Research on Vibration Reduction Compatibility for Multiple Transportation Conditions

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Abstract. Vibrations in the process of transportation are often neglected in product design. In order to avoid damage to acceleration sensitive equipment or components of spacecraft due to destructive transport load, vibration absorbers are usually installed in the container during spacecraft transportation. How to determine the vibration reduction scheme suitable for different payload weight has become the key point and technical difficulty in the design of vibration reduction in transportation. Aiming at this problem, the size and weight of the target product are analysed, and the vibration features of different sections of the satellite are analysed and tested. On this basis, a general vibration reduction subsystem of the container is designed, which is suitable for three kinds of satellite sections and four types of transportation conditions, and the loading distribution parameters are optimized. Transportation test data show that the vibration reduction effect of the new system for aerospace products meets the design requirements.

1. Introduction
Transportation is an important part of the life cycle of spacecraft. When the transport excitation frequency is equal to the natural frequency of the satellite-container, resonance will occur, which will damage the vibration sensitive components on the satellite. Spacecraft transportation experiences continuous mechanical loads [1-3]. This kind of load is characterized by random vibration (excitation frequency ranges from 5Hz to 300Hz) and periodic load [4]. In addition, impact load is inevitable. Referring to the mechanical environment conditions of satellite design, the vibration at the interface of satellite is generally no more than 0.6g and the impact is no more than 1G. In actual highway transportation, the vibration requirements are generally satisfied, but there are usually several shocks exceeding 1G.

The dynamic environment of launch vehicle is the main design basis of satellite structure design. Vibration in the process of satellite ground transportation are easily neglected in satellite design. Satellite container system usually adopts fixed fundamental frequency vibration reduction system because of its reliability, size and other constraints. [5]

How to determine the vibration reduction scheme suitable for different payload weight has become the key point and technical difficulty in the design of vibration reduction in transportation. Aiming at this problem, the size and weight of the target product are analysed, and the vibration features of different sections of the satellite are analysed and tested. On this basis, a general vibration reduction subsystem
of the container is designed, which is suitable for three kinds of satellite sections and four types of transportation conditions, and the loading distribution parameters are optimized in this paper.

2. Platform size and layout design

In order to ensure and expand the universality of the container, it is required that the payload adapter in the packing cabin should be equipped with a universal support interface to meet the mechanical constraints of all kinds of satellite and meet the matching requirements of the above three types of transport conditions. Moreover, for the smaller MEO, the task requires both single transport and two at the same time.

Based on the above conditions, the adapter should also satisfy the following constraints:

- The size should not exceed the transport limit.
- Easy to assemble and disassemble.
- The overall stiffness of the adapter is greater than that of the payload.

In addition, the transporting vibration sensor should be located in the location with better local stiffness when the size of the shock absorber loading table is large. The structural design forms satisfying the above conditions are shown in Figure 1.

![Figure 1. Compatible Transportation Support Interface](image)

The basic parameters of various transport loads are shown in Table 1.

| Load type                  | MEO | IGSO | GEO | Double MEO | Max. Envelope |
|----------------------------|-----|------|-----|------------|---------------|
| Transportation weight (kg) | 590 | 1450 | 650 | 1250       | 1500          |
| X                          | 2506| 3056 | 2630| 5412       | 3056          |
| Y                          | 1900| 2428 | 1950| 2428       | 2428          |
| Z                          | 1942| 2882 | 1870| 1942       | 2882          |
| Transverse screw spacing (mm)| 1470| 2324 | 1910| 1470       | /             |
| Longitudinal screw spacing (mm)| 200 | 200  | 200 | 200/400    | 200           |

3. Design of Vibration Absorption Scheme

3.1. Statics analysis

Constraints and Load Applying: Fixed constraints in six areas below the loading platform, and loaded 1.6t in the black area. As shown in Figure 2
Figure 2. Mechanical analysis constraints and load sketch of loading platform

The deformation analysis shows a maximum deformation of 0.12 mm and less than the required 5 mm. The maximum stress is 92.2 MPa at the upper middle area and less than steel’s breaking limit (520 MPa).

3.2. Design of vibration absorption system

In order to reduce the impact of vibration and shock on the product during transportation, the selection of shock absorber should avoid approaching the first fundamental frequency of the product, so as to ensure that the vibration and shock acceleration transmitted to the product after vibration reduction meet the established requirements.

On the premise of inheriting mature vibration reduction schemes, we should also consider saving the space occupied in the container, so we choose the way of vibration reduction inside the container. According to the different configurations of the loaded products, the number of vibration absorption modules can be adjusted flexibly.

The vibration magnitude of aerospace products in highway transportation is generally between (0,0.2) and less than the damage threshold required by the products. To assess the impact on products, the focus is on the occasional impact events in transportation, such as the impact caused by crossing railway crossings, deceleration belts, deceleration prompt marking, etc. It can be seen from the data of field transportation test that all the impact exceeding limits occur in the vertical direction (Z direction). The data are shown in Table 2. Therefore, the vertical vibration reduction is preferred in the transport vibration reduction scheme.

| Serial no. | Sampling rate/Hz | X/g  | Y/g  | Z/g  |
|------------|-----------------|------|------|------|
| 1          | 256             | 0.246| 0.366| 1.05 |
| 2          | 32              | 0.84 | 0.48 | 1.36 |
| 3          | 64              | 0.201| 0.366| 1.13 |
| 4          | 256             | 0.402| 0.456| 1.72 |
| 5          | 256             | 0.231| 0.291| 1.22 |
| 6          | 128             | 0.21 | 0.357| 1.19 |

Table 2. Shocklog RD298 recorded overload data

Figure 3 is a sketch of vertical preferential vibration isolator.
Figure 3. Vertical Preferential Vibration Isolator

The calculation of vertical first-order fundamental frequency $F_{nz}$ of wire rope isolator is as follows

$$2\pi f_{nc} = \sqrt{\frac{k_z}{\alpha M}}$$

(1)

In the formula:
- $k_z$: Static stiffness (N/mm) under load.
- $\alpha$: Dynamic stiffness coefficient (average dynamic stiffness coefficient $a = 1.5$)
- $M$: Loading mass per assembly (kg)

Formula (1) shows that increasing $M$ can reduce the first-order fundamental frequency of the vibration reduction system without reaching the maximum deformation. Therefore, the actual load of the isolator is generally larger than the nominal load. Ring wire rope isolator stiffness and damping are both non-linear. The static stiffness of working state should be calculated by first-order fundamental frequency. The use of average static stiffness will lead to inaccurate results.

The shock absorber should be arranged symmetrically under the loading platform plane according to the configuration of Figure 4. Considering that the vibration reduction effect for each satellite should be close to each other, two alternatives are proposed as options, eighteen stronger isolators (GGT220-108) or twenty weaker isolators (GGT250-137). According to the maximum envelope calculation, the total weight $G = 2220$ kg. The first order frequencies of the two schemes can be obtained respectively, as shown in Table 3.

Table 3. Optional isolator model parameters

| Vibration Isolator Model | GGT220-108 | GGT250-137 |
|--------------------------|------------|------------|
| Number of isolators      | 18         | 20         |
| Nominal load (kg)        | 60         | 66         |
| Working Static Stiffness (N/mm) | 190         | 100         |
| Maximum dynamic deformation(mm) | 58         | 90         |
| Maximum impact force(kN) | 4.5        | 4.8        |
| Capacitance (J)          | 220        | 250        |
| Height (mm)              | 108        | 137        |
| Calculated $F_{nz}$ (Hz) | ~ 10.6     | ~ 5.8      |

And the system vibration transmission rate:

$$T = \frac{Output}{Input} = \frac{1 + (2\zeta \lambda)^2}{(1 - \lambda^2)^2 + (2\zeta \lambda)^2}$$

(2)

Where:
- $\lambda$: vibration frequency ratio, $\lambda = f_i/f_n$.
- $f_i$: Frequency of external road disturbance, $f_i = \omega / (2 \pi)$
- $\zeta$: The average damping ratio of the system is 0.15.
It can be seen that the applicability of scheme 2 is better than that of scheme 1. In scheme 2, when the input excitation is greater than 8.2Hz, the system enters the vibration absorption region; when the input excitation frequency is not less than 17.4Hz, the system isolation effect is more than 80%, and the system vibration isolation design meets the design requirements. The vibration reduction arrangement of the loading platform under the maximum envelope is determined as follows: the loading platform is connected to the container through 20 wire rope dampers, and is equipped with two real-time monitoring acceleration sensors, a ShockLog298 vibration and shock recorder, which monitors the vibration state of the whole transportation process of the product, as shown in Figure 4.

Figure 4. Envelope Vibration Absorption Arrangement Scheme

Referring to the loads and constraints imposed on the system in the section of static analysis, the vibration transmission rate of the loading platform is analysed. The simulation results show that the vibration isolation performance of the system has reached more than 80% when the vibration transmission rate is about 15 Hz. The response data is the response at the vibration measuring point in Figure 2. From the curve of Figure 5, it can be seen that the first-order frequency of the vertical direction of the vibration absorber system is about 5Hz, which is slightly smaller than the calculation result (5.8 Hz) in Table 3. This is mainly due to the larger size of the loading platform, the weakened stiffness, and the reduction of the number of equivalent isolators under the specific excitation, resulting in the increase of the equivalent load of a single isolator.

Figure 5. Vibration Transmittance Simulation Curve

According to the above calculation, on the basis of the maximum equivalent shock absorber load meeting the above requirements, the isolator layout can be quickly and conveniently adjusted according to four typical loading modes.

4. Test Verification
Transportation tests were carried out using 1450 kg counterweight based on IGSO satellite. After the end of test, the data in the shock recorder are exported and the time domain waveform of the data larger than 0.9g is analysed. By comparing the shock events before and after vibration reduction, it can be found that the vibration amplitude decreases significantly. See Figures 6.
Shock response spectrum transformation is carried out where the transportation exceeds the limit point. The conclusions are as follows: The equivalent sinusoid vibration magnitude is no more than 0.3g (0Hz~100Hz accumulative), which is less than 0.6g of the satellite permission level, so the shock signal does no harm to the satellite. As shown in figure 7.

Figure 6. Overload Points in the curve

Figure 7. After Vibration Reduction Maximum Point’s Spectrum Analysis

5. Conclusion
In this paper, a technical scheme of loading multiple products in a container is proposed. Through statics analysis and dynamics analysis, the type of isolator is determined, the optimal scheme is defined, and the problem of effective vibration reduction of multiple transport loads is solved. The results of transportation test prove the validity of the scheme.

References
[1] Liu Chen, Zhu Jiantao, et al. Dynamics parameter measurement and analysis of certain satellite platform during launch and in orbit [J]. Spacecraft Environment Engineering, 2017, 34(3): 270-276.
[2] CGWIC. LM-3A Series Launch Vehicles User's Manual [R]. Issue2011.
[3] GJB 1027A-2005 the requirement for launch, upper-stage, and space vehicle[S]. Beijing: The committee of technology and industry of national defence, 2006.
[4] Shi Lixia, et al. Dynamics environment excitations during spacecraft transportations [J]. Spacecraft Environment Engineering, 2013(3): 250-255.
[5] PENG Chengrong. System design for spacecraft [M]. Beijing: China Science and Technology Press, 2011.