Optimization design of compliant focusing mechanism for space optical camera with light weight

Yan Li\textsuperscript{1,2}, Bo Liu\textsuperscript{1}, Wenjie Ge\textsuperscript{1} and Xinxing Tong\textsuperscript{3}\textsuperscript{10}

Abstract
Thermal radiation and vacuum lead to the deformation of bearing cylinder with carbon fiber composites which generates the defocus phenomenon for space optical camera. As heavier weight and higher complexity of the rigid focusing mechanism for the space optical camera, a novel compliant focusing mechanism with lighter weight and simple structure was designed and tested in this paper. In order to achieve stroke range and ensure image quality for the compliant focusing mechanism, micro driving displacement from the piezoelectric ceramics was amplified by two-stage flexible hinge lever-type mechanism. Taking the size parameters of the flexure hinges and beams as the design variables, an optimization model of the compliant focusing mechanism was established in which minimizing weight was considered as objection function under the stroke requirements. Consequently, optimal structure parameters of the compliant focusing mechanisms under different input forces can be obtained under the allowable stress and lighter weight. The effectiveness of the proposed compliant focusing mechanism was verified by the simulations and experiments. The results show that the compliant focusing mechanism can achieve a stroke of 2 mm for the focal plane assembly. The proposed method provided a new idea to design the focusing mechanism with lighter weight, simpler structure and higher reliability.

Keywords
Compliant mechanism, focusing mechanism, amplifier, space optical camera

Date received: 7 June 2022; accepted: 27 July 2022
Handling Editor: Chenhui Liang

Introduction
Space optical camera plays a very important role in the fields of resource survey, disaster warning and meteorological observation. The target image should be accurately imaged on the sensor of the space optical camera, which ensures the high-resolution image. Thermal radiation and vacuum in space environment affect the deformation of the bearing cylinder with carbon fiber composites, which results in defocusing phenomenon of the optical camera. Therefore, focusing mechanism is an important component to obtain clear image for the space optical camera.

Traditional focusing mechanism composed of the rigid mechanisms are usually used to adjust the focal plane. Such as, the 6-DOF parallel mechanism was used in the Gaia space telescope with a positioning accuracy of 5 \(\mu\)m and a resolution of 0.2 \(\mu\)m.\textsuperscript{1,2} Stepper motors and non-back drivable linear actuators were employed in the focus and alignment mechanism which can position the fused silica, spherical pick-off mirror

\textsuperscript{1}School of Mechanical Engineering, Northwestern Polytechnical University, Xi’an, China
\textsuperscript{2}Xi’an Institute of Optics and Precision Mechanics, Xi’an, China
\textsuperscript{3}School of Mechanical and Precision Instrument Engineering, Xi’an University of Technology, Xi’an, China

Corresponding author:
Wenjie Ge, School of Mechanical Engineering, Northwestern Polytechnical University, No.127, West Youyi Road, Beilin District, Xi’an City, Shaanxi 710072, China.
Email: gwj@nwpu.edu.cn

Creative Commons CC BY: This article is distributed under the terms of the Creative Commons Attribution 4.0 License (https://creativecommons.org/licenses/by/4.0/) which permits any use, reproduction and distribution of the work without further permission provided the original work is attributed as specified on the SAGE and Open Access pages (https://us.sagepub.com/en-us/nam/open-access-at-sage).
in the optimal tip, tilt and piston position for the near infrared camera of the James Webb space telescope. The traditional focusing mechanisms are usually consist of the high-precision worm gear, ball screw and stepper motor. Although it can achieve high-precision focusing, increasing complexity, higher weight and launch cost seriously limit the its application.

In order to decrease the complexity and weight of the rigid focusing mechanism, a compliant mechanism actuated by piezoelectric ceramic (PZT) was viewed as the novel method to design the focusing mechanism. The compliant mechanism provides high-precision motion and transmit force, motion and energy by virtue of its material deformation. It has the advantages of no gap, no friction and no maintenance and widely used in the micro-electro-mechanical system (MEMS), aerospace and medical equipment. Tang et al. designed a novel flexure-based 3-degree-freedom elliptical micro-positioning motion stage to obtain translation motions along the x, y, and z directions with high working bandwidth. A micro gripper based three-stage amplified compliant mechanism actuated by the PZT was achieved to grip the micro parts by Chen et al. It can be seen that the compliant mechanism is the better choice to design the high-precision focusing mechanism. In virtue of the larger output force, higher stiffness and fast response speed, the PZT actuator is usually used for ultra-precision motion and positioning. However, the output displacement of the PZT actuator is more small and cannot achieve the design requirements of large strokes. Therefore, the PZT actuator and displacement amplifier are cooperated to achieve the larger range and force for the focusing mechanism.

The multistage lever-type mechanism is the most common displacement amplifier which has been widely used in parallel robots, precision positioning platforms and other fields. Compared with the bridge-type amplifying mechanism, it has the advantages of simpler structure, higher bearing capacity and larger magnification. Accurate model of flexure hinges is the key to design and optimize the multistage lever-type amplifier. Some scholars proposed the compliance matrix of elliptical and circular hinges, which effectively described the deformation of compliant mechanisms composed of the circular, parabolic and elliptical flexible hinges. Subsequently, many novel models and methods have been developed to design the lever-type amplifying mechanisms. Ma et al. designed a new piezoelectric linear actuator with a symmetrical lever-type displacement amplifying mechanism based on a uniaxial double-slot right circular flexible hinge. It can achieve a large displacement for the driving part and sufficient clamping force for the clamping part. Jung and Kim designed a piezoelectric actuated platform which was composed of a multi-levers amplifying mechanism and had a magnification of more than 60 times. Choi et al. proposed a motion platform driven by the PZT using symmetrical planar multiple lever-type mechanism which has a magnification of more than 20 times. A piezoelectric pump with the lever-type hinge mechanism was designed by Ham et al. The small displacements could be amplified by a factor of 10 using the pump. Consequently, the lever-type mechanism is more suitable to design motion platform with simpler structure and larger magnification.

Although the lever-type mechanism has the potential to achieve higher magnification (60 times by Jung and Kim), the load-carrying capacity is more small. So, it is more suitable for the MEMS and minimally invasive medical devices. Moreover, the light weight of compliant focusing mechanism is the key to design the space optical camera. However, optimization design of integrating the structure parameters and lighter weight for the compliant focusing mechanism is rarely reported. A novel compliant focusing mechanism for the space optical camera was proposed to achieve the higher magnification ratio and lighter weight in this paper. The size parameters of the flexure hinges and beams were defined as the design variables. Optimal structure parameters of compliant focusing mechanism can be obtained by the optimization method. Simulation and experiments were implemented to verify the effectiveness of the novel compliant focusing mechanism with lighter weight and larger stroke.

Section 2 presented the simplified structure of the focusing mechanism based on the two-stage flexible hinge lever-type mechanism. Section 3 deduced the design model and optimization process of the compliant focusing mechanism. Deformed simulation and modal analysis of the proposed compliant focusing mechanism were implemented in Section 4. Experiment and evaluation of the optimal compliant focusing mechanism were described in Section 5. The conclusion was given in Section 6.

Structure description of the compliant focusing mechanism

The optical system of the space optical camera is mainly composed of four parts: the main mirror, the secondary mirror, the third mirror and the reflector. The primary and secondary mirrors are installed in the bearing cylinder with carbon fiber reinforced resin matrix composites. The simplified schematic diagram of the optical system of the space optical camera is shown in Figure 1. The overall size of the space optical camera is 1700 mm × 650 mm × 750 mm. The focal distance, focus depth and pixel size is respectively 5 m, ± 0.2 mm, and 10 µm. The size of the focal plane is 200 mm × 100 mm × 45 mm (length, width, and height). Due to the vibration in launch process and
thermal radiation in space, the bearing cylinder with carbon fiber composites is deformed, which changes the optical path and causes the defocus. Therefore, a focusing mechanism is required to adjust the focal plane to obtain a clear image. In virtue of the obvious advantages, compliant mechanism and the PZT were combined to design the focusing mechanism for space optical camera in this paper. According to the design requirements of the space optical camera, simplified structure diagram of the compliant focusing mechanism based on the two-stage flexure hinge lever-type mechanism and the PZT is shown in Figure 2.

The compliant focusing mechanism is composed of two symmetrical two-stage flexure hinge lever-type amplifiers and the PZT actuators. The two-stage flexure hinge lever-type mechanism is manufactured of aluminum alloy (7075-T651) with 455 MPa for the yield strength, 71 GPa for the elastic modulus and 0.33 for the Poisson’s ratio. The actual focal plane assembly weighs 2 kg which is composed of charge coupled devices (CCD), support frame with invar alloy, printed circuit board and heating tube. In Figure 2, a thin rectangular plate is used to replace the actual focal plane assembly. It should be noted that since the optical camera is a weightless state in space, the load of the focal plane assembly on the compliant mechanism is negligible. The symmetrical two-stage flexure hinge lever-type amplifier is adopted to eliminate the parasitic displacement. The focusing mechanism is driven by a square laminated piezoelectric ceramic (model: 150/5 × 5/20 H) with maximal output force of 1600 N and maximal displacement of 24 μm. According to deformation of the bearing cylinder and the light paths, the focal plane should be moved within 2 mm along the vertical direction to ensure a clear image. Therefore, input displacement from the PZT needs to be magnified by 100 times to achieve the clear image using the two-stage flexible hinge lever-type mechanism.

Modeling and optimizing of the compliant focusing mechanism

Amplification ratio

The amplification ratio of the two-stage flexible hinge lever-type mechanism is the key to design the compliant focusing mechanism. Four groups of identical two-stage flexible hinge lever-type mechanisms are adopted to support the focal plane and realize movement along the vertical direction in the paper. In order to simplify the calculation model, only one of the two-stage flexure hinge lever-type mechanism is investigated and its simplified structural schematic is shown in Figure 3.

The upper beam and lower beam are in contact at point 2. The symmetrical structure offsets the parasitic displacement in x direction at the output point and only the displacement in y direction is considered in the paper. $l_1$, $l_2$, $l_3$, $l_4$ represent the length of the upper and lower beams for the lever-type mechanism in x direction. $r_1$, $r_2$ and $t_1$, $t_2$ respectively denote the radius and longitudinal section thickness of flexure hinges for the
upper and lower beam. $h_1$, $h_2$ and $b_1$, $b_2$ respectively represent the height and thickness of the upper and lower beams. $F_{in}$ denotes the input force from the PZT. $u_{in}$ and $u_{out}$ denote the input and output displacements. $G$ represents the external load from the focal plane.

The deformation of the upper beam can be deduced from the deformation of the flexible hinge and rectangular beam with constant cross section. The deformation of the symmetrical semicircular flexible hinge at point 1 can be obtained by Lobiotiu’s theory\(^2\) and expressed as:

$$
\begin{bmatrix}
  u_{1y}^h \\
  \theta_{1z}^h
\end{bmatrix} = \frac{12}{\pi E b_1} \int_0^{2r_1} \frac{1}{t(x)^3} \left[ x^2 - 1 \right] dx \begin{bmatrix}
  F_r - G \\
  F_r l_1 - G(l_1 + l_2)
\end{bmatrix}
$$

(1)

where, $E$ represents the modulus of elasticity, the superscript $h$ represents the flexible hinge, $F_r$ represents the interaction force between the upper beam and lower beam, $r_1$ represents the radius of the flexure hinge of the upper beam, $u_{1y}^h$ represents the displacement of flexible hinge for the upper beam in $y$ direction at the point 1, and $\theta_{1z}^h$ represents the rotation angle of the upper beam around $z$ direction at the point 1. $t(x)$ represents the variable thickness of the longitudinal section for the flexible hinge which can be expressed as:

$$
\begin{align*}
  t(x) &= t_1 + 2 \left[ r_1 - \sqrt{x(2r_1 - x)} \right]
\end{align*}
$$

(2)

The deformation of the upper beam with constant rectangular section can be obtained by the Euler beam theory and expressed as:

$$
\begin{align*}
  u_{2y}^h &= \frac{F_r l_1^3}{6 E l_1} (2l_1) + \frac{G l_1^2}{6 E l_1} (l_1 - 3(l_1 + l_2)) \\
  u_{3y}^h &= \frac{F_r l_1^3}{6 E l_1} (3(l_1 + l_2) - l_1) + \frac{G(l_1 + l_2)^2}{6 E l_1} (-2(l_1 + l_2))
\end{align*}
$$

(3)

where, $I_1$ represents the inertia moment around $z$ axis of the upper beam with constant rectangular section; the superscript $b$ represents the beam, $u_{2y}^b$ represents the displacement of the upper beam in $y$ direction at point 2, and $u_{3y}^b$ represents the displacement of the upper beam in $y$ direction at point 3. According to the superposition theory, the displacement $u_{2y}^{upper}$ of the upper beam in $y$ direction at point 2 and the output displacement $u_{out}$ are expressed as:

$$
\begin{align*}
  u_{2y}^{upper} &= u_{1y}^h + l_1 \theta_{1z}^h + u_{2y}^b \\
  u_{out} &= u_{1y}^h + (l_1 + l_2) \theta_{1z}^h + u_{3y}^b
\end{align*}
$$

(5)

(6)

The calculation process of the lower beam is similar to the upper beam. Therefore, using the same theory, the deformations of the lower beam can be obtained (symbols notation are same as the upper beam) and expressed as:

$$
\begin{align*}
  u_{in} &= u_{4y}^b + l_3 \theta_{4z}^b + u_{5y}^b \\
  u_{lower} &= u_{4y}^b + (l_3 + l_4) \theta_{4z}^b + u_{6y}^b
\end{align*}
$$

(7)

(8)

$$
\begin{align*}
  u_{4y}^b &= \frac{F_r l_2^3}{6 E l_2} (2l_3) + \frac{F_r l_2^3}{6 E l_2} (l_3 - 3(l_3 + l_4)) \\
  u_{6y}^b &= \frac{F_r l_2^3}{6 E l_2} (3(l_3 + l_4) - l_3) + \frac{F_r (l_3 + l_4)^2}{6 E l_2} (-2(l_3 + l_4))
\end{align*}
$$

(9)

(10)

(11)

where, $I_2$ represents the inertia moment around $z$ axis of the lower beam with constant rectangular section, $u_{4y}^b$ represents the displacement of the lower beam in $y$ direction at the point 4, $\theta_{4z}^b$ represents the rotation angle of the lower beam around $z$ direction at the point 4, $u_{5y}^b$ represents the displacement of the upper beam in

---

**Figure 3.** Simplified structural schematic of the two-stage flexural hinge lever-type mechanism.
y direction at point 5, $u_{by}^b$ represents the displacement of the upper beam in $y$ direction at point 6. Since the upper and lower beams are in contact with each other at point 2, the displacement of the upper beam in $y$ direction at point 2 is equal to the displacement of the lower beam in $y$ direction at point 6. So, an additional deformation coordination equation is established and expressed as:

$$u_{by}^b + (l_3 + l_4)\theta_{ax}^b = u_{by}^b = u_{by}^h + l_1\theta_{ax}^h + u_{by}^b$$  \hspace{1cm} (12)$$

The interaction force $F_i$ can be obtained by solving equation (12). Then, $F_i$ is substituted into equations (1), (3), (4), (9), (10), and (11). Based on the above information, $u_{in}$ and $u_{out}$ can be respectively derived from equations (7) and (6). Therefore, the amplification ratio of the two-stage flexible hinge lever-type mechanism can be expressed as:

$$p = \frac{u_{out}}{u_{in}}$$  \hspace{1cm} (13)$$

**Optimization model**

The amplification ratio of the two-stage flexure hinge lever-type mechanism is closely related to the flexibility matrices of the flexure hinges. According to the Lobiotiu’s theory, the flexibility matrix of the symmetric circular flexure hinge can be determined by the size parameters $(r_1, r_2, t_1, t_2)$ of the flexure hinge. Weight of flexible hinges is very light and negligible. Therefore, the total weight of the two-stage flexure hinge lever-type mechanism is determined by the size parameters $(h_1, h_2, b_1, b_2, l_1, l_2)$ of beams. An optimization model of the two-stage flexure hinge lever-type mechanism with light weight is established and expressed as follows:

Find $x = [r_1, t_1, h_1, r_2, t_2, h_2]$

$$\text{min } f_0(x) = \rho(b_1h_1(l_1 + l_2) + b_2h_2(l_3 + l_4))$$

$$f_1(x) = u_{in}(x) - 0.02 \leq 0$$

$$f_2(x) = 2 - u_{out}(x) \leq 0$$

$$f_3(x) = t_1 + 2r_1 - h_1 \leq 0$$

$$f_4(x) = t_2 + 2r_2 - h_2 \leq 0$$

$$1 \leq r_1, r_2 \leq 15$$

$$1 \leq t_1, t_2 \leq 15$$

$$1 \leq l_1, l_2 \leq 40$$

$$1 \leq h_1, h_2 \leq 40$$

(14)

where, $x$ is the design variable vector which includes structural parameters of the flexible hinges, upper and lower beams. $\rho$ denotes the material density of the lever-type mechanism. $f_0(x)$ is the objective function which represents the weight of the entire structure. $f_1(x)$ is the constraint function of the input displacement from the PZT. $f_2(x)$ is the constraint function of the output displacement for the compliant focusing mechanism. According to the desired stroke range of the focusing mechanism, the minimal output displacement is 2 mm. $f_3(x)$ and $f_4(x)$ are the structural constraint functions which can ensure the structural integrity of the flexure hinges for the upper and lower beams. The allowable ranges of the size parameters for the flexible hinges and beams are respectively set as $[1 \text{ mm}, 15 \text{ mm}]$ and $[1 \text{ mm}, 40 \text{ mm}]$.

**Sensitivity analysis and solution**

Sensitivity analysis is the key to solve the optimization problems based on gradient optimization algorithm. According to equation (14), the sensitivities of the objective and constraint functions to the change of the design variable $x$ are respectively deduced and expressed as:

$$\frac{\partial f_0(x)}{\partial x} = \frac{\partial \rho(b_1h_1(l_1 + l_2) + b_2h_2(l_3 + l_4))}{\partial x}$$  \hspace{1cm} (15)$$

$$\frac{\partial f_1(x)}{\partial x} = \frac{\partial u_{in}(x)}{\partial x} = \frac{\partial u_{in}^h}{\partial x} + l_1\frac{\partial \theta_{ax}^h}{\partial x} + \frac{\partial u_{in}^b}{\partial x}$$  \hspace{1cm} (16)$$

$$\frac{\partial f_2(x)}{\partial x} = -\frac{\partial u_{out}(x)}{\partial x} = \frac{\partial u_{out}^h}{\partial x} + \frac{l_1\partial \theta_{ax}^h}{\partial x} + \frac{\partial u_{out}^b}{\partial x}$$  \hspace{1cm} (17)$$

$$\frac{\partial f_3(x)}{\partial x} = \frac{\partial t_1}{\partial x} + 2\frac{\partial r_1}{\partial x} - \frac{\partial h_1}{\partial x}$$  \hspace{1cm} (18)$$

$$\frac{\partial f_4(x)}{\partial x} = \frac{\partial t_2}{\partial x} + 2\frac{\partial r_2}{\partial x} - \frac{\partial h_2}{\partial x}$$  \hspace{1cm} (19)$$

The sensitivities $\frac{\partial u_{in}^h(x)}{\partial x}$, $\frac{\partial u_{in}^b(x)}{\partial x}$, $\frac{\partial t_1\theta_{ax}^h(x)}{\partial x}$, and $\frac{\partial t_2\theta_{ax}^b(x)}{\partial x}$ in the equation (16) can be deduced by analyzing equations (9) and (10). By deriving analytical equations (1) and (4), partial derivatives $\frac{\partial t_1\theta_{ax}^h(x)}{\partial x}$, $(l_1 + l_2)\frac{\partial \theta_{ax}^h(x)}{\partial x}$, and $\frac{\partial u_{in}^b(x)}{\partial x}$ are obtained. Substituting these derivatives into equations (16) and (17), the sensitivities of the objective and constraint functions to the design variable can be achieved.

A globally convergent method of moving asymptotes (GCMMA) belonging to a convex programming method was adopted to solve the optimization model of the compliant focusing mechanism in the paper.\(^\text{26}\)

The flow chart of the optimization process was shown in Figure 4 and specific steps were as follows:

**Step 1:** Optimization parameters were firstly initialized. They include design variables $x^k$, maximal input displacement $t_{\text{max}}$ and minimal output displacement $u_{\text{min}}$, the allowable ranges $(r_{\text{min}}, r_{\text{max}}, l_{\text{min}}, l_{\text{max}})$ of size parameters for the flexible hinges and $(h_{\text{min}}, h_{\text{max}})$ for the beams. The allowable error $\varepsilon$ of
Defining design variables $x^k$, maximal input displacement $u_{max}$, allowable ranges of the radius $(r_{max}, r_{min})$, thickness $(t_{min}, t_{max})$, and height $(h_{max}, h_{min})$, allowable error of iterative convergence $e$.

Solving objective function $f_0(x^k)$ and constraints $f_1(x^k)$, $f_2(x^k)$, $f_3(x^k)$.

Deducing sensitivities of objective and constraint function.

Updating design variables $x^{k+1}$

$$G_{CMA}$$

$k = k + 1$

Convergence

No

Yes

Step 3: The sensitivities of objective and constraint functions to the change of design variable $x_1$ were respectively deduced based on the current design variables $x^k$.

Step 4: The design variables vector $x^k$ was updated by the GCMMA method and new value was set as $x^{k+1}$. Based on the updated design variables $x^{k+1}$, the updated objective function $f_0(x^{k+1})$ and constraint functions $f_1(x^{k+1})$, $f_2(x^{k+1})$, $f_3(x^{k+1})$, $f_4(x^{k+1})$ can be obtained, respectively.

Step 5: If $|x^{k+1} - x^k| \leq e$ was satisfied, the optimization loop should end. Then, the optimal size parameters of the compliant focusing mechanism can be obtained and outputted. Otherwise, the superscript index will be updated to $k = k + 1$ and return to step 2.

Table 1. Structural sizes of the two-stage flexural hinge lever-type mechanism.

| Parameters               | Values |
|-------------------------|--------|
| Beam lengths $l_1$ (mm) | 10     |
| Beam lengths $l_2$ (mm) | 10     |
| Beam lengths $l_3$ (mm) | 126    |
| Beam lengths $l_4$ (mm) | 126    |
| Beam thickness $b_1$ (mm)| 5      |
| Beam thickness $b_2$ (mm)| 5      |
| Elasticity modulus $E$ (MPa) | 71e3  |
| External load $G$ (N)    | 0      |

Optimization results and simulation analysis

In order to investigate the effectiveness of the proposed method, the optimization process and results under different input forces $F_{in} = 100$ N, $F_{in} = 150$ N, $F_{in} = 200$ N, $F_{in} = 300$ N, and $F_{in} = 400$ N were implemented. The initial design variables were defined as $r_1 = r_2 = r_3 = r_4 = h_1 = h_2 = 4$ mm. Based on the previous research, the allowable range of size parameter for the flexure hinges and beams were respectively set as [1 mm, 15 mm] and [1 mm, 40 mm]. The allowable error of optimization convergence was defined as $e = 0.001$. The other structural sizes of the two-stage flexible hinge lever-type mechanism were constant in the optimization process and listed in Table 1.

Based on the optimization model, Figure 5 shown the iterative curves of the objective function $f_0$ and constraint functions $f_1$, $f_2$, $f_3$ under the different input forces. The optimization process was gradually stable and convergent. It verifies the effectiveness of the proposed method to design focusing mechanism. It can be seen from Figure 5(a) that the stable values of the objective functions are 79g, 58g, 52g, 47g, 46g under the input force $F_{in} = 100$ N, $F_{in} = 150$ N, $F_{in} = 200$ N, $F_{in} = 300$ N, $F_{in} = 400$ N. As the input force is increased, the value of the objective function gradually is decreased and the total weight of the focusing mechanism is more and more lighter. From the Figure 5(b) and (c), the input and output displacements of the two-stage flexure hinge lever-type mechanism can satisfy the constraint requirements. Table 2 listed the optimal size parameters of the two-stage flexure hinge lever-type mechanism under different input forces.

3D models of the two-stage flexure hinge lever-type mechanism under the different input forces were constructed. Subsequently, deformed simulations and model analysis were performed to verify the feasibility of the proposed compliant focusing mechanism using the finite element method (FEM). The output displacements and Von-Mises stress under the different output forces were shown in Figure 6. The output displacement under different input forces can approximately reach at 2 mm. The results are consistent with the optimization results and verify the effectiveness of the proposed compliant focusing mechanism for the space optical camera.

The curves of maximal stress and the total weight of compliant focusing mechanism under the different input force $F_{in}$ were shown in Figure 7. As the input force is increased, the maximal Von-Mises stress of the focusing mechanism is gradually increased. When the input force is increased to 200 N, the Von-Mises stress
was 239.2 MPa which has exceeded the allowable stress of the material (safety factor is 2). Moreover, the total weight (objective function $f_0$) was gradually decreased with the increase of input force. However, when the input force is increased to 300 N, the objective function is converged to about 47g. At the moment, the reduction of total weight was not obvious. Consequently, the balance between total weight and maximal stress was very important of the compliant focusing mechanism. If the input force is increased, the total weight will become lighter and the maximal stress will become larger. From the Figure 7, the output displacement under the $F_{in} = 150$ N can reach at 2.0493 mm and maximal stress is 197.54 MPa which satisfies the allowable stress of the material. Consequently, size parameters under the input force $F_{in} = 150$ N were selected to design the two-stage flexible hinge lever-type mechanism with lighter weight and allowable stress.

**Figure 5.** Iterative curves of the optimization process under different input forces: (a) objective function $f_0$, (b) constraint function $f_1$, (c) constraint function $f_2$, and (d) constraint function $f_3$.

**Table 2.** Optimal size parameters of compliant focusing mechanism with different input forces.

| Parameter                  | Case 1 | Case 2 | Case 3 | Case 4 | Case 5 |
|----------------------------|--------|--------|--------|--------|--------|
| Input force $F_{in}$ (N)   | 100    | 150    | 200    | 300    | 400    |
| Height of upper beam $h_1$ (mm) | 15.57  | 10.47  | 7.85   | 5.28   | 3.99   |
| Height of lower beam $h_2$ (mm) | 26.14  | 20.49  | 19.42  | 19.56  | 20.02  |
| Sum of beam heights $h_1 + h_2$ (mm) | 41.71  | 30.96  | 27.27  | 24.84  | 24.01  |
| Radius of flexible hinge in upper beam $r_1$ (mm) | 7.28   | 4.73   | 3.42   | 2.14   | 1.50   |
| Radius of flexible hinge in lower beam $r_2$ (mm) | 1  | 1  | 1  | 1  | 1  |
| Thickness of upper beam flexible hinge $t_1$ (mm) | 1  | 1  | 1  | 1  | 1  |
| Thickness of lower beam flexible hinge $t_2$ (mm) | 1  | 1.21  | 1.48  | 2.09  | 2.64  |
Figure 6. Output displacements and Von-Mises stresses of the two-stage flexure hinge lever-type mechanism under the different input forces: (a) \( F_{in} = 100 \) N, (b) \( F_{in} = 150 \) N, (c) \( F_{in} = 200 \) N, (d) \( F_{in} = 300 \) N, and (e) \( F_{in} = 400 \) N.
In order to avoid the resonance in space, the modal analysis of the two-stage flexure hinge lever-type mechanism under the $F_{in} = 150 \text{ N}$ was investigated to obtain the natural frequency and mode using the FEM. Figure 8 shown the first four order vibration modes and natural frequencies of the two-stage flexure hinge lever-type mechanism. The first four order natural frequencies respectively were 129, 144, 175, and 973 Hz, which were larger than the spatial excitation frequency (generally less than 100 Hz). Moreover, the vibration frequency in the emission stage was 70 Hz which was lower than the first-order frequency of 129 Hz. Moreover, the two-stage flexure hinge lever-type mechanism was closed in the launching stage. When it is working in space, the focusing mechanism is opened and can achieve different motion by adjusting the voltage of the PZT. So, the proposed focusing mechanism of the space optical camera can successfully avoid the external vibration and further ensure image quality.

**Experiment and analysis**

The compliant focusing mechanism based on the two-stage flexible hinge lever-type mechanism under the input force $F_{in} = 150 \text{ N}$ was fabricated by laser cutting...
technology and the experimental platform was established as shown in Figure 9. Displacement experiments of the compliant focusing mechanism under the input force $F_{in} = 150 \text{ N}$ were investigated.

The displacement of the focal plane in the vertical direction was generated by the combination of two-stage flexible hinge lever-type mechanism and the PZT. Electronic dial gauge (range: 0–6.5 mm, repeatability: ± 0.003 mm) and laser sensor (range: 200–1000 mm, repeatability: ± 0.05 mm) were respectively used to measure the input and output displacements. Square laminated piezoelectric ceramics (PSt 150/5 × 5/20 H) were used to generate the input force. The power supply and analog input voltages were respectively set as 15 V and 0–10 V. Output displacement from the piezoelectric ceramic can be directly measured by the electronic dial gauges.

Based on the established experimental platform, deformation of the compliant focusing mechanism in initial and target states were shown in Figure 10. In initial stage, the analog input voltage and output displacement of the PZT were set as 0. Digital indications of the electronic dial gauges and laser displacement sensor were respectively 0 and 353.5 mm. When the input voltage of the PZT was set to 10 V, the deformation of the compliant focusing mechanism was shown in Figure 10(b). Digital indications of the electronic dial gauges and laser displacement sensor were respectively as 21 μm, 22 μm, and 355.6 mm. Consequently, the displacement at the input point of the compliant focusing mechanism were respectively 21 μm and 22 μm. The displacement at the output point was reached at 2.1 mm which was approximately consistent with the theoretical analysis. Therefore, experimental results verified the effectiveness of the proposed method to design the two-stage flexure hinged lever-type mechanism.

In addition, deformation experiments of compliant focusing mechanism with different input displacements derived from the PZT were implemented by the experimental platform. The input displacements were defined as the 4, 8, 12, and 16 mm. The output displacements $u_{out}$ and the amplification ratios $p$ were investigated with the different input displacements. Comparing the theoretical, simulation and experimental results, the error results of the output displacement and amplification ratio were listed in Table 3. The error of the output displacement $u_{out}$ is equal to the error of the amplifier ratio. Its reason is the fact that the amplification ratio is proportional to the output displacement under the certain input displacement. The maximal errors of the $u_{out}$ and $p$ are respectively 5.0% and 5.5% when the input displacement is 4 μm. The maximal error of the output displacement between theoretical and experimental results is 0.076 mm. Since focal length range of the space camera is ± 0.2 mm, the maximal error 0.076 mm is less...
than the error range. So, the maximal error can’t affect the image clarity. The compliant focusing mechanism proposed in the paper can achieve design requirements of the space camera.

The experimental errors were mainly derived from manufacturing, assembly and measurement accuracy. The error from the laser cutting was key reason to influence the manufacturing accuracy. In addition, the contact type method to measure input displacement inevitably generated measurement errors. The installation accuracy of the measurement system also leaded to large measurement errors. Future research will improve the accuracy using a high-precision non-contact measurement system.

**Conclusions**

This paper proposed a novel compliant focusing mechanism for space optical camera which had the characteristics of lighter weight and larger stroke. The compliant focusing mechanism was composed of a flexible hinge lever-type amplifier and a PZT. The PZT were used to provide the driving force and output displacement was amplified by a two-stage flexible hinge mechanism. Considering the lighter weight, two-stage semicircular flexure hinge lever-type mechanism was achieved by the optimization theory. The effectiveness of the proposed method was verified by simulation and experiment. The results were discussed and as following:

1. The size parameters of the semicircular flexure hinge and the upper and lower beams were defined as design variables. The minimal total weight of the focusing mechanism was taken as the optimization function. The GCMMA optimization strategy was used to optimize the two-stage flexure hinge lever-type mechanism to achieve the lightweight and large stroke.

2. With increase of the input force from the PZT, the maximal stress was increased and the total weight of the two-stage flexure hinge lever-type mechanism was gradually decreased.

3. Comparing theoretical calculation, simulation and experimental results, the correctness and effectiveness of the proposed compliant focusing mechanism were proved. In addition, the focusing mechanism designed by the optimization method can realize the movement of the focal plane assembly within a range of 2 mm and the total weight was significantly reduced.

**Acknowledgements**

The authors would like to thank the National Natural Science Foundation of China under Grant No. 51375383 and the Natural Science Basic Research Program Foundation of Shaanxi Province of China under Grant No. 2022JM-197.

**Author contributions (Roles)**

Yan Li performed significantly the data analyses and wrote the manuscript; Bo Liu performed the experiment; Wenjie Ge contributed to the conception of the study; Xinxing Tong provided the constructive discussions.

**Declaration of conflicting interests**

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

**Funding**

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was supported by the National Natural Science Foundation of China under Grant No. 51375383 and the Natural Science Basic Research Program Foundation of Shaanxi Province of China under Grant No. 2022JM-197.

**Ethical approval/Patient consent**

There is no ethical approval /patient consent.

**Table 3.** Error results of output displacement and amplification ratio by comparing theoretical, simulation and experimental results.

| Input displacement $u_{in} \, (\mu m)$ | Output displacement $u_{out}$ and Error | Amplification ratio $p$ and error |
|----------------------------------------|-----------------------------------------|----------------------------------|
|                                        | Theory Value ($\mu m$)                  | Simulation Value ($\mu m$) Error (%) | Experiment Value ($\mu m$) Error (%) |
|                                        | Value ($\mu m$) Error (%)               | Value ($\mu m$) Error (%)         |
| 4                                      | 402 5.5 382 5.0                         | 100.5 95 95.5 5.0                 |
| 8                                      | 790 2.7 808 2.3                         | 100.4 101.4 2.6 2.2              |
| 12                                     | 1205 1.6 1150 4.6                      | 98.8 102 1.6 4.6                 |
| 16                                     | 1580 2.0 1596 1.0                      | 98.8 100.8 2.0 1.0              |
| 20                                     | 1995 1.1 2071 3.8                      | 99.8 100.9 1.1 3.8              |
References

1. Kohley R, Garé P, Vétel C, et al. Gaia’s FPA: sampling the sky in silicon. In: Proceedings of SPIE space telescopes and instrumentation, Amsterdam, The Netherlands, 1–6 July 2012, 8442, p.84421. Washington, DC: SPIE.

2. Zago L, Schwab P and Gallieni D. Development and testing of a high-precision high-stiffness linear actuator for the focus-center mechanism of the SOFIA secondary mirror. In: Proceedings of SPIE international society for optical engineering, Munich, Germany, 27–28 March 2000, 4014, pp.392–398. Washington, DC: SPIE.

3. Witherspoon B. Qualifying the flight design of the focus and alignment mechanism of the near-infrared camera on the James Webb Space Telescope. In: Proceedings of SPIE International Society for Optical Engineering, San Diego, CA, 24–25 August 2011, 8150(11), pp.81500I–81500I-10. Washington, DC: SPIE.

4. Liebe C, Pollock R and Hannah B. System for establishing best focus for the Orbiting carbon Observatory instrument. Opt Eng 2009; 48: 705–709.

5. Tong X, Ge W, Yuan Z, et al. Integrated design of topology and material for composite morphing trailing edge based compliant mechanism. Chin J Aeronaut 2021; 34: 331–340.

6. Tong X, Ge W, Gao X, et al. Simultaneous optimization of fiber orientations and topology shape for composites compliant leading edge. J Reinforced Plast Compos 2019; 38: 706–716.

7. Liu M, Zhan J, Zhu B, et al. Topology optimization of compliant mechanism considering actual output displacement using adaptive output spring stiffness. Mech Mach Theory 2020; 146: 103728.

8. Zhu B, Zhang X, Zhang H, et al. Design of compliant mechanisms using continuum topology optimization: A review. Mech Mach Theory 2020; 143: 103622.

9. Tang C, Zhang M and Cao G. Design and testing of a novel flexure-based 3-degree-of-freedom elliptical micro/nano-positioning motion stage. Adv Mech Eng 2017; 9: 168781401772524.

10. Chen X, Deng Z, Hu S, et al. Research on three-stage amplified compliant mechanism-based piezo-driven microgripper. Adv Mech Eng 2020; 12: 168781402091147.

11. Zhu D, Feng Y and Zhan W. Topology optimization of three-translational degree-of-freedom spatial compliant mechanism. Adv Mech Eng 2019; 11: 168781401982822.

12. Li R and Zhu B. An augmented formulation of distributed compliant mechanism optimization using a level set method. Adv Mech Eng 2016; 8: 168781401664448.

13. Dong W, Sun LN and Du ZJ. Design of a precision compliant parallel positioner driven by dual piezoelectric actuators. Sens Actuators A Phys 2007; 135: 250–256.

14. Shen J and Yun Z. Amplifying performance of lever-type flexure hinge amplifying mechanism. In: Proceedings of 9th international conference on electronic measurement and instruments, Beijing, China, 16–19 August 2009, pp.1294–1297. New York: IEEE.

15. Jouaneh M and Yang R. Modeling of flexure-hinge type lever mechanisms. Precis Eng 2003; 27: 407–418.

16. Smith ST, Badami VG, Dale JS, et al. Elliptical flexure hinges. Rev Sci Instrum 1997; 68: 1474–1483.

17. Lobontiu N, Paine JSN, Garcia E, et al. Design of symmetric conic-section flexure hinges based on closed-form compliance equations. Mech Mach Theory 2002; 37: 477–498.

18. Yong YK, Lu TF and Handley DC. Review of circular flexure hinge design equations and derivation of empirical formulations. Precis Eng 2008; 32: 63–70.

19. Raghavendra MRA, Kumar AS and Jagdish BN. Design and analysis of flexure-hinge parameter in microgripper. Int J Adv Manuf Technol 2010; 49: 1185–1193.

20. Chen G, Jia J, Qi H, et al. Performance optimization of elliptical flexure hinge using a modified particle swarm algorithm. In: Proceedings of the IEEE international conference on industrial technology, Hong Kong, China, 14–17 December 2009, pp.1180–1185. New York: IEEE.

21. Ma L, Xiao J, Zhou S, et al. A piezoelectric inchworm actuator of linear type using symmetrical lever amplification. Pro IMechE, Part N: JNanoengineering and Nano-systems 2015; 229: 172–179.

22. Jung HJ and Kim JH. Novel piezo driven motion amplified stage. Int J Precis Eng Manuf 2014; 15: 2141–2147.

23. Choi SB, Han SS, Han YM, et al. A magnification device for precision mechanisms featuring piez actuators and flexure hinges: design and experimental validation. Mech Mach Theory 2007; 42: 1184–1198.

24. Ham YB, Seo WS, Cho WY, et al. Development of a piezoelectric pump using hinge-lever amplification mechanism. J Electroceram 2009; 23: 346–350.

25. Lobontiu N. Compliant mechanisms: design of flexure hinges. New York, NY: CRC Press, 2002.

26. Svanberg K. A class of globally convergent optimization methods based on conservative convex separable approximations. SIAM J Optim 2002; 12: 555–573.