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Vibration Damping Design and Testing Research of Space Payload Cabinet

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ABSTRACT

Space payloads which installed on spacecraft such as satellites and airships are usually experienced random vibrations and low-frequency sinusoidal vibrations during launching. In this paper, a space payload cabinet is introduced, and the damping design is carried out by applying constrained viscoelastic damping layer to the surfaces of the cabinet to ensure that the space payloads can withstand the above-mentioned mechanical environmental conditions. A reliable connection between the space payload cabinet and the shaking table is achieved through the vibration test fixture. The basic requirements for the function and design of the vibration test fixture are presented. A method for detecting the dynamic characteristics of the fixture is proposed. The vibration characteristic of the space payload cabinet is simulated with the finite element software MSC.Nastran. Then the vibration test of the cabinet is conducted and the vibration response data is acquired by using the B&K test system. The test results show that the damping effect of the space payload cabinet is obvious after applying the viscoelastic constrained damping layer.

KEY WORDS: Space payloads; viscoelastic constrained damping layer; damping; vibration characteristics; Fixture

1 INTRODUCTION

THE space payloads are the instruments, equipments or subsystems that directly perform the mission of a specific spacecraft. It can generally be used for scientific detection experiment, information acquisition and transmission. The nature and function of the spacecraft are mainly determined by the space payloads. The space missions are accomplished through spacecraft, and the effective output of the spacecraft mainly depends on the output of the space payloads. It can be said that the space payloads are the core of the spacecraft and play a leading role in spacecraft design. With the development of large and complex spacecrafts, more and more space payloads have been launched to carry out corresponding space missions. In order to ensure that the space payloads can carry out the space missions normally in its all life cycle, it is significant to provide necessary support and guarantee to the payloads. Otherwise, the payloads cannot give full play to its due role.

The space payloads are mounted on the spacecraft and will experience vibrational environment such as wide-range random vibrations and low-frequency sinusoidal vibrations during launching (Bao, 1982; Fu, Fu, Jia, 2015; Toyoshima et al., 2010; Mastroddi, 2010). The random vibrations are mainly caused by the exhaust noise of the engine, the aerodynamic noise in the transonic flight section, and the pressure fluctuation in the combustion chamber of the engine. The low-frequency sinusoidal vibrations are mainly caused by pogo vibration, low-order modal free oscillation of the missile structure caused by engine start, flameout and inter-stage separation, transverse jitter caused by shock waves in gusty and transonic flight segments, and low-order longitudinal oscillations caused by incomplete combustion of the engine (Chen, Wang, & Xue, 2017; Shen et al, 2006; Wang & Liu, 1995). The influence of the mechanical environments on the space payloads is not negligible or even fatal, which may decrease the accuracy of the optical and electrical instruments, or lead to mechanical fatigue, circuit transient short circuit, open circuit, or failure (Lin, Lv, & Wang, 2012; Meng, Zhou, & Miao, 2016; Chen et al., 2010). Therefore, it is necessary to reduce vibration of the space payloads during the launch phase.

Vibration test is an important part of environmental test, and fixture plays a very important role in vibration

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test. Unreasonable fixture design, manufacturing and installation are prone to "over vibration" and "under vibration". Even if the requirements of the control spectrum are barely reached, the load of the vibration system will be increased, especially when the thrust capacity of the vibration table is small, the contradiction will be more prominent (Fen, 2001; Zeng, 2010; Amjad et al., 2017). The vibration fixture requires good dynamic characteristics, and the vibration table energy should be transmitted to the test product as far as possible (Ke, Sun, & Mao, 2003; Zhang, Meng, Jiang, 2017). The empirical method is one of the most popular methods in fixture design, but it has inherent shortcomings. Since the design of large and medium fixture is generally applied in aviation and military fields, there is little discussion about its design theory, while there is no systematic theory and method. With the development of modern structural dynamic design method, the combination of finite element modal analysis and vibration test used in the design of fixture will be a fast, efficient and meet the design method of engineering precision (Zheng & Chang, 2006; Xu, Gao, Yu, & Zhang, 2017).

In this paper, according to the actual engineering requirements, a thin wall space payload cabinet for placing space payloads is used as the research object and treated with constrained layer damping (CLD) for vibration reduction (Lee, 2008; Lall, Asnani, & Nakra, 1987; Liu, Fan, & Lian, 2015). In order to ensure a reliable connection between the payload cabinet and the vibrating table, the basic requirements of the design of the vibration test fixture and the testing method of its dynamic characteristics are given in this paper. The damping effect of the constrained viscoelastic layer is studied based on finite element analysis and experiment. The researches indicate that CLD treatment has the advantages of simple form, light weight, convenient adhesion, and excellent damping performance, and there is no need to change the existing structure (Gao, 2001; Gao et al., 2017; Zoghaib & Mattei, 2014). As the carrier of the aerospace precision photoelectric instruments, the thin-walled space payload cabinet equipment ensures the performance of these instruments by damping measures, which plays a key role in the development of space industry.

2 DYNAMIC MECHANICAL PROPERTIES OF CLD

CLD treatments are extensively used to damp flexural vibrations of thin-walled metal structures. It has been known for some time that the energy dissipation due to shear strain in the viscoelastic layer can be increased by constraining it with a stiffer covering layer, as shown in Figure 1.

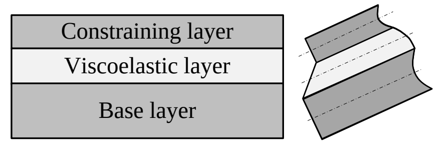


Figure 1. **CLD structure and movement relationship of each layer**

In the theory of vibration, the motion equation of the multi-degree of freedom system takes the form

$$M\ddot{x}+C\dot{x}+Kx=F\left(t\right) \tag{1}$$

where, M, C and K represent physical coordinate mass, damping, and stiffness matrices. F is vector of applied loads

The dynamic equation of the system in frequency domain becomes

$$\left(K_R+iK_I-\omega^2 M\right)X=F \tag{2}$$

where, M is the mass matrix of the composite structure, K_R is the real part of the complex stiffness matrix, K_I is the imaginary part of the complex stiffness matrix, F is the exciting force vector, and X is the displacement vector.

The imaginary parts of the complex stiffness matrix can be further expressed as

$$K_I=\eta_v K_{vR} \tag{3}$$

where η_v is the material loss factor of the viscoelastic materials, and K_{vR} is the real part of the complex shear modulus of the viscoelastic damping material.

For the classical modal strain energy method (Johnson & Kienholz, 1982; Ro & Baz, 2002), the real part K_R of the complex stiffness matrix is adopted to construct characteristic equation, which is used to solve the real mode and can be expressed as

$$\left(K_R-\omega_r^2 M\right)\varphi_r=0 \quad \left(r=1,2\cdots n\right) \tag{4}$$

where, ω_r, φ_r are the r-th order natural frequency and modal shape, respectively. Further, the loss factor of composite structure can be solved using the obtained real modal shape and the solving equation is

$$\eta_r=\eta_v\frac{\varphi_r^T K_{vR}\varphi_r}{\varphi_r^T K_R\varphi_r} \qquad \left(r=1,2\cdots \mathrm{n}\right) \tag{5}$$

It can be seen that when the damping material is selected, the modal loss factor of the structure is directly proportional to the strain energy of the viscoelastic layer damping material. Therefore, the damping material with high loss factor should be applied to the part of the structure with large modal strain energy to obtain a good damping effect.

3 THE FUNCTION AND BASIC REQUIREMENTS OF FIXTURE

3.1 Fixture Function

THE vibration test diagram is shown in Figure 2. The fixture in the vibration test is the member that realizes the connection between the test piece and the vibration table, and the vibration test fixture has two main functions: Fix the test piece on the vibrating table; Transmit the mechanical energy of the vibration table to the test piece.

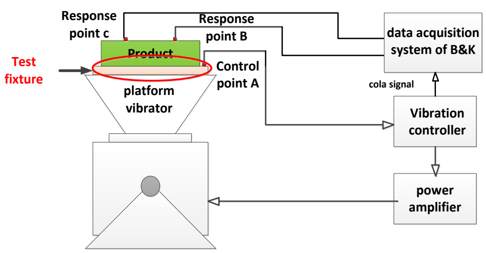


Figure 2. Schematic diagram of vibration test

In vibration test, the most ideal condition is to allow the energy to be transferred without loss to the test piece, in order to reduce energy loss, a nearly rigid connection is to be formed between the moving ring and the fixture. The vibration table motion ring has resonance frequency with or without test piece, which depends on the stiffness and total mass of the moving ring. The designed fixture should avoid resonance under the conditions of minimum weight and maximum stiffness. The transmission characteristics of the fixture are the key to the success and accuracy of the test.

3.2 The Basic Requirements of Fixture

Requirements for mechanical properties of fixtures: the frequency response characteristics of the fixture should be flat and the first-order natural frequency of fixture should be higher than the highest test frequency, for large fixture, to make the first-order natural frequency higher than first-order natural frequency of the test pieces of 3 ~ 5 times, avoid fixture resonance coupling with the test piece on the test direction.

The requirements of the physical characteristics of the fixture: the total mass of the fixture and the attached table are inversely proportional to the square of the resonant frequency. In order to prevent the resonance of the fixture and the test piece within the test frequency range, the quality of the fixture should be as light as possible, and the ratio of stiffness to mass of fixture should be as large as possible.

Requirements for characteristics of fixture materials: in order to meet the high frequency characteristics of the fixture, the factor of the natural frequency of the fixture is E/ρ, where E is the young's modulus and the ρ is the material density. For most metals, the ratio of E/ρ is roughly the same, but the weight is one of the key parameters in the design of the fixture, and for metals of the same size, aluminum is 1/3 heavier than magnesium, steel is 4 times heavier than magnesium, and aluminum, Magnesium has better damping properties than steel, so aluminum and magnesium are commonly used as fixture design materials.

Manufacturing requirements for fixture processing: the commonly used fixture processing and manufacturing methods are casting, block material machining, welding, bolt connection, bonding, etc. Compared with other manufacturing methods, casting manufacturing methods not only have the characteristics of various shapes, it also has high damping, which will help to reduce the amplitude of resonance.

Requirements for fixture installation and connection: Fixture should be able to simulate the actual installation of the test piece, and should be able to prevent non-related failures caused by different installation status; The response of each joint point of the fixture and the test piece should be as same as possible in order to ensure the consistency of the excitation input during the test.

3.3 Testing of Dynamic Characteristics of Fixture

In order to ensure the correctness of vibration and impact test, the fixture should be tested dynamically before use, and its main dynamic characteristics should be checked to determine whether it is necessary to take technical measures to compensate.

The dynamic measurement method of the fixture is as follows: the acceleration sensor is installed on the table of the shaking table for closed-loop control. Set its exciting direction as a_{oy}, select several response points near the connecting hole between the fixture and the test piece. A three-axis acceleration sensor is installed at each response point. The acceleration in the direction of excitation and the acceleration in the lateral response are measured by open-loop a_{iy} and a_{ix} and a_{iz} respectively. A sinusoidal sweep test was carried out according to the frequency range of test parts, and the acceleration of the control points and the response points was recorded with the frequency variation curve. The a_{iy}/a_{oy} curve is obtained and the transfer rate is calculated to check the transfer characteristics of the fixture in the exciting direction, The acceleration ratio in the direction of transverse vibration and excitation is obtained as $\sqrt{a_{ix}^2 + a_{iz}^2} / a_{oz}$.

The vibration control system, as shown in Figure 3, amplifies the driving signal generated by the controller step by step through the power amplifier, drives the shaking table to work, and at the same time, the response signal of the control point is fed back to the controller, which is compared with the reference spectrum. Then the new driving spectrum is obtained, and the control spectrum can meet the requirements of the test control precision.

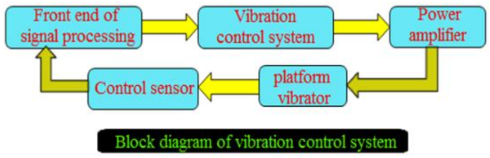


Figure 3. Block diagram of vibration control system

The measurement system, as shown in Figure 4, uses a built-in accelerometer to measure the acceleration response of each point of the product in a vibrating environment. The acceleration sensor measures the acceleration of the X, Y, Z axial system. In the process of the test, the signal analysis system monitors the response of the key points in order to find the problem in time. After each test, the key data are analyzed and the next test is decided according to the measurement results.

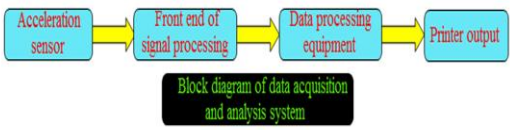


Figure 4. Diagram of data acquisition and analysis system

The laboratory uses the VR controller to control the vibration of the shaking table, at the same time output cola signal The data acquisition system of B&K collects the response point data and cola signal. As shown in Figure 5.

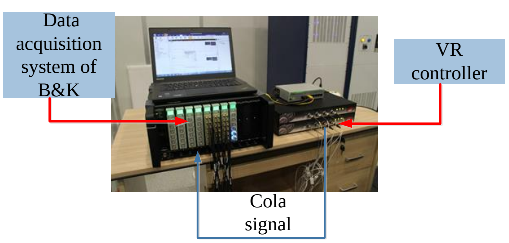


Figure 5. Data acquisition and analysis equipment

Open PULSE Time Data Recorder, and collect the response point and cola time domain signal, the collected signal is in pti format. Open pti file recorded by Time Data Recorder, convert the file to uff format. Then use the offline sine data reduction module of LMS software to process and analyze the data of the uff format file.

4 SIMULATION AND TESTING RESEARCH OF SPACE PAYLOAD CABINET

4.1 Composite structure

THE whole structure is composed of space payload cabinet, fixed backplane and fixture device. The three-dimensional model of space payload cabinet and its fixture device are shown in Figure 6. The space payload cabinet is a thin plate structure, hollow inside. The lower cover plate of the cabinet is fixedly connected with the fixed backplane, and then the fixed backplane is fixedly connected with the fixture. The quality of the cabinet is 15 kg, and the material is hard aluminum alloy 7075.

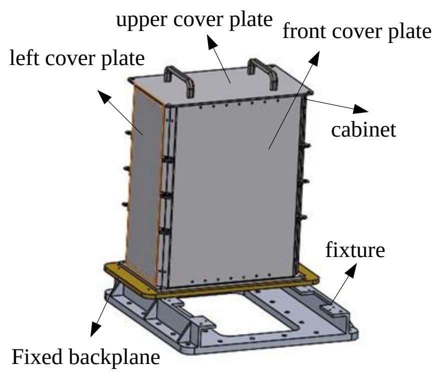


Figure 6. Space payload cabinet and fixed tool

The viscoelastic damping layers are placed in the middle surface of the front and rear cover plate and the left and right cover plate of the cabinet. The thickness of the viscoelastic layers are 0.8 mm and the thickness of the constrained layers are 2 mm. The material properties of viscoelastic layers and constrained layers are shown in Table 1.

4.2 Simulation Analysis

4.2.1 Modal analysis

Modal analysis is a prerequisite for kinetic analysis, because the natural frequency and the modality are the essential parameters of kinetic analysis, reflecting the vibrational properties of the structure. Modal analysis is divided into theoretical modal analysis and experimental modal analysis. In the finite element software, the theoretical modal analysis is carried out, which is also called computational modal analysis. The principle is to use the finite element method to discrete the structure, and then establish the vibration differential equation to solve the eigenvalue and eigenvector of the equation, namely the natural frequency and modal mode of the structure. The basic equation of modal analysis is:

$$[K]\{\Phi_i\}=\omega_i^2[M]\{\Phi_i\} \tag{6}$$

In this case, [M] and [K] are the mass matrix and the stiffness matrix, and ωi and $\{\Phi_i\}$ are the natural frequencies and modalities of the first model of the structure. Modal analysis is to solve the equation (6) and obtain the natural frequency and modal mode of the structure.

The cabinet finite element model uses a right-handed coordinate system. The X-axis is along the length of the cabinet, and the positive direction is from the right cover plate to the left cover plate. The Y-axis is along the width of the cabinet, and the positive direction is from the back cover plate to front cover plate. The positive direction of the Z-axis is upward. The method of automatic mesh generation and manual mesh division is adopted, and the triangular element and quadrilateral element are used. The viscoelastic layers adopt solid element, and the constraint layers adopt shell element with a bias. After modal analysis, the first 6 natural frequencies and mode shapes of the cabinet are shown in Table 2 and Figure 7 respectively.

4.2.2 Harmonic response analysis

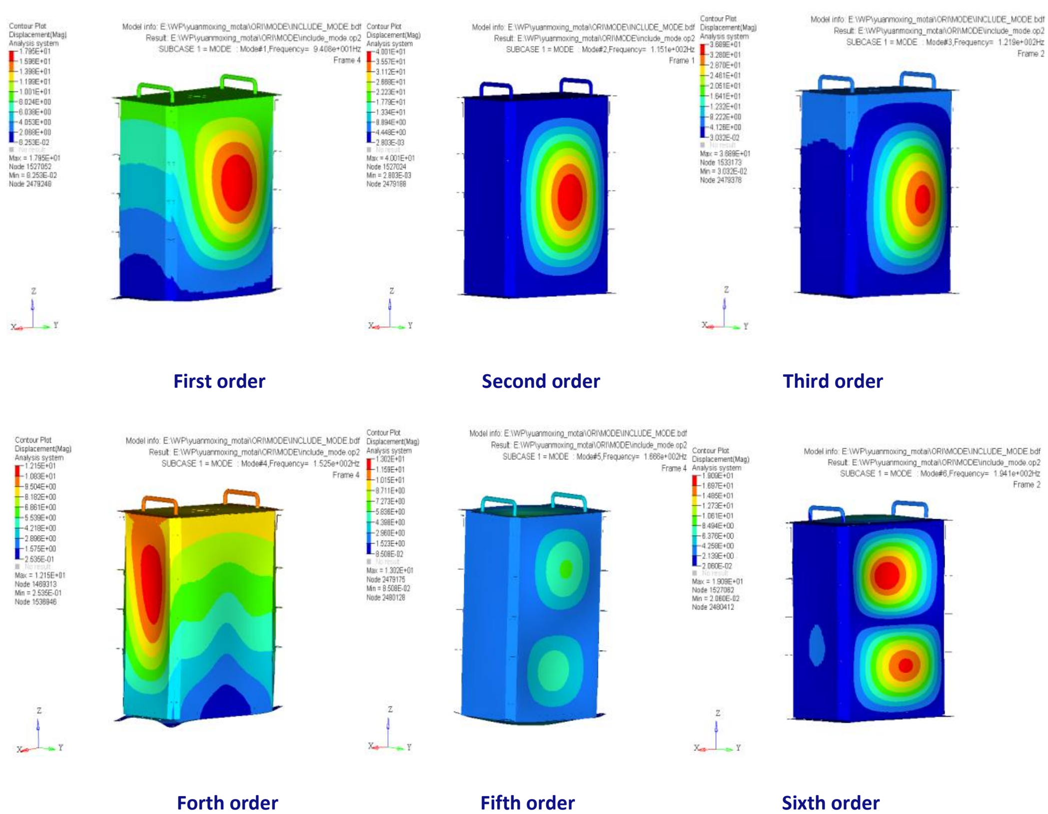
The sinusoidal swept frequency vibration analysis of the original model and the new model with damping material was performed in MSC. Nastran finite element analysis software. The frequency range was 0-100 Hz and the acceleration level was 2 g. The modal expansion method was used for frequency response analysis to improve the calculation speed and accuracy. The simulation results are shown in Figure 8 and Figure 9. It can be seen that the X-direction maximum acceleration value occurs at the left and right cover plates, and the Y-direction maximum acceleration value occurs at the front and rear cover plates. The Z-axis maximum acceleration value occurs at the upper and lower cover plates. The maximum acceleration

Table 1. Material parameters of CLD

	Shear modulus(Mpa)	Elastic modulus(Mpa)	Poisson ratio	Loss factor
Viscoelastic layer	1.5	—	0.45	1.05
Constraining layer	—	7.2×10^4	0.33	—

Table 2. Natural frequency of space payload cabinet

Mode	Original model	new model	Mode	Original model	new model
	fn/Hz			fn/Hz	
1	94.1	85.1	4	152.5	143.4
2	115.1	98.5	5	166.6	164.4
3	121.9	111.2	6	194.1	180.4



First order   Second order   Third order

Forth order   Fifth order   Sixth order

Figure 7. The first six orders of mode shapes

values of the original model and the new model and their corresponding frequency comparison results are shown in Table 3. It can be seen that the damping effect of the cabinet model in the Y direction is obvious, the vibration reduction effect is close to 30%. The damping effect in the X and Y directions is not significant, mainly because no resonance occurs in these two directions from 0 to 100 Hz, and the effect of the damping material is not fully exerted.

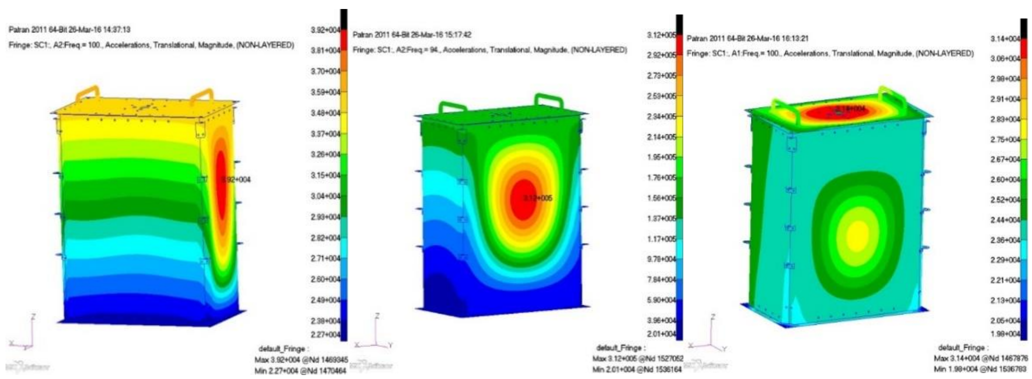
Furthermore, after applying the constrained damping layer, the basic frequency of the cabinet model was reduced from the original 94 Hz to 84 Hz.

4.3 VIBRATION TEST OF CABINET

The test object is the space payload cabinet which is mounted on the vibratory table through fixture device. The test system includes excitation system, test system, data acquisition system and data processing system. The test apparatuses include a vibration table, a controller, a power amplifier, a 64-channel B&K 3660-D data acquisition device, a B&K 4508-B acceleration sensor, and a computer as shown in Figure 10. The sine sweep signals generated from the vibration controller are amplified by the power amplifier and transmitted to the vibration table. The vibration tables excite the vibration of the cabinet along the direction of X, Y and Z axis respectively. The acceleration response signals of each measurement channel are filtered and amplified, then collected and delivered to the computer. After average processing and analysis, the acceleration curves of each test point are obtained.

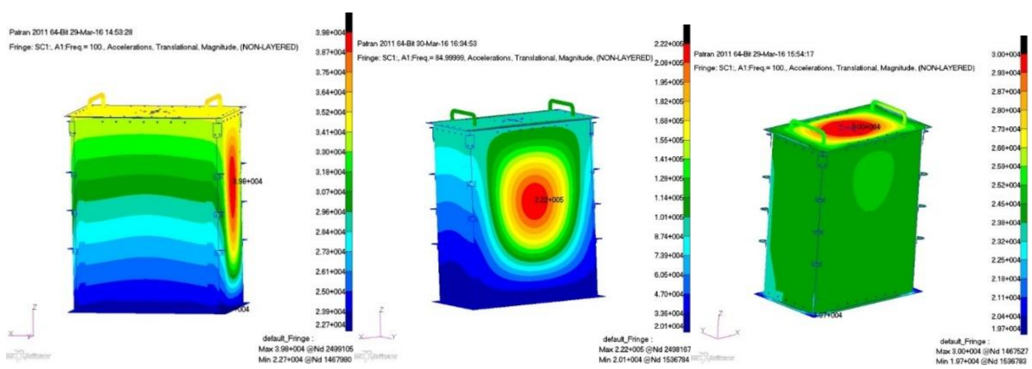
Table 3. Acceleration amplitude and frequency

Direction	Original model	new model	Original model	new model
	Maximum acceleration(g)		Frequency(HZ)	
X	3.92	3.98	100	100
Y	31.2	22.2	94	84
Z	3.14	3	100	100



X Y Z

Figure 8. Acceleration nephogram of the original model



X Y Z

Figure 9. Acceleration nephogram of the new model

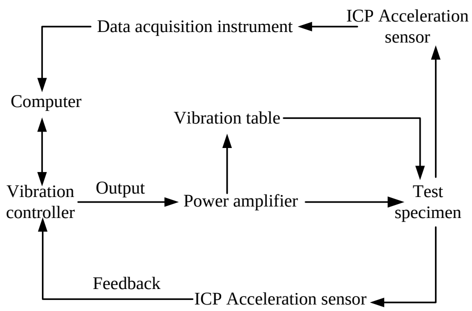


Figure 10. Vibration test data processing flow chart

As shown in Figure 11, the geometric center of each cover plate of the cabinet is selected as the test point. Then the sine sweep vibration tests in the direction of X, Y, and Z axis are performed on the original structure and the new structure with CLD treatments to evaluate the vibration reduction effect. The test points 3, 1, 4, 2, 5 and 6 are correspond to the left, right, front, rear, upper and lower cover plate respectively, as shown in Figure 12.

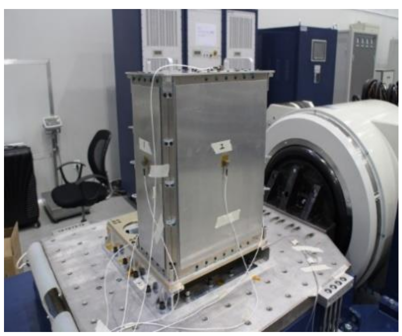
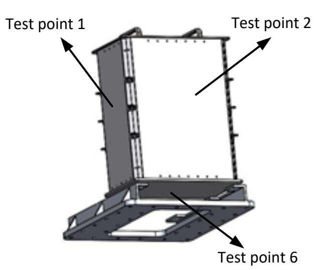
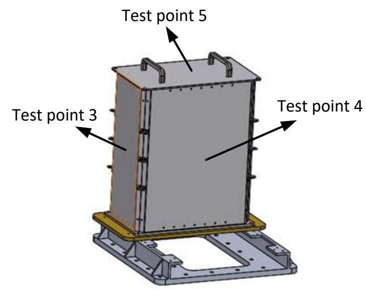


Figure 11. Vibration test of cabinet



(a)



(b)

Figure 12. Location distribution of testing point

The acceleration curves of each test point in the direction of X, Y and Z axis in the range of 0-100Hz can be obtained, and the maximum acceleration amplitude of the original structure and the new structure can be extracted, as shown in Figure 13-14. The acceleration responses of the original structure simulation of the space payload cabinet without viscoelastic damping layers are very close to the testing results. After applying viscoelastic damping layers, the maximum amplitude of acceleration responses of the new structure are lower than that of the original structure.

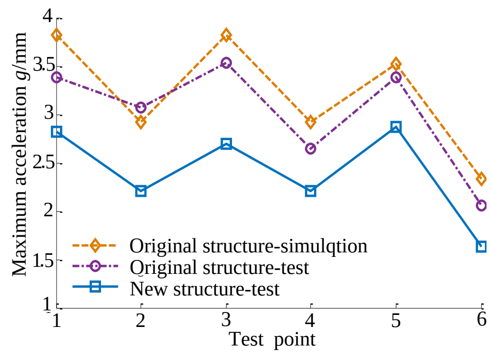


Figure 13. Maximum acceleration amplitude of the test points in the direction of X axis

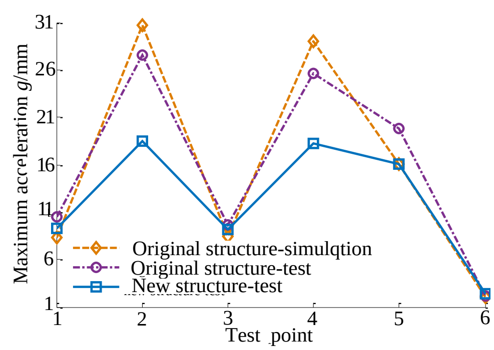


Figure 14. Maximum acceleration amplitude of the test points in the direction of Y axis

By comparing the test data of these test points, it can be seen that the maximum acceleration of the new cabinet in the direction of X, Y and Z axis is lower than that of the original structure. After applying constrained layer damping material, the vibration damping effect of the cabinet in the direction of Y axis is the most obvious, the amplitude of the maximum acceleration response of the test points 2 and 4 are reduced to about 30%, and the test points 1 and 5 are about 15%. The vibration reduction effect in the direction of X axis is also obvious. The response amplitude of the test points 2, 3, and 6 are reduced by more than 20%, and the test points 1, 4 and 5 are about 15%. The vibration reduction effect in the direction of Z axis is poor, and the response amplitude of each test point is reduced below 10%.

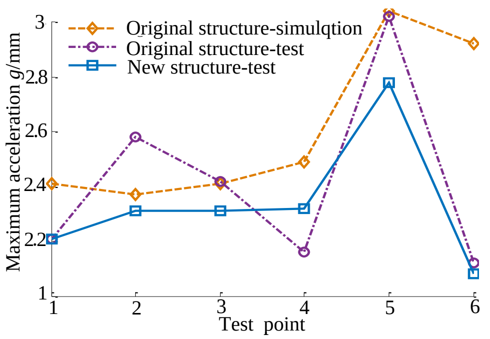


Figure 15. Maximum acceleration amplitude of the test points in the direction of Z axis

5 CONCLUSION AND FUTURE WORK

IN this paper, a thin-walled cabinet for carrying space payload is introduced, and the damping design is carried out by applying constrained viscoelastic damping layer. The functions and fixture design requirements of vibration test fixture are put forward. The method of testing the dynamic characteristics of fixture is put forward, and the method and principle of test and test are explained. A test method based on the vibration response point of the B&K test system is proposed, this method is of great significance to the signal processing of vibration test, and the fixture meets the basic requirements of the fixture. According to actual work conditions, based on the commercial finite element analysis software, the modal analysis and the harmonious response analysis of the original model and the new model are carried out, and the simulation results are verified by the vibration test. The simulation and test results show that the damping effect of the space payload cabinet with CLD treatment is obvious, and the greater the deformation of the position, the more obvious the damping effect of the constrained viscoelastic damping layer. In addition, through analysis of the vibration test data, there are more intuitive and accurate evaluation on the vibration damping effect on the space payload cabinet, which is of great significance for reference and guidance for the applications on similar spacecraft. In the future, topology optimization theory will be introduced to optimize the layout of constrained layer damping material for improving material utilization and reducing weight.

6 ACKNOWLEDGMENTS

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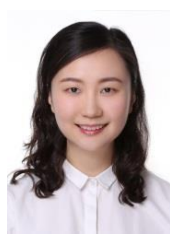

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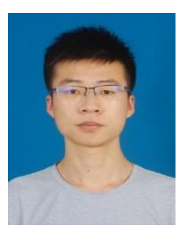

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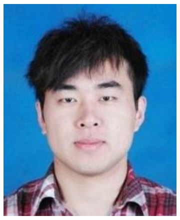

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