Vibration research of the AC dipole-girder system for CSNS/RCS

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Abstract: The China Spallation Neutron Source (CSNS) is a high intensity proton accelerator based facility. Its accelerator complex includes two main parts: an H^- linac and a rapid cycling synchrotron (RCS). The RCS accumulates the 80 MeV proton beam and accelerates it to 1.6 GeV, with a repetition rate of 25 Hz. The AC dipole of the CSNS/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 magnet. The dipole magnet of CSNS/RCS is an active vibration equipment, which is different from the ground vibration accelerator. It is very important to design and study the dynamic characteristics of the dipole-girder system. This paper takes the AC dipole and girder as a specific model system. A method for studying the dynamic characteristics of the system is put forward by combining theoretical calculation with experimental testing. The ANSYS simulation method plays a very important role in the girder structure design stage. With this method, the mechanical resonance phenomenon was avoided in the girder design time. At the same time the dipole vibratory force will influence the other equipment through the girder. Since it is necessary to isolate and decrease the dipole vibration, a new isolator was designed to isolate the vibratory force and decrease the vibration amplitude of the magnet.

Key words: AC dipole, girder, vibration, modal analysis, testing modal, vibration isolation

PACS: 29.25.Dz, 46.40.-f **DOI:** 10.1088/1674-1137/38/6/067005

1 Introduction

The China Spallation Neutron Source (CSNS-Ⅰ) accelerators consist of an 80 MeV H^- linac and a rapid cycling synchrotron of 1.6 GeV [1, 2]. The rapid cycling synchrotron (RCS) ring is a four-folded symmetrical topological structure that consists of four arc zones and four line segments. There are 24 sets of 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 can cause severe vibration, especially at a frequency of 25 Hz, such as in the J-PARC AC dipole magnets. The vibration may influence other equipment through the magnet girder system.

The AC dipole-girder system, with a complex structure and high precision adjustment, is one of the most important pieces of equipment of the CSNS/RCS. Because of the self-excited vibration, the comprehensive technical index of its requirement is different from other accelerators whose vibration is caused by ground vibration. Consequently, it is necessary to study the dynamic characteristics 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 study the relationship among the excitation, system and response. Domestic and foreign scholars have obtained many achievements thanks to the use of 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 characteristics of the girder are very important. This paper adopts the AC dipole-girder system as the research object. Both theoretical and testing methods are used to study the dynamic characteristics of the system. A new isolator was then designed to improve the dynamic characteristics of the system, decrease the vibratory force, and avoid the resonance phenomenon.

2 Vibration testing of the AC dipole

The CSNS/RCS old dipole magnet is placed on the magnetic measurement girder, and the dipole is operated at a 25 Hz sinusoidal alternating current of 1100 DC with

Received 15 July 2013
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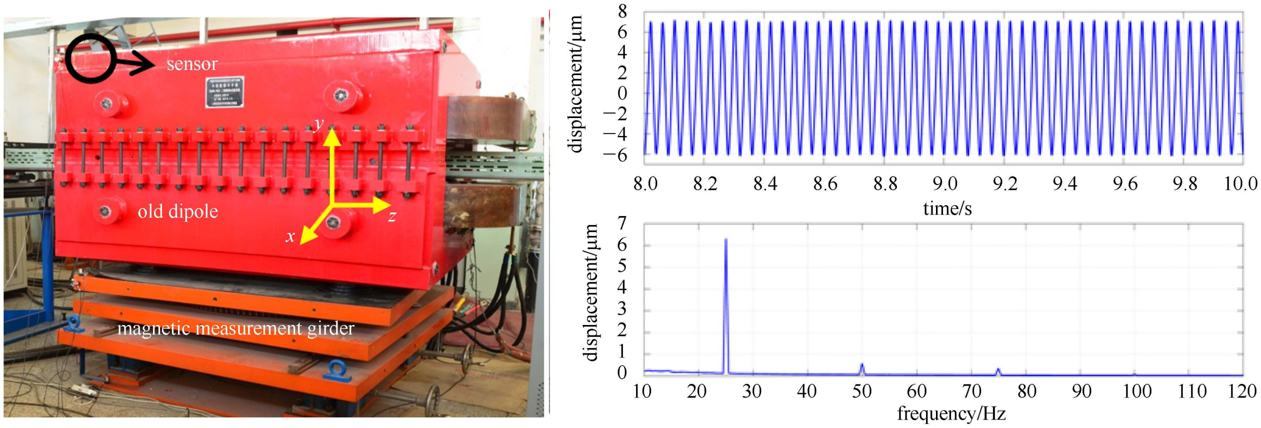

Fig. 1. The old dipole vibration testing of the CSNS/RCS.

816 AC, which causes severe vibration. An acceleration sensor will be used to measure the vibration of the dipole. The testing project and results are shown in Fig. 1. The maximum amplitude is 7.16 μm at the vertical direction (y). The main amplitude frequency is 25 Hz, and the maximum amplitude is 6.31 μm. The amplitude frequency doubling of 25 Hz is much lower than the exciting frequency. At the same time, the vibration influences the other equipment through the magnetic measurement girder. The iron core and coil of the old CSNS/RCS dipole cracked in the magnetic measurement phase. The sixth natural frequency of the dipole-magnetic measurement girder system is 24.75 Hz, which is quite close to the exciting frequency of 25 Hz [6]. The sixth natural frequency may amplify the vibration amplitude of the dipole. So, the final girder of the AC dipole must avoid resonance phenomenon through structural design or the use of a vibration isolator.

3 Modal analysis of the dipole-girder system

3.1 Theoretical modal analysis simulation

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

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

where $\boldsymbol{M}$ is the system mass matrix, $\boldsymbol{C}$ and $\boldsymbol{K}$ are the damping and stiffness matrices, $\boldsymbol{X}''$ 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. The nonzero solution condition of the constant-coefficient-linear-homogeneous differential of Eq. (1) is

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

The natural frequency ω_i^2 and main vibration mode $\{\varphi_i\}$ can be obtained from Eq. (2), where i=1, 2,$\cdots$, n. The finite element calculation is the main method to simulate the dynamic characteristic of the structure. This paper uses ANSYS software to simulate the AC dipole-girder system of CSNS/RCS. The dipole girder was designed with split type structure to increase the stiffness and decrease the adjustment coupling of the girder. The main structure of the girder is composed of the reinforcing steel plate and three direction adjustment system, as shown in Fig. 2.

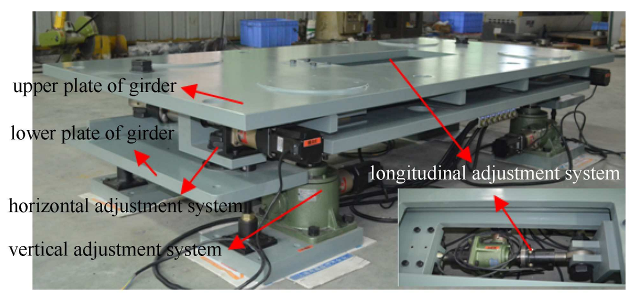

Fig. 2. The AC dipole-girder structure of CSNS/RCS.

The finite element structure (FE) consists of a new dipole, girder system. The dipole magnet is composed of silicon steel sheets, steel plate, stainless plate and aluminum coil. The girder is composed of steel plate and an adjusting mechanism. Considering the FE accuracy and computational cost, some measures are used to simplify the FE. The structure's physical properties are listed in Table 1.

The FE is constructed with elements of solid 186. In the progress of analysis, the Block Lanczos Method is used to calculate the natural frequency and vibration mode of the system. The top six step modalities are obtained, and the results are shown in Fig. 3 and Table 2.

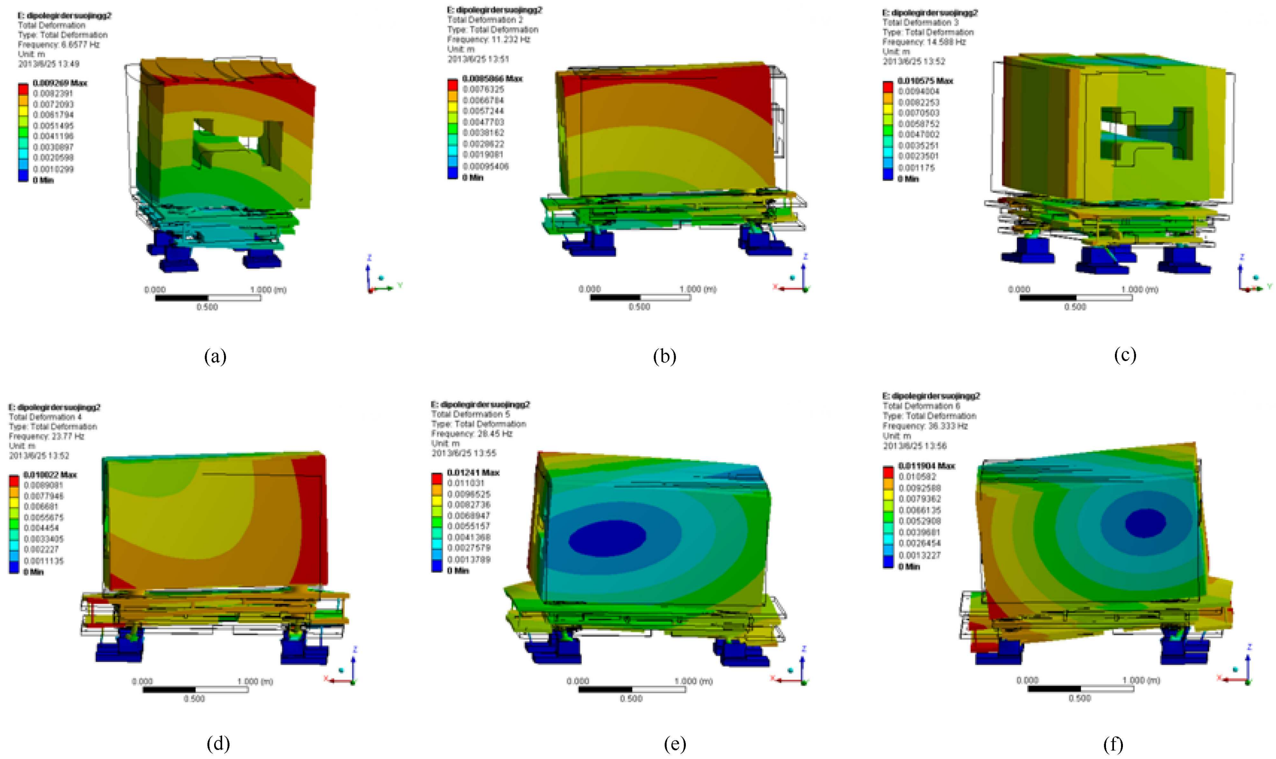

Fig. 3. (color online) (a), (b), (c), (d), (e) and (f) are respectively the top six rank modal shapes of the dipole-girder system.

Table 1. Parameters of material.

material	young's modulus/Pa	poisson's ratio	density/ (kg·m^{-3})
Q345D steel	2.09E11	0.269	7890
silicon steel sheet	1.97E11	0.26	7650
aluminum	6.9E10	0.34	2700
stainless steel	1.93E11	0.31	7750

Table 2. The top six step natural frequency.

modal order i	natural frequency f/Hz	modal shape
1	6.658	x direction bend
2	11.23	z direction bend
3	14.59	rotate at y axis
4	23.77	up-down vertical motion at x-z plane
5	28.45	pitch motion at one diagonal
6	36.33	pitch motion at the other one diagonal

3.2 Testing modal analysis

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.). With the assumption of the zero initial state of system, Eq. (1) is Fourier transformed. The frequency response function can be obtained based on the orthogonality condition of the real symmetric matrix [7].

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

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

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 the force hammer excitation system. The modal parameters identification method of MIMO is also taken. There are 60 measuring points arranged around the whole system according to the selecting principle. A total of 24 points are arranged on the dipole to measure the x, y and z direction acceleration of the 12 corner points, and 36 points are 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. 4.

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

modal order	1	2	3	4	5	6
natural frequency f/Hz	8.66	11.09	15.33	23.21	28.44	36.22
damping ratio(%)	1.772	2.602	2.647	5.519	2.786	3.922

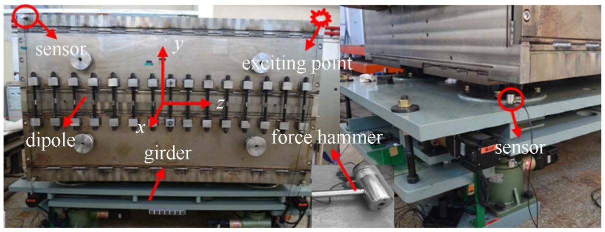

Fig. 4. The experiment layout of the testing modal.

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 measured 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 shapes [8].

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

where $\boldsymbol{\Psi}$ 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 MAC value should be very small. Fig. 5 is the MAC matrix of the dipole-girder system, which satisfies the theory of testing modal analysis. It shows that the experimental modal testing results are correct.

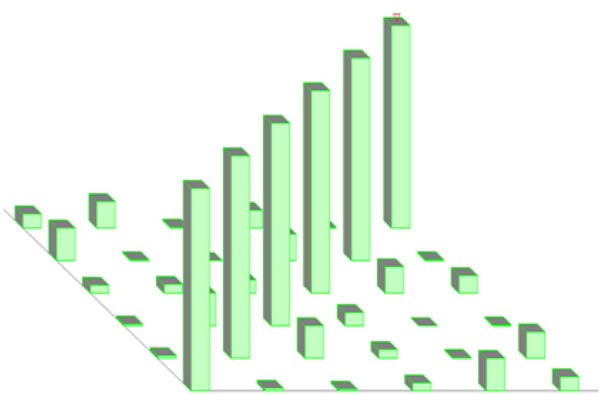

Fig. 5. The MAC matrix of modal testing.

Figure 6 shows the natural frequencies contrast between simulations and testing. The simulation results are almost identical with the testing results, which indicates the modal analysis of the structure and the FE of the system is reasonable. According to these results, the resonance phenomenon can be avoided at the design stage through ANSYS simulation if the FE is suitable and correct. There is not a natural frequency close to the exciting frequency, so the system will not amplify the vibration amplitude of the dipole. At the same time, with the improvement of the new dipole manufacturing technique, the new dipole vibration amplitude is decreased to 4.42 μm, as shown in Fig. 7. The dipole vibration can be improved by a suitable dynamic characteristic design.

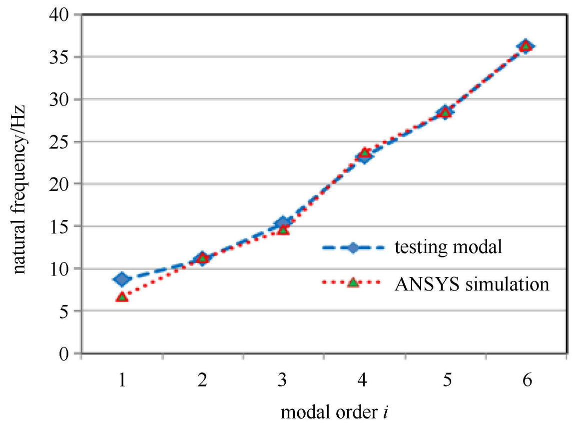

Fig. 6. The natural frequencies contrast.

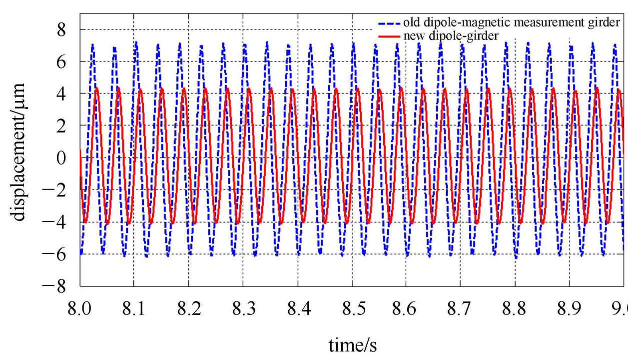

Fig. 7. The vibration amplitude contrast of the dipole.

4 Vibration testing with a vibration isolator

The dipole vibratory force still influences the other equipment in long time operation, although the AC dipole-girder system's dynamic characteristics have been changed and the vibration amplitude is less than the old dipole-magnetic measurement girder system. To improve the dynamic characteristics and decrease the dipole vibratory force influence, this paper fixes four vibration isolators between the girder and the dipole. The mass

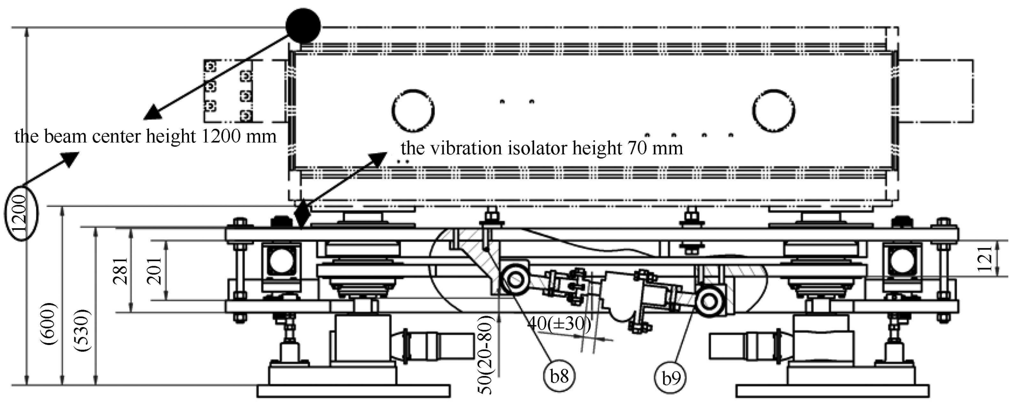

Fig. 8. The size of the beam and vibration isolator.

of the dipole is 24.5 t and the center beam height is 1200 mm. There is only 70 mm height for the vibration isolator design, which is shown in Fig. 8. The vibration isolator will have fatigue resistance, creep resistance (<0.1 mm per year), and irradiation resistance. These factors make the design of the vibration isolator a bigger challenge.

In this paper a new absorber is designed for the CSNS/RCS AC dipole with special material. Then, the acceleration sensor will be used to measure the vibration of the dipole. With vibration isolation, the testing project has the same testing points as Fig. 1. The maximum amplitude is 1.81 μm in the vertical direction (y), which is shown in Fig. 9.

From Fig. 10, the main vibration is not at 25 Hz with vibration isolator. The frequency doubling of 25 Hz also has the greatest contribution to the dipole vibration. The dynamic response of the AC dipole has been changed with the vibration isolator. The exciting vibration (25 Hz) of the dipole was reduced, so the dipole vibration amplitude with isolator is decreased 60% compared the vibration amplitude without the isolator.

At the same time, the acceleration of the vibration isolator has also been measured, which was used to indicate the vibratory force transmission efficiency decrease. The up sensor acceleration value is used to indicate the vibratory force of the dipole, and the down sensor value is used to indicate the vibratory force after it is weakened. From Fig. 11, the up sensor maximum value is 0.17 m/s^2 and the down sensor maximum value is 0.057 m/s^2, so the vibratory force vibration isolation efficiency of the isolator is 66.2%.

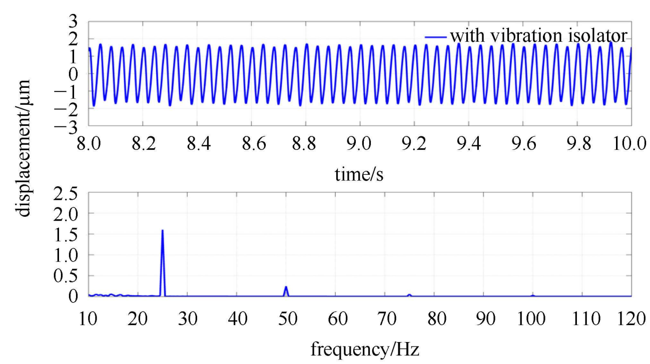

Fig. 9. The vibration amplitude (y direction) with isolator of dipole.

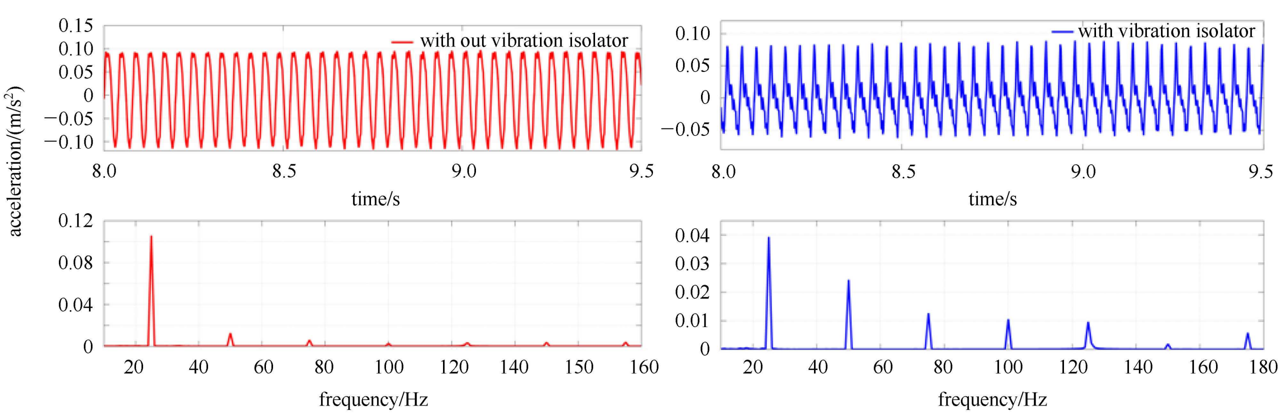

Fig. 10. The acceleration amplitude spectrum of dipole (y direction) comparison.

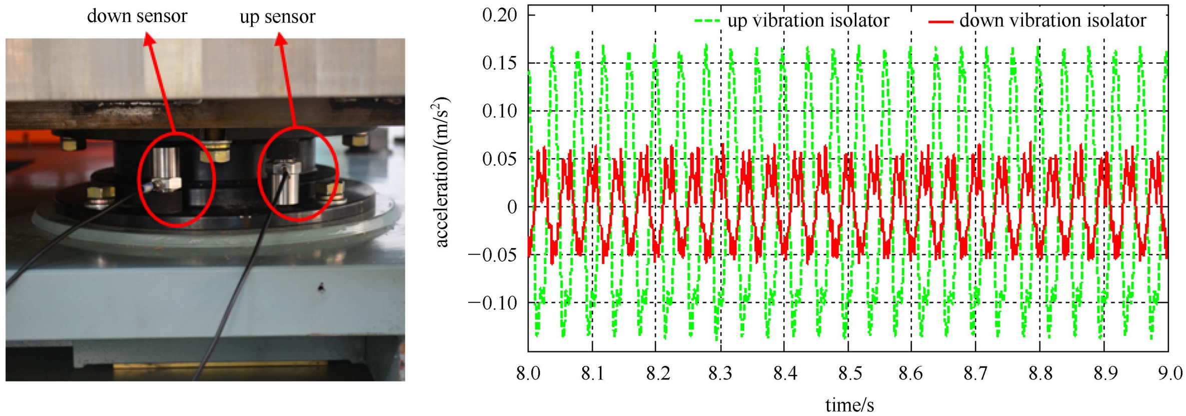

Fig. 11. The acceleration comparison between the upside-downside of the vibration isolator.

5 Conclusions

The AC dipole and girder system play a very important role in the accelerator of CSNS/RCS. Consequently, it is necessary to study the vibration of the system. This paper has established the suitable finite element structure of the magnet girder system. A method for analysing and studying the dynamic characteristic of the system is put forward by combining theoretical calculation (ANSYS simulation) with experimental testing. Consequently, the resonance phenomenon can be avoided and the structure vibration resistance can be improved before manufacture. A new isolator was designed to decrease the vibration amplitude and the vibratory force transmission of the AC dipole. With the vibration isolator, the AC dipole vibration amplitude decreased 60% and the vibratory force transmission decreased 66.2%. The active vibration of the magnet is different to passive vibration that is caused by ground vibration. Consequently, this paper can provide a reasonable way to design equipment that has self-excited vibration, such as the AC dipole, AC quadrupole [9] and superconductor cavity system of CSNS, etc.

The authors would like to thank their CSNS colleagues who kindly gave their help and advice.

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