

Article

Design, Dynamic Verification, and Multi-Objective Optimization of a Passive Multi-Link Deployable Support Mechanism for Lunar Surface Solar-Concentrating Systems

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Abstract

Lunar in situ resource utilization requires solar concentrating systems that can be launched in a compact configuration and deployed reliably on the lunar surface. This paper presents a multi-link coupled deployable support mechanism for a reflector-Fresnel concentrating system. The mechanism adopts a shape-memory-alloy rigid release for the stowed state and passive spring hinges for autonomous deployment, aiming to reduce drive complexity while maintaining a high deployment ratio. To avoid interference caused by coupled link motion, a motion-envelope model is established for joint trajectory planning. The deployment process is then analyzed through vector-based kinematic modeling, D'Alembert force analysis, and Lagrange dynamic equations. The analytical predictions are corroborated through high-fidelity multibody dynamic simulations: the predicted driving torque of Link 3 is 0–0.68 N·m, close to the simulated range of 0–0.70 N·m, with a relative peak-value error of 2.8%; the maximum angular acceleration is 0.08 rad/s². Finite-element modal analysis gives a first locked-state natural frequency of 54.969 Hz. NSGA-II optimization further reduces the maximum driving torque by 10.9%, reduces torque fluctuation by 9.7%, and increases the maximum deployment ratio from 5.6 to 7.2. The results provide a quantified design and simulation basis for passive deployable concentrating mechanisms intended for lunar surface concentrating systems.

Keywords: deployable mechanism; lunar surface concentrating system; multi-link coupling; kinematic modeling; dynamic modeling; multi-objective optimization



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1. Introduction

The Moon is a key target for sustained deep-space exploration and the development of extraterrestrial resources. As lunar missions gradually progress toward sustained surface operations, lunar in situ resource utilization (ISRU) is becoming increasingly important. For long-term lunar operations, ISRU can reduce Earth-Moon transportation demand and support the local production of energy, materials, and consumables [1,2]. The continued advancement of lunar exploration programs, including China's Chang'e missions [3], further demonstrates the growing demand for lunar ISRU technologies. Among the available lunar energy resources, solar energy is one of the most readily accessible sources on the lunar

surface, with a space solar irradiance of approximately 1367 W/m^2 [4–6]. Concentrated solar energy can provide thermal input for lunar regolith processing, water-ice extraction, thermal storage, and other ISRU tasks. Accordingly, a lunar concentrating system must satisfy launch-volume, landing-load, and environmental constraints while remaining capable of compact stowage and reliable deployment into a large, stable optical working configuration on the lunar surface.

Solar concentration concepts and solar energy-concentrating mechanisms have been investigated in both space solar power and lunar resource-utilization contexts. Representative studies include the SPS-ALPHA concentrator concept [7], Fresnel-lens concentrators for lunar water-ice extraction [8], solar-assisted lunar-soil forming platforms [9], lightweight solar concentration and tracking systems [10], high-uniformity Fresnel concentrators [11], and reflector-optical fiber systems for lunar regolith melting [12]. More recently, deployable segmented solar concentrators have also been investigated to improve concentration performance while satisfying space deployment requirements [13]. These studies mainly demonstrate optical concentration, energy conversion, or process-level feasibility. For a deployable lunar concentrating system, the support mechanism that positions, deploys, and stabilizes the reflector is equally critical, because its deployment accuracy and locked-state stiffness directly affect the subsequent optical working state.

Space deployable mechanisms provide an effective solution to the conflict between launch-volume constraints and large operational dimensions [14]. For space applications, deployable mechanisms have been widely used in antenna reflectors, deployable booms, and solar-array systems, where compact stowage, deployment coordination, driving strategy, and locked-state stiffness are key design concerns [15–18]. Passive deployment elements, such as tape springs and torsion-spring hinges, are particularly attractive because they can store elastic energy in the folded state and release it during deployment, thereby reducing the need for multiple active actuators [19,20]. Analytical methods such as loop-closure equations and kinematic modeling are commonly used to evaluate mobility, deployability, and deployment stability [15,21]. These studies show that configuration synthesis, driving strategy, kinematic coordination, deployment stability, and locked-state performance are common core issues in the design of space deployable mechanisms. And for lunar surface solar-concentrating systems, the deployable support must not only achieve compact stowage and reliable deployment but also avoid reflector-link interference and provide a stable terminal posture for subsequent optical tracking.

Dynamic modeling is equally important because deployment is a transient process involving inertial effects, joint constraints, driving torque, and possible synchronization errors. Existing studies have emphasized that the design of deployable structures should be supported not only by configuration synthesis, but also by coordinated kinematic and dynamic analyses [22,23]. For deployable booms, Chu and Lei [24] established the design theory and dynamic analysis framework of a lenticular deployable boom, while Zhang et al. [25] developed a Lagrange-form dynamic model for a two-segment deployable beam system. For antenna support mechanisms, Guo et al. [26] carried out degree-of-freedom, kinematic, and dynamic analyses of a spatial support mechanism for planar deployable antennas, and Han et al. [27] further investigated the dynamic behavior of a scissors hoop-rib truss deployable antenna mechanism through theoretical modeling, simulation, and ground experiments. For large-diameter truss mechanisms, Han et al. [15,28] analyzed the kinematic and dynamic characteristics of multi-ring and double-ring truss deployable antenna mechanisms. Existing studies have also shown that joint clearance, damping, friction, gravity, and structural flexibility may affect deployment dynamics [29]. Therefore, kinematic feasibility alone is insufficient for a space deployable mechanism; deployment torque, ac-

celeration response, motion synchronization, and locked-state modal characteristics should also be quantified, as addressed in the subsequent analysis of the proposed mechanism.

For a lunar concentrating reflector, these issues become more coupled than in many conventional antenna or boom applications. The support mechanism must be compact during launch and landing, release with limited impact, avoid link-reflector interference during deployment, and finally provide a reflector pose suitable for optical tracking. Therefore, the mechanism-level problem is not merely to design a foldable support, but to establish a quantitative design and verification route for a passively driven coupled linkage. To address these issues, this paper proposes a multi-link coupled deployable support mechanism for a lunar reflector-Fresnel concentrating system and develops a modeling-simulation-optimization framework for its deployment performance assessment. The main contributions are as follows.

- (1) A deployable reflector support configuration is developed by integrating a shape-memory-alloy rigid release concept with passive spring-hinge deployment. Candidate locking/unlocking schemes are compared according to release impact, structural constraint stability, and potential failure modes.
- (2) A motion-envelope-based trajectory planning method and analytical kinematic/dynamic models are established for the coupled linkage. The theoretical torque prediction is checked using an independent multibody dynamic simulation model.
- (3) A multi-objective optimization model based on NSGA-II is formulated to balance deployment smoothness, synchronization error, and lightweight design. The optimized configuration is evaluated using deployment ratio, driving torque, torque fluctuation, and locked-state modal frequency.

The remainder of this paper is organized as follows. Section 2 presents the design requirements, system configuration, and locking/unlocking scheme of the deployable concentrating mechanism. Section 3 establishes the kinematic model and analyzes the deployment trajectory. Section 4 develops the dynamic model and verifies the deployment response and modal characteristics. Section 5 conducts multi-objective parameter optimization based on NSGA-II. Section 6 discusses the main findings, engineering implications, limitations, and conclusions of this study.

2. Overall Structural Design of the Energy-Concentrating Deployable Mechanism

2.1. Design Philosophy and Goal-Oriented Analysis

The structural design of the deployable concentrating mechanism is treated as a goal-oriented configuration synthesis problem rather than a simple engineering arrangement. The overall configuration of the proposed lunar surface solar concentrating system is illustrated in Figure 1. For lunar surface operation, the mechanism must satisfy three coupled requirements: compact stowage under launch-volume constraints, autonomous deployment after landing, and stable reflector positioning for subsequent optical concentration. To resolve the conflict between the compact stowed dimension required during launch and landing and the large operational span required by the reflector-Fresnel concentrating system, a deployable topological configuration is required.

For the lunar surface scenario, the application requirements are translated into mechanism-level constraints, including compact stowage within the launch and landing envelope, low-shock non-pyrotechnic release, passive deployment under limited power availability, tolerance to a possible landing attitude deviation, and locked-state stiffness for maintaining the reflector posture.

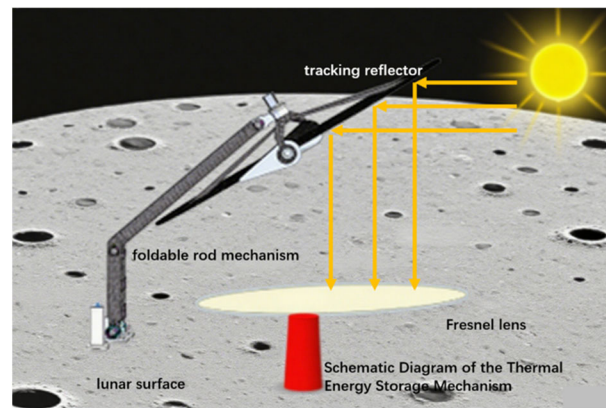


Figure 1. Schematic Diagram of the Concentrating Foldable Mechanism.

Based on structural synthesis theory, a multi-link serial open-chain configuration is selected as the basic deployable topology. This topology decomposes the large operational workspace of the reflector into several revolute-link motions, thereby improving folded compactness while maintaining the required deployed reach. To reduce structural mass and avoid heavy active driving units at each joint, a “rigid locking + passive hinge drive” strategy is adopted. During launch and landing, the rigid locking device constrains the folded structure to withstand mechanical disturbances. After release on the lunar surface, the elastic strain energy stored in the passive spring hinges provides the deployment torque, enabling autonomous deployment without continuous external power input.

The deployment function and the optical tracking function are further decoupled in the configuration design. The planar reflector is supported by an L-shaped bracket, which separates the deployment trajectory workspace from the optical tracking workspace and reduces the risk of motion interference. After the support mechanism reaches the deployed state and is constrained by terminal positioning locks, the dual-axis azimuth-pitch motors are used only for solar tracking. In this way, passive deployment and optical operation are assigned to different functional stages of the mechanism.

2.2. Release Mechanism Design

The release mechanism determines the transition from the locked launch configuration to the deployable configuration, and therefore must satisfy both high-restraint and low-shock requirements. Because the optical concentrating reflector is sensitive to dynamic impact, traditional pyrotechnic separation devices are unsuitable for this application, as their high-frequency shock may lead to reflector misalignment or degradation of optical accuracy. A flexible thermal-knife restraint was considered during the preliminary screening stage; however, its release repeatability depends on rope pretension, thermal cutting reliability, and post-release retraction. These uncertainties may alter the initial deployment boundary condition of a passively driven multi-link mechanism. Therefore, this scheme is not adopted as the final release design.

SMA-based release devices have potential advantages of compact structure and low release shock, which is consistent with the low-disturbance requirement of the lunar concentrating system. Accordingly, this study introduces an SMA-based locking concept as a preliminary hold-down and release approach for the folded deployable mechanism. The overall configuration of the energy-concentrating deployable mechanism with the SMA-based locking concept is shown in Figure 2. During launch and landing, the locking interface is intended to constrain the reflector and the foldable rod mechanism as a stable folded assembly. After activation on the lunar surface, the SMA element undergoes phase-transformation-induced contraction, thereby releasing the constraint through a low-shock

path. It should be noted that the present study does not provide a detailed hold-down and release mechanism design. In future prototype-level development, the degrees of freedom to be constrained or released, positioning repeatability, tolerance allocation, and deterministic constraint interfaces should be further investigated.

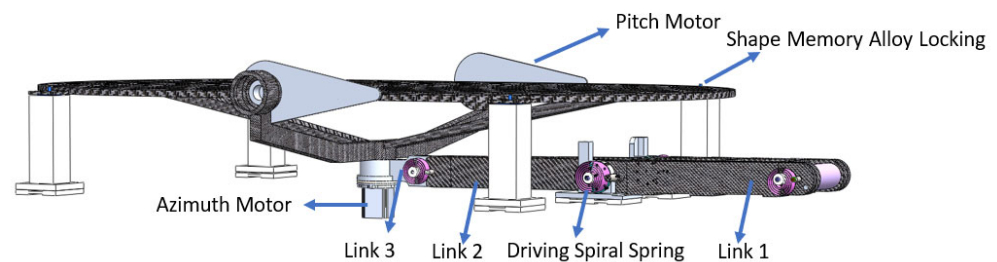


Figure 2. Configuration of the energy-concentrating deployable mechanism with an SMA-based locking concept.

Following release, the spiral spring hinges convert stored elastic potential energy into deployment torque and drive the foldable rod mechanism to rotate toward the working configuration. When the prescribed terminal positions are reached, positioning pins constrain the deployment joints and transform the mechanism from a reconfigurable linkage into a locked support structure. Therefore, the SMA release device, passive spring hinges, and terminal positioning locks jointly define the launch restraint, deployment actuation, and post-deployment support boundary conditions of the mechanism.

2.3. Detailed Design and Mobility Analysis

The detailed configuration of the concentrating foldable mechanism comprises three primary foldable links, namely Link 1, Link 2, and Link 3, which are interconnected by passive torsional spring hinges. The geometric lengths of the links are determined according to the folded-volume constraint and the required deployed operational dimensions, and are further optimized in Section 5. The folded-state structural design is shown in Figure 3. The numbered arrows in Figure 3 indicate the deployment motion directions of Link 1, Link 2, and Link 3 driven by their corresponding torsional spring hinges, while arrow 4 indicates the motion tendency of the reflector-supporting assembly driven by the L-shaped link. During this passive deployment stage, the azimuth and pitch tracking axes remain locked; therefore, the reflector is transported from the folded position toward the working posture together with the L-shaped link, rather than being actively rotated by the tracking axes. Figure 4 further illustrates the intermediate deployment state and the fully deployed operational configuration of the concentrating foldable mechanism.

To provide a theoretical basis for the subsequent kinematic modeling, the mobility of the deployment stage is evaluated using an equivalent planar open-chain model. During deployment, the azimuth-pitch tracking axes are locked, and Link 3, the L-shaped bracket, and the reflector are treated as one rigid body. Therefore, the deployable support can be simplified as a planar serial chain composed of a fixed base, three moving links, and three revolute pairs.

$M = 3(N - 1) - 2J_1 - J_2$, where N is the total number of links including the fixed base, J_1 is the number of lower pairs, and J_2 is the number of higher pairs. For the equivalent deployment model, $N = 4$, $J_1 = 3$, and $J_2 = 0$. Thus, $M = 3(4 - 1) - 2 \times 3 - 0 = 3$.

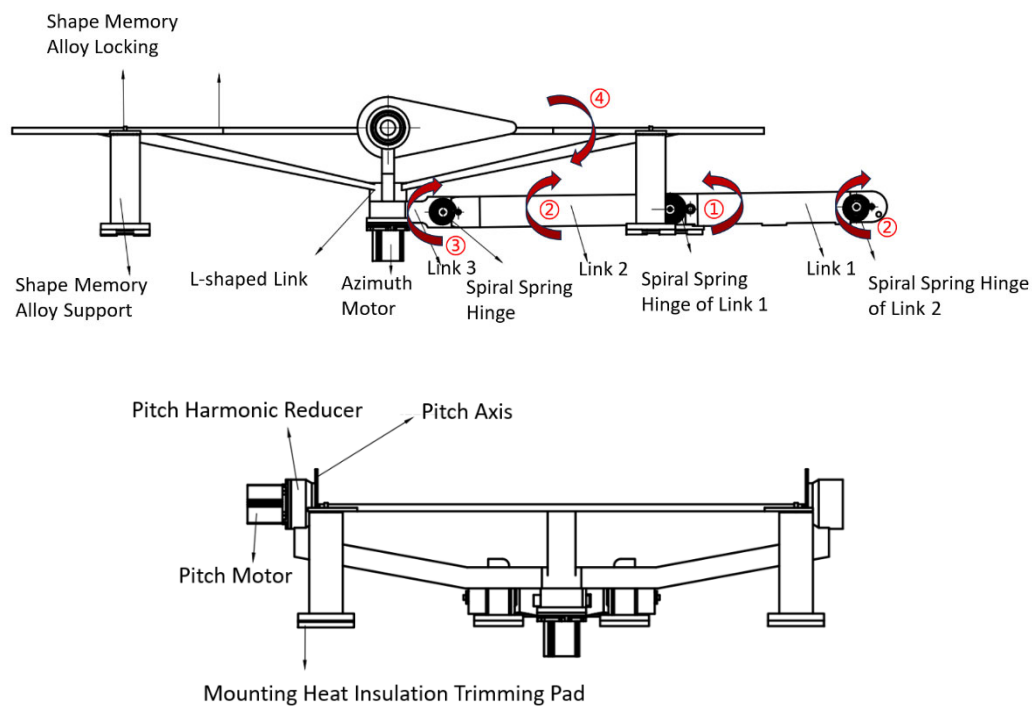


Figure 3. Folded-state structural design and passive deployment motion directions of the concentrating foldable mechanism.

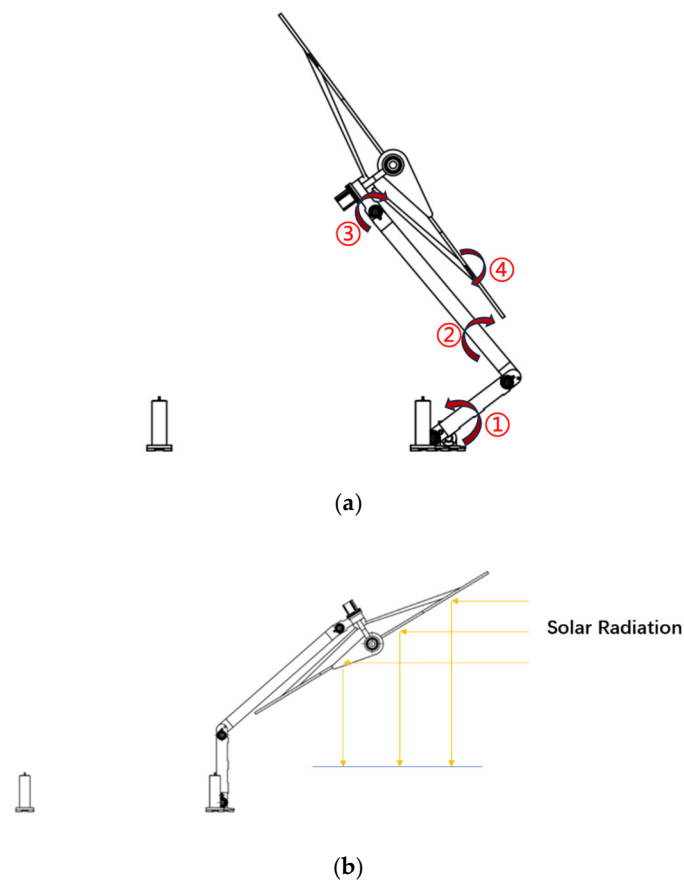


Figure 4. Deployment states of the concentrating foldable mechanism: (a) Intermediate deployment state of the concentrating foldable mechanism; (b) Deployed operational configuration of the concentrating foldable mechanism.

This result indicates that the deployment stage can be described by three independent generalized coordinates corresponding to the rotations of the three passive spring hinges. The passive hinges provide the associated driving torques, while the terminal positioning locks constrain these deployment degrees of freedom after full extension. The mechanism is therefore transformed from a reconfigurable open-chain system into a locked support structure for subsequent dual-axis solar tracking.

3. Kinematic Analysis of the Mechanism

The purpose of the kinematic analysis is to establish the mapping relationship between the hinge rotations of the deployable support and the spatial position of the concentrating reflector. This analysis provides the theoretical basis for deployment trajectory planning, dynamic modeling, and subsequent parameter optimization. During the deployment stage, the azimuth-pitch tracking axes are locked, and the reflector, L-shaped bracket, and terminal link are treated as a rigidly connected body. Therefore, the deployable mechanism can be simplified as an equivalent planar serial linkage composed of revolute joints and rigid links. In this section, the deployment process is first constrained by motion-envelope analysis, and then the kinematic model is established using vector-loop formulation and forward kinematic verification.

3.1. Deployment Analysis of the Concentrating Mechanism

The proposed concentrating mechanism is passively driven by spring hinges during deployment. Unlike an actively controlled multi-joint system, the passive deployment process is more sensitive to the initial release state, joint coordination, and link-reflector interference. Therefore, the deployment path must be planned before dynamic modeling to ensure that the moving links remain within the allowable motion envelope.

Several representative deployment configurations are shown in Figure 5. Configurations (a)–(d) correspond to possible interference states, in which the reflector or supporting links violate the allowable deployment envelope. Configuration (e) represents the desired deployment state. These configurations indicate that the deployment problem should not be treated only as a geometric unfolding process but as a constrained motion-planning problem for a coupled linkage.

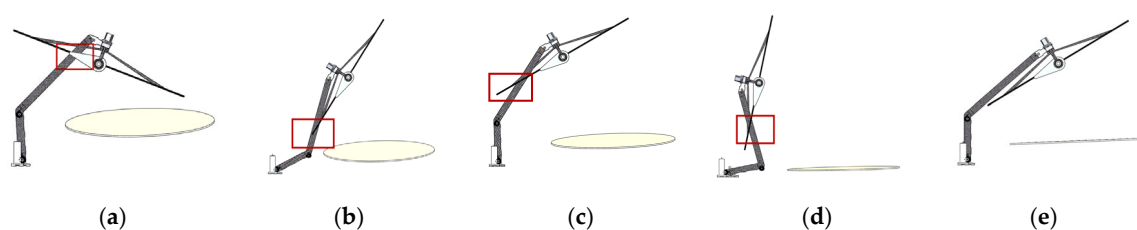


Figure 5. Several representative Deployment Configurations of the Concentrating Foldable Mechanism: (a–d) Interfering configurations; (e) correct deployment configuration.

Taking the interference-free configuration as the target state, the deployment trajectory of the mechanism is planned by constraining the motion envelope of the terminal link and the reflector. The planned deployment path is shown in Figure 6, where the end-joint trajectory and the reflector displacement envelope are used to evaluate the feasibility of the deployment motion. The corresponding joint-angle planning curves are shown in Figure 7. These curves provide the prescribed kinematic inputs for the subsequent vector analysis and dynamic simulation.

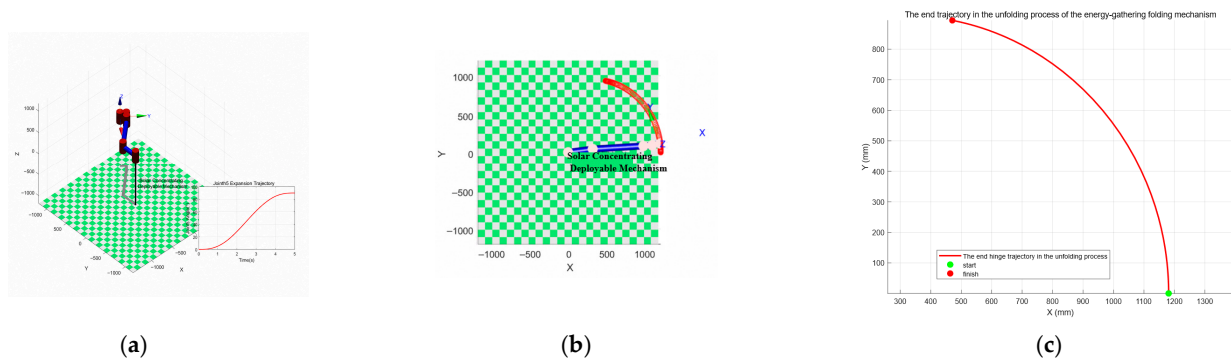


Figure 6. Deployment Trajectory and Motion Envelope of the Concentrating Foldable Mechanism (a) Three-dimensional deployment trajectory and end-joint time-angle curve; (b) Top-view deployment trajectory of the end joint and reflector; (c) Planned end-joint displacement trajectory.

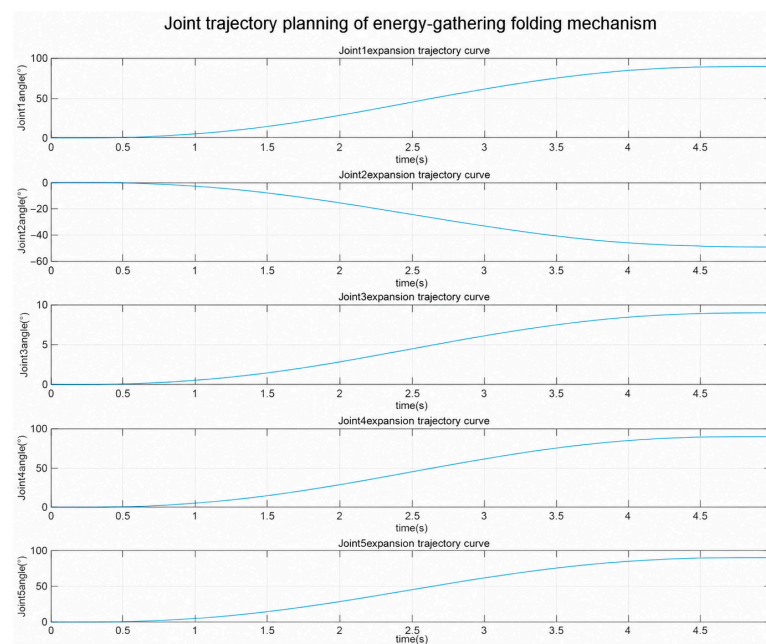


Figure 7. Joint-Angle Planning Curves During Deployment.

3.2. Deployment Process Simplification and Vector-Based Kinematic Model

To establish an analytical kinematic model, the deployable support is simplified according to the following assumptions: the links are treated as rigid bodies, the passive hinges are modeled as ideal revolute joints, the geometric dimensions of the hinge connectors are neglected, and the tracking motors remain locked during deployment. The resulting simplified model of the concentrating foldable mechanism in the folded state is shown in Figure 8, and its top view is presented in Figure 9. Under these assumptions, Rod 1, Rod 2, and Rod 3 are represented by line segments AB, BC, and CD, respectively. The L-shaped bracket connecting Rod 3 and the reflector is represented by segment DE, and the reflector is treated as a rigid body attached to the terminal link.

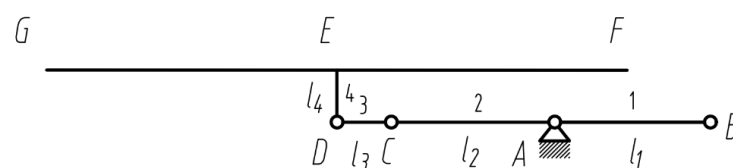


Figure 8. Simplified Diagram of the Concentrating Foldable Mechanism in the Folded state.

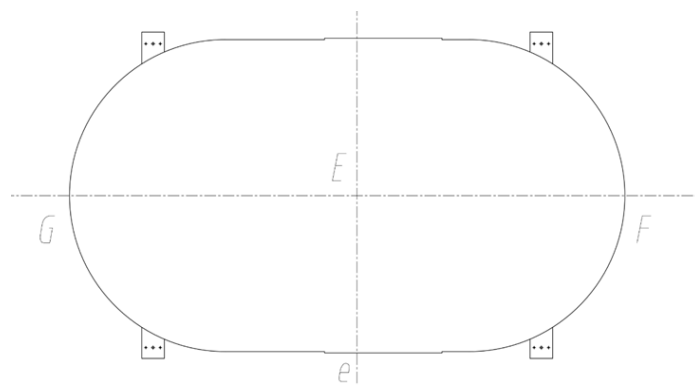


Figure 9. Top View of the Concentrating Reflector.

The intermediate and final deployment states are shown in Figure 10 and Figure 11, respectively. These two states define the geometric boundary conditions of the deployment process and are used to determine the initial and terminal configurations of the kinematic model.

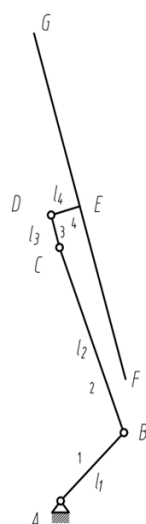


Figure 10. Intermediate Deployment State of the Concentrating Foldable Mechanism.

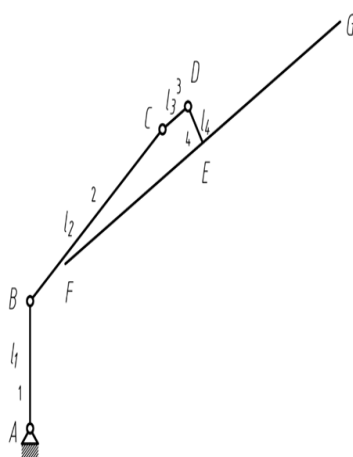


Figure 11. Final Deployment State of the Concentrating Foldable Mechanism.

Taking point A as the origin, a Cartesian coordinate system is established for the equivalent planar linkage. The vectors, link orientation angles, and key points of the

mechanism are defined in Figure 12, and the coordinates of the main points are listed in Table 1.

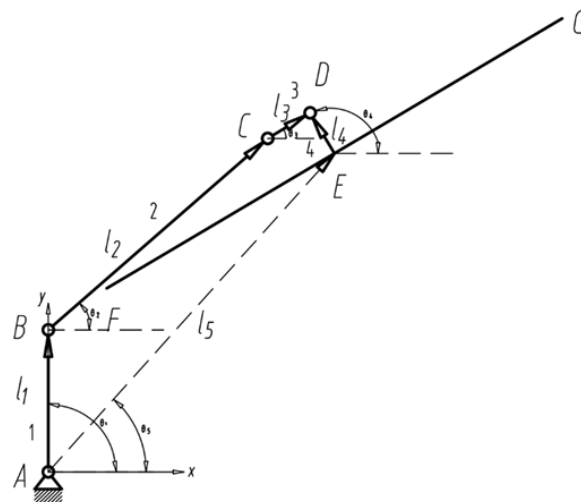


Figure 12. Simplified Vector Analysis Diagram of the Concentrating Foldable Mechanism.

Table 1. Key Points and Their Coordinates of the Concentrating Foldable Mechanism.

Key Points	Coordinates/mm
A	(0, 0)
B	(0, 330)
C	(509.5, 445.1)
D	(607.9, 504.6)
E	(664.8, 410.4)

The vector-loop method is then used to describe the position relationship among the links. By connecting points A, B, C, D, and E, the vector loop of the equivalent linkage can be written as:

$$\vec{l}_1 + \vec{l}_2 = \vec{l}_5 + \vec{l}_4 + \vec{l}_3 \quad (1)$$

It should be noted that this vector loop is an auxiliary geometric closure constructed to determine the centroid position of the reflector. It does not represent an actual closed-chain constraint of the deployable support, which remains an equivalent planar open-chain mechanism during deployment. In this formulation, $\vec{l}_5$ is the position vector from the fixed base point A to the reflector centroid E, rather than a physical structural link. Therefore, l_5 and θ_5 are dependent variables that vary with the prescribed link orientation angles.

The above equation is expressed in the complex form as:

$$l_1 e^{i\theta_1} + l_2 e^{i\theta_2} = l_5 e^{i\theta_5} + l_4 e^{i\theta_4} + l_3 e^{i\theta_3} \quad (2)$$

where all vector orientation angles are measured counterclockwise from the positive direction of the global X-axis. By expanding the complex equation using Euler's formula and equating the real and imaginary parts, the scalar position equations of the mechanism can be obtained as:

$$l_1(\cos \theta_1 + i \sin \theta_1) + l_2(\cos \theta_2 + i \sin \theta_2) = l_5(\cos \theta_5 + i \sin \theta_5) + l_4(\cos \theta_4 + i \sin \theta_4) + l_3(\cos \theta_3 + i \sin \theta_3) \quad (3)$$

$$l_1 \cos \theta_1 + l_2 \cos \theta_2 = l_5 \cos \theta_5 + l_4 \cos \theta_4 + l_3 \cos \theta_3 \quad (4)$$

$$l_1 \sin \theta_1 + l_2 \sin \theta_2 = l_5 \sin \theta_5 + l_4 \sin \theta_4 + l_3 \sin \theta_3 \quad (5)$$

In these equations, $l_1 - l_4$ are known structural dimensions, and $\theta_1 - \theta_4$ are the prescribed absolute orientation angles of the corresponding vectors. The magnitude l_5 and orientation angle θ_5 of the auxiliary position vector $\vec{l}_5$ are dependent variables to be determined.

To avoid ambiguity in the subsequent derivation, the auxiliary position vector $\vec{l}_5$ is decomposed into its Cartesian components. By rearranging Equations (4) and (5), these components are obtained as

$$\begin{aligned} X &= l_1 \cos \theta_1 + l_2 \cos \theta_2 - l_3 \cos \theta_3 - l_4 \cos \theta_4 \\ Y &= l_1 \sin \theta_1 + l_2 \sin \theta_2 - l_3 \sin \theta_3 - l_4 \sin \theta_4 \end{aligned} \quad (6)$$

where X and Y are the horizontal and vertical components of $\vec{l}_5$, respectively. Therefore, the orientation angle and magnitude of the auxiliary position vector can be obtained as:

$$\begin{aligned} \theta_5 &= \text{atan2}(Y, X) \\ l_5 &= \sqrt{X^2 + Y^2} \end{aligned} \quad (7)$$

Accordingly, the position components of the reflector centroid are given by:

$$\begin{cases} s_5 = X = l_5 \cos \theta_5 \\ h_5 = Y = l_5 \sin \theta_5 \end{cases} \quad (8)$$

By differentiating the displacement equation with respect to time, the velocity and acceleration of the reflector centroid are obtained as:

The velocity of the centroid is:

$$\begin{cases} v_x = \dot{l}_5 \cos \theta_5 - l_5 \dot{\theta}_5 \sin \theta_5 \\ v_y = \dot{l}_5 \sin \theta_5 + l_5 \dot{\theta}_5 \cos \theta_5 \end{cases} \quad (9)$$

The acceleration of the centroid is:

$$\begin{cases} a_x = \ddot{l}_5 \cos \theta_5 - 2\dot{l}_5 \dot{\theta}_5 \sin \theta_5 - l_5 \ddot{\theta}_5 \sin \theta_5 - l_5 \dot{\theta}_5^2 \cos \theta_5 \\ a_y = \ddot{l}_5 \sin \theta_5 + 2\dot{l}_5 \dot{\theta}_5 \cos \theta_5 + l_5 \ddot{\theta}_5 \cos \theta_5 - l_5 \dot{\theta}_5^2 \sin \theta_5 \end{cases} \quad (10)$$

To examine the consistency of the analytical kinematic solution, a corresponding numerical model is established using the same geometric parameters and prescribed joint-angle inputs. In this model, the passive spring–hinge joints are represented by equivalent revolute joint drives at points A , B , and C . The displacement, velocity, and acceleration responses of the reflector centroid are extracted and compared with the analytical predictions obtained from Equations (8)–(10). As shown in Figure 13, the numerical responses show the same variation trends as the theoretical results, indicating that the established kinematic model can describe the main deployment motion of the reflector centroid. The continuous centroid responses also suggest that the planned trajectory does not introduce abrupt kinematic changes during deployment.

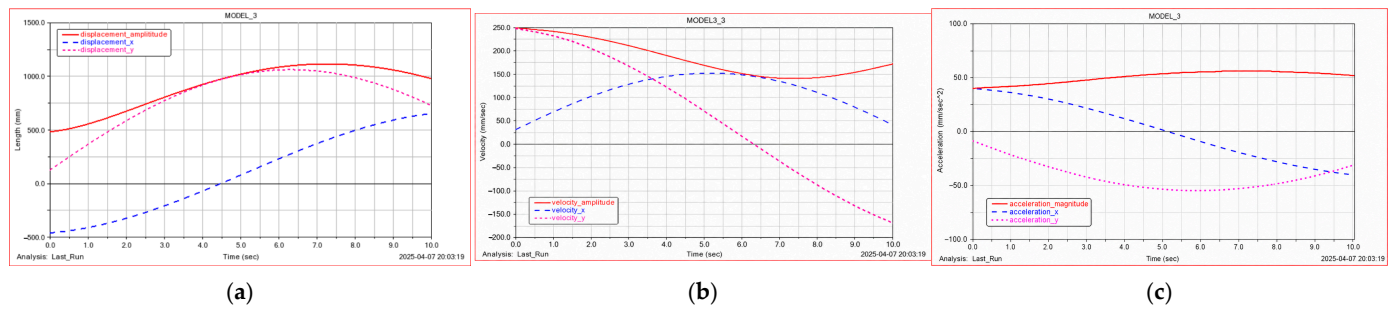


Figure 13. Numerical Kinematic Responses of the Reflector Centroid During Deployment. (a) Velocity components; (b) Displacement components; (c) Acceleration components.

3.3. Forward Kinematic Model and Numerical Verification

To further verify the consistency of the vector-based kinematic formulation, an equivalent forward kinematic model is established. The deployable support is modeled as a planar open-chain linkage with revolute joints, and joint coordinate systems are assigned along the deployment chain. The Denavit–Hartenberg (D-H) parameters of the equivalent joints are listed in Table 2, and the corresponding coordinate systems are shown in Figure 14.

Table 2. D-H Parameters Table of the Joints.

Joint i	Link Length a_{i-1}	Torsion Angle α_{i-1}	Offset d_i	Joint Angle θ_i	Joint Angle Range ($^\circ$)
1	0	0°	0	θ_1	$0^\circ \sim 90^\circ$
2	l_1	0°	0	θ_2	$48.8^\circ \sim 180^\circ$
3	l_2	0°	0	θ_3	$0^\circ \sim 10^\circ$
4	l_3	0°	0	θ_4	90°
5	l_4	0°	0	θ_5	90°

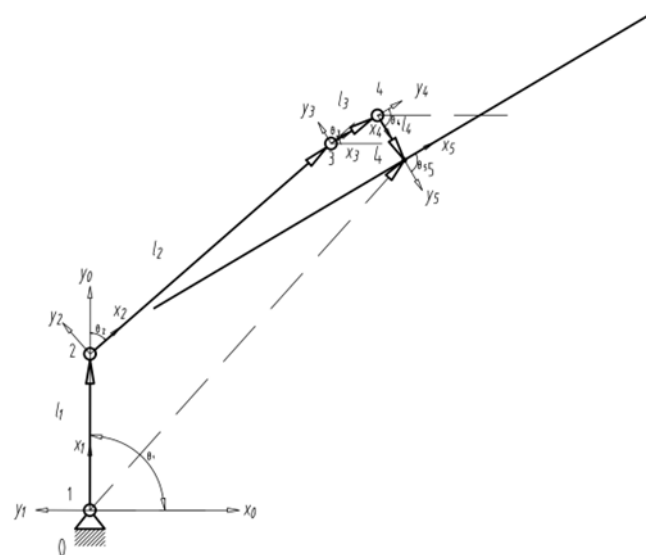


Figure 14. Joint Coordinate Systems of the Concentrating Foldable Mechanism.

The homogeneous transformation matrix between adjacent coordinate systems can be written as:

Joint Coordinate System {1} can be obtained by rotating from Joint Coordinate System {0}, and the homogeneous transformation matrix 0_1T A between them is:

$${}^0_1T = \begin{bmatrix} \cos \theta_1 & -\sin \theta_1 & 0 & 0 \\ \sin \theta_1 & \cos \theta_1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (11)$$

Joint Coordinate System {2} can be obtained by rotating from Joint Coordinate System {1}, and the homogeneous transformation matrix 1_2T between them is:

$${}^1_2T = \begin{bmatrix} \cos \theta_2 & -\sin \theta_2 & 0 & l_1 \\ \sin \theta_2 & \cos \theta_2 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (12)$$

Joint Coordinate System {3} can be obtained by rotating from Joint Coordinate System {2}, and the homogeneous transformation matrix 2_3T between them is:

$${}^2_3T = \begin{bmatrix} \cos \theta_3 & -\sin \theta_3 & 0 & l_2 \\ \sin \theta_3 & \cos \theta_3 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (13)$$

Joint Coordinate System {4} can be obtained by rotating from Joint Coordinate System {3}, and the homogeneous transformation matrix 3_4T between them is:

$${}^3_4T = \begin{bmatrix} \cos \theta_4 & -\sin \theta_4 & 0 & l_3 \\ \sin \theta_4 & \cos \theta_4 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (14)$$

Joint Coordinate System {5} can be obtained by rotating from Joint Coordinate System {4}, and the homogeneous transformation matrix 4_5T between them is:

$${}^4_5T = \begin{bmatrix} \cos \theta_5 & -\sin \theta_5 & 0 & l_4 \\ \sin \theta_5 & \cos \theta_5 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (15)$$

Joint Coordinate System {5} can be obtained by rotating from Joint Coordinate System {0}, and the homogeneous transformation matrix 0_5T between them is:

$${}^0_5T = {}^0_1T {}^1_2T {}^2_3T {}^3_4T {}^4_5T \quad (16)$$

By multiplying the adjacent homogeneous transformation matrices, the pose of the terminal link and the reflector can be obtained. This forward kinematic model provides an independent verification route for the vector-loop solution and confirms whether the planned joint-angle inputs can reproduce the desired deployment trajectory.

The folding, intermediate, and fully deployed states obtained from the forward kinematic model are shown in Figure 15. The simulated deployment sequence is consistent with the planned motion path, indicating that the equivalent kinematic model can describe the main deployment behavior of the mechanism.

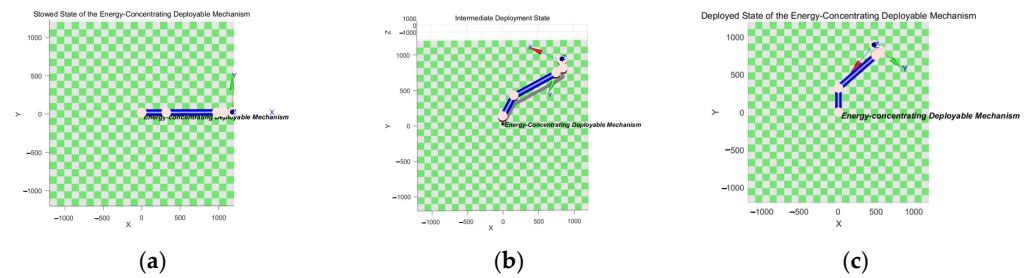


Figure 15. Folding and Deployment States of the Concentrating Foldable Mechanism. (a) Folded State; (b) Intermediate Deployment State; (c) Fully Deployed State.

The motion curves of the terminal joint are shown in Figure 16. The curves provide displacement-level verification of the planned trajectory and serve as the input basis for the subsequent dynamic analysis.

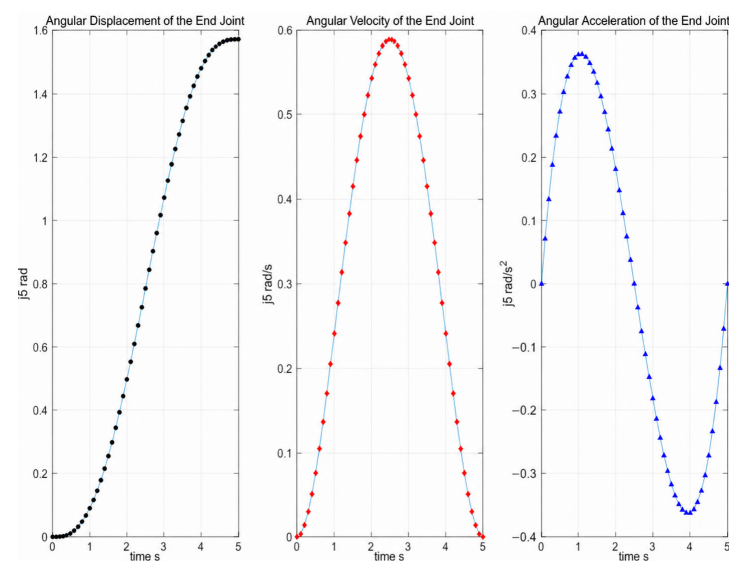


Figure 16. Motion Curves of the End Joint.

The first three deployment joints correspond to the passive spring–hinge joints of the mechanism. Their angular acceleration curves are shown in Figure 17. The absence of abrupt discontinuities in the acceleration curves indicates that the planned trajectory does not introduce sudden kinematic changes during deployment. Therefore, the kinematic model provides a feasible basis for the dynamic response analysis and driving-torque evaluation in Section 4.

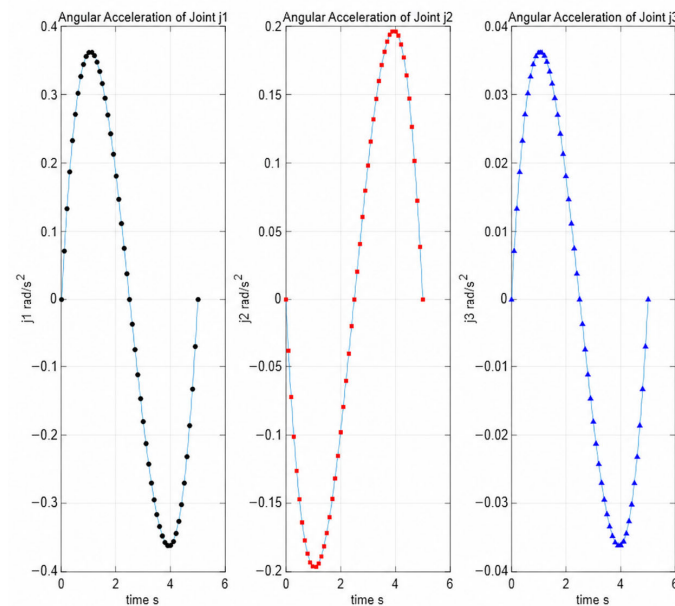


Figure 17. Angular Acceleration Curves of the First Three Deployment Joints.

4. Dynamic Modeling and Verification of the Deployable Mechanism

The deployment of the proposed concentrating mechanism is a transient passive motion driven by spring hinges after release. Although the kinematic analysis in Section 3 provides the planned deployment trajectory, kinematic feasibility alone is insufficient to evaluate the torque demand, joint reaction forces, acceleration response, and locked-state stiffness of the mechanism. Therefore, dynamic modeling and numerical verification are carried out in this section. First, the force equilibrium of each link is established based on D'Alembert's principle. Then, the deployment dynamics are formulated using the Lagrange equation. Finally, multibody dynamic simulation and finite-element modal analysis is used to examine the deployment response and locked-state dynamic characteristics.

4.1. Dynamic Modeling of the Deployment Process

4.1.1. Force Analysis Based on D'Alembert's Principle

During deployment, the azimuth and pitch tracking motors are locked, and the mechanism is driven by the passive spring hinges. The foldable support is simplified as a rigid multi-link system. The L-shaped bracket and the reflector are treated as Link 4, and the reflector mass is included in the dynamic analysis of this terminal link. At this stage, the dominant mechanical quantities considered in the equivalent dynamic equilibrium include gravity, joint reaction forces, hinge torque, and the equivalent inertial force and inertial moment introduced according to D'Alembert's principle. The acceleration values of each link have been obtained through the aforementioned kinematic analysis, so the inertial force and inertial couple of each link can be determined.

The force analysis of the concentrating foldable mechanism is shown in the Figure 18 below:

The mass of Link 1 is 0.588 kg, Link 2 is 0.837 kg, Link 3 is 0.3 kg, and Link 4 is 1.3 kg. $M_i (i \geq 1)$ is the inertial couple on the link, and M_0 is the balancing couple acting on Link 1. According to D'Alembert's principle, the dynamic equilibrium of each link can be expressed by setting the resultant force in the x- and y-directions and the resultant moment about the centroid equal to zero:

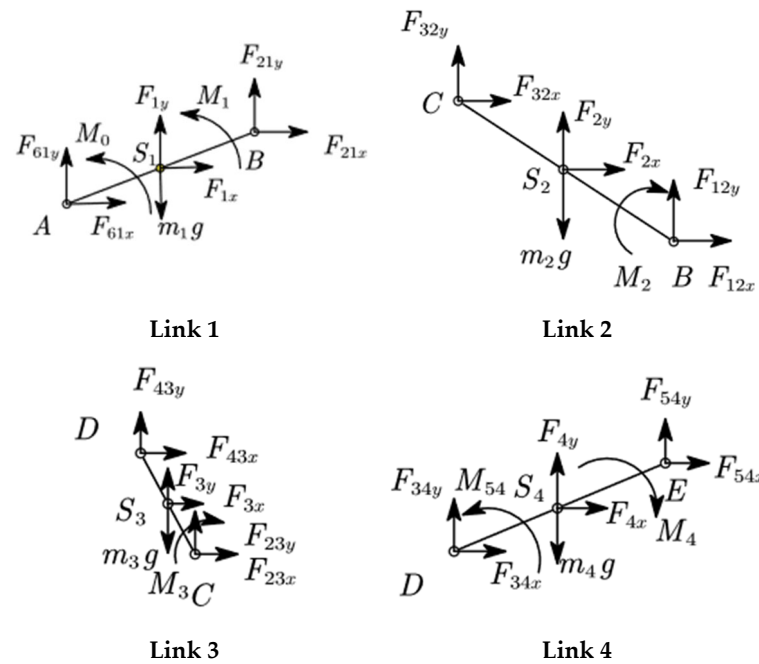


Figure 18. Force Analysis of the Links.

For Link 1:

$$F_{61x} + F_{21x} + F_{1x} = 0 \quad (17)$$

$$F_{61y} + F_{21y} + F_{1y} - m_1g = 0 \quad (18)$$

$$F_{21x}(y_B - y_{S1}) - F_{21y}(x_B - x_{S1}) + F_{61y}(x_{S1} - x_A) - F_{61x}(y_{S1} - y_A) - M_1 - M_0 = 0 \quad (19)$$

For Link 2:

$$F_{12x} + F_{32x} + F_{2x} = 0 \quad (20)$$

$$F_{12y} + F_{32y} - m_2g + F_{2y} = 0 \quad (21)$$

$$F_{32x}(y_C - y_{S2}) + F_{32y}(x_C - x_{S2}) - F_{12x}(y_{S2} - y_B) - F_{12y}(x_B - x_{S2}) + M_2 = 0 \quad (22)$$

For Link 3:

$$F_{23x} + F_{43x} + F_{3x} = 0 \quad (23)$$

$$F_{23y} + F_{43y} + F_{3y} - m_3g = 0 \quad (24)$$

$$F_{43x}(y_D - y_{S3}) + F_{43y}(x_{S3} - x_D) - F_{23x}(y_{S3} - y_C) - F_{23y}(x_C - x_{S3}) + M_3 = 0 \quad (25)$$

For Link 4:

$$F_{34x} + F_{54x} + F_{4x} = 0 \quad (26)$$

$$F_{34y} + F_{54y} + F_{4y} - m_4g = 0 \quad (27)$$

$$F_{34y}(x_{S4} - x_D) - F_{34x}(y_{S4} - y_D) + F_{54x}(y_E - y_{S4}) - F_{54y}(x_E - x_{S4}) + M_4 - M_{54} = 0 \quad (28)$$

After sorting out the aforementioned equilibrium equations, the unknown quantities include the reaction forces of each kinematic pair, the balancing couple M_0 and the reaction couple M_{54} . These unknowns are organized into a system of linear equations, which is expressed in matrix form as follows:

$$A_1 F_q = B_1 \quad (29)$$

where A_1 is the coefficient matrix; F_q is the column matrix composed of unknown quantities; B_1 is the column matrix composed of known parameters.

Among them:

$$A_1 = \begin{bmatrix} 0 & 1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -1 & -(y_{S1} - y_A) & (x_{S1} - x_A) & (y_B - y_{S1}) & -(x_B - x_{S1}) & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -(y_{S2} - y_B) & -(x_B - x_{S2}) & (y_C - y_{S2}) & (x_C - x_{S2}) & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -(y_{S3} - y_C) & -(x_C - x_{S3}) & (y_D - y_{S3}) & (x_{S3} - x_D) & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & -(y_{S4} - y_D) & (x_{S4} - x_D) & (y_E - y_{S4}) & -(x_E - x_{S4}) & -1 \end{bmatrix} \quad (30)$$

$$F_q = \begin{bmatrix} M_0 \\ F_{61x} \\ F_{61y} \\ F_{21x} \\ F_{21y} \\ F_{23x} \\ F_{23y} \\ F_{34x} \\ F_{34y} \\ F_{45x} \\ F_{45y} \\ M_{54} \end{bmatrix} \quad B_1 = \begin{bmatrix} -F_{1x} \\ m_1 g - F_{1y} \\ M_1 \\ -F_{2x} \\ m_2 g - F_{2y} \\ -M_2 \\ -F_{3x} \\ m_3 g - F_{3y} \\ -M_3 \\ -F_{4x} \\ m_4 g - F_{4y} \\ -M_4 \end{bmatrix}$$

4.1.2. Lagrange Dynamic Model of Passive Deployment

To further describe the coupled deployment process, the Lagrange equation is used to establish the system-level dynamic model. The concentrating foldable mechanism is modeled as a constrained multibody system composed of four rigid components. Before the mechanism reaches the fully deployed configuration, the tracking motors are inactive, and the deployment motion is generated by the passive spring hinges. Because the generalized coordinates of the links are constrained by the revolute joints, the system coordinates are not independent.

$$A\ddot{q} + \phi_q^T \lambda = B \quad (31)$$

$$\phi(q, t) = 0$$

Among them, A and B are coefficient matrices, ϕ is the constraint equation of the position coordinate q , ϕ_q is the Jacobian matrix of the constraint equation, and λ is the Lagrange multiplier. This equation is the Euler-Lagrange equation. From Equation (31), the Euler-Lagrange equation of the system can be expressed in more detail as follows:

$$I(q, t)\ddot{q} + \phi_q^T(q, t)\lambda - Q(q, \dot{q}, t) = 0 \quad (32)$$

$$\phi(q, t) = 0$$

where q , $\dot{q}$, and $\ddot{q}$ represent the displacement, velocity, and acceleration of the system; $I \in R^{n \times n}$ is the inertia matrix of the concentrating foldable system; and $Q \in R_n$ is the vector of external forces acting on the system.

By calculating the first-order and second-order derivatives of $\phi(q, t) = 0$, the expressions for velocity and acceleration can be obtained as follows:

$$\dot{\phi}(q, \dot{q}, t) = \phi_q(q, t)\dot{q} - v(q, t) = 0 \quad (33)$$

$$\ddot{\phi}(q, \dot{q}, \ddot{q}, t) = \phi_q(q, t)\ddot{q} - \eta(q, \dot{q}, t) = 0$$

From Equation (33):

$v(q, t) = \phi_q(q, t)\dot{q} - \dot{\phi}(q, \dot{q}, t)$, $\eta(q, \dot{q}, t) = \phi_q(q, t)\ddot{q} - \ddot{\phi}(q, \dot{q}, \ddot{q}, t)$, we have where are the velocity expression and acceleration expression, respectively. The initial conditions of the concentrating foldable system are as follows:

$$q(0) = q_0 \quad (34)$$

$$\dot{q}(0) = \dot{q}_0$$

This dynamic model provides the theoretical basis for calculating the required hinge torque and evaluating the deployment response in the following simulation analysis.

4.1.3. Modal Analysis

After deployment and terminal locking, the mechanism is transformed from a re-configurable linkage into a locked support structure. The locked-state stiffness directly affects the reflector's ability to maintain its optical working attitude. Therefore, modal analysis is carried out to evaluate the dynamic characteristics of the mechanism in the locked configuration.

The free vibration equation of the finite-element model can be expressed as:

$$M\ddot{y}(t) + Ky(t) = 0 \quad (35)$$

Among them, M is the overall mass matrix of the mechanism, and K is the overall stiffness matrix. The global mass matrix M and stiffness matrix K are assembled from the element mass matrices and element stiffness matrices, respectively. $y(t)$ represents displacement, and $\ddot{y}(t)$ represents acceleration. The solution to Equation (35) above is:

$$y(t) = \Psi \sin(\omega t + \varphi), \ddot{y}(t) = -\omega^2 \Psi \sin(\omega t + \varphi) \quad (36)$$

Among them, Ψ represents the amplitude vector, ω is the circular frequency, and φ is the initial phase angle. Substituting Equation (36) into Equation (35), the characteristic equation of Equation (35) is obtained as follows:

$$(K - \omega^2 M)\Psi = 0 \quad (37)$$

The natural frequencies and mode shapes can then be obtained by solving the characteristic equation. In this study, the modal solution is used to examine whether the locked mechanism avoids low-frequency flexible modes under the assumed boundary conditions.

4.2. Dynamic Simulation and Response Verification

4.2.1. Multibody Dynamic Simulation of Deployment

The planned joint trajectories obtained in Section 3 are used as reference deployment trajectories for the multibody dynamic analysis. The dynamic model incorporates the passive spring hinges, gravitational effects, inertial effects, and joint constraints to evaluate the corresponding torque response during deployment. The angular displacement, angular velocity, and angular acceleration curves of the first three spring-hinge joints are shown in Figure 19. These curves define the reference motion conditions for evaluating the deployment torque response.

A multibody dynamic model is established using the same geometric parameters, mass properties, and joint definitions as the theoretical model. The passive spring-hinge configuration and the tilted lunar-landing condition are further considered in this model. Six planar spiral springs are symmetrically arranged in pairs at the three deployment hinges A , B , and C . Each deployment hinge is modeled as a revolute joint equipped with an equivalent torsional spring. If k_i denotes the stiffness of a single spiral spring, the equivalent hinge stiffness is $k_{eq,i} = 2k_i$, $i = A, B, C$.

The passive spring hinges are the primary driving components responsible for autonomous deployment after the SMA-based release. Therefore, an accurate representation of the spring torque is essential for evaluating the deployment feasibility. The torque generated by each spring hinge is expressed as:

$$T_{s,i} = k_{eq,i}(\theta_{0,i} - \theta_i) \quad (38)$$

where $T_{s,i}$ represents the torque generated by the i -th spring hinge, $k_{eq,i}$ is the equivalent torsional stiffness, $\theta_{0,i}$ is the preloaded or reference angular position, and θ_i is the instantaneous joint angle during deployment.

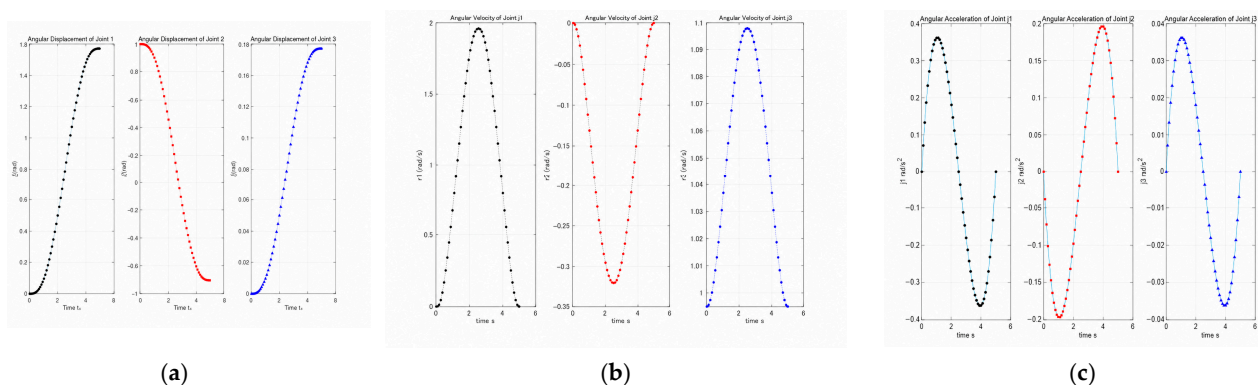


Figure 19. Kinematic parameters of the first three hinge joints (a) Angular displacements; (b) Angular velocities; (c) Angular accelerations.

To account for a possible landing attitude deviation, a 10° landing-tilt condition is introduced by resolving the lunar gravitational acceleration into the local coordinate system of the mechanism. With $g_m = 1.62 \text{ m/s}^2$, the lateral and axial acceleration components are calculated as: $a_{\parallel} = g_m \sin 10^\circ = 0.281 \text{ m/s}^2$, $a_{\perp} = g_m \cos 10^\circ = 1.595 \text{ m/s}^2$.

Since Link 3 is directly connected to the reflector-supporting terminal part, its driving torque is selected as a representative index for evaluating the deployment torque demand. The simulated driving torque of Link 3 is shown in Figure 20.

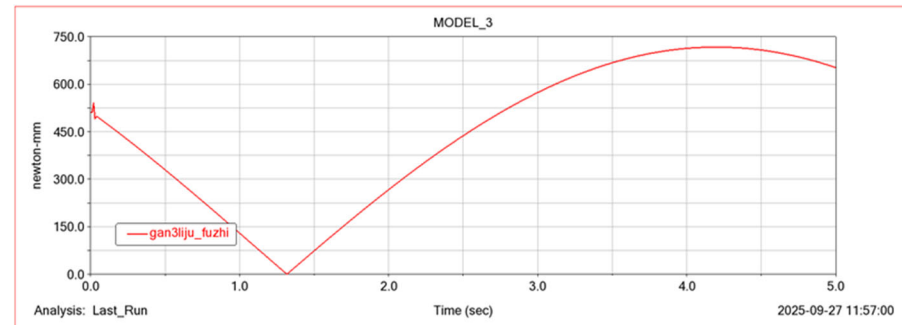


Figure 20. Driving Torque Diagram of Link 3.

For comparison, the analytical driving torque of Link 3 is obtained from the dynamic model, as shown in Figure 21.

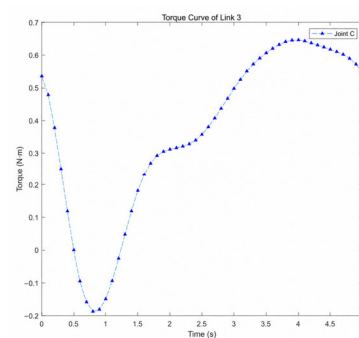


Figure 21. Analytical Driving Torque of Link 3.

The simulated driving torque of Link 3 varies within 0–0.70 N·m, while the analytical prediction gives a range of 0–0.68 N·m. The relative peak-value error between the analytical and simulated torque results is 2.8%, indicating that the dynamic model can capture the main torque response of the deployment process. In addition, the maximum angular acceleration of the hinge joints is 0.08 rad/s^2 . Under the assumed inertial parameters, the maximum equivalent external force is 0.7 N. These results indicate that the planned deployment trajectory leads to a continuous dynamic response and a relatively low torque demand under the present modeling assumptions. Since the multibody dynamic model incorporates the equivalent torsional spring joints, the consistency between the simulated and analytical torque responses supports the applicability of the spring–hinge representation under the prescribed deployment trajectory.

For a practical motorization and deployment-margin assessment, additional non-ideal effects must be considered when the concept is further matured. These effects include hinge friction, spring hysteresis, manufacturing and assembly tolerances, joint clearance, harness stiffness associated with the azimuth and pitch motors, and possible magnetic or thermal-vacuum effects. In the present theoretical stage, the calculated torque demand is therefore interpreted as the ideal baseline value. Prototype-level tests will be required to determine the applicable torque margin and to update the spring preload and stiffness values under realistic boundary conditions.

4.2.2. Finite-Element Modal Verification of the Locked Mechanism

To evaluate the stiffness of the mechanism after deployment and locking, a finite-element model of the locked configuration is established. The material properties, mass distribution, and terminal locking constraints are assigned according to the designed structure. The first six mode shapes of the concentrating foldable mechanism are shown in Figure 22.

The first six natural frequencies and mode shapes of the mechanism are extracted and summarized in Table 3.

Table 3. First Six Order Modes and Mode Shapes of the Concentrating Foldable Mechanism.

Order	Frequency (Hz)	Mode Shape
1	54.969	Vertical yaw of the mirror and Link 2
2	69.903	Torsion of the mirror around the X-axis
3	105.46	Yaw of the mirror around the Z-axis
4	118.15	Butterfly like yaw of the mirror around the Z-axis
5	125.47	Torsion of the mirror around the Y-axis
6	134.57	Torsion of the mirror along the X-axis

The first natural frequency of the locked mechanism is 54.969 Hz. This result indicates that no low-frequency global flexible mode appears in the locked configuration under the present boundary conditions. Therefore, the modal analysis provides a stiffness-level verification for the subsequent optical working state of the concentrating mechanism.

It should be noted that the finite-element modal analysis in this study was conducted using nominal material properties under the assumed locked-state boundary conditions. The lunar thermal-vacuum environment was not directly coupled into the current modal model. In the actual lunar surface environment, the absence of atmospheric convection and the dominant radiative heat transfer may result in significant temperature variations, which can affect the elastic modulus, joint stiffness, and consequently the dynamic characteristics of the mechanism.

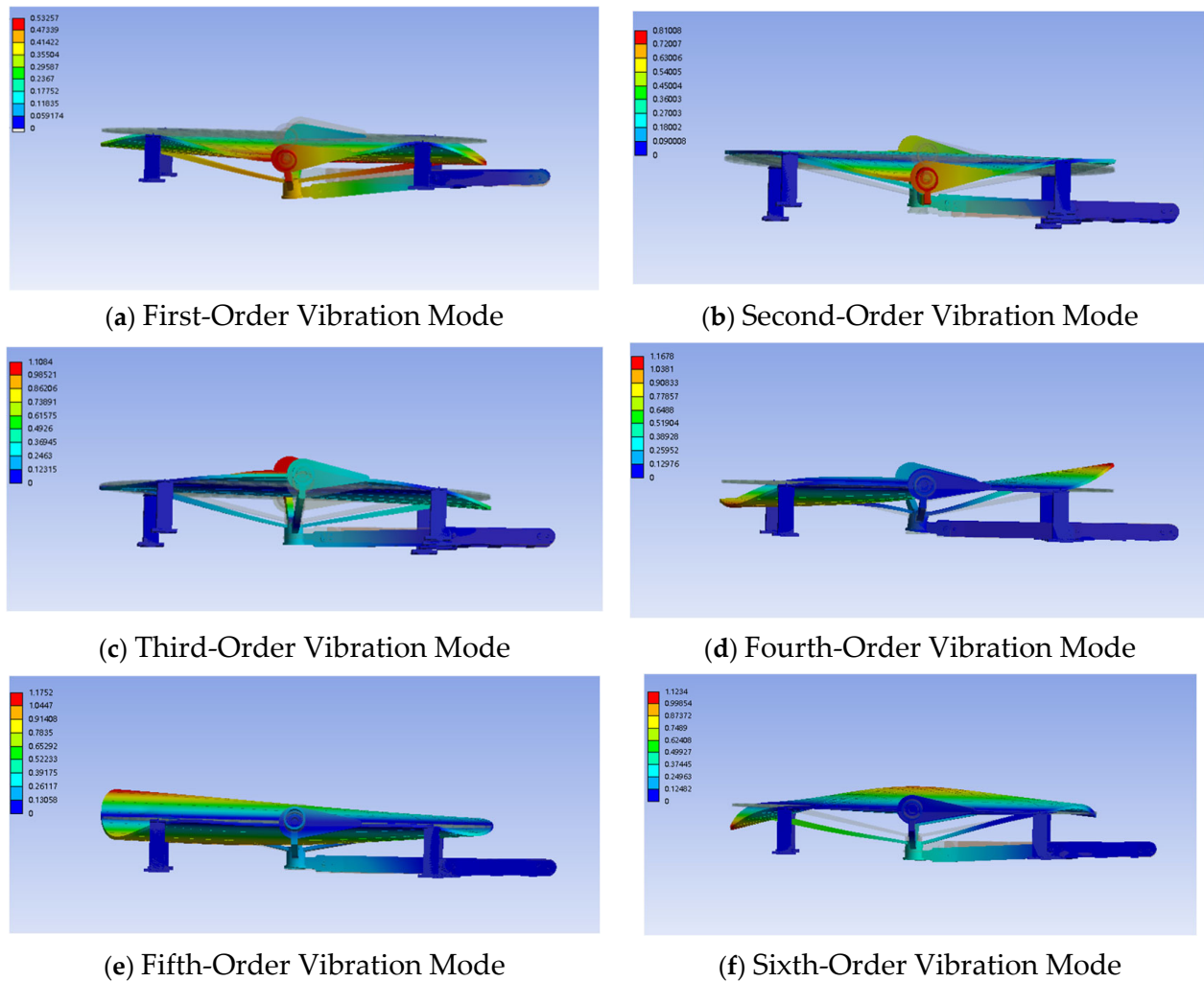


Figure 22. First Six-Order Modal Analysis of the Concentrating Foldable Mechanism.

Since the natural frequency is approximately proportional to the square root of the structural stiffness-to-mass ratio, a first-order sensitivity estimation can be conducted based on the variation in the effective elastic modulus. Assuming that the effective stiffness decreases by 10% and 20% due to temperature-induced stiffness variation, the first natural frequency would decrease from 54.969 Hz to approximately 52.1 Hz and 49.2 Hz, respectively. Therefore, the current modal result should be regarded as a nominal stiffness verification, while a fully coupled thermal-structural modal analysis will be considered in future prototype-level investigations.

5. Multi-Objective Parameter Optimization Design of the Mechanism Based on the Non-Dominated Sorting Genetic Algorithm (NSGA-II)

The core function of the energy-concentrating deployable mechanism is to achieve reliable deployment and energy concentration in space. The deployment performance of the concentrating foldable mechanism is affected by the geometric parameters of the links and the stiffness parameters of the passive spring hinges. The link lengths determine the folded envelope, deployed configuration, and reflector centroid trajectory, while the spring stiffness parameters influence the equivalent hinge torque and motion smoothness during deployment. In the present optimization model, deployment smoothness and synchronization accuracy are selected as the primary optimization objectives. Specifically, the acceleration response of the reflector centroid is used to characterize deployment

smoothness, and the joint-angle deviation from the planned deployment trajectory is used to evaluate synchronization accuracy. The hinge torque response, torque fluctuation, deployment ratio, and structural compactness are further evaluated as post-optimization performance indicators rather than being directly included in the objective functions. Under the prescribed geometric and dynamic constraints, the non-dominated sorting genetic algorithm II (NSGA-II) is adopted to obtain a set of Pareto-optimal solutions that balance smooth deployment and synchronized motion.

5.1. Optimization Objectives and Design Variables

According to the structural configuration and dynamic response analyzed in the previous sections, the link lengths and spring stiffness parameters are selected as design variables. The link lengths determine the folded envelope, deployed reflector position, and centroid trajectory, while the spring stiffness parameters affect the hinge driving torque and deployment smoothness. Therefore, these variables are used to optimize the deployment performance of the mechanism, as listed in Table 4. The spring stiffness variables listed in Table 4 correspond to the equivalent torsional stiffness parameters used in the spring–hinge joint model described in Section 4.2.1.

Table 4. Design Variables.

Design Variables	Physical Meaning	Value Range	Unit
l_1	Length of Link 1	[0.32, 0.34]	m
l_2	Length of Link 2	[0.660, 0.686]	m
l_3	Length of Link 3	[0.091, 0.125]	m
$k_{eq,1}$	Torque Coefficient (Equivalent torsional stiffness of hinge A)	[0.5, 5]	Nm/rad
$k_{eq,2}$	Torque Coefficient (Equivalent torsional stiffness of hinge B)	[0.5, 5]	Nm/rad
$k_{eq,3}$	Torque Coefficient (Equivalent torsional stiffness of hinge C)	[0.5, 5]	Nm/rad

In the optimization process, only the effective lengths of Link 1, Link 2, and Link 3 are selected as design variables, while the link cross-sectional profiles, material properties, and joint configurations are kept unchanged. Therefore, the link profiles do not scale linearly with the optimized lengths. For each candidate design, the mass properties and inertia parameters are recalculated according to the updated link lengths under the assumption of uniform material distribution and constant cross-sectional dimensions, and the interference-free deployment envelope is checked using the same geometric model described in Section 3. This assumption allows the influence of link-length variation on the deployment performance to be evaluated independently.

A multi-objective function is established to achieve the optimization objectives of improving the trajectory smoothness and motion stability of the concentrating foldable mechanism while reducing the synchronous deployment error. To evaluate the deployment smoothness, the acceleration response of the reflector centroid is selected as one objective index. A smaller centroid acceleration indicates a smoother deployment process and lower inertial disturbance to the reflector.

$$f_{1smooth} = \min \sqrt{\frac{1}{T} \int_0^T [\ddot{x}_c^2(t) + \ddot{y}_c^2(t)] dt} \quad (39)$$

Among them, $\ddot{x}_c(t)$ and $\ddot{y}_c(t)$ are the centroid accelerations of the concentrating reflector in the X and Y directions, respectively, and T is the total deployment time of the mechanism. Where the acceleration components of the reflector centroid and the total deployment time are defined according to the kinematic model in Section 3.

The synchronization error of the mechanism reflects the deviation between the actual deployment trajectory and the ideal trajectory, which is defined as the integral of the sum of squares of the angular displacement deviations of the first three joints:

$$f_{2error} = \min \left(\int_0^T [(\theta_1(t) - \theta_{1,ref}(t))^2 + (\theta_2(t) - \theta_{2,ref}(t))^2 + (\theta_3(t) - \theta_{3,ref}(t))^2] dt \right) \quad (40)$$

Among them, $\theta_{i,ref}(t)$ is the ideal angular displacement of the i -th joint, and $\theta_i(t)$ is the actual angular displacement. These are used to quantify the trajectory tracking deviation during deployment.

Meanwhile, to ensure the physical feasibility and deployment reliability of the concentrating foldable mechanism, the following constraint conditions are set:

Geometric constraints: The length of each link must meet the mechanism dimensions after deployment, match the installation space, and avoid mutual interference: $l_1 + l_2 \geq 0.8$, $l_2 + l_3 \geq 0.6$. In addition to the length constraints, the interference-free condition is checked using the deployment envelope established in Section 3; Dynamic constraints: To limit inertial disturbance during deployment, the peak angular acceleration of each joint is constrained as: $|\ddot{\theta}_i(t)| \leq 1.0 \text{ rad/s}^2 (i = 1, 2, 3)$.

5.2. Algorithm Implementation Process

The NSGA-II procedure is implemented to solve the above multi-objective optimization problem. The initial population is generated within the feasible ranges of the design variables. Each individual contains the link-length and spring-stiffness parameters, and its objective values are calculated using the kinematic and dynamic models established in the previous sections. Non-dominated sorting and crowding-distance calculation are then used to rank the individuals and maintain population diversity. Tournament selection, crossover, mutation, and elite retention are performed iteratively until the maximum generation number is reached.

In this study, the population size is set to 100, the crossover probability is 0.7, the mutation probability is 0.2, and the maximum number of generations is 50. The optimization procedure is shown in Figure 23.

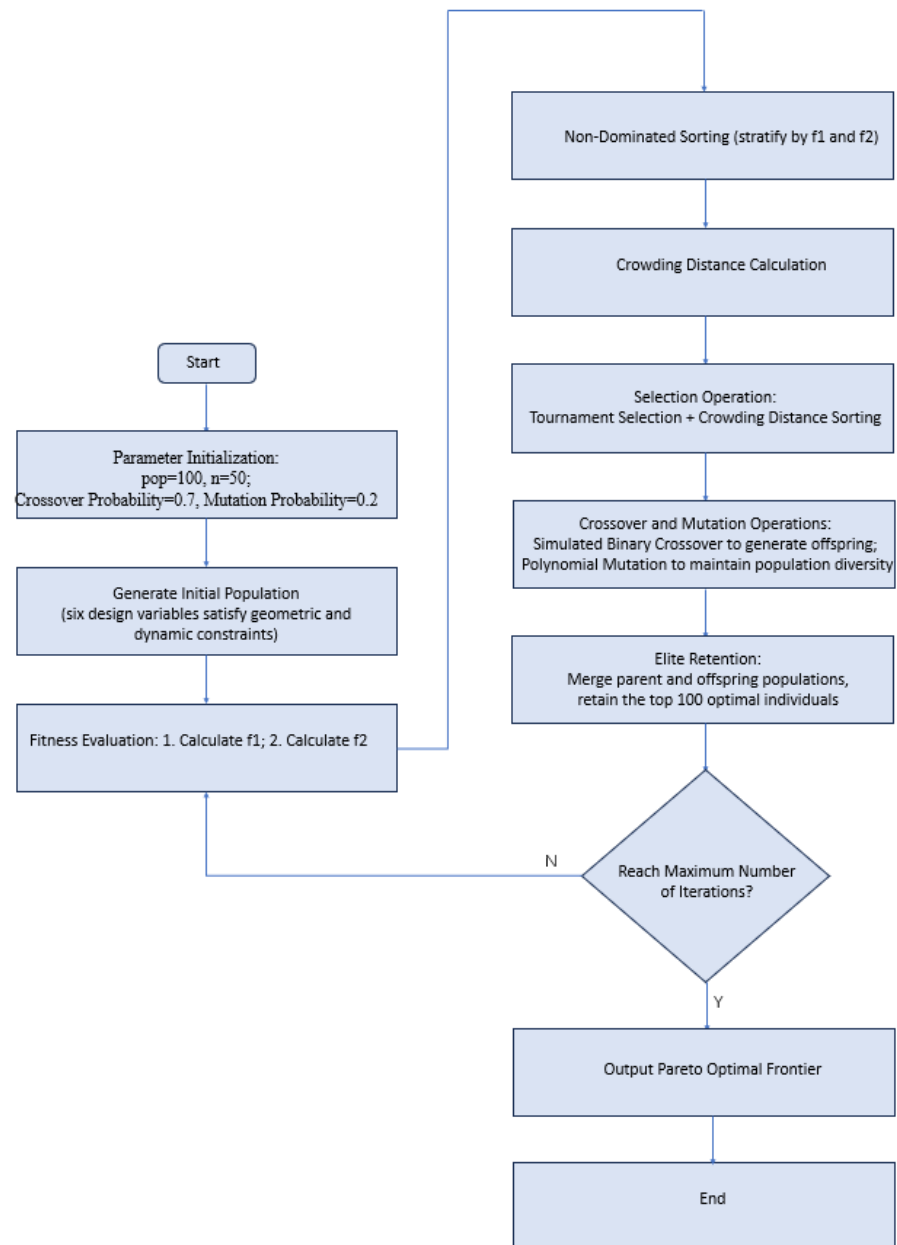


Figure 23. Flow Chart of Algorithm Implementation.

5.3. Pareto-Front Analysis and Solution Selection

The Pareto frontier obtained from the multi-objective optimization is shown in Figure 24. The frontier contains 14 non-dominated solutions, indicating that no single solution can simultaneously minimize all objective functions. Therefore, the final design should be selected by balancing deployment smoothness, synchronization accuracy, and structural feasibility.

Among the Pareto-optimal solutions, the third solution is selected as the final optimized parameter set because it provides a balanced compromise between centroid acceleration and synchronization error while satisfying the geometric constraints of the folded and deployed configurations. The selected optimization result is listed in Table 5.

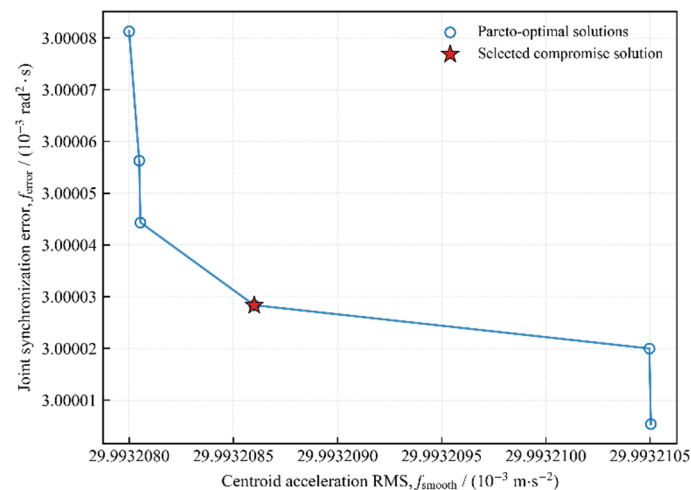


Figure 24. Pareto front obtained by NSGA-II optimization.

Table 5. Selected Pareto-Optimal Design Parameters.

$l_1(\text{m})$	$l_2(\text{m})$	$l_3(\text{m})$	$k_{eq,1}$ (Nm/rad)	$k_{eq,2}$ (Nm/rad)	$k_{eq,3}$ (Nm/rad)	Centroid Acceleration m/s ²	Synchronization Error rad ² ·s
0.32	0.66	0.091	0.99	1	1.1	0.02999320	0.003000028

5.4. Verification of the Optimized Parameters

The original and optimized parameters are compared in Table 6. After optimization, the folded and deployed dimensions of the mechanism are recalculated to evaluate the deployment ratio. The maximum deployment ratio increases from 5.6 to 7.2, corresponding to an increase of 28.57%.

Table 6. Comparison Table of Optimized Parameters and Original Parameters.

Optimization Indicator	Initial Design	Optimized Design	Modification Range
$l_1(\text{m})$	0.33	0.32	3.03%
$l_2(\text{m})$	0.676	0.660	2.36%
$l_3(\text{m})$	0.115	0.091	20.8%
$k_1(\text{Nm/rad})$	0.5	0.99	98%
$k_2(\text{Nm/rad})$	0.5	1	100%
$k_3(\text{Nm/rad})$	1	1.1	10%

Before optimization, the maximum deployment ratio of the mechanism occurs in the H direction, which is 5.6. After optimization, the dimensions of the concentrating deployable mechanism in the folded state are: 1431.49 mm × 719.81 mm × 174.95 mm; and the dimensions in the deployed state are 2165.25 mm × 719.81 mm × 1242.76 mm; the maximum deployment ratio appears in the H direction at 7.2, the minimum deployment ratio appears in the V direction at 1, and the maximum deployment ratio is increased by 28.57%.

The driving torque responses of Link 3 before and after optimization are compared in Figure 25: the peak torque reduction amplitude $\rho_{\max}$ is calculated according to Equation (41), decreasing by 10.9%.

$$\rho_{\max} = \frac{T_{\max,pre} - T_{\max,post}}{T_{\max,pre}} \times 100\% = 10.9\% \quad (41)$$

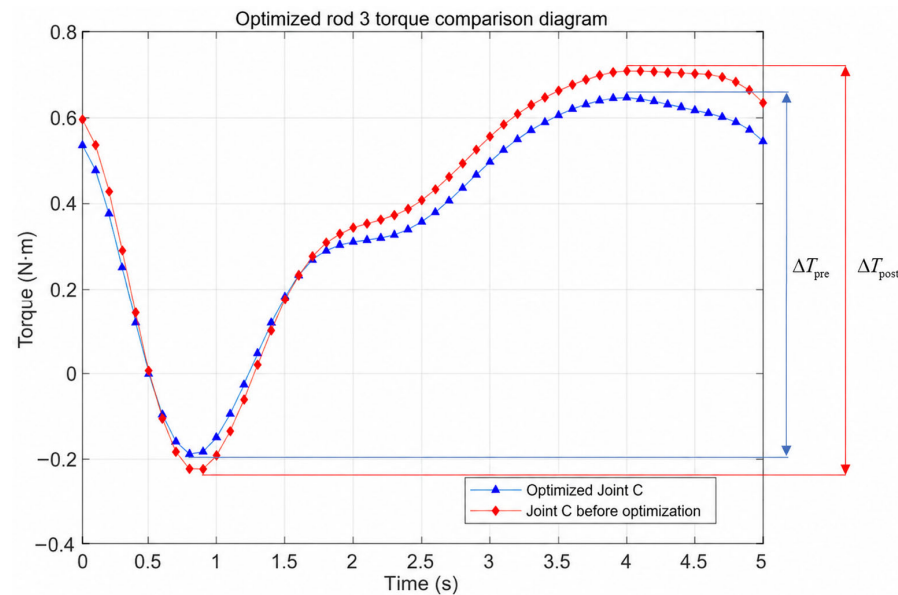


Figure 25. Comparison of driving torque responses of Link 3 before and after optimization.

The torque fluctuation amplitude ρ_{amp} , calculated according to Equation (42), is reduced by 9.7%.

$$\rho_{amp} = \frac{(T_{\max,pre} - T_{\min,pre}) - (T_{\max,post} - T_{\min,post})}{(T_{\max,pre} - T_{\min,pre})} \times 100\% = 9.7\% \quad (42)$$

The total link-length reduction is denoted by ρ_l and calculated as $p_l = 4.46\%$ using the following formula:

$$\rho_{li} = \frac{\sum_{i=1}^3 L_{i,pre} - \sum_{i=1}^3 L_{i,post}}{\sum_{i=1}^3 L_{i,pre}} \quad (43)$$

where $L_{i,pre}$ and $L_{i,post}$ are the lengths of Link i before and after optimization, respectively. According to the optimized parameters, the total link length decreases from 1.121 m to 1.071 m, corresponding to a total link-length reduction of

$$\rho_l = \frac{1.121 - 1.071}{1.121} \times 100\% = 4.46\% \quad (44)$$

The multi-objective optimization results show that the optimized mechanism achieves better deployment performance under the prescribed trajectory and idealized joint assumptions. The maximum deployment ratio in the H -direction increases from 5.6 to 7.2, corresponding to an improvement of 28.57%. The peak driving torque of Link 3 is reduced by 10.9%, and the torque fluctuation amplitude is reduced by 9.7%. Meanwhile, the total link length decreases from 1.121 m to 1.071 m, corresponding to a total link-length reduction of 4.46%. These improvements indicate that the optimized design reduces the deployment torque demand and enhances the compactness of the deployable mechanism, while maintaining smooth and synchronized deployment behavior.

6. Discussion and Conclusions

This study proposed a passive multi-link deployable support mechanism for lunar surface solar-concentrating systems. The mechanism was designed to satisfy the requirements of compact stowage, passive deployment, terminal locking, and stable reflector support. A hybrid reflector-Fresnel concentrating configuration was considered, and the foldable

support was formulated as a reconfigurable linkage system connecting the stowed state and the optical working state.

The results show that the “rigid locking + passive spring–hinge drive” strategy can provide a feasible deployment scheme with reduced drive complexity. Based on the motion-envelope planning and vector-based kinematic analysis, the deployment trajectory of the reflector centroid was obtained, and the relationship between the link motion and reflector working posture was clarified. This provides a theoretical basis for avoiding link-reflector interference during deployment.

Dynamic modeling and numerical simulation further verified the deployment performance of the mechanism. The simulated driving torque of Link 3 varied within 0–0.70 N·m, while the analytical result varied within 0–0.68 N·m, giving a relative peak-value error of 2.8%. The maximum angular acceleration during deployment was approximately 0.08 rad/s², indicating that the planned deployment trajectory does not introduce severe inertial disturbance under the adopted modeling assumptions. The finite-element modal analysis showed that the first natural frequency of the locked mechanism was 54.969 Hz, suggesting that no obvious low-frequency global flexible mode appears in the modeled locked-state configuration.

The multi-objective optimization results further improved the deployment performance and structural compactness. Based on the selected Pareto-optimal solution, the maximum deployment ratio increased from 5.6 to 7.2, corresponding to an improvement of 28.57%. The maximum driving torque of Link 3 was reduced by 10.9%, the torque fluctuation amplitude was reduced by 9.7%, and the total link length decreased from 1.121 m to 1.071 m, corresponding to a reduction of 4.46%. These results indicate that the optimized design can reduce deployment torque demand while maintaining smooth and synchronized deployment behavior.

From an engineering perspective, the proposed mechanism provides a compact and passively driven support scheme for lunar deployable concentrating systems. The analytical modeling, dynamic verification, and parameter optimization establish a preliminary design route for prototype-level deployable concentrating mechanisms. However, the present study is still limited to theoretical modeling and numerical verification. Future work will focus on prototype fabrication, deployment experiments, hold-down/release constraint allocation, joint clearance and friction modeling, harness routing for the azimuth and pitch motors, thermal-vacuum effects, lunar dust influence, and optical-mechanical integrated verification under lunar surface operating conditions.

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