Design of High-lightweight Space Mirror Component Based on Automatic Optimization

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Abstract. Aiming at the problems of high weight, low cost and extremely short development cycle faced by commercial aerospace space remote sensing cameras, a mathematical model with modal fundamental frequency as the objective function under the constraints of weight and size has been established. Through the automatic optimization method, the lightening rate of the mirror can reach 71.8%. Using the “2 + 2 + 2” quasi-static constraint as the criterion, the flexible support structure of the mirror is optimally designed to realize the high rigidity of the component and ensure the high bearing capacity of the force and thermal environment. Static simulation analysis of the design results of the mirror assembly using hypermesh shows that under the combined influence of gravity field and temperature field, the change of the mirror surface shape (RMS) is within 0.0056λ. At the same time, the modal analysis of the design results shows that the first-order natural frequency of the component can reach 146.52 Hz, which has extremely high dynamic stiffness. It can effectively adapt to the dynamic environment, and the design of the mirror assembly can fully meet the needs of commercial aerospace space remote sensing cameras.

1. Introduction

With the continuous development of science and technology and the increasing requirements of space-to-ground observation, high-resolution space cameras have gradually become the development trend of optical remote sensing satellites. Besides, with the development of the domestic commercial aerospace field, the market has put forward higher requirements for the lightweight, low cost and short-cycle development of high-resolution space cameras. As the spatial resolution increases, the aperture of the entrance pupil of the camera optical system increases linearly, and the weight ratio of the space mirror assembly in the space remote sensing camera is further increased, the light weight and low-cost design of the large aperture space mirror will be of great significance.

As a core component in the optical system, the large-aperture mirror will be subjected to the comprehensive influence of various environmental factors such as gravity field changes, temperature field changes, and harsh mechanical environment during its manufacturing and assembly, ground test, launch and launch, and orbit work [1]. However, these complex environmental factors will cause varying degrees of mirror surface shape (RMS) and position accuracy to deteriorate, resulting in deterioration of the imaging quality of the space camera. Therefore, designing a space mirror component that meets the requirements is one of the difficulties and key technologies in the development of space remote sensors.
2. Mirror structure design
This article studies the main reflector of a commercial space camera. According to the design requirements of the optical system, the effective clear aperture is 520 mm, the central aperture is 140 mm, the mirror radius of curvature is 1798 mm, the mirror shape accuracy is $PV \leq 1/10\lambda$, $RMS \leq 1/50\lambda$ ($\lambda = 632.8$ nm), and the operating temperature level varies from $20^\circ$C to $3^\circ$C.

2.1. Mirror material
In order to achieve light weight and high stability design of the mirror, high specific stiffness ($E/\rho$) and high thermal stability ($\lambda/\alpha$) materials should be selected. At present, the materials commonly used as mirrors are mainly metal beryllium (Be), ultra-low expansion fused silica (ULE), silicon carbide (SiC), glass-ceramics (Zerodur) and aluminum alloy (Al) [2], etc. Considering physical parameters, silicon carbide material is very suitable as a large-aperture mirror material, but in order to meet the low cost and short cycle requirements, the mirror material used in this article is glass ceramic (Zerodur).

| Table.1 Physical parameters of common optical material |
|-----------------|-----------------|-----------------|-----------------|-----------------|
|                 | Density $\rho$(kg/m$^3$) | Modulus of elasticity E/GPa | Thermal conductivity $\lambda$(W/mk) | Thermal expansion coefficient $\alpha/(10^{-6}/K)$ | Poisson's ratio $\mu$ |
| Be              | 1850            | 287             | 216             | 11.4            | 0.043           |
| ULE             | 2190            | 72              | 1.4             | 0.03            | 0.17            |
| SiC (RB)       | 3050            | 370             | 155             | 2.5             | 0.24            |
| Zerodur         | 2530            | 91              | 1.64            | 0.05            | 0.24            |
| Al              | 2700            | 68              | 167             | 23.6            | 0.33            |

2.2. Mirror thickness ratio
The mirror thickness ratio of the mirror has a great influence on the rigidity of the mirror body, and it is one of the important factors affecting the accuracy of its own surface shape. For pie-shaped mirrors, the empirical formula of the diameter-thickness ratio $D/t$ and self-weight deformation of the circular mirror is given by Roberts [3]:

$$\delta = \frac{3\rho ga^4}{16Et^2} = \frac{3\rho g(D/t)^2 D^2}{256E}$$

$\delta$ is the maximum self-weight deformation of the mirror surface (m), $\rho$ is the material density (kg/m$^3$), $g$ is the acceleration of gravity (m/s$^2$), $a$ is the radius of the disk (m), $a = D/2$, $E$ is the elastic modulus of the material Quantity (Pa), and $t$ is the thickness of the disc. According to equation (1), the thickness of the mirror under the condition that the maximum deformation of the self-weight is $\lambda/10$ is 61 mm.

2.3. Support points and position determination
Mirror support methods are mainly divided into center support, circumferential support and back support. The center support is mainly used for small-diameter mirrors. The circumferential support is mainly used for mirror systems with little change in ambient temperature because of no flexible components. For round large-diameter mirrors, the back-support scheme is mainly used. After the diameter-thickness ratio of the mirror is determined, the number and position of the support points of the mirror need to be determined, because the specific number and position of the corresponding support points of the mirror will directly affect the lightweight structure of the mirror, and the number
and position of these support points. It also directly determines the subsequent support structure design of the mirror. Hall gives the relationship between the calculation of the minimum number of support points \( N \) and the mirror deformation PV value (peak valley) for the disk mirror, and gives the empirical formula \[4\]:

\[
N = \frac{1.5r^2}{t} \sqrt{\frac{\rho g}{E\delta}}.
\]

\( r \) is the radius of the disk, \( t \) is the thickness of the lens body, \( \rho \) is the material density, \( g \) is the acceleration of gravity, and \( E \) is the elastic modulus. According to the above formula, the thickness of the mirror body is selected to be 61 mm, the maximum deformation of the mirror is 63.2 nm (\( \lambda/10 \)), and the calculated support points are 3.46 points, respectively. That is, the theoretical calculation according to the above formula requires 4 points of support. In the assembly process of large-diameter mirrors, if the 4-point support method is used, there is a problem of over-positioning. This paper proposes to use a 3-point support structure on the back for optimal design.

For a solid disc-shaped mirror, Hindle gives an empirical formula for the radius of support \[5\]:

\[
D = 289.063 D^{2/3}.
\]

According to formula (3), the radius of the support position \( R = 150 \) mm is initially determined, and the specific value should be comprehensively determined according to the lightweight form combined with the results of the finite element analysis.

2.4. Lightweight design of mirror

For this type of mirror, the structure of the mirror body mainly includes open back, semi-closed back and closed back \[6-9\]. Aiming at the requirements of the processability of the glass-ceramic material and the high lightweight rate, the mirror in this paper adopts an open back type, and the lightweight hole adopts a triangular hole with high rigidity. According to the estimated parameters of the front mirror structure, the parametric optimization analysis of the lightweight design of the mirror is performed. The constraints are the weight of the mirror, the thickness range of the ribs, and the height of the ribs. The objective function is the modal fundamental frequency. The optimization result cloud is shown in Fig.1.

![Fig.1 Optimization result of open back type mirror](image)

Through optimization analysis, the thickness of the reinforcing rib of the mirror is 6 mm, and the local wall thickness at the support point is 10 mm. Refer to the optimization results to build a three-
dimensional model of the mirror, as shown in Fig.2. The optimized fundamental frequency of the mirror is 716.35Hz, and the first-order modal cloud is shown in Fig.3.

![Fig.2 Mirror model](image1)

Deform: mode, A1: Mode 7 : Freq. = 716.35, Eigenvectors,

![Fig.3 First order modal cloud image of mirror](image2)

### 3. Mirror support structure design

In order to ensure that the mirror assembly can meet the requirements of ground installation, micro-weight field and mechanical environment, the mirror assembly must have high dimensional stability, thermal stress unloading and high dynamic stiffness characteristics. In the design of the supporting structure, the low-expansion alloy steel (4J32) is used to nest the material directly in contact with the mirror. The thermal stress unloading process is realized by hinge flexures, and the size and position of the flexures are optimized to achieve “2+2+2” quasi-static constraints to meet thermal stress unloading and high dynamic stiffness characteristics of components [10, 11]. The hinge is made of titanium alloy (TC4) with higher rigidity, and the back plate is made of medium-weight aluminum-based silicon carbide (AL / Sic). The structure of the mirror assembly is shown in Fig.4.
When bonding the steel nest and the fixing hole of the reflector, the optical epoxy glue is used, and the single-sided gap is calculated according to the following formula:

$$h = \frac{r_0(\alpha_c - \alpha_0)}{\alpha_b - \alpha_c + \frac{\nu}{1-\nu} \left( 2 - \frac{h}{2L} \right) \alpha_b - \frac{3}{4} \left( \alpha_0 + \alpha_c \right)}$$

(4)

$r_0$ and $\alpha_0$ are the nesting radius and thermal expansion coefficient, $\alpha_b$ and $\nu$ are the thermal expansion coefficient and Poisson’s ratio of the glue, and $\alpha_c$ is the thermal expansion coefficient of the mirror.

When the nesting thickness is 5 mm, the thickness of the adhesive layer is calculated by formula (4) to be 0.05 mm.

4. **Finite element analysis**

The Hypermesh software is used to perform accurate finite element analysis and calculation of the mirror components. The mesh uses a tetrahedral second-order finite element model, as shown in Fig.5.

4.1. **Static analysis**

Considering that the mirror assembly is affected by the gravity field and the temperature field after entering the rail during the assembly and testing process, the finite element model is analyzed by loading 1 g gravity, 4°C temperature rise and the combined effect of the two conditions. In the
experiment, the change of mirror shape and optical axis orientation under different working conditions was analyzed. Because the primary mirror is supported by three evenly distributed supporting points, the 1g gravity field is calculated according to the two directions of X direction (optical axis level) and 60° rotation (optical axis level), so we choose the better direction as the detection direction of the mirror assembly product installation. The results of the finite element analysis of the mirror components in the above working conditions are shown in Table.2:

Table.2 Calculation results of Mirror

| Condition                        | Surface accuracy (nm) | R/(") |
|----------------------------------|-----------------------|--------|
|                                  | PV        | RMS    | X    | Y    | Z    |
| X-direction gravity              | 0.014 λ   | 0.002 λ | 0.004 | -0.42 | 0.42 |
| X direction rotation 60° gravity | 0.013 λ   | 0.003 λ | -0.05 | -0.72 | 0.71 |
| 3°C temperature rise             | 0.039 λ   | 0.005 λ | 1.04  | 0.007 | -1.03|
| Gravity field and 3°C temperature rise | 0.048 λ | 0.0056 λ | 1.05  | -0.41 | -0.69|

The simulation software analyzes the change of the mirror surface shape under various working conditions, and performs the fitting calculation through Matlab, and obtains the mirror deformation cloud diagram as shown in Fig.6, Fig.7, Fig.8, and Fig.9.
It can be seen from the analysis results in Table 2 that the X-direction gravity surface shape result is better, and it is recommended to select this direction for the detection of the installation and adjustment of the reflector assembly. Under the action of X-direction gravity, 3°C temperature rise, and X-direction gravity and 3°C temperature rise, the surface shape change and the inclination angle change of the mirror assembly all meet the design index requirements.

4.2. kinetics analysis

The modal calculation of the mirror assembly reflects the dynamic stiffness characteristics of the assembly. The mode shape of the mirror assembly needs to be staggered with other external excitations to ensure that it does not resonate with other components and cause damage risks under harsh mechanical environments. Perform modal analysis on the mirror assembly to verify whether the mirror assembly has sufficient dynamic stiffness. The fundamental frequency of the design of the reflector assembly must be greater than 120Hz. The fundamental frequency of the mirror assembly is 146.52Hz, and the vibration modes are shown in the Table 3 below:

| One Step  | Two Step  | Three Step | Four Step | Five Step | Six Step |
|-----------|-----------|------------|-----------|-----------|----------|
| 146.52Hz  | 146.67Hz  | 182.05Hz   | 323.63Hz  | 380.58Hz  | 380.8Hz  |
5. Conclusion
For the 520mm mirror component in this article, the material of the mirror is glass-ceramic. Through the lightweight design of the mirror, the automatic optimization and the flexible support of the component are optimized. The weight reduction rate of the mirror reaches 71.8%. According to the gravity field, temperature field and coupling field conditions, the surface shape analysis is carried out. Under its comprehensive influence, the maximum change of the mirror shape is 0.0056λ, which meets the design requirements. Through modal analysis, the first-order natural frequency of the mirror assembly reaches 146.52 Hz, which has high dynamic stiffness characteristics and can effectively adapt to the dynamic environment. The design method and structure of the mirror assembly adopted in this article can meet the low cost and short cycle requirements of commercial aerospace, and provide reference and reference for the same type of products.

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