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Full Text

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Design of the support structure for a space-based concave thin mirror

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Abstract: The neutral surface of a concave thin mirror is too close to the mirror surface, which makes it difficult to effectively mount the flexible structure and increases the mirror surface shape error. To address this problem, we design a flexible support structure including connectors, a support plate, and flexible structures, and construct an equivalent mirror by installing connectors and a support plate on the back of the mirror. While ensuring that the neutral surface of the equivalent mirror is moved away from the mirror surface, we optimize the support structure so that the rotary center of the flexible structure is located on the neutral surface of the equivalent mirror, avoiding the tilting moment. Following design and modeling of the structure, we analyze the static and dynamic characteristics using a finite element simulation, finding a root-mean-square (RMS) value for the surface shape error of 9.28 nm under the coupled effects of 1g gravity load, 4°C temperature rise, and 0.005 mm unevenness assembly error, with a fundamental frequency of 170.75 Hz, which all meet the design requirements. Finally, we carry out a surface shape error test of the mirror assembly, confirming it to meet the design index requirement of the mirror assembly. Simulation and test results verify the reliability and effectiveness of our proposed support structure.

Keywords: Space optics; Thin mirror; Flexible support; Neutral surface; Surface shape error

1. INTRODUCTION

Along with the continuous development of optical technology, space-based remote sensors are used increasingly widely in the field of astronomy. The surface shape error of any given mirror is closely related to the imaging quality of the system in which it is employed, and the manufacturing of mirrors and their supporting structures have attracted much attention. Under complex and variable external loads, a good support structure ensures the stability of the mirror surface shape error. Mirror support structures are important for spatial positioning and maintaining surface shape error, ensuring the normal operation of a mirror assembly and affecting the imaging quality of the optical system. This is a key area of research in the field of space-based remote sensors[1,2].

In response to the demand for high-performance applications of mirrors, much work has been focused on the optimization and improvement of support structures. Guo et al.[3] optimized a design for the support structure of a 1.5 m aperture space-based mirror in response to the problem of residual stresses from processing and assembly, which damage the imaging performance of the system. Li et al.[4] designed a biaxial 2-degree-of-freedom flexible hinge for a 0.5 m aperture mirror assembly structure, and determined a relationship between the flexible hinge installation position and the mirror surface shape error in the loading and adjustment direction. Liu[5] proposed an adjustable bipod flexible support for a 1.2 m aperture space-based mirror assembly. Based on the principle of centripetal or undershirt motion of a bidirectional ball screw, the position of the intersection point of the flexible rod is changed to adjust the relative position of the rotary center of the support and the neutral surface. This effectively reduces surface shape error caused by positional deviation. Zhai et al.[6] designed a flexible structure consisting of two flexible modalities connected in series for a 2.02 m aperture space mirror assembly. This was combined with a typical flexible support structure to form a composite flexible support structure suitable for a large-aperture mirror support and pendulum-swept mirror support.

In this study, we design a three-point back flexible support structure for the high-performance support requirements of a concave thin mirror with an $ $814 mm aperture. We theoretically examine how to support a mirror with a horizontal optical axis, design a support structure,

and describe its characteristics in detail. Then, we model the three-dimensional structure and set load cases to simulate and verify the static and dynamic characteristics. Finally, we verify the feasibility and validity of the support structure using a surface shape error test.

2. DESIGN INDEX REQUIREMENT

2.1. Operational Environment and Index Analysis

The operational environments of the mirror assembly are mainly categorized into ground-based, launch, and space environments, and the external loads of different environments make various requirements for the mirror assembly. Since the fabrication, assembly, and testing of the mirror assembly are carried out in a ground-based environment, gravity is the main cause of mirror deformation. Along with the ignition, acceleration, and separation phases of the rocket, the mirror assembly is subjected to severe impact and vibration during launch. When the mirror assembly is launched into space and enters the microgravity environment, any deformation caused by gravity is eliminated, degrading the surface shape accuracy and position accuracy, meaning that the effect of gravity load release on the mirror must be taken into account. At the same time, the mirror assembly will be subjected to complex and variable external heat flux in space. Under various load cases, a reliable support structure is required to maintain the stability of the mirror.

We examine support technology for an existing concave thin mirror composed of silicon carbide (SiC), with an effective aperture of Φ814 mm and an apex radius with a curvature of 4540 mm. To reduce the cost of launching it into orbit, the mass of the mirror assembly should be limited to make it as light as possible. To ensure the imaging quality of the optical system, the mirror should have good position accuracy and surface shape accuracy. A support structure should maintain the mirror surface shape as much as possible while ensuring the spatial positioning of the mirror. From the ground to space, the mirror assembly is subject to the effects of gravity, temperature, and assembly errors, requiring the support structure to improve the static characteristics of the mirror to reduce the effects of external loads. During launch, the mirror assembly is subjected to adverse impact and vibration conditions, requiring it to have good dynamic characteristics under the effect of various external excitations. In summary, the index requirements of the mirror assembly are given in Table 1.

Table 1. Design index requirements for the mirror assembly

Item	Requirement
Effective aperture/mm	$\Phi 814$
Mass/kg	≤ 80
Temperature/°C	20 ± 4
Surface shape error/nm	$\leq \lambda/45(\lambda = 632.8$ nm)
Displacement/μm	≤ 10
Tilt/(“)	≤ 2
Frequency/Hz	> 100

2.2. Index Allocation

When designing a support structure, it is usually necessary to allocate indices such as surface shape error, mass, rigid body displacement, and inclination[7]. Among these, the surface shape error is the most critical index in the design of the support structure, requiring a more complicated error analysis. Consequently, index allocation is carried out after analyzing various sources of error that cause mirror deformation.

Surface shape error of a mirror usually includes the residual error of mirror surface processing, the error caused by gravity load, the error caused by temperature load, and the error caused by assembly stress. Because these errors are not related to each other, the calculation between the total error and all component errors is described as

$$\sigma = \sqrt{\sum_{i=1}^{s} u_i^2 + \sum_{i=1}^{q} \sigma_i^2}, \tag{1}$$

where u_i is the single undefined systematic error, and σ_i is the random error.

Combining the surface processing capability of the optical mirror and the installation capability of the assembly, the index allocation considers various errors and tolerance allowance (see Fig. 1). Based on the concave thin mirror and back support theory, we determine the support form to be a three-point back support. In the design process of the support structure, we optimize structural parameters to ensure that the RMS value of each load case is lower than the index component.

3. SUPPORT STRUCTURE DESIGN

For a large-aperture mirror with horizontal optical axis, the neutral surface balances the gravitational moment of each mirror per unit volume[8]. A mirror with a support structure is subjected to gravity and support force and moment (i.e., the force and moment provided by the support structure to the mirror) on the ground, and the moments should be balanced on the neutral surface. Therefore, to reduce the deformation and surface astigmatism, the support structure should be designed to ensure as much as possible that the support point is located on the neutral surface.

3.1. Support Point and Neutral Surface

The flexible structure is an important component of the support structure, which mainly relies on its elastic deformation to isolate as much as possible the effect of external loads on the mirror. In the support structure of the mirror assembly, the flexible structure is usually the flexure hinge, which has a variety of structural forms which can be categorized into single-axis flexure, two-axis flexure, and multiple-axis flexure hinges (see Fig. 2).

Figure 1 flowchart text:

- Total surface shape error: RMS $\lambda/45$ (14.06 nm)
  - Error caused by mirror processing: RMS $\lambda/70$ (9 nm)
  - Error caused by gravity load: RMS $\lambda/70$ (9 nm)
  - Error caused by temperature load: RMS $\lambda/158$ (4 nm)
  - Error caused by assembly stress: RMS $\lambda/211$ (3 nm)
  - Margin (tolerance allowance): RMS $\lambda/211$ (3 nm)

Fig. 1. Surface shape error allocation of the mirror assembly.

The rotary center of the flexible hinge is the support point of the mirror assembly, which is the center point of the elastic bending deformation of the structure, and its position depends on the design structure and material distribution. For a flexible hinge, with a simple structure and regular shape, it is generally close to the geometric center (as shown in Fig. 2C).

Figure 2 panel labels visible: A, B, C. Panel C label: Rotary center.

Fig. 2. Types of flexure hinge. (A) Single-axis flexure hinge. (B) Two-axis flexure hinge. (C) Multiple-axis flexure hinge.

The neutral surface is the surface on which the gravitational moments are balanced, and the center of mass is the average position of the mass distribution, both of which are related to the mass distribution and are therefore located close together. There are many factors affecting the neutral surface, which vary with the complexity of the structure. Therefore, the neutral surface of a mirror cannot be measured with analysis software or the actual test, while the position of the center of mass can be obtained through the modeling function of the Unigraphics NX (UG) software. In this study, we define the center of mass plane as the plane perpendicular to the optical axis of the mirror that passes through the center of mass point. In designing the support structure, the position change of the neutral surface can be described by the center of mass plane, and the position of the ideal support point is determined by finding the relationship between the distance from the rotary center of the flexible structure to the center of mass plane and the surface shape error.

3.2. Equivalent Mirror Formation and Support Structure Design

We consider a concave thin mirror, with a center of mass plane too close to the mirror surface, so that there is not enough support space near the center of mass plane. This results in the rotary center of the flexible structure not falling on the center of mass plane of the mirror, as shown in Fig. 3A. The proximity of the center of mass plane to the neutral surface makes it difficult to ensure that the support point is located on the neutral surface. Consequently, to ensure that the flexible structure has sufficient mounting space near the neutral surface, the position of the neutral surface should be far away from the mirror surface, with a similar situation for the center of mass plane. In the support structure, we add a support plate for mounting flexible structures and connectors for adjusting the distance between this plate and the mirror. This forms a new equivalent mirror, as shown in Fig. 3B, with an increased distance between the center of mass and the mirror surface because of the increased mass on the back of the mirror.

Expanding on this idea, we propose a flexible support structure for a concave thin mirror, consisting of connectors, a support plate, and flexible structures. With the coaction of the connectors and the support plate, the position of the neutral surface is moved further away from the surface in the equivalent mirror than it is in the single mirror. In the design process, we carry out parameter optimization of the support structure to ensure that the support point is located on the neutral surface.

The use of appropriate materials is critical to the imaging performance of an optical system[9,10]. The mirror in this assembly is composed of SiC, with high specific stiffness and good thermal stability. To avoid extrusion resulting from different expansion rates, the connectors and support plate are made from invar (4J32) material with the same linear expansion coefficient as SiC. As a key support component, the flexible structures are made from titanium alloy (TC4) with stable performance. The material properties of the components are shown in Table 2.

The three-dimensional structure of the mirror assembly is shown in Fig. 4. The back of the mirror is an open structure with holes to reduce the total weight (such as triangular holes and fan-shaped holes). The outer edges of the mirror are also cut to reduce weight, and the mounting holes for matching the connectors are located at the intersection between stiffeners and the center of the mirror. The connectors for the mirror and the support plate

Figure labels: A; Center of mass plane; Mirror. B; Center of mass plane; Support plate; Mirror; Connector.

Fig. 3. Schematic diagram of the positional relationship between the mirror and the center of mass plane. (A) Mirror and center of mass plane. (B) Equivalent mirror and center of mass plane.

Table 2. Material and physical properties of the mirror assembly components

Property	Mirror	Connector, support plate	Flexible structure
Material	SiC	Invar (4J32)	TC4
Density $\rho/(10^3\ \mathrm{kg\ m^{-3}})$	3.05	8.1	4.4
Elastic modulus E/GPa	330	141	109
Poisson' s ratio μ	0.25	0.25	0.29
Linear expansion coefficient $\alpha/(10^{-6}\mathrm{K}^{-1})$	2.5	2.5	9.1

contain a mandrel in the center and six connecting rods at the outer edges, which are bonded to the sidewall of the mirror mounting holes with epoxy resin adhesive at one end and screwed to the support plate at the other end. The support plate is designed to be lightweight and has mounting holes for affixing it to the flexible structures. Three flexible structures are symmetrically distributed at angles of 120°, which achieve stable support of the mirror by improving adaptation to gravity, releasing the thermal stress and alleviating the assembly stress.

Figure labels: Mirror; Connecting rod; Mandrel; Connector; Flexible structure; Support plate.

Fig. 4. Three-dimensional modeling of the mirror assembly structure.

3.3. Support Position Optimization

Our initial design is based on the position where the rotary center of the flexible structure is located on the center of mass plane, and we further optimize this by adjusting the flexible structure along the optical axis of the mirror at a fixed distance. To ensure an optimal surface shape, the rotary center of the flexible structure should be located on the neutral surface, and we find the distance from the ideal support point to the center of mass plane to give the ideal location for the flexible structure.

We define the distance from the rotary center of the flexible structure to the center of mass plane of the equivalent mirror as a variable, x (see Fig. 5A). The center of mass plane is determined with the UG software, and the position of the flexible structure is adjusted along the optical axis of the mirror (with distances at intervals of 1 mm) to analyze the relationship between x and the surface shape error (see Fig. 5B).

The results show that the mirror has an optimal RMS value under a gravity load of 1g when $x = 22$ mm, at which point an increase or decrease in distance (at intervals of 1 mm) increases the deformation and degrades the accuracy. This gives the position of the neutral surface, with the distance from the ideal support point to the center of mass plane being 22 mm. Optimizing the structural parameters gives the mass of the mirror assembly as 78.66 kg, which meets the design index requirement of the total mass being under 80 kg.

4. ENGINEERING ANALYSIS

During transport from the ground to space, the mirror is subjected to various disturbing factors such as gravity release, impact, vibration, thermal load, and assembly error. Consequently, it requires a reliable support structure to maintain the surface shape error. To evaluate the effectiveness of the support structure, the finite element mesh of the assembly is generated using the Abaqus software (see Fig. 6). To verify the static characteristics of the structure, we analyze the deformation under different load cases, such as a 1g gravity load, 4°C temperature rise, and 0.005 mm unevenness assembly error. To verify the dynamic characteristics of the structure and ensure that the assembly has sufficient dynamic stiffness under external excitation, we carry out dynamic analysis.

Fig. 5. Position of the rotary center of the flexible structure and its influence on the mirror surface shape error. (A) Schematic diagram showing positions of the rotary center of the flexible structure and the center of mass plane. (B) Relationship between the distance between the rotary center of the flexible structure to the center of mass plane, x, and the RMS value of the surface shape error.

Fig. 6. Finite element model of the mirror assembly.

4.1. Static Analysis

Combined with the operational environment of the mirror assembly, we apply the load cases of 1g gravity, 4°C temperature rise, and 0.005 mm unevenness assembly error to the finite element model. These conditions are set in turn to simulate and analyze the mirror surface shape under the corresponding load cases. We also perform a static analysis under a coupled load of 1g gravity, 4°C temperature rise and 0.005 mm unevenness assembly error. The mirror surface shape nephograms under various load cases are shown in Fig. 7, and the corresponding surface shape values with each load case are shown in Table 3.

The RMS values of the surface shape error under the separate effects of gravity, temperature, and assembly error are better than the corresponding index allocations. This indicates that the surface shape error meets the design index requirement (as shown in the simulation result for the coupled load).

During transport to space, the mirror is subjected to released gravity load and variable temperature load. These differ from assembly error, which is formed during installation. Consequently, the deformation caused by assembly error is fixed and the correction of the deformation caused by assembly error is clearly a one-time effect. As a result, the effect of assembly error on the displacement and inclination can be ignored, and the values of displacement and inclination caused by assembly error are not within the range of design index requirement. Later in the analysis, the rigid body displacements and inclinations are described by taking the absolute values as their signs only represent the direction. The displacement δ_Z in the optical axis direction is not included in the finite element analysis since it is generally corrected by the focusing mechanism.

Since there is a linear superposition between the total rigid body displacement and each displacement component, we calculate the synthesized displacement in the direction of gravity caused by gravity load and temperature rise load to be 6.60 μm, which is better than the design index requirement of 10 μm. Similarly, the synthesized inclination angle in the X direction, caused by gravity and temperature, is calculated as $0.990''$, which is better than the design index requirement of $2''$.

4.2. Dynamic Analysis

During the ignition, acceleration, and separation of the rocket, the mirror assembly is subjected to severe impact, vibration, and overload, which may result in plastic deformation or even fracture failure in extreme cases. To evaluate the dynamic characteristics of the assembly, we analyze the natural frequency and vibration mode[^11-13]. The natural frequency of the structure can be described as

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}}, \tag{2}$$

where f_n is the natural frequency, K is the stiffness of the structure, and M is the mass of the structure. For a structure with a fixed mass, a higher natural frequency means a higher dynamic stiffness.

To avoid the resonance phenomenon due to the natural frequency of the mirror assembly being close to the external interference frequency, we carry out dynamic analysis of the mirror assembly to obtain the low-order natural frequency to ensure that it is not in the range of excitation frequency. We find the first three natural frequen-

Fig. 7. Mirror surface shape contour maps of the mirror assembly under various load cases. (A) The load case of 1 g gravity load ($-Y$ direction). (B) The load case of 4°C temperature rise load. (C) The load case of 0.005 mm unevenness assembly error (of mounting surface 1). (D) The coupled load case of 1 g gravity load, 4°C temperature rise load and 0.005 mm unevenness assembly error.

Table 3. Mirror surface shape values of the mirror assembly under different load cases

Load case	Surface shape error/nm: RMS	Surface shape error/nm: Index allocation	Displacement/μm: δ_X	Displacement/μm: δ_Y	Displacement/μm: δ_Z	Inclination angle/(“): θ_X	Inclination angle/(“): θ_Y	Inclination angle/(“): θ_Z
1 g gravity load	8.54	$\lambda/70$ (9 nm)	0	−6.54	0	−0.990	0	−0.005
4 °C temperature rise load	3.32	$\lambda/158$ (4 nm)	−0.04	−0.06	−7.66	0	0	−0.002
0.005 mm unevenness assembly error	0.87	$\lambda/211$ (3 nm)	0	−1.82	−1.67	−3.513	0	0
Coupled load	9.28	$\lambda/45$ (14.06 nm)	−0.04	−8.41	−9.33	−4.503	0	−0.007

cies of the mirror assembly to be at 170.75 Hz, 189.87 Hz, and 189.90 Hz, with the fundamental frequency higher than the design index requirement of 100 Hz. The first three natural vibration modes are shown in Fig. 8.

5. ASSEMBLY AND TEST

5.1. Mirror Assembly Installation

For the high-performance supported mirror assembly, to minimize the effect of gravity release on the mirror surface, the support point must be located on the neutral surface during installation. Since the neutral surface of the concave thin mirror is too close to the mirror surface to effectively install the support structure, the equivalent mirror is created to keep the neutral surface away from the mirror surface, and the structural parameters are optimized to find the position of an ideal support point. The mirror is made by casting, which causes its geometric parameters to deviate from the simulation model. At the same time, there are some errors in the actual geometric parameter test of the mirror. Therefore, during the actual assembly of the support structure, the installation position should be corrected using a combination of actual measurements and a simulation model.

The neutral surface is not measurable, the center of mass plane is measurable, and both are related to the mass distribution. They are located relatively close to each other. In Section 3.2, we analyze and determine the axial distance from the ideal support point to the center of mass, which is defined as x. Because of the deviation of the actual parameters from the simulation model, a testing apparatus for centroid and rotational inertia (shown in Fig. 9A) is used to measure the center of mass position of the equivalent mirror with a precision of \$±\$0.02 mm. The actual axial distance from the center of mass to the apex

Fig. 8. The first three natural vibration modes of the mirror assembly. (A) The first natural vibration mode. (B) The second natural vibration mode. (C) The third natural vibration mode.

of the mirror surface is measured as D during assembly. To eliminate the effect of the difference between the actual measurement and the simulation model, the actual axial distance from the ideal support point to the apex of the mirror surface should be described as $D + X$, based on which the assembly is carried out, as shown in Fig. 9B.

5.2. Surface Shape Error Test

To evaluate whether the surface shape error meets the design index requirement, we conduct a test with the ZYGO interferometer[14–16]. In the test, the mirror with horizontal optical axis is subjected to a $1g$ gravity load, 4°C temperature rise load (in an ambient room temperature of 20°C), and 0.005 mm unevenness assembly error. The test environment is shown in Fig. 10A, and the surface shape error nephogram of this test is shown in Fig. 10B.

The RMS value of the surface shape error is measured to be 10.11 nm under these testing conditions, which is lower than the design index requirement of $\lambda/45$ (14.06 nm). The assembly is maintained in the same state as in the test after entering space. Based on this, we combine the tested surface shape error with a released gravity load of $-1g$ to calculate the maximum limiting surface shape error of the mirror assembly in space.

The limiting surface shape error in space is calculated to be 13.23 nm by the RMS method (which is the result of $\sqrt{10.11^2 + 8.54^2}$, from Table 3 and Fig. 10B). The result is lower than $\lambda/45$ (14.06 nm), and we infer that the mirror assembly meets the design index requirement.

Fig. 9. Measurement of the center of mass and installation of the mirror assembly. (A) Testing apparatus for centroid and rotational inertia. (B) Structure of the mirror assembly after installation.

6. DISCUSSION

For the mirror assembly, to avoid the tilting moment, it should be ensured that the support point is located on the neutral surface of the mirror. In this study, we consider a space-based concave thin mirror with a neutral surface too close to the mirror surface to effectively mount the support structure. We propose a flexible support structure in which the support plate and connectors with the mirror form an equivalent mirror with a neutral surface further away from the mirror surface because of the increased mass of the mirror back. The structural parameters can be optimized to ensure that the rotary center of the flexible structure is located on the neutral surface of the equivalent mirror for stable support.

7. CONCLUSIONS

We present a theoretical study of a support for a space-based concave thin mirror and propose a flexible support structure that consists of connectors, a support plate, and flexible structures. The connectors and support plate are affixed to the mirror to form a new equivalent mirror, which alters the center of mass, causing the neutral surface to be further from the mirror surface. By optimizing structural parameters, the rotary center of the flexible structure is designed to be on the neutral surface of the equivalent mirror, avoiding the appearance of tilting moment and reducing surface deformation.

Using finite element simulation, under the coupled load effects of 1g gravity, 4°C temperature rise, and 0.005 mm unevenness assembly error, we find the RMS value of the surface shape error to be 9.28 nm, and the displacement and inclination angle that meet the design index requirements. We also carry out a dynamic analysis on the mirror assembly, finding the fundamental frequency to be 170.75 Hz, which is higher than the design index requirement of 100 Hz. Finally, we create an installation of the assembly and conduct tests, measuring the RMS value to be 10.11 nm, which corresponds to the maximum limiting surface shape error of 13.23 nm, all of which meet the design index requirements. Our proposed flexible support structure is feasible and effective.

Fig. 10. Surface shape error test of the mirror assembly. (A) Environment of surface shape error test. (B) Results of surface shape error test under the coupled load.

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AI DISCLOSURE STATEMENT

AI-assisted technology was not used in the preparation of this work.

AUTHOR CONTRIBUTIONS

Ming Bu collected and analyzed the data, and conceptualized and wrote the paper. Kejun Wang conceived the ideas, implemented the study, and revised the paper. All authors read and approved the final manuscript.

DECLARATION OF INTERESTS

The authors declare no competing interests.

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Note: Figure translations are in progress. See original paper for figures.