Simulation design method of free boundary simulator for high precision spacecraft on orbit

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Abstract. The reproduction of on-orbit free boundary is one of the key technologies for highly accurate ground micro-vibration test of high precision optical spacecraft. This paper focuses on the simulation design method of the free boundary simulator based on finite element method (FEM). The preliminary design is modeled by one-dimensional spring elements, which is effective to optimize the critical design parameters. Then the detailed design model is established to verify the capability of strength, the stability and the modal performance of the free boundary simulator. The actual simulator is manufactured according to the simulation design. And the test of the actual simulator indicates that, the simulation results match well with the experimental results. The development of the free boundary simulator of spacecraft can be instructed effectively by the simulation method in this paper.

1. Introduction

The micro-vibration test is an important approach to evaluate the effect of the moving mechanism on the spacecraft with high precision optical payloads in orbit, and to obtain the structural response transmission characteristics from the vibration source to the payloads, to improve the imaging quality of optical payloads on the spacecraft. One of the key technologies of the micro-vibration test is the reproduction of the “free-free” boundary condition of the spacecraft on orbit.

Several methods have been proposed to reproduce free boundary condition for spacecraft during micro-vibration test. Such as air bags were used to give flight-like boundary conditions for Solar Dynamic Observatory [1]. With the goal of getting as close as possible to a spacecraft free-free condition, the Artemis satellite was mounted on top of a system of pneumatic isolators. The six resonant modes of the system were kept below 2.8 Hz [2]. A ground based test adapter was developed to safely support a large payload and isolate it from ground borne vibration during system level stability testing. The finite element model based simulation was used for predicting the isolation system rigid body modes [3]. The simulation is an effective way for the design of the free boundary simulator, however, the simulation method is not comprehensive presented in the literature.

This paper introduces the two-step design method of the free boundary simulator based on FEM. Firstly the preliminary design is modeled by spring elements, which is effective to optimize the critical design parameters. Then the detailed design model is established to verify the strength capability, the stability and the modal performance of the free boundary simulator. The actual simulator is manufactured according to the simulation design. And the test of the actual simulator indicates that, the simulation results match well with the experimental results. It is concluded that the development of the free boundary simulator of spacecraft can be instructed effectively by the simulation method in this
2. Preliminary simulation of on-orbit free boundary simulator

The spacecraft is in a free state without any external constraints during its flight on orbit due to the lack of gravity. By the theory of structural dynamics, the spacecraft possesses six rigid body modes of zero frequency. However, external constraints have to be applied to hold the spacecraft at rest due to the gravity of the earth on ground. The goal of on-orbit free boundary reproduction on ground is to make the spacecraft possess six quasi rigid body modes after the external constraints applied. Therefore, the free boundary simulator shown in figure 1 is proposed. The difficulties of the design are heavy payload bearing, high stability, and low suspension frequencies. Effective simulation method is of necessary to accomplish the design.

![Figure 1](image1.png)

Figure 1. The sketch of the on-orbit free boundary simulator.

2.1. Modeling of preliminary design

In the stage of preliminary design, the emphasis is to simulate the mechanical parameters of the metal coil spring based on the configuration of the free boundary simulator, so that the requirements of load-bearing and suspension frequencies are satisfied. For the convenience of analysis and optimization of the stiffness parameters of the coil springs, the one-dimensional spring element of NASTRAN is used to simulate the coil springs. And the lumped mass element is accepted to represent the spacecraft with mass and moment of inertia. The spacecraft and the springs are connected by RBE2 elements to simplify the bolt connection. The combined model of the spacecraft and the free boundary simulator is illustrated in figure 2.

![Figure 2](image2.png)

Figure 2. The preliminary model of spacecraft and free boundary simulator.

2.2. Sensitivity analysis of design parameters

A group of initial stiffness parameters of the springs are set for the preliminary design, as shown in table 1. Modal analysis results show that the suspension frequencies are high. The bounce modal
frequency is 2.6 Hz, and the second rocking modal frequency is 3.2 Hz. The initial parameters do not satisfy the design requirements. Modification of the initial parameters should carry on.

Table 1. The initial stiffness parameters.

| Stiffness parameters         | Parameter value |
|-----------------------------|-----------------|
| Lateral stiffness(K_X, N/m) | 40000           |
| Lateral stiffness(K_Y, N/m) | 40000           |
| Axial stiffness(K_Z, N/m)   | 80000           |
| Bending stiffness(K_RX, Nm/rad) | 1000     |
| Bending stiffness(K_RY, Nm/rad) | 1000     |
| Torsional stiffness(K_RZ, Nm/rad) | 3000    |

As seen that there are six stiffness parameters for the spring element in the preliminary design. In order to achieve high efficiency of design, the sensitivity analysis is performed, so that the most sensitive parameters to the suspension frequencies can be identified. The six stiffness parameters are defined as design variables, and the first six modal frequencies of the system are set as response objective. The sensitive matrix is obtained by sensitivity analysis, which is in table 2. It shows that, the first rocking modes are mainly affected by lateral stiffness and axial stiffness, the torsion mode is basically affected by lateral stiffness, and the bounce mode and the second rocking modes are mainly affected by axial stiffness. It is also seen that the bending and torsional stiffness of the spring have weak influence on the suspension modes. The sensitivity results are used to instruct the optimization of the parameters.

Table 2. The sensitivity matrix of stiffness parameters.

| Mode                | K_X | K_Y | K_Z | K_RX | K_RY | K_RZ |
|---------------------|-----|-----|-----|------|------|------|
| First rocking (Y)   | 0.00| 6.23| 3.01| 0.03 | 0.00 | 0.00 |
| First rocking (X)   | 6.89| 0.00| 2.70| 0.00 | 0.02 | 0.00 |
| Torsion             | 5.83| 8.24| 0.00| 0.00 | 0.00 | 1.69 |
| Bounce              | 0.00| 0.00| 28.67| 0.00 | 0.00 | 0.00 |
| Second rocking (X)  | 8.82| 0.00| 23.36| 0.00 | 0.11 | 0.00 |
| Second rocking (Y)  | 0.00| 14.65| 30.60| 0.23 | 0.00 | 0.00 |

2.3. Results of preliminary design

Based on the sensitivity analysis results, more targeted selection of parameters are made for the design optimization of suspension frequencies. The optimized parameters of stiffness are listed in table 3. It shows that obvious modification is produced for lateral and axial stiffness. And the optimized suspension frequencies are shown in table 4. The shapes of the first and second rocking mode are illustrated in figure 3. The suspension frequencies satisfy the design requirements. The preliminary design is accepted for detailed design.

Table 3. The optimized parameters of stiffness.

| Stiffness parameters         | Parameter value |
|-----------------------------|-----------------|
| Lateral stiffness(K_X, N/m) | 24000           |
| Lateral stiffness(K_Y, N/m) | 24000           |
| Axial stiffness(K_Z, N/m)   | 48800           |
| Bending stiffness(K_RX, Nm/rad) | 1000    |
| Bending stiffness(K_RY, Nm/rad) | 1000     |
| Torsional stiffness(K_RZ, Nm/rad) | 3000    |
Table 4. The suspension frequencies of preliminary design.

| Mode order | Frequency (Hz) | Mode shape          |
|------------|----------------|---------------------|
| 1          | 0.43           | First rocking (Y)   |
| 2          | 0.48           | First rocking (X)   |
| 3          | 0.8            | Torsion             |
| 4          | 2.0            | Bounce              |
| 5          | 2.1            | Second rocking (X)  |
| 6          | 2.5            | Second rocking (Y)  |

Figure 3. The shapes of the first and second rocking mode.

3. Detailed simulation and test validation of on-orbit free boundary simulator

3.1. Modeling of detailed design

Instructed by the preliminary design simulation results, the particular model of coil spring can be selected, and the diameter of the thread of the spring, the diameter of the coil, the height of the spring, and the pitch of the spring are all determined. The detailed design model is established. The beam element is used to present the coil spring. The spacecraft can be modelled by lumped mass element as same as in preliminary design. And if a detailed spacecraft model is given, the lumped mass element can be replaced for more detailed simulation. The mechanical interface structure is represented by solid element. The spacecraft and the interface are connected by RBE2 elements as same as in preliminary design to simplify the bolt connection. The combined model of the spacecraft and the free boundary simulator is illustrated in figure 4.

Figure 4. The detailed model of spacecraft and free boundary simulator.
3.2. Analysis of the design
Comprehensive analyses of static strength, stability, and mode are performed by using the detailed model. The deadweight of the spacecraft is loaded on the model with fixed base ring of the free boundary simulator as boundary condition. The maximum displacement of the system is 71.8 mm shown in figure 5(a), which is the linear deformation of the spring. And the maximum stress of the structure is 36.6 MPa, located at the supporting structure of the spring, as presented in figure 5(b). The deformation satisfies the operating requirements of spring, and the stress satisfies the structure strength requirements.

![Figure 5](image)

(a) (b)

**Figure 5.** The static analysis results of the detailed model. (a) static deformation (mm) and (b) static stress (MPa).

The stability of the free boundary simulator loaded by spacecraft deadweight is shown in figure 6. The first buckling mode shape of the system is toppling deformation of the springs. And the assurance factor is 1.6 which is satisfying for the engineering application.

![Figure 6](image)

**Figure 6.** The first buckling mode deformation.

| Mode order | Frequency (Hz) | Mode shape            |
|------------|----------------|-----------------------|
| 1          | 0.45           | First rocking (Y)     |
| 2          | 0.48           | First rocking (X)     |
| 3          | 0.8            | Torsion               |
| 4          | 2.0            | Bounce                |
| 5          | 2.06           | Second rocking (X)    |
| 6          | 2.08           | Second rocking (Y)    |
Modal analysis results of the system are shown in table 5. The highest suspension of the system is 2.08 Hz. The mode shapes of the first and second rocking mode are presented in figure 7. The suspension of the system satisfies the design requirements.

![Figure 7. The shapes of the first and second rocking mode.](image)

### 3.3. Test validation

Based on the simulation of the detailed design, the free boundary simulator is manufactured, as shown in figure 8. The base ring of the simulator is fixed on the test hall floor. And the upper ring is used to attach spacecraft.

![Figure 8. The manufactured free boundary simulator.](image)

![Figure 9. The test configuration of the system.](image)
The test configuration is illustrated in figure 9. Suspension frequencies measurement is performed by application of external force and torque along the X, Y and Z direction of the spacecraft to make the spacecraft deviate from the static balance position. Then the applied force is removed to make the system freely oscillate, and the free oscillation response of the system is measured by acceleration sensor and laser displacement meter. The oscillation period is extracted by analyzing the measured response curves to obtain the suspension frequencies.

The test results are listed in table 6. It is found that there are certain errors between the simulation and test by comparing tables 5 and 6. One of the reasons is that the lumped mass model is used in the simulation, which is different than the real spacecraft. However the simulation results match well with the experimental results. It is satisfied for engineering application.

| Mode order | Frequency (Hz) | Mode shape       |
|------------|----------------|------------------|
| 1          | 0.35           | First rocking (Y)|
| 2          | 0.37           | First rocking (X)|
| 3          | 0.91           | Torsion          |
| 4          | 1.98           | Bounce           |
| 5          | 1.99           | Second rocking (X)|
| 6          | 1.99           | Second rocking (Y)|

4. Conclusion
A simulation method of the free boundary simulator based on FEM is proposed for the design of on-orbit free boundary simulator of spacecraft. The method basically consists of two steps. The first step is to simulate the design by one-dimensional spring elements, with focuses on the identification of sensitive design parameters for effective optimization. The second step sets up the detailed simulation model to analysis the accurate static and dynamic performances of the free boundary simulator, for the purpose of engineering application. The simulation is validated by the test of the actual simulator. It is concluded that the proposed method provides effective instruction to the design of the on-orbit free boundary simulator of spacecraft.

References
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