Modal analysis of CSNS/RCS dipole magnet and magnetic measurement girder

LIU Renhong* ZHANG Junsong QU Huamin WANG Haijing

Institute of High Energy Physics, Chinese Academy of Sciences, Beijing 100049, China

Abstract The dipole magnet of the China Spallation Neutron Source (CSNS) Rapid-cycling Synchrotron (RCS) will be 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 dipole magnet. By taking the dipole magnet and magnetic measurement 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. This paper established the mechanical model of the system, and the top six step natural frequency and vibration mode were obtained through theoretical modal analysis (ANSYS). Then according to testing modal analysis, the natural frequency, damping ratios and vibration mode of the system structure were obtained too. The theoretical modal analysis results coincide with the experimental testing results. Besides, the 6th step natural frequency is close to the exciting frequency of the magnet, so the resonance phenomenon may take place at the actual working conditions. The dynamic characteristic data of the structure can provide an analysis basis for the further study and the formal dipole magnet girder optimal design of RCS.

Key words Dipole magnet, Magnetic measurement girder, Modal analysis, Testing modal, Natural frequency

1 Introduction

The CSNS-I accelerators consist of an 80-MeV H^- linac and a rapid cycling synchrotron of 1.6 GeV[1]. The RCS ring is a four-folded symmetrical topological structure which consists of four arc zones and four line segments. There are 24 sets dipole magnets uniformly 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 report of the CSNS dipole magnet[2]. At the same time the vibration influenced other equipments through the magnetic measurement girder.

Dipole 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 than other accelerator which vibration was caused by the ground vibration. So 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 base 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 formal dipole magnet of CSNS/RCS is designing now, and the dynamic characteristic of the girder is very important. This paper adopts the dipole magnet & magnetic measurement girder system as research object. The theoretical and testing methods are used to study the dynamic characteristic of the system.

* Corresponding author. *E-mail address:* liurh@ihep.ac.cn

Received date: 2012-04-26

2 Theoretical Modal Analysis

2.1 Theoretical Modal Theory

The system was suffered the vibrating force which comes from the dipole magnet. The whole system is a multi-degree-of-freedom system, and the vibration differential equation can be expressed as Eq.(1)

$$\boldsymbol{MX}''+\boldsymbol{CX}'+\boldsymbol{KX}=\boldsymbol{F}(t) \quad (1)$$

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 dipole magnet.

The response of the whole system can be regarded as the superposition of the natural frequency and vibration mode parameters in the state of non-damping free vibration, then Eq.(1) changes into

$$\boldsymbol{MX}''+\boldsymbol{KX}=0 \quad (2)$$

The nonzero solution condition of the constant -coefficients-linear-homogeneous differential is expressed by Eq.(2)

$$|\boldsymbol{K}-\boldsymbol{\omega}^2\boldsymbol{M}|=0 \quad (3)$$

The natural frequency ω_i^2 and main vibration mode $\{\varphi_i\}$ can be obtained from equation (3), where I =1, 2,…, n.

2.2 Finite Element Modal Analysis (FEMA)

The finite element structure (FE) consists of dipole magnet, magnetic measurement girder and two-layer rubber plate. The dipole 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, such as ignore all kinds of fabrication holes and small holes, ignore some transition fillet, etc. 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}
Steel	2.09×10^{11}	0.3	7850
Silicon Steel Sheet	2×10^{11}	0.26	7650
Rubber Sheet	2×10^{7}	0.4	1200

FEMA software ANSYS will be utilized. The FE is constructed with element of Solid186. In the progress of analysis, the Block Lanczos Method is used to calculate the natural frequency and vibration mode of the system. The top 6 step modalities are obtained, as shown in Fig.1 and Table 2.

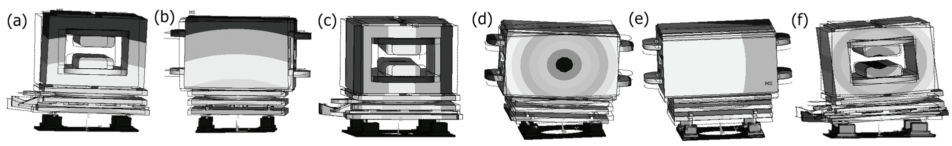

Fig.1 (a), (b), (c), (d), (e), (f) are the respectively top 6 rank modal shape, x axis is the longitudinal. Direction of the structure, y is the transverse direction and z is the vertical direction.

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). At the assumption of the zero initial state of system, the Eq.(1) is Fourier transformed. And the frequency response function can be obtained based on the orthogonality condition of the real symmetric matrix[6].

$$H_{ij}(\omega)=\sum_{r=1}^{n}\frac{\phi_{ir}\phi_{jr}}{\left(k_r-\omega^2 m_r\right)+j\omega c_r}=\sum_{r=1}^{n}\frac{\phi_{ir}\phi_{jr}}{k_r\left(1-\lambda_r^3+j2\zeta_r\lambda_r\right)} \quad (4)$$

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 *i*s the r step modal damping ratio, ω_r and $ø_r$ are the r step natural frequency and main modal shape vector.

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

Table 2 Top 6 step natural frequency.

Modal Order (i)	Natural Frequency	Modal Shape
1	4.9631	y Direction Bend
2	6.1108	x Direction Bend
3	7.8933	Rotate at z Axis
4	19.225	Rotate at y Axis
5	21.387	Rotate at x-y Plane Diagonal
6	23.506	Rotate at x Axis

3.2 Testing Modal Measurement

In this paper, the testing scheme was based on the theoretical modal analysis results of ANSYS. The natural frequency distribution range of the system is estimated that the main modal concentrates on less than 100 Hz. This test takes force hammer excitation system. The method of single-point excitation and multipoint vibration picking are taken too. The 68 measuring points are arranged around the whole system according to selecting principle, 24 points, on the dipole magnet to measure the X, Y and Z direction acceleration of the 8 corner points, and 24 points, on the girder to measure the three direction acceleration of the first and third plate's corner points. The 20 points are used to measure the acceleration of the magnet coil. The testing system and the acceleration sensor arrangement are shown in Fig.2.

The force hammer and vibration response signal are acquired by the intelligent analyzer of INV3032C. After testing, we got the system testing modal parameters (natural frequency and damping ratio) and the top 3 step mode shape through the data processing analysis system of DASP. The results of the testing are shown Fig.3 and Table3. And the Modal Assurance Criterion (MAC) was used to estimate the correctness of different mode shape.

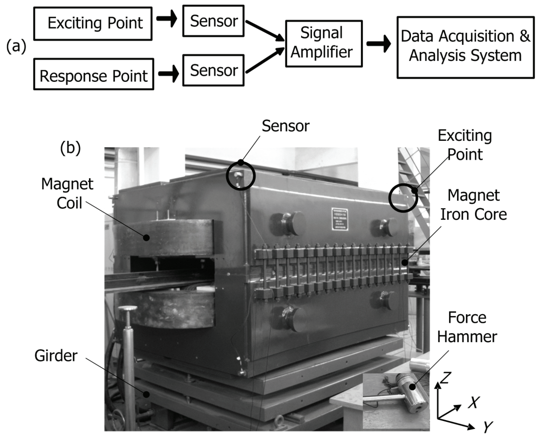

Fig.2 Experiment layout of experimental modal testing. (a) Block diagram of the testing system, (b) Layout of the testing experiment

$$MAC\left(\{\boldsymbol{\Psi}\}_r,\{\boldsymbol{\Psi}\}_s\right)=\frac{\left|\{\boldsymbol{\Psi}\}_r^{*T}\{\boldsymbol{\Psi}\}_s\right|^2}{\left(\{\boldsymbol{\Psi}\}_r^{*T}\{\boldsymbol{\Psi}\}_r\right)\left(\{\boldsymbol{\Psi}\}_s^{*T}\{\boldsymbol{\Psi}\}_s\right)} \tag{5}$$

where $\{\boldsymbol{\Psi}\}$ is the mode shape vector. The MAC value of each order modality is less than 15% from Fig.3(d) which satisfies the theory of testing modal analysis.

Table 3 Natural frequency (Hz) and damping ratio of the modal testing.

i	1	2	3	4	5	6
Natural Frequency	3.998	6.222	7.502	18.189	19.568	24.699
Damping Ratio%	5.830	3.296	0.659	1.482	4.019	0.433

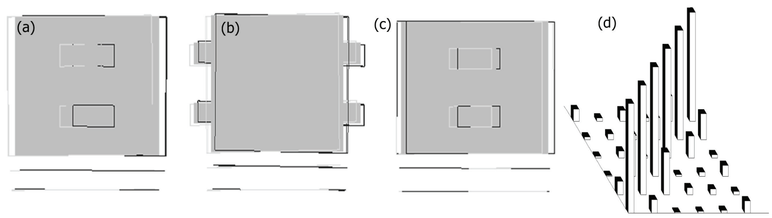

Fig.3 (a), (b) and (c) are the top 3 step modal shape, (d) is the MAC value of the testing.

4 Theory and Experimental Analysis

Figure 4 shows the top 6 step natural frequency comparison graph of testing and theoretical calculation. The theoretical calculation results are almost identical with the test results, indicating the modal analysis of the structure and the FE of system is reasonable.

Fig.4 also shows that: the minimum natural frequency is 3.998 Hz, the maximum is 24.699 Hz, and the distribution of frequency is relatively concentrated. The sixth frequency of the system is close to the exciting frequency (25 Hz) of the dipole magnet, which demonstrates why the dipole magnet vibrated severely, and leading the iron core cracked. Two factors should be considered in the future design: the optimizing of the 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 dipole magnet vibration to other equipments.

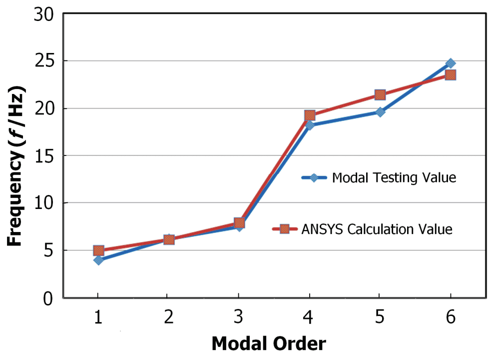

Fig.4 The frequencies contrast between FEA and testing.

The modal shape manifests the whole vibration of the structure. 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 studying. The mass of the experiment object mainly concentrates in the top of the structure, and the bottom of the structure is AM which have less mass than other parts. It is found that the AM lead to low stiffness of the system through ANSYS simulation. The structure stiffness can be improved through changing the position of the adjusting mechanism and increasing some auxiliary to support structure[7].

5 Conclusions

The dipole magnet and girder play a very important role in accelerator of CSNS/RCS. This paper established the suitable finite element structure of the dipole magnet and magnetic measurement girder, used ANSYS to simulate the vibration mode of the system and expound the principle and the method of testing modal.

The testing and simulation results show that: the natural frequency of the system is low and the sixth order frequency is close to exciting frequency which may cause resonance phenomenon in the future operation; the vibration isolator must be designed and used in the magnet-girder system; the auxiliary support structure should be used to improve the stiffness of the system. The theoretical calculation of natural frequency and ANSYS modal shape are almost identical with the testing results, the simulation method can be used to estimate the dynamic characteristic of the system before manufacture. This paper provides a theoretical basis for the CSNS/RCS formal dipole magnet girder structure design, manufacture and reconstruction. The dynamic characteristic of the system can be easily obtained by ANSYS simulation, and the resonance phenomenon can be avoided in the design phase.

Acknowledgements

The authors would like to thank other CSNS colleagues and the technicians of the China Orient Institute of Noise & Vibration.

References

1 Wei J, Fu S N, Fang S X. China spallation neutron source accelerators: design, research, and development. Proceedings of EPAC, 2006.

2 Yu C H. The amplitude-measurement of the dipole magnet report, Internal-report, 2008, IHEP.

3 Doose C, Sharma S. Investigation of passive vibration damping methods for the advanced photon source storage ring girders, MEDSI02, APS, Chicago 2002, 133–139.

4 Wang X, Yan Z B, Du H W. Nucl Sci Tech, 2012, **23:** 1–6.

5 Hermans L, Vender Auweraer H. Mechanical systems and signal processing, 1999, **13:** 193–216.

6 Ward H, Stefan L, Paul S. Modal analysis theory and testing. Beijing, Beijing Institute of Technology Press, 2001, 1–70.

7 Wang X, Yang Z B, Zhang M, *et al.* Design and testing of the girder support system for SSRF, http://www. Light-source.ca/ medsi-sri 2008/pdf/papers/ (March 25, 2012).