Design and Optimization of Vibration Isolation System for Spacecraft Transport Package Box

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Abstract. The damping effect of vibration isolation system for spacecraft transport package box is directly associated with superior or inferior mechanical environment for spacecraft transportation. In this Article, the design process and program for vibration isolation system for spacecraft transport package box were introduced first; followed by analysis on excitation and response during spacecraft transportation by train; finally, an optimization program of vibration isolation system applicable to current working conditions for transportation was given. The results showed that: the frequency of railway road spectrum excitation was concentrated at 1.8Hz and 6.2Hz, indicating that the current vibration isolation system for package box did not have better damping effect. After reducing first-order inherent frequency by about 3Hz, acceleration root mean square may reduce by 40%.

1. Introduction
After research and development, spacecrafts may be delivered to launching site for launching missions by road, train, and ship or air transportation. During transportation, spacecrafts may be subject to excitation caused by disturbance from the environment and means of transportation, including bumps in the road, wave impact, flow disturbance and emergency braking [1-2]. The vibrations and shocks induced by these factors are transmitted to the spacecraft. If not effectively suppressed and attenuated, the spacecraft will be adversely affected or even damaged [3]. Damping is generally achieved by mounting dampers on the base of the package box. For different types of spacecraft, in case of changes on spacecraft mass and center of mass, the number and layout of dampers may be adjusted to minimize the damage caused by vibration during transportation.

From consideration of vibration isolation, the softer the vibration isolation system is, the better the vibration is. But for consideration of the support and the limit, the harder the vibration isolation system is, the better it is to ensure the compactness of the space structure. Obviously these two are contradictory, so in actual application, we should find a compromise solution that can better meet the two requirements [4-5].

This Article introduces the design method of vibration isolation system for spacecraft transport package box, analyzes the data of train running test under true working conditions, evaluates the vibration isolation system of the current transport package box, and finally gives the optimization program for of vibration isolation system.
2. Design of Vibration Isolation System

2.1 Design Process
The vibration isolation system must fully consider the transportation conditions during design, and combine the quality characteristics of the vibration reduction object and the type of damper to finally confirm a reasonable and feasible damping program. For vibration isolation system of spacecraft transport package box, wire rope dampers are generally used. In order to balance vertical and lateral stiffness of the vibration isolation system, wire rope dampers are generally subject to 45° layout [6]. See Figure 1 for design process for vibration isolation system of spacecraft transport package box.

![Design Process Diagram](image_url)

**Figure 1.** Design Process of vibration isolation system

M1: Determine parameters of vibration reduction object, including weight and dimensions of spacecraft; M2: Determine installation method of dampers: the number of wire rope dampers shall meet requirements on spacecraft stability, installation location shall be distributed evenly, and the stiffness center of the wire rope damper coincides with the whole-spacecraft position in vertical direction as much as possible; M3: Determine static load of dampers; M4: Analyze excitation from external disturbance, and determine peak frequency of external disturbance source; M5: Analyze excitation from external shock wave, shock waves are mainly from working conditions such as emergency braking, turning and hump or dip in road, determine maximum amplitude of the shock wave; M6: Design peak response frequency of system, and determine inherent frequency of vibration isolation system based on excitation from external disturbance, which shall be \(1/\sqrt{2}\) less than the frequency of external disturbance; M7: Determine capacity parameters for wire rope dampers based on maximum peak of external shock wave; M8: Select suitable type of wire rope dampers based on number of wire rope dampers, inherent frequency of vibration isolation system and capacity parameters of wire rope dampers; M9: Calculate actual dynamic characteristics (peak response frequency, vibration and shock transmissibility, shock resistance of system) based on actual load-deformation characteristics of wire rope dampers, verify whether meeting with requirements.

2.2 Design Program
Transportation status of model-x spacecraft is shown in Figure 2, the vibration isolation system is supplied with 18 dampers, and L bracket, docking flowerpot and spacecraft above the damper. During transportation, external excitation may be transferred to L bracket through the vibration isolation system before finally transferring to the spacecraft. Controlling the transfer function of vibration isolation system can effectively reduce the transmissibility of vibration power flow.


3 Analysis of Excitation and Response

3.1 Sensor layout
Train running test under true working conditions may be required to objectively evaluate the efficiency of vibration reduction during spacecraft transportation [7]. During training running test, model-x spacecraft was supplied with 6 vibration sensors and 3 shock sensors as shown in Figure 3. For which, shock 3, vibration 5 and vibration 6 were set at base of package box, the measured data was regarded as road spectrum excitation data of working conditions for transportation. Vibration 2, vibration 3, vibration 4 and shock 2 were set at lower surface of the docking flowerpot. Vibration 1 and shock 1 were set at upper surface of the L bracket, and the measured data was regarded as response data after vibration reduction. Direction X shown in Figure was forward direction of transportation, direction y was lateral direction and direction z was vertical direction.

3.2 Analysis of excitation and response
The train running test was carried out according to true path of spacecraft transportation and the typical section on such path was selected for analysis. Figure 4 gives the vibration acceleration power spectral density curve of measuring point 5 and 6. From which we can see that, vertical acceleration power spectral density at vertical direction was far greater than that at forward and lateral direction. The frequency components of the acceleration power spectral density at three directions were centered at 1.8Hz and 6.2Hz.

Figure 5 gives the vibration acceleration power spectral density curve of measuring point 1 and 2. The curve at measuring point 3 and 4 was basically the same with that of measuring point 2. Measuring point 1 was relatively close to base of package box, making vertical vibration greater than other two directions significantly; measuring point 2 was located at docking flowerpot and far away...
from base of package box, vibration amplitude in the forward direction increased significantly due to pitching angular speed. The frequency components of the acceleration power spectral density at three directions were centered at 1.8Hz, 6.2Hz and 10–11Hz. Based on road spectrum excitation data and measuring point response data, it can be seen that frequency of railway road spectrum excitation were mainly at 1.8Hz and 6.2Hz, while 10–11Hz shall be a first-order frequency of the vibration isolation system.

![Figure 4. Acceleration Power Spectral Density of Vibration Measuring Point 5 (Left) and 6 (Right)](image-url)

![Figure 5. Acceleration Power Spectral Density of Vibration Measuring Point 1 (Left) and 2 (Right)](image-url)

![Figure 6. Acceleration Power Spectral Density of Shock Measuring Point 3 (Left) and 1 (Right)](image-url)

Acceleration root mean square of each measuring point may be obtained by integration processing of acceleration power spectral density curve, as shown in Table 1 and Table 2. For measuring point at vibration sensors, data was amplified averagely by about 2.5 times at direction x, about 1.5 times at direction and slightly decreased at direction z, making integrated acceleration amplified averagely by about 1.2 times. For measuring point at shock sensors, data was amplified averagely by about 2.2 times.
at direction x, about 1.53 times at direction and slightly decreased by about 1/2 at direction z, making integrated acceleration consistent substantially. Hence, current vibration isolation system did not have better damping effect and required optimization.

**Table 1. Vibration Acceleration Root Mean Square by Measuring Point**

| RMS          | Excitation Point (m/s²) | Response Point (m/s²) |
|--------------|-------------------------|-----------------------|
|              | Measuring Point 5       | Measuring Point 6     | Mean Value | Measuring Point 1 | Measuring Point 2 | Measuring Point 3 | Measuring Point 4 | Mean Value |
| x-forward direction | 0.2027                    | 0.2068                | 0.2048     | 0.2802            | 0.5499              | 0.667              | 0.6647                | 0.5405    |
| y-lateral direction   | 0.2236                    | 0.2063                | 0.2150     | 0.3134            | 0.5034              | 0.4107              | 0.3443                | 0.3930    |
| z-vertical direction   | 0.6423                    | 0.7181                | 0.6802     | 0.7231            | 0.6853              | 0.3908              | 0.2796                | 0.5197    |
| Integrated          | 0.7097                    | 0.7752                | 0.7425     | 0.8364            | 1.0127              | 0.8753              | 0.7991                | 0.8809    |

**Table 2. Shock Acceleration Root Mean Square by Measuring Point**

| RMS          | Excitation Point (m/s²) | Response Point (m/s²) |
|--------------|-------------------------|-----------------------|
|              | Measuring Point 3       | Measuring Point 1     | Measuring Point 2 | Mean Value |
| x-forward direction | 0.2075                    | 0.3096                | 0.6565              | 0.4831    |
| y-lateral direction   | 0.3426                    | 0.3789                | 0.5396              | 0.4593    |
| z-vertical direction   | 0.9987                    | 0.7145                | 0.4173              | 0.5659    |
| Integrated          | 1.076                     | 0.866                 | 0.9467              | 0.9064    |

**4 Optimization Program**

Without changing the overall program of the vibration isolation system, adjusting the number of dampers is the most simple and easy solution for optimizing the vibration isolation system. Table 3 gives the strategy for stiffness of the vibration isolation system adjustment. After adjusting the number of dampers, the positions of other dampers will be adjusted accordingly so that the bearing capacity of each damper is uniform.

**Table 3. Stiffness Adjustment Strategy**

| No.          | Adjustment Method       | Adjusted First-Order Frequency (Hz) |
|--------------|-------------------------|-----------------------------------|
| Increase system stiffness | Strategy 1 | Increase 4 dampers | 12.08                  |
|               | Strategy 2 | Increase 12 dampers | 16.37                  |
| Decrease system stiffness | Strategy 3 | Remove dampers 3, 6, 15 and 18 | 6.67                   |
|               | Strategy 4 | Remove dampers 1, 2, 4, 5, 9, 10, 11, 12, 16, 17 | 3.69                   |
|               | Strategy 5 | Remove dampers 1, 2, 4, 5, 9, 10, 11, 12, 15, 16, 17 | 3.09                   |
|               | Strategy 6 | Remove dampers 1, 2, 4, 5, 6, 7, 9, 10, 11, 12, 14, 15, 16, 17 | 2.21                   |

For simulation calculation, the measured data curve of the acceleration power spectrum is generally used as the stimulus input for simulation [8]. That is, the measured data of measuring point 5 and 6 in train running test are used as input for different adjustment strategies, and the vibration responses at measuring point 1, 2, 3, and 4 in the train running test are obtained. Figure 7 gives the acceleration power spectral density at measurement point 2. Table 4 gives the acceleration root mean square of each strategy.

We can see from Figure 7 and Table 4 that, Strategies 1-3 had no significant impact on acceleration power spectral density; acceleration root mean square of strategy 4 decreased by -20.99%, with peak amplified at frequency of 2Hz and reduced significantly at frequency of about 4.4Hz. Acceleration root mean square of strategy 6 decreased by -44.30%, with peak amplified at frequency of 2Hz and reduced more significantly at frequency of about 4.4Hz, indicating better damping effect.

Without changing the model of dampers, strategy 5 was recommended, for which damper deformation was 13mm and meeting capacity requirements. Acceleration root mean square may be reduced by 31.28% within scope of damper application. For strategy 6, though damping effect was more significant, the deformation was 30mm and above, which was close to capacity limit of damper,
thus such strategy was not recommended.

Strategy 5 reduced the inherent frequency of vibration isolation system by about 3Hz, which may lead to greater displacement of spacecraft easily in case of large shock. And once the maximum displacement exceeded the capacity limit of damper, spacecraft may be subject to secondary shock. Hence, after implementation of strategy 5, limit blocks and retarders may be recommended on the vibration isolation system to prevent secondary shock to spacecraft in special circumstances. The layout after adding of limit blocks and retarders is shown in Figure 8. Damping of hydraulic retarder may be adjusted as required. And if vertical damping coefficient was adjusted to as twice the previous value, the acceleration root mean square may be reduced by 50% and above as compared with the previous program.

Figure 7. Acceleration Power Spectral Density at Measuring Point 2 in Different Dampers Number

| No. | RMS before Adjustment (m/s²) | RMS after Adjustment (m/s²) | Variation  |
|-----|-----------------------------|-----------------------------|------------|
| Increase system stiffness | Strategy 1 | 0.4393 | 0.4709 | -7.19% |
| | Strategy 2 | 0.4369 | 0.55% |
| Decrease system stiffness | Strategy 3 | 0.4393 | 0.4244 | -3.39% |
| | Strategy 4 | 0.3471 | -20.99% |
| | Strategy 5 | 0.3019 | -31.28% |
| | Strategy 6 | 0.2447 | -44.30% |
Figure 8. Recommended Layout of Limit Block and Retarder

5 Conclusions
Following conclusions were obtained in this Article after calculation and analysis:

1) Frequency of railway road spectrum excitation was mainly concentrated at 1.8Hz and 6.2Hz;
2) Damping system with first-order frequency of 10~11Hz did not have better damping effect;
3) Damping effect was obvious and acceleration root mean square may be reduced by 40% when inherent frequency of damping system was reduced by about 3Hz;
4) For design of damping system with low frequency, limit blocks and retarders are recommended to prevent secondary shock.

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