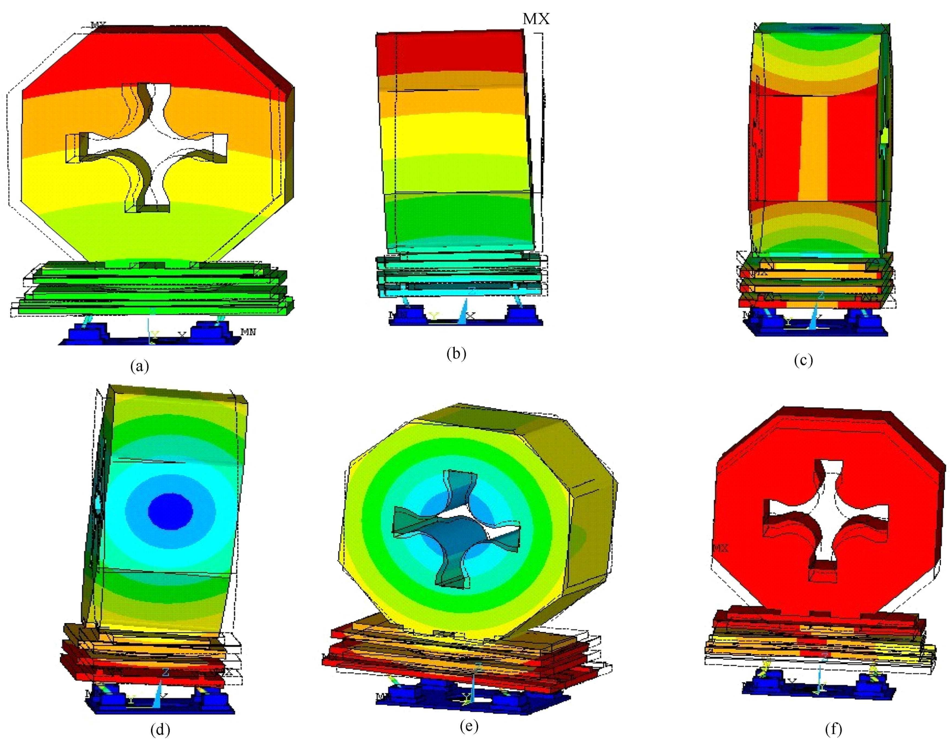

Fig. 2. (a)–(f) are respectively the top 6 rank modal shapes.

Table 2. The top 6 step natural frequency.

modal order i	natural frequency f/Hz	modal shape
1	4.591	x direction bend
2	5.116	z direction bend
3	7.662	Rotate at y axis
4	19.519	Rotate at x axis
5	20.954	Rotate at z axis
6	27.034	up-down translational motion at x-z plane

3 Testing modal analysis

3.1 Testing modal theory

Testing modal analysis can obtain the dynamic performance parameters of the system with the curve fitting analyses of the transfer function of the structure's excitation and response (such as acceleration, velocity, displacement, etc.). At the assumption of the zero initial state of system, Eq. (1) is Fourier transformed. And the frequency response function can be obtained based on the orthogonality condition of the real symmetric matrix [6].

$$H_{ij}(\omega)=\sum_{r=1}^{n}\frac{\phi_{ir}\phi_{jr}}{(k_r-\omega^2 m_r)+\mathrm{j}\omega c_r}=\sum_{r=1}^{n}\frac{\phi_{ir}\phi_{jr}}{k_r(1-\lambda_r^3+\mathrm{j}2\zeta_r\lambda_r)}, \tag{3}$$

where $\lambda_r=\omega/\omega_r$, $\omega_r=(k_r/m_r)^{0.5}$, $\zeta_r=c_r/(2m_r\omega_r)$, m_r is the r step modal mass, k_r and c_r are the r step modal stiffness and modal damping, ζ_r is the r step modal damping ratio, ω_r and ϕ_r are the r step natural frequency and main modal shape vector.

In testing modal, the transfer function can be calculated from the exciting point and detecting point parameters. The different order modal parameters can be calculated from any one row or one column elements.

3.2 Testing modal measurement

In this paper, the testing scheme is based on the theoretical modal analysis results of ANSYS. The natural frequency distribution range of the system is estimated. This test takes force hammer excitation system. The modal parameters identification method of MIMO is taken too. There are 72 measuring points arranged around the whole system according to the selecting principle, 48 points arranged on the magnet to measure the x, y and z direction acceleration of the 16 corner points, and 24 points arranged on the girder to measure the three direction acceleration of the first and third plate's corner points. The testing system and the acceleration sensor arrangement are shown in Fig. 2.

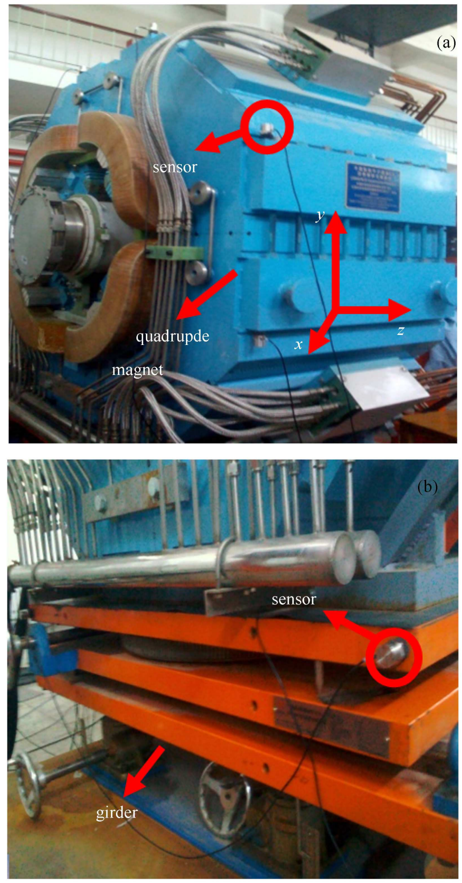

Fig. 3. (a), (b) are the layout of the experimental modal testing.

The force hammer and vibration response signals are acquired by the intelligent analyzer of INV3032C. After testing, the system testing modal parameters (natural frequency and damping ratio) have been got through the data processing analysis system of DASP. The results of the testing are shown in Table 3.

The modal assurance criterion (MAC) is used to estimate the correctness of different mode shape [7].

$$\mathrm{MAC}(\{\Psi\}_r,\{\Psi\}_s)=\frac{\left|\{\Psi\}_r^{*\mathrm{T}}\{\Psi\}_s\right|^2}{\left(\{\Psi\}_r^{*\mathrm{T}}\{\Psi\}_r\right)\left(\{\Psi\}_s^{*\mathrm{T}}\{\Psi\}_s\right)}. \tag{4}$$

where Ψ is the mode shape vector. The MAC matrix is one of the important estimate methods in modal parameter identification. The same physical modal MAC value should be close to 1, and the different physical modal

Table 3. The natural frequency and damping ratio of the modal testing.

modal order	1	2	3	4	5	6
natural frequency f/Hz	4.501	5.572	6.628	17.383	22.247	26.502
damping ratio(%)	4.674	4.949	8.262	3.943	3.742	3.177

MAC value should be very small. Fig. 4 is the MAC matrix of the quadrupole magnet system which satisfies the theory of testing modal analysis. It shows that the experimental modal testing results are correct.

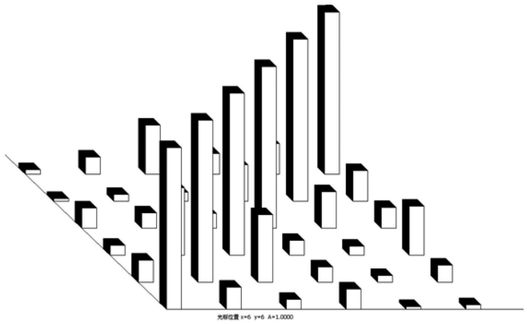

Fig. 4. The MAC matrix of modal testing.

4 Theoretical and experimental analyses

Figure 5 shows the top 6 step natural frequencies comparison graph of testing and theoretical calculation (FEMA). The theoretical calculation results are almost identical with the test results, which indicates the modal analysis of the structure and the FE of the system is reasonable.

Figure 5 also shows that the minimum natural frequency is 4.501 Hz, the maximum is 26.502 Hz, and the distribution of frequency is relatively concentrated. The sixth frequency of the system is close to the exciting frequency (25 Hz), so the girder isn't suitable for the quadrupole, the girder should be optimized. And the top 6 step frequencies are very low, which is bad for the stiffness of system. The frequency and modal shape are related to the mass distribution after in-depth study. The mass of the experimental object mainly concentrates at the top of the structure, and the bottom of the structure is AM, which has less mass than the other parts. It is found that AM leads to low stiffness of the system through ANSYS simulation. Because of this problem, there are some measures taken such as the optimizing of the girder structure to make the natural frequency far away from the exciting frequency (25 Hz); the adoption of effective isolation method to reduce the influence of quadrupole magnet vibration to other equipment; changing the position of the adjusting mechanism and increasing some auxiliary support structure to increase the stiffness of the girder [8].

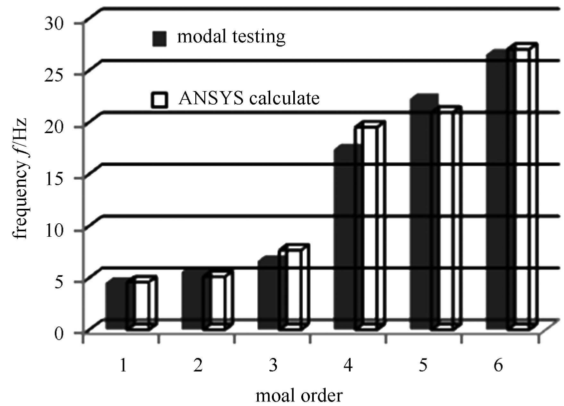

Fig. 5. The frequencies contrast between FEA and testing.

5 Conclusions

The quadrupole magnet and girder play a very important role in the accelerator of CSNS/RCS, so studying the vibration of the system is necessary. This paper establishes the suitable finite element structure of the magnet girder system, and uses ANSYS to simulate the vibration mode of the system and expounds the principle and method of the testing modal. The results provide a theoretical basis for the CSNS/RCS quadrupole magnet girder structure optimization design. The resonance phenomenon can be avoided and the structure vibration resistance can be improved before manufacture. So this paper can provide a reasonable way to design the equipment which has self-excited vibration, such as dipole, quadrupole magnet and superconductor cavity system of CSNS, etc.

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