

Article

Design and Validation of a Compact Focusing Mechanism for Space Optical Cameras with MTF-Based Defocus Analysis

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Abstract

A compact encoder-based focusing mechanism is proposed to compensate for axial focal-plane displacement in a space optical camera. The mechanism converts the rotation of a stepper motor into linear motion of the CCD focal plane assembly through a worm gear pair, a gear pair, and a cylindrical cam pair, while an encoder provides position feedback for closed-loop control. A displacement-count transmission model was developed and verified through MTF simulation, positioning tests, autofocus experiments, finite element analysis, and vibration tests. The mechanism achieved a single-step control resolution of 1.38 μm and an encoder-based displacement measurement accuracy of 0.486 μm . The positioning test showed a maximum absolute error of 7 μm within a ± 2 mm travel range, satisfying the ± 7.5 μm requirement. Optical simulation indicated that axial defocus broadened the PSF and reduced the MTF near the 50 lp/mm Nyquist frequency. Autofocusing experiments using a resolution target showed that the normalized stripe contrast recovered to its maximum near the best-focus position, supporting the practical image-quality recovery trend predicted by the MTF simulation. The measured first natural frequency was 174.64 Hz, and the random vibration amplification ratios were below 3 in all tested directions. The results demonstrate that the proposed mechanism can provide accurate and mechanically robust compensation of focal-plane displacement.

Keywords: CCD focal plane assembly; encoder-based closed-loop control; focal plane compensation; positioning accuracy; mechanical verification



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1. Introduction

Space optical cameras are key payloads for Earth observation satellites and provide high-resolution images for land resource surveys, mapping, agriculture and forestry monitoring, environmental observation, disaster assessment, and defense reconnaissance [1]. The increasing use of small satellites, compact electro-optical payloads, agile imaging platforms, and very low Earth orbit missions has made high-resolution imaging more accessible, but it has also imposed stricter requirements on payload size, structural stability, and long-term image quality [2–4]. For long-focal-length space cameras, even a small displacement of the detector from the best image plane can degrade the final image. Such displacement may be caused by launch vibration, thermal vacuum effects, orbital temperature variation,

and thermoelastic deformation during service [5]. In imaging theory, defocus broadens the point spread function (PSF) and reduces contrast transfer at high spatial frequencies. This degradation is commonly evaluated by the modulation transfer function (MTF), which has become an important indicator for monitoring the actual imaging performance of optical remote sensing satellites [6]. Recent studies have also shown that MTF degradation must be considered in high-resolution and agile satellite imaging, since it directly affects the usable spatial resolution and observation quality [7,8]. Therefore, focus compensation is not only a mechanical positioning problem but also a necessary means of maintaining image quality in space optical cameras.

In linear imaging theory, the MTF represents the magnitude of the optical transfer function and describes how object contrast is transferred to the image plane at different spatial frequencies [9]. It therefore connects three quantities that are central to space imaging: contrast, spatial frequency, and resolvable detail. This makes it more suitable than a displacement index alone for evaluating whether a focal plane correction is effective. MTF-based evaluation has been standardized in digital camera resolution testing and is also widely used in optical remote sensing payloads. Recent studies have further shown its practical value for on-orbit monitoring. Wang et al. [6] estimated the MTF of a Jilin-1 satellite imager directly from remote sensing images and used the MTF at the Nyquist frequency to assess imaging stability. Valenzuela et al. [7] also emphasized that satellite spatial resolution should be evaluated through image plane response rather than by ground sampling distance alone. In addition, MTF degradation has been considered in the mission planning of super agile satellites, showing its direct influence on usable image quality during dynamic imaging [8]. These studies support the use of MTF as a bridge between mechanical focus compensation and final imaging performance.

Existing focusing mechanisms for space optical cameras can be broadly divided into rigid transmission mechanisms and compliant mechanisms. Rigid mechanisms based on screws, gears, cams, or their combinations are widely used because they can provide a relatively large stroke, high load capacity, and clear motion transmission. This design philosophy has also been adopted in flight instruments. For example, the Kepler focus mechanism employed a stepper motor, gears, a ball screw, bearings, a lever arm, and flexures to adjust the telescope focus [10]. The Mars 2020 Mastcam-Z cameras also use motor-driven focus and zoom mechanisms, in which the focus motor drives a lead screw to provide a 9 mm travel range for the focus group [11]. In the OSIRIS-REx mission, the PolyCam camera used a focus mechanism to support observations from long-range acquisition to subcentimeter surface imaging [12]. Jia et al. [13] further summarized space camera focusing mechanisms into cam and screw configurations. Cam mechanisms can convert eccentric cam rotation into linear motion, but wear and the lack of self-locking capability may reduce their reliability. Screw mechanisms usually provide large-stroke and accurate transmission, although their size and transmission complexity may become limiting factors. To reduce friction, backlash, and lubrication-related uncertainty, compliant and piezoelectric focusing mechanisms have attracted increasing attention. Lu et al. [14] proposed a three degrees of freedom focusing system for a three-mirror anastigmat (TMA) telescope using piezoelectric ceramic actuation and flexible support, achieving a fundamental frequency of about 121.5 Hz. Li et al. [15] developed compliant focusing mechanisms in which the small displacement of piezoelectric actuators was amplified by a two-stage flexure hinge lever structure, achieving either high amplification and load capacity or a 2 mm focal plane stroke. More recently, Huo et al. [16] optimized a flexible hinge focusing mechanism and verified by vibration testing that its first fundamental frequency reached 163.28 Hz, while the tilt error was significantly reduced. These studies have advanced the stroke, positioning accuracy, lightweight design, structural stiffness, and launch vibration

adaptability of space-focusing mechanisms. However, most existing work still evaluates focusing performance mainly through mechanical indicators, such as displacement error, tilt error, natural frequency, and vibration survivability. The quantitative relationship among focus displacement, MTF variation, and image quality recovery has not yet been fully incorporated into the design and validation process.

Recent studies have shown that image information and MTF evaluation are effective for focus assessment in space cameras. Peng et al. [17] proposed an on-orbit autofocus method based on multi-star images, in which the optimal detector position was estimated from star images and verified through ground tests, prototype experiments, and on-orbit data. Yan et al. [18] reported the performance test of the high-resolution imaging camera onboard Tianwen 1, showing that MTF can be used to evaluate the focusing state after adjustment. For operational image quality monitoring, Wang et al. [6] estimated the on-orbit MTF of optical remote sensing satellites from acquired images, and Yu et al. [19] analyzed the slanted edge method for MTF measurement of remote sensing cameras. Meanwhile, recent focusing mechanism studies have used finite element analysis and vibration tests to verify stiffness and launch environment adaptability [16]. However, image quality evaluation, positioning accuracy, and mechanical environment validation are still often treated separately, rather than integrated into a unified focusing mechanism design framework.

In this study, a compact encoder-based cylindrical-cam focusing mechanism is proposed for focal plane compensation in space optical cameras. The mechanism converts motor rotation into axial motion of the CCD (charge-coupled device) focal plane assembly through a worm gear pair, a gear pair, and a cylindrical cam pair, while encoder feedback is used for closed-loop control positioning. The core contribution is that the mechanism is evaluated from four complementary aspects: mechanical positioning accuracy, MTF sensitivity to defocus, practical imaging recovery during focusing, and mechanical environmental adaptability. This integrated validation route is intended to support the application of compact focal plane compensation mechanisms in space optical cameras.

2. Design and Methods

2.1. MTF-Guided Design and Autofocusing Strategy

A space optical camera can tolerate a limited amount of defocus without obvious degradation of image resolution. In this study, the half depth of focus is used to estimate the allowable focal plane displacement:

$$\delta_{allow} = 2F^2\lambda \quad (1)$$

where F is the F-number of the optical system and λ is the central wavelength of the operating spectral band. In this design, $F = 10.8$ and $\lambda = 587.6$ nm. The allowable defocus δ_{allow} calculated from Equation (1) is approximately 137 μm .

The positioning requirement of the focusing mechanism is determined by manufacturing errors of opto-mechanical components, assembly errors, electronic control errors, and transmission errors. To provide sufficient margin for focus compensation, the closed-loop positioning accuracy of the mechanism is specified as ± 7.5 μm . This value is much smaller than the allowable defocus of approximately 137 μm and is used as a constraint for mechanical design, resolution analysis, and positioning verification. The MTF simulation shows that the image quality can still approach the selected reference level when the focal-plane displacement is reduced to approximately ± 50 μm . However, this ± 50 μm range represents an acceptable image-quality neighborhood rather than the allowable residual positioning error of the mechanism. Therefore, the ± 7.5 μm requirement keeps the mechanical positioning residual much smaller than the optical defocus tolerance and helps preserve margin

for assembly residuals, calibration uncertainty, thermoelastic drift, and repeated focusing operations. The $\pm 7.5\ \mu\text{m}$ requirement also leaves margin above the $\pm 3\ \mu\text{m}$ inner-loop deadband, ensuring that the controller can stop stably while still satisfying the required positioning accuracy.

Considering structural strength, stiffness, and compensation margin, the focusing stroke is designed to be $\pm 2\ \text{mm}$. This stroke is defined as the bidirectional mechanical travel relative to the calibrated working position, corresponding to a total travel of 4 mm. The stroke should be interpreted as the available mechanical compensation capability of the prototype rather than as the expected routine focusing displacement during normal operation. Static focal-plane offsets caused by manufacturing and assembly errors are mainly absorbed during ground alignment and focus calibration and are converted into the initial working position of the mechanism. A one-time focal-plane shift caused by gravity release and interface-state changes can be corrected by initial on-orbit refocusing. After this initial calibration, the remaining travel is mainly reserved for reversible thermoelastic drift, slow long-term dimensional variation under the space environment, control and search allowance, and mechanical limiting margin. Since the present study does not include a complete thermal-structural-optical coupled analysis for a specific mission temperature range, material combination, and orbital lifetime, the $\pm 2\ \text{mm}$ stroke is treated as the prototype design and experimentally verified mechanical travel range, rather than as a universal in-orbit displacement envelope for all space missions. For a specific flight mission, the remaining positive and negative travel margins should be checked separately according to the actual opto-mechanical configuration, thermal environment, material system, and calibration strategy to avoid reaching either physical limit.

2.1.1. Mechanical Design

The proposed focusing mechanism consists mainly of a stepper motor, a worm gear pair, a gear pair, a cylindrical cam pair, an encoder, and a CCD focal plane assembly, as shown in Figure 1. The worm gear pair consists of a worm and a worm wheel, while the gear pair is composed of a transmission gear and a cam gear. The cylindrical cam pair consists of a cylindrical cam and a cam follower. The rotary motion generated by the stepper motor is transmitted to the cylindrical cam through the worm gear pair and the gear pair, where speed reduction and torque amplification are achieved. The guide groove of the cylindrical cam converts the rotary motion into the axial displacement of the CCD focal plane assembly along the optical axis, thereby compensating for focal plane shift. The encoder is coupled to the transmission chain to measure the actual position of the CCD focal plane assembly and provides position feedback to the controller for closed-loop positioning. The power-off holding capability is provided by the combined transmission chain rather than by the cylindrical cam pair alone. Under the motor power-off condition, an axial inertial load acting on the CCD focal plane assembly would first be transferred to the cam follower and the cylindrical cam groove. The groove constrains the follower motion, so the axial vibration load is mainly converted into alternating normal contact forces on the groove surfaces rather than into free axial sliding of the CCD focal plane assembly. For irreversible axial slippage to occur, the load would have to drive the cam follower along the groove, rotate the cylindrical cam, transmit reverse torque through the gear pair, and further back-drive the worm gear pair and motor side. The worm gear pair arranged at the front end of the transmission chain therefore provides the main anti-back-driving resistance, while the cylindrical cam groove, bilateral guide structure, mechanical end stops, and anti-loosening fasteners provide additional geometric constraint and auxiliary interlocking. Therefore, the work does not simply claim that the cam pair is self-locking; instead, the

power-off holding behavior is explained by the combined effects of reverse transmission resistance, cam-groove geometric constraint, and mechanical limiting/interlocking.

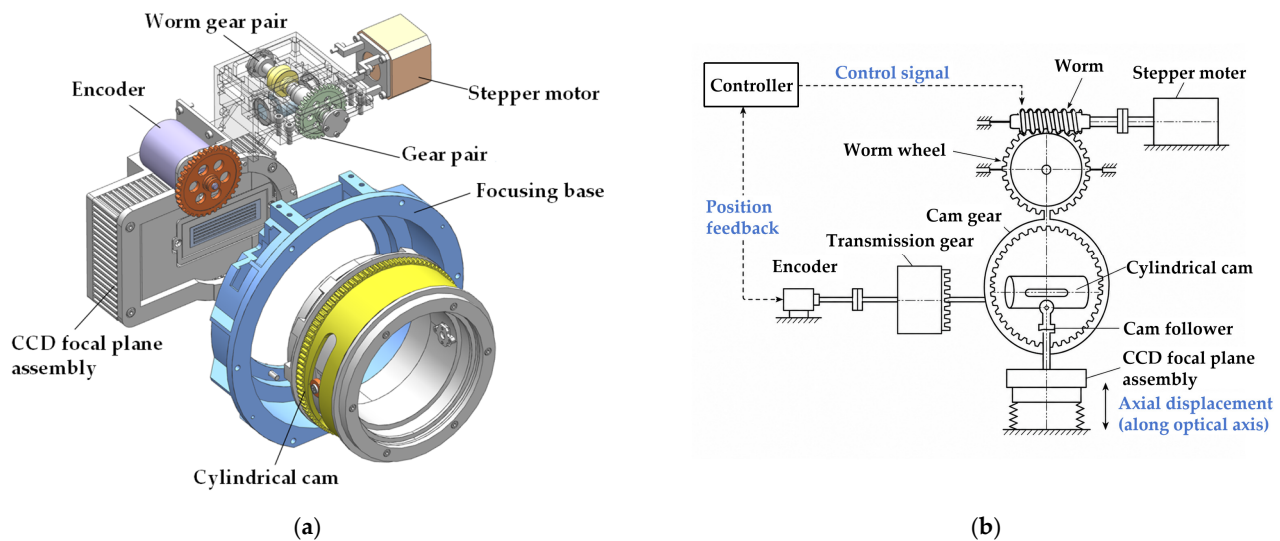


Figure 1. Compact focusing mechanism: (a) structural configuration of the focusing mechanism; (b) transmission chain, encoder-based closed-loop feedback, and anti-back-driving load path under the motor power-off condition.

The transmission ratio from the stepper motor to the cam gear is expressed as:

$$i_{14} = \frac{z_2 z_4}{z_1 z_3} \quad (2)$$

where z_1 is the number of worm starts, z_2 is the number of worm wheel teeth, z_3 is the number of transmission gear teeth, and z_4 is the number of cam gear teeth. In this mechanism, $z_1 = 1$, $z_2 = 22$, $z_3 = 45$, and $z_4 = 144$. This ratio determines the relationship between the motor rotation and the cam rotation. The lead of the cylindrical cam is determined by the linear stroke of the cam follower and the corresponding cam rotation angle, which can be written as:

$$L = \frac{360s_c}{\theta_c} \quad (3)$$

where s_c is the linear stroke of the cam follower and θ_c is the corresponding cam rotation angle in degrees. With $s_c = 7$ mm and $\theta_c = 65^\circ$, the cam lead is calculated as $L = 38.77$ mm. The transmission ratio from the stepper motor to the encoder is given by:

$$i_{15} = \frac{z_2 z_5}{z_1 z_3} \quad (4)$$

where z_5 is the number of encoder gear teeth. With $z_5 = 65$, the transmission ratio i_{15} is 31.78. This ratio is used to establish the feedback relationship between the encoder counts and the focal plane displacement. When the stepper motor rotates by one revolution, the axial displacement of the CCD focal plane assembly is:

$$s_1 = \frac{L}{i_{14}} \quad (5)$$

The calculated displacement is $S_1 = 0.55$ mm. The corresponding change in encoder counts is:

$$c_1 = \frac{N}{i_{15}} \quad (6)$$

where N is the total encoder count per revolution. In this study, $N = 2^{16}$, and c_1 is approximately 2062 counts. Thus, the relationship between the encoder count c and the displacement s of the CCD focal plane assembly can be written as:

$$s = \frac{s_1}{c_1}(c - c_0) \quad (7)$$

where c_0 is the reference encoder count at the initial focal plane position. After encoder zeroing, c_0 can be set to zero. Equation (7) provides the basis for converting a target focal plane displacement into a target encoder count during the closed-loop control process.

The focusing mechanism adopts encoder-based closed-loop control. The controller first converts the target displacement into the corresponding target encoder count and then drives the stepper motor until the feedback count reaches the prescribed tolerance range. The focusing control resolution corresponding to one motor step is:

$$r_s = \frac{s_1 \alpha}{360^\circ} \quad (8)$$

where α is the step angle of the stepper motor. With $\alpha = 0.9^\circ$, the focusing control resolution r_s is 1.38 μm .

The equivalent displacement identification accuracy caused by the angular measurement error of the encoder can be expressed as:

$$r_e = \frac{Li_{15}\varepsilon_e}{360^\circ i_{14}} \quad (9)$$

where ε_e is the angular accuracy of the encoder. With $\varepsilon_e = 36$ arcsec, the encoder-based displacement measurement accuracy is calculated as $r_e = 0.486$ μm .

The encoder identification accuracy is better than the single-step control resolution of the motor, and both values are much smaller than the required closed-loop control positioning accuracy of ± 7.5 μm and the allowable defocus of 137 μm . Therefore, the proposed transmission and feedback scheme provides sufficient actuation and feedback resolution for subsequent MTF-guided focusing. These two values define the actuation and feedback resolution of the mechanism. However, the inner-loop controller does not perform zero-error tracking. The final stopping condition is governed by the position deadband specified in the next section.

2.1.2. Autofocusing Method

To establish a quantitative relationship between mechanical focusing displacement and optical image quality, the MTF at the Nyquist frequency is used as the main image quality metric. If the detector pixel size is p , the Nyquist frequency f_N is given by:

$$f_N = \frac{1}{2p} \quad (10)$$

At the i -th candidate focal plane position s_i , the MTF at the Nyquist frequency can be expressed as:

$$MTF_N(s_i) = MTF(f_N, s_i) \quad (11)$$

Here, $s = 0$ denotes the nominal best focus position determined by optical design or calibration. In the actual autofocusing process, the best focus position is unknown in advance and is estimated from the image quality feedback. It is denoted as s^* . When the CCD focal plane assembly deviates from the best focus position, the point spread function becomes broadened, and the medium and high spatial frequency responses decrease. As a result, MTF_N decreases as the defocus amount increases.

When a standard target, a high-contrast edge, or calibrated imagery is available, MTF_N can be directly calculated and used as the focusing metric. For in-orbit natural scenes or images with unstable texture conditions, MTF_N may not be reliably obtained. In this case, a normalized sharpness function, such as gradient energy, Laplacian response, or high-frequency energy, can be used as a substitute metric. Therefore, the focusing metric $Q(s_i)$ is defined as:

$$Q(s_i) = \begin{cases} MTF_N(s_i), & \text{when MTF can be directly measured} \\ S_N(s_i), & \text{when an image sharpness function is used} \end{cases} \quad (12)$$

where $S_N(s_i)$ denotes the normalized sharpness function. The objective of autofocus is to find the axial displacement that maximizes the image quality metric:

$$s^* = \arg \max_{s_i} Q(s_i) \quad (13)$$

The focusing control adopts a dual closed-loop structure consisting of an image-quality outer loop and an encoder-based position inner loop. The outer loop estimates the best focusing displacement according to the image quality metric, while the inner loop drives the CCD focal plane assembly to the specified axial position using encoder feedback. According to the previously established displacement-count mapping relationship, the target encoder count c_i corresponding to the candidate displacement s_i is:

$$c_i = c_0 + \frac{c_1}{s_1} s_i \quad (14)$$

where c_0 is the encoder count at the reference position. After the controller drives the stepper motor, the actual position should first satisfy:

$$|s_{actual} - s_i| \leq \varepsilon_s \quad (15)$$

where ε_s is the allowable position error. In the implemented inner position loop, ε_s was set to $\pm 3 \mu\text{m}$. According to the displacement-count conversion relationship, this deadband corresponds to approximately 11 encoder counts. The value is larger than the encoder-based displacement identification accuracy of $0.486 \mu\text{m}$ and the single-step focusing control resolution of $1.38 \mu\text{m}$ but remains smaller than the required closed-loop positioning accuracy of $\pm 7.5 \mu\text{m}$. Therefore, the controller does not attempt to track a zero residual error. When $|s_{actual} - s_i| \leq \varepsilon_s$, the target position is regarded as reached, and no additional forward or reverse motor pulse is issued. This deadband-based stopping logic prevents repeated direction reversal when the target position lies between two discrete motor steps and helps suppress low-frequency micro-amplitude hunting in the stationary state. After the mechanism becomes stable, the image is acquired and $Q(s_i)$ is calculated. To reduce the influence of gear backlash and reverse hysteresis, candidate positions should preferably be approached from the same direction. If reverse motion is required, the mechanism can first overshoot by a small displacement and then return to the target position from the unified approach direction.

In the present experimental validation, no independent absolute image quality threshold was calibrated. Therefore, the autofocus procedure is treated as a relative search and peak estimation process. The position corresponding to the maximum sampled image quality metric, or the fitted local peak around it, is used as the estimated best focus position.

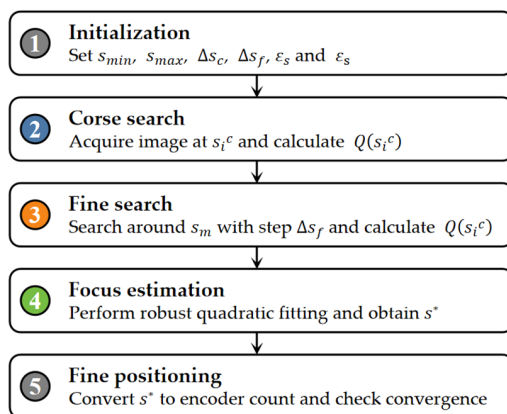
To avoid the low efficiency caused by small step traversal over the full focusing range, the coarse-to-fine adaptive focusing strategy shown in Figure 2 is used. The focal plane displacement relative to the nominal best focus plane is denoted by maintaining the best focus, whereas the fine search increases the sampling density near the peak. The coarse

search step should be much larger than the positioning error of the mechanism, but it should not be so large that the peak region of the metric curve is skipped. Its selection should consider the focusing range, depth of focus, the width of the MTF defocus peak, image noise, and focusing time. Considering the ± 2 mm design travel of the proposed focusing mechanism, the coarse search step Δs_c can be set to 50–200 μm , with about 100 μm preferred in engineering applications.

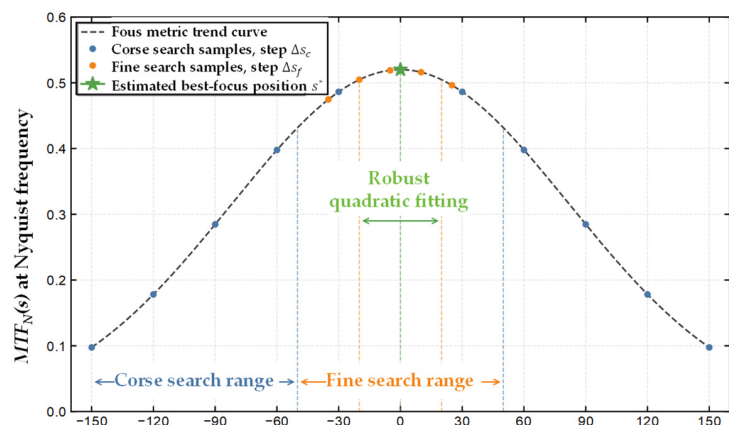
$$s_i^c = s_{min} + i\Delta s_c, i = 0, 1, \dots, N_c \quad (16)$$

where

$$N_c = \text{floor} \left[\frac{s_{max} - s_{min}}{\Delta s_c} \right] \quad (17)$$



(a)



(b)

Figure 2. MTF-based autofocusing strategy: (a) workflow of the autofocusing procedure; (b) coarse-to-fine search and local quadratic fitting for best focus estimation.

At each coarse search position, the controller calculates the target encoder count according to Equation (14) and drives the CCD focal plane assembly to the target position. After the position inner loop becomes stable, an image is acquired and $Q(s_i^c)$ is calculated. The coarse search peak is obtained as:

$$m = \arg \max_i Q(s_i^c) \quad (18)$$

$$s_m = s_{min} + m\Delta s_c \quad (19)$$

A fine search interval is then constructed around s_m . The fine search step should be matched to the repeatability of the mechanism, image quality noise, and focusing time. For the proposed mechanism, if the position loop is stable and the metric fluctuation is small, Δs_f can be set to 5–10 μm . If image noise is high or the target texture is weak, it can be relaxed to 10–20 μm to prevent metric fluctuation from masking the true peak trend. The fine search positions are:

$$s_j^f = s_m + j\Delta s_f, j = 1 - M, \dots, 0, \dots, M \quad (20)$$

After the fine search, local samples near the peak are used to fit a quadratic function. This step reduces the influence of the discrete sampling interval and provides a sub-step estimate of the best focus position. The fitting is applied only in the peak neighborhood and is not used as a global model of the complete defocus curve.

$$Q_s = as^2 + bs + c, a < 0 \quad (21)$$

where a , b , and c are the fitting coefficients, and $a < 0$ indicates that the metric has a single downward opening peak near the best focus position. Since image acquisition may be affected by noise, vibration, scene texture variation, or individual outliers, ordinary least squares fitting can be influenced by outliers. To improve the stability of focus estimation, a Huber robust loss is used for weighted quadratic fitting:

$$\min_{a,b,c} \sum_{j=1}^n p_{\delta} \left[Q_j - \left(a (\Delta s_j^f)^2 + b \Delta s_j^f + c \right) \right] \quad (22)$$

where $p_{\delta}(\cdot)$ is the Huber loss function and δ is the robustness threshold. In this study, local robust quadratic fitting is used only to reduce the influence of isolated fluctuations in the image quality metric. No separate outlier rejection procedure is claimed in the experimental validation.

The estimated best focus position is obtained from the vertex of the fitted curve:

$$s^* = -\frac{b}{2a} \quad (23)$$

If $a < 0$ and s^* lies inside the fine search interval, the fitted estimate is considered valid; otherwise, the fine search sample with the maximum metric value is used as the final focus position. After s^* is obtained, the controller converts it into the target encoder count and executes final positioning. The autofocus process is considered converged if:

$$|s_{actual} - s^*| \leq \varepsilon_s \quad (24)$$

Therefore, convergence in this study is determined by the encoder-based residual position error in Equation (24). The fitted peak is used when it lies inside the fine search interval; otherwise, the fine search sample with the maximum metric value is used for final positioning. The later imaging experiment is, therefore, interpreted as a practical verification of image quality recovery during focal plane compensation, rather than as a complete quantitative validation of every algorithmic branch shown in Figure 2.

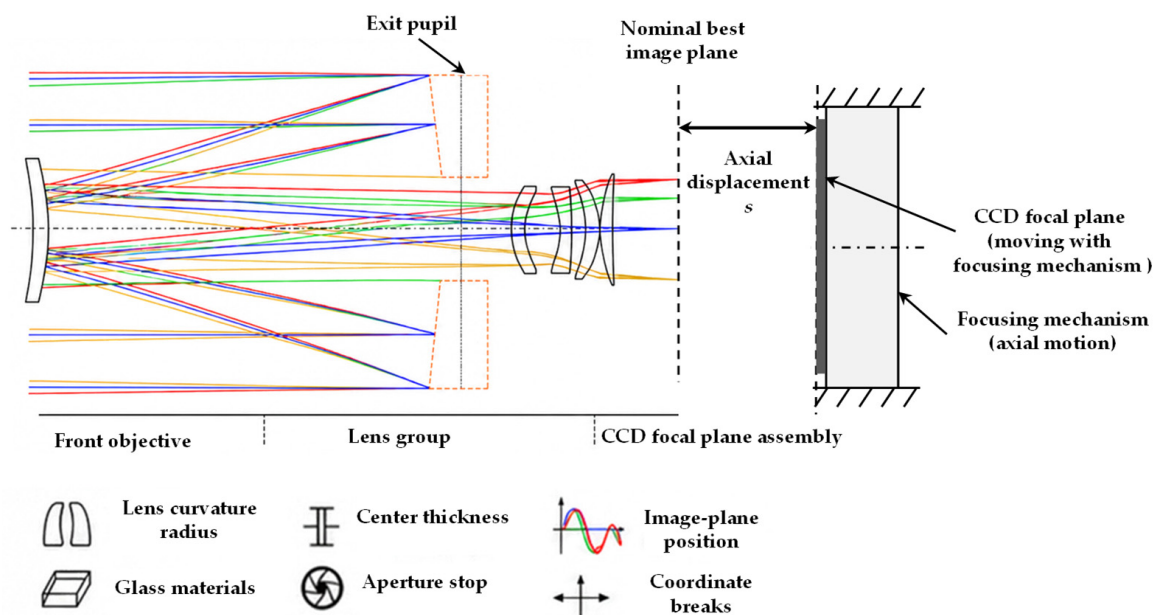
2.2. Numerical Simulation

2.2.1. MTF Simulation

To evaluate the effect of axial focal plane defocus on image quality, a sequential optical model was established in Zemax OpticStudio. During the simulation, the field points, wavelength set, aperture, and sampling parameters were kept unchanged, and only the axial position of the CCD focal plane was varied. The detailed simulation parameters, including their specific values, are listed in Table 1. This setting isolated the PSF and MTF changes caused by the prescribed focal plane displacement. The focal-plane displacement relative to the nominal best-image plane is denoted by s , as illustrated in Figure 3. Here, $s = 0 \mu\text{m}$ refers to the nominal best-image plane rather than the paraxial Gaussian image plane. The focal plane displacement relative to the nominal best-image plane is denoted by s , as illustrated in Figure 3. Here, $s = 0 \mu\text{m}$ does not refer to the paraxial Gaussian image plane but to the nominal best-image plane determined in the optical model. In Zemax OpticStudio, the image surface was axially adjusted until the average of the tangential and sagittal MTF values near the 50 lp/mm Nyquist frequency reached its maximum, while the corresponding local spot spreading remained small. This image surface was then used as the reference focal plane for the defocus simulation. The simulated focal plane displacements were 0, ± 50 , ± 100 , and $\pm 150 \mu\text{m}$. For each defocus condition, automatic refocusing was disabled, and only the axial position of the CCD focal plane was changed.

Table 1. Main parameters used in the MTF simulation.

Parameter	Symbol	Value or Description
CCD pixel pitch	p	$10\ \mu\text{m}$
Nyquist frequency	f_N	$50\ \text{lp/mm}$
Focal plane displacement	s	$0, \pm 50, \pm 100, \pm 150\ \mu\text{m}$
Spatial-domain output	FWHM	Width of the simulated PSF
Frequency-domain output	$\text{MTF}_N(s)$	MTF at $50\ \text{lp/mm}$
Directional MTF output	$\text{MTF}_T, \text{MTF}_S$	Tangential and sagittal MTF

**Figure 3.** Sequential optical model used for MTF simulation in Zemax OpticStudio, showing the nominal best-image plane, CCD focal plane, and prescribed axial displacement s of the focal plane assembly. The reported MTF-defocus simulation was performed at the central on-axis field.

For each focal plane displacement, the tangential and sagittal MTF values were recorded, and their average value was calculated as:

$$\overline{\text{MTF}}_{50}(s) = \frac{\text{MTF}_T(50, s) + \text{MTF}_S(50, s)}{2} \quad (25)$$

where $\text{MTF}_T(50, s)$ and $\text{MTF}_S(50, s)$ are the tangential and sagittal MTF values at $50\ \text{lp/mm}$ when the focal plane displacement is s . The simulation outputs included the PSF distributions, MTF frequency response curves, and the variation in MTF at $50\ \text{lp/mm}$ with focal plane displacement.

2.2.2. Finite Element Analysis

To evaluate the structural stability and dynamic stiffness of the focusing mechanism under launch-induced random vibration, a finite element model of the assembled mechanism was established in commercial software COMSOL Multiphysics® 6.4. As shown in Figure 4, the model was generated from the three-dimensional assembly model. The main load-transfer paths, including the base, supporting frame, encoder bracket, worm support, motor mounting structure, and mechanical interfaces, were retained. The refined mesh regions were determined according to three criteria: the main mechanical load-transfer path from the mounting base to the CCD focal plane assembly, geometric discontinuities such as mounting holes and abrupt section changes, and local stress-sensitive regions iden-

tified from preliminary modal and stress-sensitivity inspections. Based on these criteria, local mesh refinement was applied around the base mounting holes, encoder bracket and transmission support, worm gear support, motor interface, and other supporting transition regions. Representative refined mesh regions are indicated in Figure 4. Non-critical small geometric features were simplified while preserving their equivalent mass, centroid position, and global stiffness contribution.

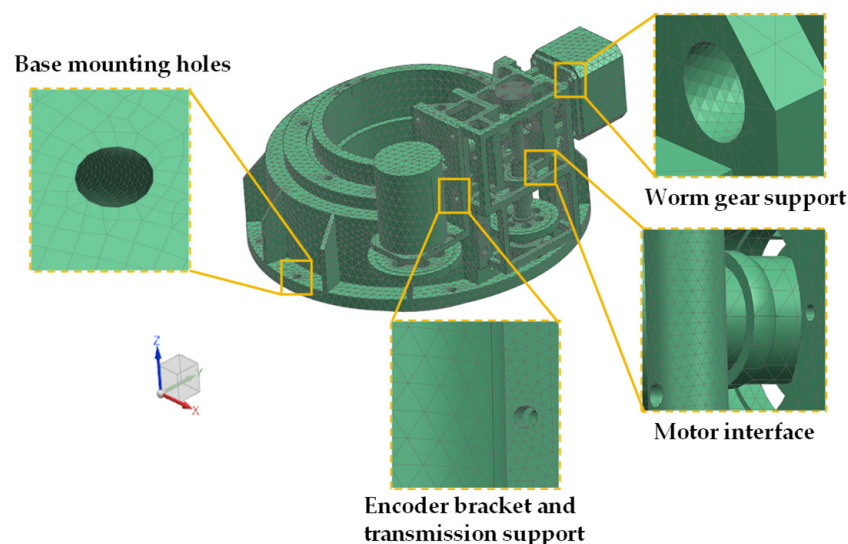


Figure 4. Finite element model and representative local mesh-refinement regions of the focusing mechanism. The enlarged views show refined mesh areas at the base mounting holes, encoder bracket and transmission support, worm gear support, and motor interface.

Modal analysis was performed with the focusing base fixed to reproduce the actual mounting constraint. The first six natural frequencies and mode shapes were extracted to identify low-order flexible regions and dynamic-response-sensitive components. The design requirement for the first natural frequency was higher than 100 Hz. Random vibration analysis was then conducted using the acceleration power spectral density input listed in Table 2 over 20–2000 Hz. A damping ratio of 0.03 was used, and excitations were applied in the X, Y, and Z directions, where X and Y are the radial directions and Z is the vertical direction.

Table 2. Random vibration input conditions.

Frequency range/Hz	20–50	50–100	100–240	240–500	500–2000
Power spectral density	+3 dB/oct	0.4 g ² /Hz	−9 dB/oct	0.029 g ² /Hz	−9 dB/oct
Overall RMS acceleration ¹			7.7 Grms		
Directions			X, Y, and Z		

¹ Root Mean Square (RMS).

2.3. Experimental Setup

2.3.1. Focusing Accuracy Experiment

The experimental setup is shown in Figure 5. The focusing mechanism was horizontally fixed on a dedicated fixture, and the dial indicator probe was aligned with the axial motion direction of the CCD focal plane assembly to measure the actual displacement. Before the test, the initial encoder count was recorded, and the corresponding axial displacement was defined as 0 μm.

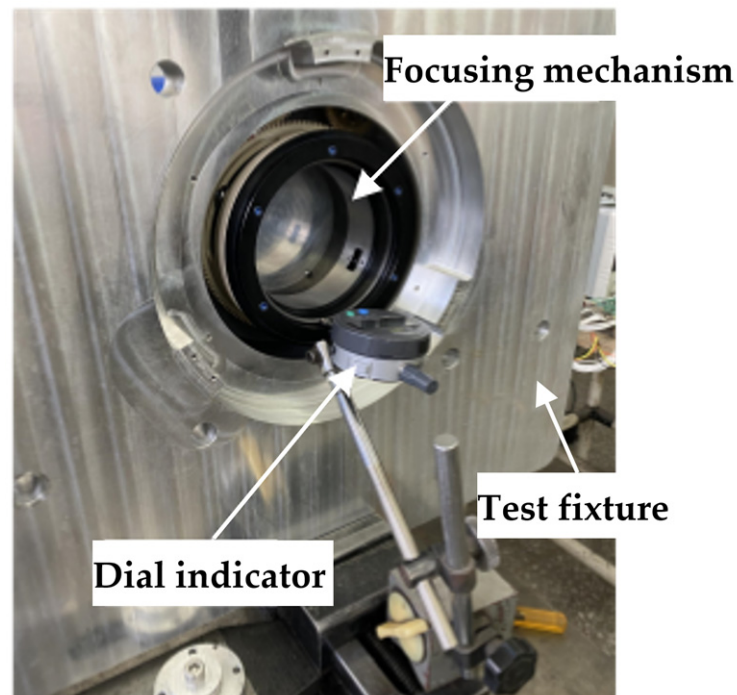


Figure 5. Experimental setup for the focusing accuracy test.

2.3.2. Autofocusing Verification Method

The experimental setup is shown in Figure 6. The camera and the resolution target were adjusted so that the evaluated stripe region was located near the image center. Therefore, the autofocus verification was evaluated at the central near-axis field of the camera. For all focal-plane positions, the same fixed central-field ROI was used, and illumination, exposure time, camera gain, and image acquisition region were kept unchanged. This setting ensured that the same field point and the same target features were compared during the defocusing and refocusing process. During the experiment, illumination, exposure time, camera gain, and the image acquisition region were kept unchanged to avoid non-focusing factors affecting the image quality evaluation.

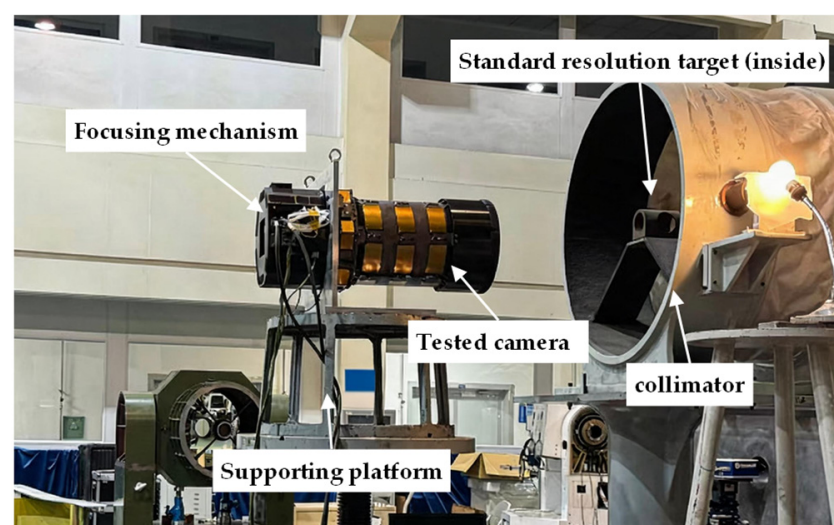


Figure 6. Experimental setup for autofocus process verification.

The focal plane displacement relative to the nominal best focus plane is denoted by s , where $s = 0 \mu\text{m}$ represents the nominal best focus position. Representative sampling

positions of $s = -150, -100, -50, 0, 50, 100$, and $150 \mu\text{m}$ were selected to observe the image quality variation when the focal plane moved from a defocused state toward the best focus plane and then away from it. For each sampling position, the controller converted the target displacement into the corresponding target encoder count according to the displacement-count mapping relationship. The CCD focal plane assembly was then driven to the target position by the encoder-based position-control loop. At each target position, the target image was acquired only after the encoder feedback entered the $\pm 3 \mu\text{m}$ position deadband and no further motor pulse was commanded, ensuring that the image was recorded under a stable stationary condition. A fixed region of interest was used for image evaluation to ensure that the same target features were compared at different focal plane positions. For quantitative comparison, the same central-field region of interest (ROI) was used at all focal-plane positions. A black-and-white stripe group with the same position, size, and spatial frequency was selected in each image. The mean gray values of the bright and dark stripes were calculated as $I_{\text{white}}(s)$ and $I_{\text{black}}(s)$, respectively. The stripe contrast was calculated as:

$$C(s) = \frac{I_{\text{white}}(s) - I_{\text{black}}(s)}{I_{\text{white}}(s) + I_{\text{black}}(s)} \quad (26)$$

and then normalized by the value at the nominal best-focus position:

$$C_{\text{norm}}(s) = \frac{C(s)}{C(0)} \quad (27)$$

Mean gray values were used instead of single maximum or minimum pixel values to reduce the influence of isolated noisy pixels. This indicator represents the relative contrast recovery of the selected resolution-target stripe group rather than an independently calibrated absolute system MTF. If the target edges or bar patterns satisfy the requirement for frequency response analysis, MTF or spatial frequency response can be extracted as the image quality metric. For rapid verification of the focusing process, a normalized sharpness function $Q(s)$ can also be used. In this section, the target image sequence is used to demonstrate the improvement in image quality caused by axial focal plane compensation.

2.3.3. Mechanical Environment Test

To validate the dynamic stiffness predicted by the finite element model and to assess the structural stability after random vibration loading, sweep frequency, random vibration, and post-vibration sweep tests were conducted. The specimen was rigidly fixed to an electrodynamic shaker using the dedicated adapter fixture shown in Figure 7 to reproduce the actual mounting constraint. Piezoelectric accelerometers were attached to key locations, including the base and main supporting frame, to measure the structural dynamic response. The sweep frequency test was used to capture the first natural frequency of the focusing mechanism and to compare it with the modal analysis result. Random vibration tests were performed along the X, Y, and Z directions using the same input spectrum as that listed in Table 2. The response curves were recorded, and the maximum total RMS acceleration response and the corresponding amplification ratio were extracted for each direction.

After random vibration testing, a post-vibration sweep was performed. The frequency response curves before and after random vibration were compared to examine whether the loading caused joint loosening, stiffness degradation, or modal frequency shift. This pre- and post-vibration frequency response comparison is commonly used in mechanical verification of space mechanisms. Similar studies on space payloads and space mechanisms also use low-level sine sweep or post-test frequency checks to assess structural integrity after vibration loading [20].

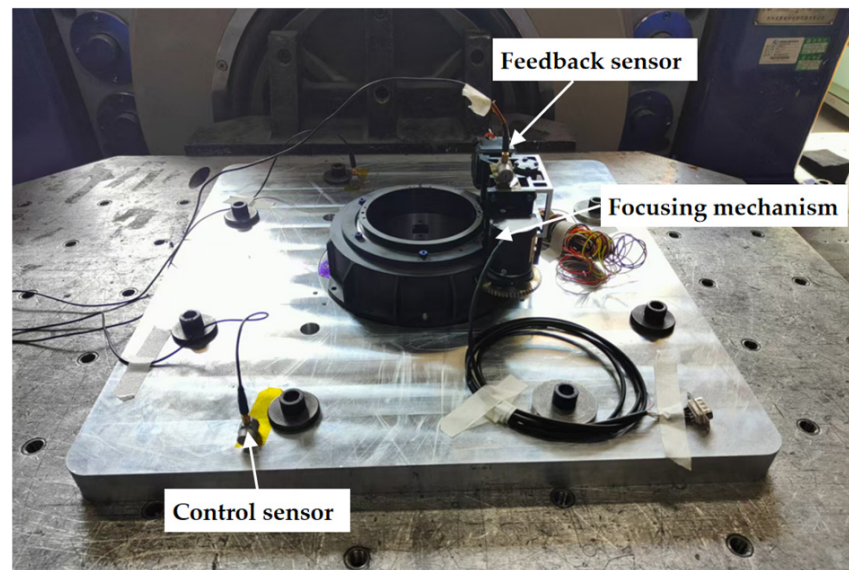


Figure 7. Experimental setup for the mechanical environment test.

The random vibration response was evaluated using two metrics: maximum equivalent stress and acceleration amplification ratio. The acceleration amplification ratio η was defined as the ratio between the total RMS acceleration response at a critical location and the input total RMS acceleration. A criterion of $\eta < 3$ was adopted to screen excessive local dynamic amplification at critical response locations [21].

$$\eta = \frac{a_{RMS, response}}{a_{RMS, input}} < 3 \quad (28)$$

where η is the acceleration amplification ratio, $a_{RMS, response}$ is the total RMS acceleration response at the critical location, and $a_{RMS, input}$ is the input total RMS acceleration.

3. Results and Discussion

3.1. Focusing Accuracy Results and Error Analysis

The displacement test results of the focusing mechanism, including the comparison between the theoretical and measured displacements, are shown in Figure 8. Within the test range of ± 2 mm, the measured displacement varies linearly with the encoder count and agrees well with the theoretical displacement. Linear fitting of the test data shows that the fitted slope is close to unity, indicating that the transmission model can accurately describe the relationship between the encoder count and the axial displacement of the CCD focal plane assembly.

The positioning error is calculated as:

$$e_i = s_{m,i} - s_{t,i} \quad (29)$$

where $s_{m,i}$ is the measured displacement at the i -th test point, $s_{t,i}$ is the corresponding theoretical displacement, and e_i is the positioning error. The error curve is shown in Figure 8b. According to the test data, the maximum absolute positioning error is $7 \mu\text{m}$, the mean error is $1.52 \mu\text{m}$, the mean absolute error is $3.52 \mu\text{m}$, and the root-mean-square error is $4.17 \mu\text{m}$. The positioning errors of all test points are within the required accuracy of $\pm 7.5 \mu\text{m}$, demonstrating that the proposed focusing mechanism satisfies the closed-loop control positioning accuracy requirement.

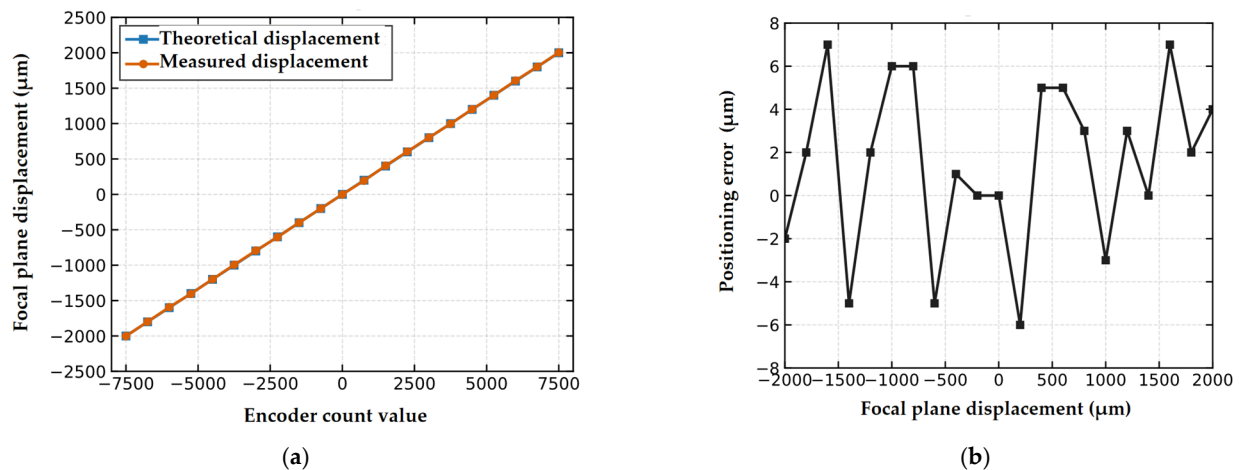


Figure 8. Focusing accuracy test results: (a) theoretical and measured focal plane displacements; (b) positioning error of the focusing mechanism.

The error distribution fluctuates in both positive and negative directions, without a clear monotonic trend over the full travel range. This indicates that the positioning error is not mainly caused by a systematic scale error in the transmission model. Instead, it is more likely related to local transmission errors, gear backlash, cam profile error, assembly error, dial indicator reading error, and initial zero-setting error. The fitting result also shows that the scale error is small, while the zero offset and local mechanical errors are the main factors affecting the positioning accuracy. To further clarify the error sources, an error budget was established for the focusing mechanism, as summarized in Table 3.

Table 3. Error budget for the focusing mechanism.

Error Source	Influence on Positioning Accuracy	Evaluation Method
Motor step resolution	Determines the minimum controllable displacement	Calculated from the transmission model
Encoder identification error	Affects position feedback accuracy	Converted from encoder angular accuracy
Gear backlash and hysteresis	Affects bidirectional positioning consistency	Forward and reverse positioning tests
Cylindrical cam profile error	Affects displacement linearity	Cam profile inspection or residual analysis
Assembly and guiding error	Introduces tilt, runout, or local motion resistance	Assembly inspection and full-travel test
Dial indicator measurement error	Affects measured displacement values	Instrument accuracy and calibration certificate
Initial zero-setting error	Introduces global displacement offset	Repeated zero-setting tests

The total positioning error is mainly composed of the motor step resolution error, encoder identification error, gear transmission error, cylindrical cam profile error, assembly and guiding error, measurement error, and zero-setting error. If we assume these error sources are approximately independent, the total error can be estimated using the root-sum-square method:

$$e_{RSS} = \sqrt{e_s^2 + e_e^2 + e_g^2 + e_c^2 + e_a^2 + e_m^2 + e_0^2} \quad (30)$$

where e_s is the error caused by the motor step resolution, e_e is the encoder identification error, e_g is the gear transmission error, e_c is the cylindrical cam profile error, e_a is the assembly and guiding error, e_m is the measurement error, and e_0 is the zero-setting error. According to the theoretical analysis, the single-step focusing control resolution is $1.38\text{ }\mu\text{m}$, and the equivalent encoder-based displacement measurement accuracy is approximately $0.486\text{ }\mu\text{m}$. Both values are much smaller than the required positioning accuracy of $\pm 7.5\text{ }\mu\text{m}$. Therefore, the micrometer-level errors observed in the experiment are mainly attributed to mechanical transmission, assembly, and measurement factors.

3.2. Optical Defocus Response and Autofocusing Verification

The optical defocus response and autofocusing verification in this section correspond to the central near-axis field. The PSF and MTF results in Figure 9 correspond to the central on-axis field, with $s = 0\text{ }\mu\text{m}$ defined as the nominal best-image plane determined in Section 2.2.1. The axial focal plane displacement was varied from -150 to $+150\text{ }\mu\text{m}$, corresponding to a total simulated compensation range of approximately $300\text{ }\mu\text{m}$. Figure 9a shows the PSF energy distributions at different defocus positions. When $s = 0\text{ }\mu\text{m}$, the optical system is located at the nominal best focus position. The PSF is highly concentrated, and the FWHM is approximately $2.5\text{ }\mu\text{m}$. As $|s|$ increases, the PSF gradually broadens and the energy concentration decreases. At $s = \pm 150\text{ }\mu\text{m}$, the FWHM increases to approximately $18.0\text{ }\mu\text{m}$, indicating a significant spreading of the point image caused by axial defocus. Figure 9b presents the normalized MTF curves under different defocus conditions. The low spatial frequency response changes only slightly with defocus, whereas the mid and high spatial frequency responses are more sensitive to the axial position of the CCD focal plane. Since the CCD pixel pitch is $10\text{ }\mu\text{m}$, the Nyquist frequency is 50 lp/mm . Therefore, the MTF at 50 lp/mm was selected as the main frequency domain indicator for evaluating the influence of defocus. Figure 9c shows the tangential, sagittal, and averaged MTF values at 50 lp/mm as functions of the focal plane displacement. All three curves reach their maximum values near $s = 0\text{ }\mu\text{m}$ and decrease as $|s|$ increases. At the nominal best focus position, the MTF at 50 lp/mm is approximately 0.48 – 0.49 . This value is higher than the reference level of 0.40 used in this study. The value 0.40 is adopted as an engineering reference criterion for acceptable imaging quality at the detector Nyquist frequency, indicating that the system still preserves about 40% contrast transfer capability at the limiting sampling frequency [22,23]. It should be noted that this value is used as a reference image quality level rather than a universal physical threshold, since the required MTF depends on the mission resolution, detector sampling, target type, and image processing strategy.

With increasing defocus, the MTF at 50 lp/mm decreases markedly. At $s = \pm 150\text{ }\mu\text{m}$, the MTF decreases to approximately 0.17 – 0.20 , showing that large axial focal plane errors strongly suppress high-frequency image information. When the displacement is reduced from $\pm 150\text{ }\mu\text{m}$ to approximately $\pm 50\text{ }\mu\text{m}$, the PSF broadening is significantly weakened, and the MTF at 50 lp/mm approaches the reference image quality level. This result indicates that a coarse search over a relatively wide defocus range can effectively move the focal plane into the neighborhood of acceptable imaging quality. After this stage, a smaller search step is required to further approach the peak image quality region near $s = 0\text{ }\mu\text{m}$.

Figure 10 shows representative target images acquired at different focal plane displacements. The image sequence exhibits a clear focus-dependent change in edge sharpness and central detail. At $|s| = 150\text{ }\mu\text{m}$, the target is obviously defocused, and fine structures are poorly resolved. At $|s| = 100\text{ }\mu\text{m}$, the main features begin to recover, but edge spreading remains visible. At $|s| = 50\text{ }\mu\text{m}$, most bar patterns are distinguishable, and the local edge contrast improves. Near $s = 0\text{ }\mu\text{m}$, the black-and-white bar edges are sharpest and the

central details are most complete, indicating that the CCD focal plane assembly is close to the best focus position. The observed image sequence is consistent with the previous MTF simulation and displacement control results. When the focal plane is far from the best focus position, the broadened point image reduces the high-frequency texture information of the target. When the focal plane approaches the best focus position, the energy concentration increases, and the target edges and fine structures are recovered. These results demonstrate that the proposed focusing mechanism can perform effective axial focal plane compensation according to encoder feedback and can improve the target image quality under practical imaging conditions. The normalized stripe contrast extracted from the selected central-field ROI is shown in Figure 10b. The value of C_{norm} reached 1.000 at $s = 0 \mu\text{m}$, where the stripe edges were the sharpest and the local contrast was the highest. When the focal plane was displaced from the best-focus position, C_{norm} decreased to approximately 0.31 at $\pm 50 \mu\text{m}$ and approximately 0.29 at $\pm 100 \mu\text{m}$. At $\pm 150 \mu\text{m}$, C_{norm} decreased further to approximately 0.016, indicating that the stripe contrast was strongly suppressed under large defocus. This quantitative trend is consistent with the visual change in the captured resolution-target images and with the MTF degradation predicted by the Zemax simulation. Therefore, the ground experiment verifies the relative image-contrast recovery trend during focal-plane compensation, while the absolute MTF-defocus relationship is provided by the optical simulation.

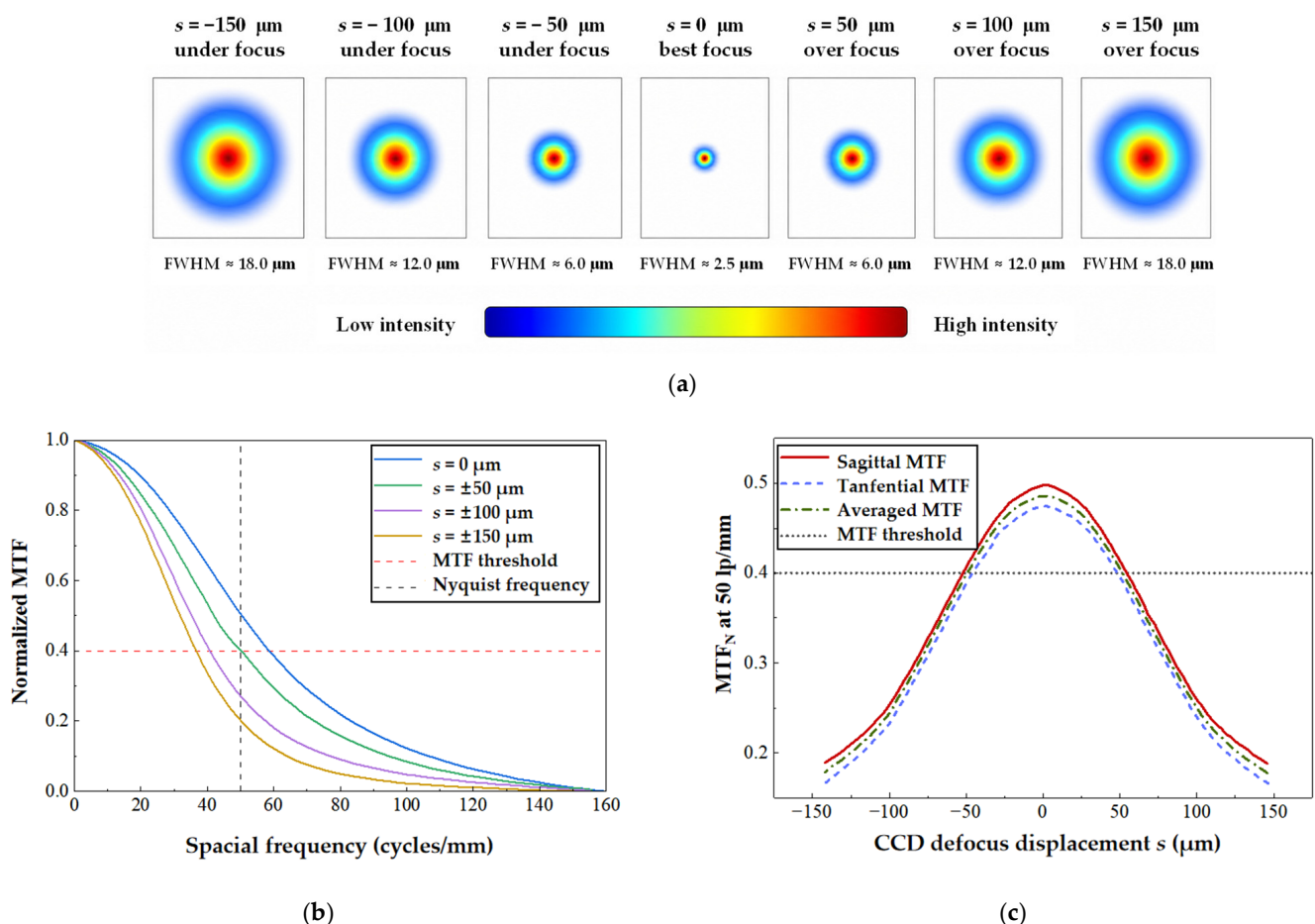


Figure 9. PSF and MTF simulation results under CCD focal plane defocus: (a) PSF energy distributions; (b) normalized MTF curves; (c) tangential, sagittal, and average MTF at 50 lp/mm.

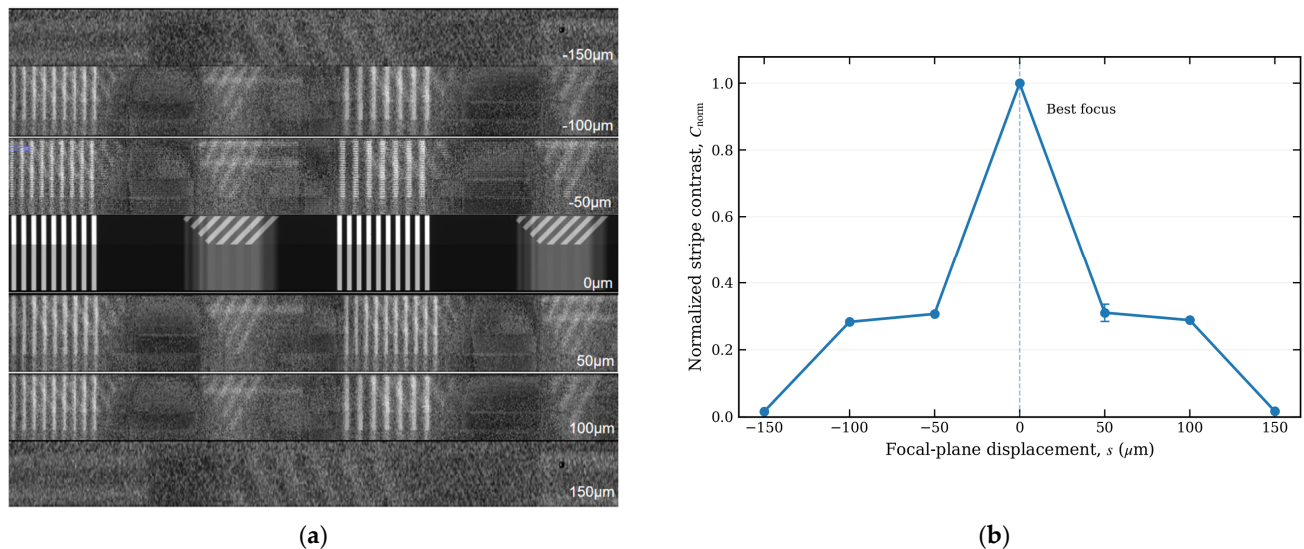


Figure 10. Resolution-target imaging verification at different focal-plane displacements: (a) captured target images; (b) normalized stripe contrast extracted from the same fixed central-field ROI.

Overall, the autofocusing process verification experiment shows that the axial displacement compensation of the CCD focal plane assembly can effectively change image sharpness. As the focal plane moves from a defocused position to the best focus region, the target edges and fine details are gradually recovered. After passing the best focus position, the image quality decreases again. This result verifies the closed-loop positioning capability of the focusing mechanism and the rationality of autofocusing based on image-quality feedback from the perspective of practical imaging.

3.3. Mechanical Properties and Environmental Validation

The first six natural frequencies and the corresponding dominant vibration modes are shown in Figure 11. The first natural frequency obtained from the finite element analysis is 163.98 Hz, which is higher than the design requirement of 100 Hz. The low-order modes are mainly associated with local rotations of the encoder, bracket, worm support, and motor, rather than global low-frequency deformation of the main supporting structure or the CCD focal plane assembly. This indicates that the main load-bearing frame has relatively high stiffness, while the low-order flexibility is concentrated around externally mounted components, transmission supports, and local interfaces. The modal analysis is used here as a dynamic stiffness verification rather than only a frequency listing. The simulated first natural frequency gives a margin of approximately 64% relative to the 100 Hz requirement, and the mode shapes indicate that the dominant flexibility is concentrated in local transmission and support regions rather than in the main load-bearing frame.

The sweep frequency test result is shown in Figure 12. The measured first natural frequency is 174.64 Hz, which also satisfies the design requirement of 100 Hz. Using the test result as the reference, the relative deviation of the simulated first natural frequency is 6.1%. The simulated value is slightly lower than the measured value, indicating that the finite element model gives a conservative estimate of the global stiffness. This difference may be related to the equivalent treatment of joint stiffness, bolt preload, local contact conditions, material parameters, and the constraint stiffness of the test fixture. Overall, the simulation and test results show good consistency and support the use of the finite element model for identifying response trends in the subsequent random vibration assessment.

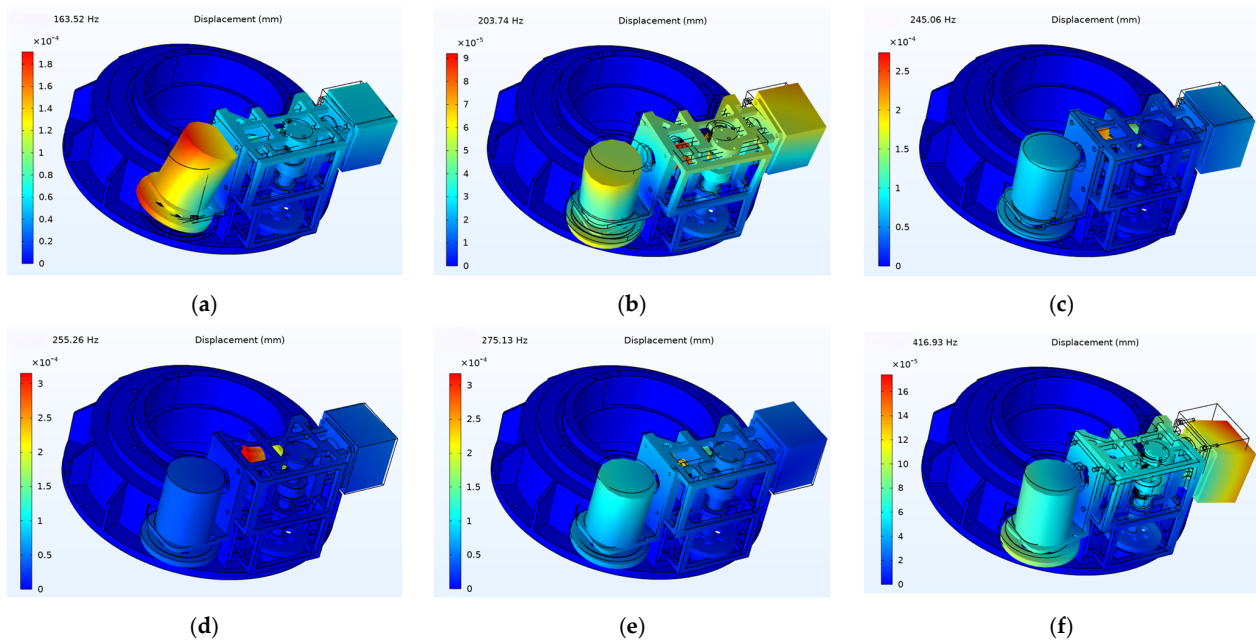


Figure 11. First six mode shapes of the focusing mechanism: (a) first mode; (b) second mode; (c) third mode; (d) fourth mode; (e) fifth mode; (f) sixth mode.

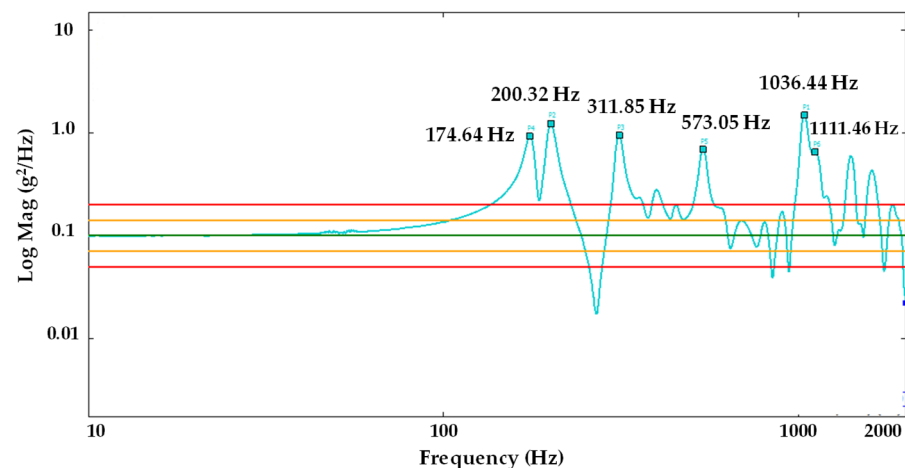


Figure 12. Sweep frequency test result.

Random vibration simulation and testing were both conducted along the X, Y, and Z directions. The finite element simulation was mainly used to evaluate the stress distribution and local strength risk under random vibration loading, while the vibration test was used to verify the acceleration response and environmental adaptability of the assembled mechanism.

The simulation results in Figure 13 show that the maximum equivalent stresses under X, Y, and Z direction excitations are 37.96 MPa, 32.93 MPa, and 43.38 MPa, respectively. The maximum stresses mainly occur around local joints and supporting transition regions, indicating that the random vibration load is transferred through the mounting and supporting structures and induces local stress concentration. Taking the yield strength of 7075-T6 aluminum alloy as 503 MPa [24], the minimum yield safety factor is calculated from the maximum simulated stress as $n = 503/43.38 = 11.6$. This result indicates that the stress level is well below the yield strength of 7075 aluminum alloy under the simulated random vibration condition. If a different 7075 temper or material state is used, the material allowable stress should be updated accordingly.

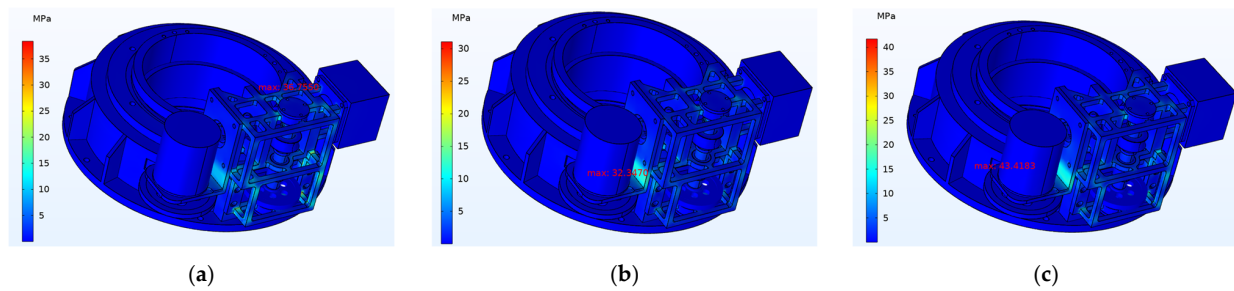


Figure 13. Stress responses from random vibration simulation: (a) X-direction excitation; (b) Y-direction excitation; (c) Z-direction excitation.

The random vibration test results in Figure 14 show that the maximum acceleration responses in the X, Y, and Z directions are 16.6778 Grms, 16.1567 Grms, and 14.1202 Grms, respectively. The corresponding amplification ratios are 2.16, 2.10, and 1.83. All measured amplification ratios are lower than the criterion value of 3, indicating that no unacceptable dynamic amplification occurs under the random vibration test condition. The X and Y direction responses are close to each other and are higher than the Z direction response, which suggests that the radial constraints and local transmission support regions are more sensitive to vibration excitation. This observation is consistent with the modal analysis, where the low-order modes are concentrated around the encoder, bracket, and worm support regions.

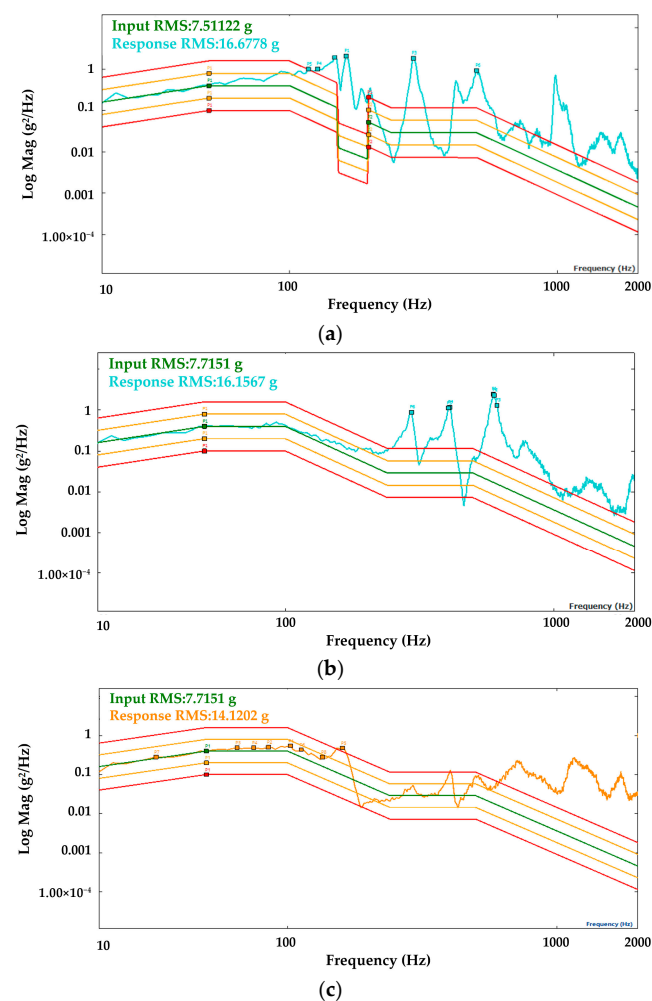


Figure 14. Measured acceleration response curves under random vibration: (a) X-direction excitation; (b) Y-direction excitation; (c) Z-direction excitation.

After the random vibration test, a post-vibration sweep was performed. The frequency response curves before and after random vibration are shown in Figure 15. The main response peaks remain nearly unchanged, and no obvious frequency shift or abnormal response change is observed. This result, together with the post-test inspection showing no visible deformation, loosening, or damage, indicates that random vibration loading did not cause global stiffness degradation or joint degradation. Therefore, the mechanical validation is supported by three consistent observations: sufficient modal frequency margin, low simulated stress level with a yield safety factor of 11.6 for 7075-T6 aluminum alloy, and stable pre- and post-vibration sweep responses. No axial displacement drift, visible loosening, or mechanical damage of the CCD focal plane assembly was observed after the random vibration test. The stable pre- and post-vibration sweep responses also indicate that the transmission chain and supporting interfaces did not experience detectable stiffness degradation or irreversible slippage. These observations support the effectiveness of the power-off holding and interlocking design.

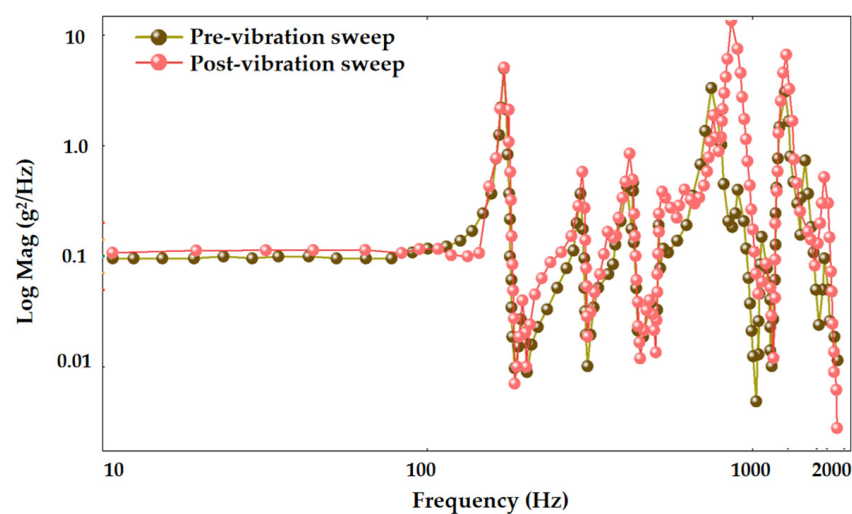


Figure 15. Comparison of sweep frequency responses before and after random vibration testing.

4. Conclusions

This study proposed and verified a compact encoder-based cylindrical cam focusing mechanism for focal plane compensation in a space optical camera. The mechanism converts motor rotation into axial motion of the CCD focal plane assembly and uses encoder feedback for closed-loop control positioning. The positioning experiment showed a maximum absolute error of 7 μm within a ± 2 mm travel range, satisfying the ± 7.5 μm requirement. The MTF simulation showed that axial defocus broadens the PSF and reduces the MTF near the 50 lp/mm Nyquist frequency, supporting the need for accurate focal plane compensation. The imaging verification experiment further showed that the normalized stripe contrast reached its maximum near the best-focus position and decreased under bidirectional defocus, confirming the relative image-quality recovery trend during focal-plane compensation. Mechanical validation showed a measured first natural frequency of 174.64 Hz, a maximum simulated equivalent stress of 43.38 MPa, a yield safety factor of 11.6 based on 7075-T6 aluminum alloy, and random vibration amplification ratios below 3 in all tested directions. The pre- and post-vibration sweep curves showed no obvious structural degradation. These results indicate that the proposed mechanism is feasible for compact focal plane compensation in space optical cameras from the viewpoints of positioning accuracy, MTF defocus sensitivity, practical imaging recovery, and mechanical environmental adaptability.

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Abbreviations

The following abbreviations are used in this manuscript:

CCD	Charge-coupled device
MTF	Modulation transfer function
PSF	Point spread function
FWHM	Full width at half maximum
RMS	Root mean square
PSD	Power spectral density
ROI	Region of interest
SFR	Spatial frequency response
TMA	Three mirror anastigmat
Grms	Root mean square acceleration in units of g
lp/mm	Line pairs per millimeter

References

1. Zhao, Q.; Yu, L.; Du, Z.; Peng, D.; Hao, P.; Zhang, Y.; Gong, P. An Overview of the Applications of Earth Observation Satellite Data: Impacts and Future Trends. *Remote Sens.* **2022**, *14*, 1863. [\[CrossRef\]](#)
2. Gadisa, D.; Bang, H. Small Satellite Electro-Optical System (EOS) Technological and Commercial Expansion. *Acta Astronaut.* **2023**, *213*, 355–372. [\[CrossRef\]](#)
3. de Sousa, F.L. Are smallsats taking over bigsats for land Earth observation? *Acta Astronaut.* **2023**, *213*, 455–463. [\[CrossRef\]](#)
4. Crisp, N.H.; Roberts, P.C.E.; Livadiotti, S.; Oiko, V.T.A.; Edmondson, S.; Haigh, S.J.; Huyton, C.; Sinpetru, L.; Smith, K.; Worrall, S.; et al. The benefits of very low earth orbit for earth observation missions. *Prog. Aerosp. Sci.* **2020**, *117*, 100619. [\[CrossRef\]](#)
5. Li, W.; Lv, Q.; Zhao, N.; Wang, J.; Liu, Y.; Zheng, T. Adaptive thermal refocusing system for a high-resolution space camera. *Appl. Opt.* **2022**, *61*, 699–709. [\[CrossRef\]](#) [\[PubMed\]](#)
6. Wang, Y.; Zhong, X.; Qu, Z.; Li, L.; Wu, S.; Zeng, C. On-Orbit Modulation Transfer Function Estimation Based on the Refined Image Kernel. *Sensors* **2023**, *23*, 4362. [\[CrossRef\]](#) [\[PubMed\]](#)
7. Valenzuela Quinteros, A.; Reinke, K.; Jones, S. A new metric for the assessment of spatial resolution in satellite imagers. *Int. J. Appl. Earth Obs. Geoinf.* **2022**, *114*, 103051. [\[CrossRef\]](#)
8. Lu, Z.; Shen, X.; Li, D.; Cheng, S.; Wang, J.; Yao, W. Super-agile satellites imaging mission planning method considering degradation of image MTF in dynamic imaging. *Int. J. Appl. Earth Obs. Geoinf.* **2024**, *131*, 103968. [\[CrossRef\]](#)
9. Boreman, G.D. *Modulation Transfer Function in Optical and Electro-Optical Systems*, 2nd ed.; SPIE: Bellingham, WA, USA, 2021. Available online: <https://spie.org/Publications/Book/2581372> (accessed on 13 May 2026).

10. Koski, K. Focus Mechanism for Kepler Mission. In *Proceedings of the 39th Aerospace Mechanisms Symposium*; NASA Marshall Space Flight Center: Huntsville, AL, USA, 2008.
11. Bell, J.F.; Maki, J.N.; Mehall, G.L.; Ravine, M.A.; Caplinger, M.A.; Bailey, Z.J.; Brylow, S.; Schaffner, J.A.; Kinch, K.M.; Madsen, M.B.; et al. The Mars 2020 Perseverance Rover Mast Camera Zoom (Mastcam-Z) Multispectral, Stereoscopic Imaging Investigation. *Space Sci. Rev.* **2021**, *217*, 24. [[CrossRef](#)] [[PubMed](#)]
12. Rizk, B.; Drouet d'Aubigny, C.; Golish, D.; Fellows, C.; Merrill, C.; Smith, P.; Walker, M.S.; Hendershot, J.E.; Hancock, J.; Bailey, S.H.; et al. OCAMS: The OSIRIS-REx Camera Suite. *Space Sci. Rev.* **2018**, *214*, 26. [[CrossRef](#)]
13. Jia, B.; Wang, L.; Wang, R.; Jin, J.; Wu, D. Theoretical modeling and experimental investigation on a novel screwed-type piezoelectric focusing mechanism for space cameras. *Mech. Syst. Signal Process* **2022**, *171*, 108844. [[CrossRef](#)]
14. Lu, Y.; Chen, C.; Zhu, J. Design of a Focusing Mechanism Actuated by Piezoelectric Ceramics for TMA Telescope. *Sensors* **2023**, *23*, 4610. [[CrossRef](#)] [[PubMed](#)]
15. Li, Y.; Ge, W.; Zhang, X.; Tong, X. Optimization and Experiment of a Novel Compliant Focusing Mechanism for Space Remote Sensor. *Sensors* **2020**, *20*, 6826. [[CrossRef](#)] [[PubMed](#)]
16. Huo, Z.; Li, G.; Tan, L.; Yang, T.; Tian, D.; Li, J. Optimal Design of High-Precision Focusing Mechanism Based on Flexible Hinge. *Machines* **2024**, *12*, 627. [[CrossRef](#)]
17. Peng, Y.; Chen, C.; Bi, G.; Sun, B.; Sha, W. On-Orbit Auto-Focusing Method of Space Camera Based on Multistar Image. *IEEE Trans. Geosci. Remote Sens.* **2024**, *62*, 5636121. [[CrossRef](#)]
18. Yan, W.; Liu, J.; Ren, X.; Li, C.; Fu, Q.; Wang, D.; Dong, J.; Zhang, X.; Chen, W.; Tan, X.; et al. Detection Capability Verification and Performance Test for the High Resolution Imaging Camera of China's Tianwen-1 Mission. *Space Sci. Rev.* **2021**, *217*, 71. [[CrossRef](#)]
19. Yu, J.; Zhang, Y.; Qi, B.; Bai, X.; Wu, W.; Liu, H. Analysis of the Slanted-Edge Measurement Method for the Modulation Transfer Function of Remote Sensing Cameras. *Appl. Sci.* **2023**, *13*, 13191. [[CrossRef](#)]
20. Yang, Z.; Li, C.; Hu, B.; Yin, Z.; Cai, Y. Design and verification of space shutter mechanism. *J. Phys. Conf. Ser.* **2025**, *3032*, 012047. [[CrossRef](#)]
21. Calvi, A. Reduction of overtesting during base-drive random vibration tests for the euclid spacecraft hardware. In *Proceedings of the 4th International Conference on Computational Methods in Structural Dynamics and Earthquake Engineering (COMPdyn 2013)*, Kos Island, Greece; Institute of Structural Analysis and Antiseismic Research School of Civil Engineering National Technical University of Athens (NTUA) Greece: Athens, Greece, 2014; pp. 167–180. [[CrossRef](#)]
22. Zhong, X.; Wang, T.; Yan, J.; Jin, G. Aberrations correction of tilted plane-parallel plate in convergent rays. *Appl. Opt.* **2015**, *54*, 1758–1764. [[CrossRef](#)]
23. Yang, S.; Zhang, Y.; Zhao, B.; Meng, Y.; Ren, D.; Zhang, J.; Sun, G.; Du, Z.; Zhang, G. Simulation method for multi-source information fusion space target. *Opt. Express* **2024**, *32*, 21820–21836. [[CrossRef](#)] [[PubMed](#)]
24. Aluminum 7075-T6; 7075-T651. Available online: <https://matweb.com/search/DataSheet.aspx?MatGUID=4f19a42be94546b686bbf43f79c51b7d&ckck=1> (accessed on 27 May 2026).

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