Research on the stability of large aperture mirror mounts using cross-flexure pivots in ICF lasers

Hongbiao Huang (黄宏彪)[1,2], Taiying Zeng (曾台英)[1,2], Tao Zhang (张 涛)[1,2], and Jianqiang Zhu (朱健强)[1]

[1]*Laboratory of High Power Laser Physics, Shanghai Institute of Optics and Fine Mechanics, Chinese Academy of Sciences, Shanghai 201800*

[2]*Graduate School of the Chinese Academy of Sciences, Beijing 100039*

Structural design and stability analysis of ϕ570 mm mirror mount are performed. Under the constraint conditions, the geometric parameters of cross-flexure pivot in large aperture mirror mount are determined. With finite element analysis software (ANSYS), the influences of cross-flexure pivot's setting angle on center drifting and stress distribution of flexure, and the dynamic performance of the whole mirror mount, are analyzed. When the setting angle between the direction of gravity and the setting direction of flexure is 0° or 90°, the center drifting of mirror mount is minimum and the stress distribution of flexure is relatively uniform. The nature frequency is 23 Hz, and the maximum amplitude of angular vibration response to random excitation input is 0.91 μrad, which is consonant with experiment results. The performance of mirror mount can satisfy the requirement of precision positioning in Shenguang (SG) II laser system.

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In inertial confinement fusion (ICF) experiments, in order to perform the laser shooting successfully, the short-pulse, high-energy laser outputted from the laser driver must be focused onto the target accurately. As the important component directing lasers, the stability of the large aperture mirror mount is critical to the positioning accuracy of lasers[1−5].

A large aperture mirror mount is shown in Fig. 1. The mirror is mounted in the cell, which is connected to the support base with a cross-flexure pivot. For the cross-flexure pivot is a flexural component in large aperture mirror mount, its performance is critical to the stability of large aperture mirror mount[2], thus, the research on the stability of large aperture mirror mount is focused on the influence of cross-flexure pivot on the mount[6]. In recent years, some scholars performed the research on the performance of cross-flexure pivot, for example, Vukobratovich *et al.* brought forward the method to calculate the torsional stiffness of cross-flexure pivot under the effect of radial force[7], Rundle analyzed the thermal hysteresis of a mirror mount using cross-flexure pivots[8], Zelenika *et al.* got the method to calculate center drifting of cross-flexure pivot with the effect of radial force[9]. However, the above researches did not mention the influence of cross-flexure pivot's setting angle on center drifting and stress distribution, and there are fewer researches on precision adjusting facility, which is large in dimension. In Shenguang (SG) II upgrading laser facility, with the requirement of experiment, the aperture of mirror is changed from ϕ270 to ϕ570 mm, and the mirror has the weight up to 400 N. In order to satisfy the requirement of precision positioning, it is important to study the stability of large aperture mirror mount using cross-flexure mirror mount.

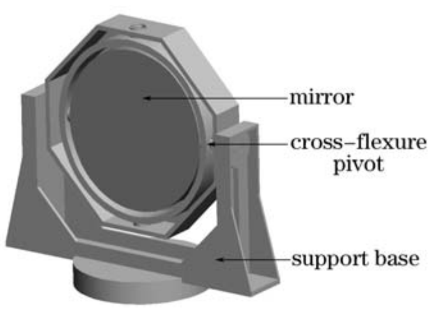

Fig. 1. Large aperture mirror mount.

In this paper, structural design and stability analysis of ϕ570 mm mirror mount is performed. Under the constraint conditions, the geometric parameters of cross-flexure pivot in large aperture mirror mount are determined. With finite analysis element software (ANSYS), the influences of cross-flexure pivot's setting angle on center drifting and stress distribution of flexure are analyzed. The dynamic performance of whole mirror mount is also analyzed. The experiment test is performed.

The stability of a structure is determined by dynamic stiffness (or dynamic compliance), the numerical value of dynamic stiffness is equal to the dynamic force needed for unit amplitude of the structure. When a structure is excited by dynamic force, the larger the dynamic stiffness, the smaller the amplitude, and *vice versa.* For a single degree-of-freedom (DOF) system excited by a harmonic force, the amplitude-frequency response characteristic of dynamic stiffness K_{D0} (namely, $|K(\omega)|$) is expressed by

$$K_{\mathrm{D0}} = \frac{P_0}{A} = k\sqrt{(1-\lambda^2) + (2\zeta\lambda)^2}, \tag{1}$$

where P_0 is force amplitude, A is response amplitude, k is static stiffness, λ is frequency ratio, and ζ is damping ratio.

According to Eq. (1), we can deduce that the dynamic stiffness of structure is related to its static stiffness, mass, damping and frequency. Thereby, with a reasonable structural design, the dynamic stiffness can be improved, namely, to improve the static stiffness, nature frequency and damping ratio. In this way, the stability of structure can be improved.

The cross-flexure pivot is shown in Fig. 2. Its geometric parameters include length L, width B, and thickness H. In order to get reasonable geometric parameters, the torsional stiffness must be as large as the cross-flexure pivot can provide under several constraint conditions. As the influence of radial load on the torsional stiffness can be ignored, the torsional stiffness of cross-flexure pivot can be shown as[7]

$$K = \frac{4EI}{L}, \tag{2}$$

where E is the Young's modulus of 65Mn steel, $E = 200$ GPa; I is inertia moment of flexure cross section, $I = \frac{BH^3}{12}$; L is the length of flexure.

In analysis, the constraint conditions are as follows. 1) The static torque of stepping motor (M_{motor}) is 1.51 N·m; 2) the adjustment range of large aperture mirror mount (θ_{adjust}) is ±40 mrad; 3) in order to ensure the adjusting accuracy, the mechanical character of four flexures must be consistent, thereby, we ensure not only the consistent physical dimension, but also the consistency and uniformity in rigidity after heat processing.

The relationship between torque (M) and rotation angle (θ) of cross-flexure pivot is

$$M = K\theta. \tag{3}$$

Inserting Eq. (2) into Eq. (3), we obtain

$$M = \frac{4EI}{L}\theta. \tag{4}$$

According to the constraint condition 1), we can obtain the maximum torque:

$$M_{\max} = \frac{2\pi\eta N_1 M_{\text{motor}}}{P} R, \tag{5}$$

where η is the transmission coefficient of screw gearing, here $\eta = 0.45$; N_1 is the transmission ratio of screw gearing, $N_1 = 52$; P is the pitch of lead screw, $P = 20$ mm; R is the distance between driving point of lead screw and center of mirror. Substituting the above values into Eq. (5), we can obtain

$$M_{\max} = \frac{2\pi \times 0.45 \times 52 \times 1.51}{0.002} \times 0.3 = 33302 \quad (\text{N} \cdot \text{m}).$$

According to the constraint condition 2), we can obtain the maximum rotation angle, $\theta_{\max} = 10|\theta_{\text{adjust}}| = 400$ mrad[10].

According to the constraint condition 3), we can obtain the thickness of flexure, $H = 2.5$ mm. Substituting the values above into Eq. (4), we obtain: $B/L = 2/3$. Let $B = 20$ mm, then $L = 30$ mm.

After obtaining the geometry parameters of the cross-flexure pivot, analysis of structure stability of the whole mirror mount should be performed. Here, we analyze the influences of cross-flexure pivot's setting angle on the center drifting and the stress distribution of flexure, then we perform the modal analysis and power spectral density (PSD) response analysis to random excitation input.

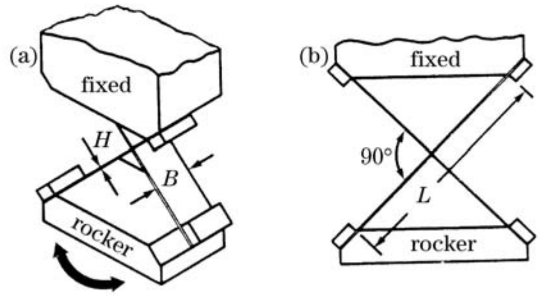

Fig. 2. Cross-flexure pivot.

The center drifting caused by gravity has an important effect on the adjusting precision. The cross-flexure pivot, as the important supporting component of large aperture mirror mount, has various stress state for different mounting position, so it has a critical effect on the center drifting.

Here, we consider the setting angle between the direction of flexure and gravity as the parameter to analyze the relationship between the setting angle and center drifting. According to the symmetry of cross-flexure pivot, let the setting angle change from 0° to 90° which is divided into six divisions by 15°. Then the center drifting of large aperture mirror mount in different setting angles is analyzed with the finite element analysis software (ANSYS). The results are shown in Table 1.

From Table 1, we can get that as the setting angle of cross-flexure pivot is 0° or 90°, the center drifting is minimum.

For the different setting angles of cross-flexure pivot, the stress status is different, so the stress distribution is also different. Non-uniform stress distribution would reduce the adjusting precision and the life of cross-flexure pivot[11]. We adopt the similar analysis method as above, and the results are shown in Table 2, the nodes in Table 2 are obtained along the length direction of flexure.

From Table 2, we can obtain that as the setting angle of cross-flexure pivot is 0° or 90°, the stress distribution of flexure is rather uniform.

In order to obtain the nature frequency of large aperture mirror mount and the response amplitude to random excitation input, we perform the modal analysis and PSD analysis for large aperture mirror mount. For the modal analysis is included in PSD analysis in ANSYS, we only perform the PSD analysis. The setting angle of cross-flexure pivot is 90°, the acceleration PSD measured on

Table 1. Influence of Cross-Flexure Pivot's Setting Angle θ on Center Drifting Δ_{drift}

θ (deg.)	0	15	30	45
Δ_{drift} (mm)	0.05854	0.08539	0.08924	0.08691
θ (deg.)	60	75	90	
Δ_{drift} (mm)	0.08921	0.08542	0.05848	

Table 2. Influence of Cross-Flexure Pivot's Setting Angle on Stress Distribution of Flexure (Unit: MPa)

θ (deg.)	Node 1	Node 2	Node 3	Node 4	Node 5	Node 6
0	15.183	14.966	13.716	13.408	13.279	12.806
15	14.810	15.038	14.162	16.140	13.849	7.325
30	13.335	14.076	13.702	16.013	13.497	7.180
45	10.864	12.134	14.709	12.331	6.401	3.300
60	7.537	9.276	9.995	12.509	10.382	5.219
75	5.029	7.345	7.985	9.342	8.236	3.321
90	0.112	0.136	0.155	0.143	0.121	0.109

the ground of target area in SG II upgrading facility is considered as the random excitation input, the acceleration PSD is shown in Fig. 3.

The first modal frequency and mode shape of large aperture mirror mount are shown in Fig. 4. The displacement response to random excitation input PSD is shown in Fig. 5.

Observing from Fig. 4, it is shown that the first mode shape of large aperture mirror mount behaves as the rotation of mirror around the cross-flexure pivot. The nature frequency is 23 Hz (Fig. 5), the maximum displacement response of large aperture mirror mount is 0.273 μm, which is equivalent to 0.91 μrad angle vibration amplitude.

To verify the results of numerical analysis, we performed a test of single transfer function and a test of acceleration response to random excitation input for the ϕ570 mm mirror mount prototype in SG II laser facility with a precision vibration measurement instrument. The scheme of experiment test is shown in Fig. 6.

In the test of single transfer function, the acceleration transducer is fixed on the mirror cell. The excitation input is created by a hammer, which knocks on the mirror mount. The acceleration response signal to the shock

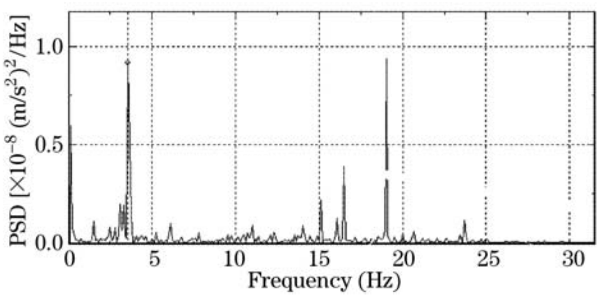

Fig. 3. Acceleration PSD plot for random vibration in vertical direction.

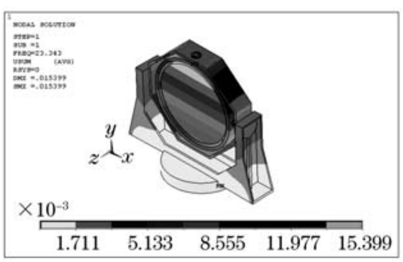

Fig. 4. First modal frequency and mode shape.

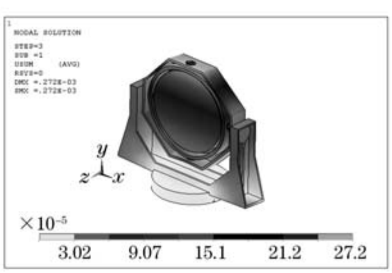

Fig. 5. Displacement response to random excitation input PSD.

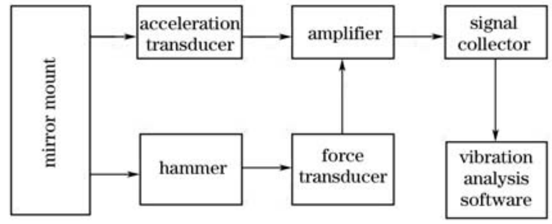

Fig. 6. Scheme of experiment test for ϕ570 mm mirror mount prototype.

excitation input is translated into voltage signal by acceleration transducer, and the shock excitation input signal produced by hammer is translated into voltage signal by force transducer, then the two signals are transmitted into amplifier and transformed into digital signals through signal collector. The single transfer function is obtained after the digital signals are analyzed by vibration analysis software. In the test of response to random excitation input, the procedures are similar to the test of single transfer function, but the hammer is not used, and the response signal is produced by random excitation input.

The test results are shown in Figs. 7 and 8. From Fig. 7, we can obtain that the first modal frequency of ϕ570 mm mirror mount is 16 Hz and the maximum acceleration response to random excitation input is 0.003 m/s^2, which can be converted into angular vibration amplitude A_2 through

$$A_2 = \mathrm{arctg}\left[\frac{A_1}{(2\pi f)^2 R_1}\right], \tag{6}$$

where A_1 is acceleration response amplitude to random excitation input, $A_1 = 0.003$ m/s^2; f is the first modal frequency, $f = 16$ Hz; R_1 is the distance between driving point of lead screw and center of mirror, $R_1 = 300$ mm. Substituting above values into Eq. (6), we can obtain $A_2 = 1.12$ μrad.

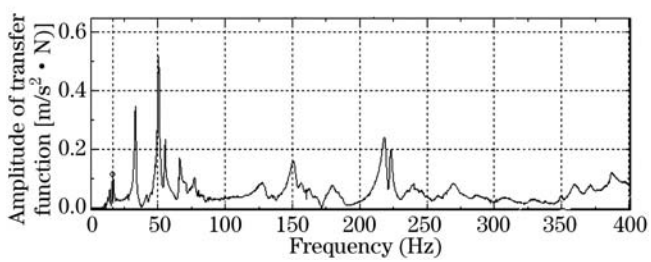

Fig. 7. Single transfer function for ϕ570 mm mirror mount prototype.

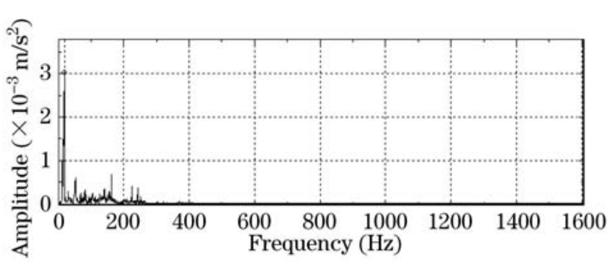

Fig. 8. Acceleration response to random excitation input.

Comparing the test results with the numerical analysis results carefully, it is shown that the angular vibration amplitude of test is rather consistent with that of numerical analysis, but the first modal frequency of test is smaller than that of numerical analysis. The reason might be that we neglected the flexibility of the foundation.

In conclusion, according to the theory of stability and the method of finite element analysis, we performed structural design and stability analysis for large aperture mirror mount, which is supported by the cross-flexure pivot and has the aperture up to ϕ570 mm. It is shown that when the setting angle between the direction of gravity and the setting direction flexure is 0° or 90°, the center drifting of mirror mount is minimum and the stress distribution of flexure is relatively uniform. The angular vibration amplitude of test (0.91 μrad) is rather consistent with that of numerical analysis (1.12 μrad), but the first modal frequency of test (16 Hz) is smaller than that of numerical analysis (23 Hz). The results can satisfy the requirements (the nature frequency is not less than 15 Hz and the angular vibration amplitude is not more than 4.8 μrad) of stability for precision positioning in SG II upgrading facility. Furthermore, it can provide important reference information for stability design of correlative structure in SG II upgrading laser facility.

H. Huang's e-mail address is alexande007@163.com.

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