Design and Optimization of Multi-petal Damping Structure for Vibration Control of Satellite Antenna Component

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Abstract. A preloaded multi-petal friction damping structure is proposed for deployment mechanism of large-scale flexible deployable satellite antenna. The damping structure is designed to suppress transverse vibration for promising stability and accuracy in the deployment process. In order to study the damping property of multi-petal damping structure and facilitate theoretical modelling, the model of deployment component with damping structure is simplified as double-layer cantilever beam model to study mechanical properties of friction and damping energy dissipation mechanism. The optimum parameters of damping structure are determined by numerical analysis of the simplified modal and the hysteresis curve can be obtained by finite element simulation to calculate loss factor. In the dynamic simulation, the loss factor can reach 0.2685 and the vibration amplitude can be reduced by more than 65%, which indicates the excellent effect for vibration suppression and the feasibility of the proposed structure.

1. Introduction
With the development of satellite technology, the application of satellites equipped with large-aperture antennas are increasing in many fields such as earth observation, telecommunications, scientific researches and remote sensing missions [1-2]. The deployment process stability and accuracy of large-scale flexible deployable satellite antenna are in high demand. Therefore, it is indispensable to develop a vibration control method with the characteristics of high working reliability, obvious vibration suppression effect and high vibration control stability to improve the precision and retention of antenna deployment process. In recent years, based on the development of artificial intelligence and intelligent materials including piezoelectric actuators with the feature of large rigidity, fast response and simple control structure, a large number of active control methods have been adopted to realize vibration control at home and abroad[3-5]. Due to the limitations that active control needs energy input and small noisy may lead to vibration aggravation, passive control is still the most extensive method of controlling vibration. Passive control restrains vibration mainly by increasing energy dissipation of structural vibration in damping system[6-7]. Xu has carried on the dynamic characteristic analysis of Additional Constrained Damping Layer. With the addition of constrained damping layer, the natural frequencies of the structure do not change much, but the vibration damping effect is nearly 20%[8]. At present, the aircraft mainly uses tuned mass dampers (TMD) to suppress vibration[9-10]. But for large flexible deployable antennas with low natural frequency, the design of
mass and elastic components is difficult. The control effect of TMD is unstable and its frequency band is narrow, which cannot meet the requirements of reliability and stability for deployment mechanism vibration control. Therefore, the paper designs a new damping structure suitable for large flexible deployable antenna.

In this paper, taking advantage of the characteristics that nonlinear friction vibration has large energy dissipation and control bandwidth[11], it is considered to adopt nonlinear friction damping structure to improve effectiveness, stability and reliability of vibration control in deployment component. Firstly, the model of deployment component with damping structure is simplified. Then, in terms of the simplified model to study the mechanical properties of friction and damping energy dissipation mechanism. The optimum parameters of damping structure are obtained by theoretical analysis. Finally, the hysteresis curve is determined by finite element simulation. Loss factor is achieved to prove rationality of the proposed structure.

2. Structure design and preliminary optimization of friction damper

Under the vibration characteristics of large-scale flexible satellite antenna and the stability requirements of deployment process, a preloaded thin-walled multi-petal friction damping structure is designed. The energy dissipation mechanism of the damping structure is that microscopic relative sliding friction of the contact surfaces between the outer wall of the petals and the inner wall of the deployment component produces energy loss. The relative sliding state of the contact surface at different positions along the deployment component axis changes with different payload of the deployment component, including micro-slip zone and bonding zone. When effective bending load of the deployment component exceeds a certain threshold, the contact area between the deployment component and the petals of damping structure is in a micro-slip state. The mathematical model of micro-slip energy dissipation on the contact surface is established to obtain the maximum damping capacity of the preloaded thin-walled multi-petal structure. Through the numerical analysis of the model, the relationship among the loss energy, the preloading force and the deployment component structure is determined, and the optimal design of the preloading multi-petal damping structure is realized.

![Figure 1. Schematic diagram of satellite antenna deployment component with damping structure.](image1)

![Figure 2. Thin-walled multi-petal damping structure.](image2)

In general, the structure of deployment component is a thin-walled circular ring cross-section beam structure with two ends hinged as shown in Figure 1, and the material is light and high rigid, such as carbon fiber. As shown in Figure 2, the preloaded thin-walled multi-petal friction damping structure is a multi-petal thin-walled ring cross-section structure. one end of the damping structure fixed with the deployment component has no petal and the other end preloaded by the tensioner ring on the inner wall of deployment component has petals. The friction loss energy is related to the bending stiffness and the structure of the petals, the preloading force and the external load of the expanded component. In order to verify the damping effect of damping structure, the model of deployment component with damping structure is simplified as double-layer cantilever beam model for theoretical analysis. Then
an example is given to receive the appropriate parameters of the friction damping structure.

2.1. Calculation for loss factor of cantilever beam model with rectangular section

The contact surface between the deployment component and the damping structure is cylindrical, which can be simplified as plane contact in a tiny region. Therefore, it is reduced to double-layer cantilever beam model. The subscript 1 represents the upper beam and the subscript 2 represents the lower beam. The distance between the slip interface and the neutral plane of the beams is \( h \). For rectangular section beams, the beam thickness is twice the neutral surface distance. The length of two beams is defined as \( l \). The load consists of total preloaded distribution force \( p \) between two beams and concentrated load \( P \) at the free end. The friction at the connection interface is defined as \( f \). Because of the external load, the double-layer cantilever beam will produce relative displacement \( \Delta u(x) \) at the interface.

![Figure 3. Double-layer cantilever beam (the simplified model).](image)

The curvature at any position \( x \) of the upper and lower beams is obtained by force balance and the relationship between bending moment and curvature. Thus, the relationship between end load \( P \) and end deformation \( v \) is achieved:

\[
v = \frac{P l^3 - f l^2 (h + h_2)}{3 (E_1 I_1 + E_2 I_2)}
\]

where \( E \) is elastic modulus of the beam, \( I \) is the cross-section moment of inertia of the beam. From the axial displacement \( u_{1,2} \) of the upper and lower beams, the relative slip of the contact surface is obtained:

\[
\Delta u(x) = u_2 - u_1 = \left( \frac{1}{E_1} \int_0^x \sigma_{1x} dx - h_1 \frac{dv_1}{dx} \right) - \left( \frac{1}{E_2} \int_0^x \sigma_{2x} dx - h_2 \frac{dv_2}{dx} \right)
\]

\[
= \frac{l}{2} \left[ - \frac{F_f}{E_1 A_1} - \frac{F_f}{E_2 A_2} + \left( h_1 + h_2 \right) \frac{P l - h_1 F_f - h_2 F_f}{E_1 I_1 + E_2 I_2} \right] \left[ 2 \left( \frac{x}{l} \right) - \left( \frac{x}{l} \right)^2 \right]
\]

where the \( \sigma_{1x} \) and \( \sigma_{2x} \) are the axial stresses at any position of the upper and lower beams. To simplify, some dimensionless parameters are defined as shown in table 1. So the simplified result is received:

\[
\Delta u(\psi) = \frac{P}{2} \left[ 3 \epsilon \beta \left( 1 - \frac{\beta}{\gamma} \right) \delta_\alpha - \frac{1}{\sigma'} \lambda_\alpha \right] \left[ 2 (\psi) - (\psi)^2 \right]
\]

From the relative slip, the friction energy can be acquired by working integral on the friction of the whole contact surface:
\[ E_{\text{loss}} = 4F \int_0^l \Delta u \sqrt{\nabla \psi} \, d\psi_x = \frac{4F^2}{3} \left[ 3\varepsilon \beta \left( \gamma - \beta \right) \delta_n - \frac{1}{\sigma} \lambda_n \right] \]  

(4)

The energy storage of the system is composed of the energy storage \( E_{ne1} \) produced by bending moment and the energy storage \( E_{ne2} \) of interfacial friction force:

\[ E_{ne1} = \frac{1}{2} \left( P - \frac{v_c}{P} \right) \]  

(5)

\[ E_{ne2} = 4F \left( \frac{1}{E_1} \int_0^l \int_0^\delta \sigma \, d\sigma_x + \frac{1}{E_2} \int_0^l \int_0^\delta \sigma_x \, d\sigma_x \right) \]  

(6)

Where \( v_c \) is the critical end deformation load, \( P \) is the minimum end concentrated load causing the initial relative slip. Therefore, loss factors of the system derived from friction energy dissipation and system energy storage:

\[ \eta = \frac{E_{\text{init}}}{2\pi E_{ne}} = \frac{4}{3\pi} \frac{3\varepsilon \beta \gamma - 3\varepsilon \beta^2 - \kappa}{\kappa + 3\varepsilon \beta^2 \gamma^2 - \varepsilon \beta \gamma + \frac{8}{3} \kappa} \]  

(7)

Table 1. Dimensionless parameters and implications.

| Dimensionless parameters | Implications |
|--------------------------|--------------|
| \( \beta = (h_i + h_s) / l \) | Ratio of the sum of neutral surface distance of upper and lower beams to the whole length |
| \( \gamma = P / F_i \) | Ratio of end concentration force to interfacial friction force |
| \( \varepsilon = (E_i A_i) / (E_i A_i + E_s A_s) \) | Ratio of tensile stiffness of upper beam to the sum of upper and lower beams |
| \( \delta_n = l^3 / (3E_i I_i) \) | Static bending deformation at the end of the upper beam under unit tensile load |
| \( \psi_x = x / l \) | Ratio of any position of beam to the whole length |

2.2. Numerical simulation of cantilever beam with rectangular section

To ensure consistency with subsequent finite element simulations, the double-layer cantilever beam interface is defined as a rectangular section, so that its length \( l = 500 \text{mm} \), the thickness of the upper layer beam are \( 1 \text{mm} \) (the distance from neutral surface to friction interface \( h_i = 0.5 \text{mm} \)). The material of the upper and lower beams is carbon fiber \( t700 \), and the elastic modulus \( E_1 = E_2 = 210 \text{GPa} \).

To simplify the formula, the neutral plane distance ratio of double-layer cantilever beams is defined as \( \alpha = h_s / h_i \). Thus, \( \beta = (h_i + h_s) / l = (\alpha + 1) / 500 \), \( \sigma = (E_i A_i) / (E_i A_i + E_s A_s) = \alpha / (\alpha + 1) \), \( \varepsilon = (E_i I_i) / (E_i I_i + E_s I_s) = 1 / (\alpha^3 + 1) \), \( \kappa = \lambda_n / \delta_n \approx 4 \times 10^{-6} \). The formula of loss factor is obtained:
The expression of the resulting loss factor $\eta$ is biased about $\gamma$ and makes it equal to zero. As a result, the maximum damping effect is obtained, that is, the optimum force ratio $\gamma_{op}$ (the ratio of the end concentrated force to the interfacial friction force) when the loss factor is maximum:

$$\gamma_{op} = \left(\frac{3000\alpha + 1}{\pi 225 \times 10^7 \alpha^2 \gamma^2 - 750\alpha (\alpha + 1)^4 + 2 (\alpha^3 + 1)(\alpha + 1)^3} \right) \frac{(\alpha + 1)^3 + \sqrt{8(\alpha + 1)^3 (\alpha^3 + 1)}}{3000\alpha}$$

The expression of $\gamma_{op}$ is brought into the expression of $\eta$ (loss factor), and the optimal damping effect can be obtained in the case of $\alpha$ (neutral surface distance ratio) of different double-layer cantilever beams. The relationship between $\eta$ and $\alpha$ is shown in Figure 4, and the relationship between $\gamma_{op}$ and $\alpha$ is in Figure 5. When the thickness of the two beams is the same ($\alpha = 1$), the maximum loss factor ($\eta = 0.3326$) and the optimum force ratio ($\gamma_{op} = 0.006438$) can be obtained. Figure 6 shows the relationship between loss factor and force ratio when the neutral plane distance ratio is 1, 0.5 or 2 respectively. The relationship between loss factor and force ratio and neutral plane ratio is as shown in Figure 7.

Through the theoretical analysis of the double-layer cantilever beam model, the parameters of multi-petal damping structure are preliminarily determined, which provides the foundation for finite element dynamic simulation.
3. Finite element simulation and analysis of multi-petal cantilever model

The multi-petal damping structure and the deployment component are hollow cylinders, and the length of them is \( l \). The damping structure is inserted into the hollow cylindrical deployment component and is pressed on the inner wall of it. It is assumed that the gap of the contact surfaces between the damping structure and the deployment component is negligible \( (r_1 = r_2) \). The geometric parameters of the antenna-damping structure model are shown in Table 2 and the cross section of the multi-petal cantilever model in figure 8.

![Figure 8. The cross section of the multi-petal cantilever model.](image)

| Implications of parameters | Measurement |
|---------------------------|-------------|
| **Damping structure**     |             |
| Inner diameter            | \( r_1 = 10.5 \text{mm} \) |
| Outer diameter            | \( r_2 = 11.5 \text{mm} \) |
| **Deployment component**  |             |
| Inner diameter            | \( r_3 = 11.5 \text{mm} \) |
| Outer diameter            | \( r_4 = 12.5 \text{mm} \) |
| Length of component       | \( l = 500 \text{mm} \) |
| Number of petals          | \( n = 6 \) |
| Friction factor           | \( \mu = 0.15 \) |

In order to validate the validity of the theoretical analysis, undamped deployment component (the finite element models of the cantilever beam with ring section) and deployment component with damping structure are established, and the loss factor is obtained by hysteresis curve to verify the accuracy of the numerical analysis. Using ANSYS finite element analysis software, the material is carbon fiber t700, and the parameters are shown in Table 3. During the process of ANSYS analysis, the two models are established and the grid is divided firstly. Then, the modal analysis is carried out. Finally, the harmonic response analysis is carried out and the two modals are analyzed to obtain their hysteresis curves respectively.

3.1. Finite element analysis of deployment component.

The model of deployment component is shown in Figure 9. By modal analysis, the first order natural vibration frequency is 197.17 Hz. When the load is 100N, through its harmonic response analysis, the hysteresis curve is obtained as shown in Figure 10. Its loss factor is 0.018 and the amplitude can reach 2.449 mm. And the first three modes are obtained in Figure 11.

![Figure 9. Deployment component without damping structure.](image)

![Figure 10. Hysteresis curve of deployment component.](image)
Figure 11. First three modes of deployment component.

3.2. Finite element analysis of deployment component with damping structure.

The model of deployment component with damping structure is shown in Figure 12. By modal analysis, the first three modes are obtained in Figure 13, and the first natural vibration frequency is 393.37Hz. When the load is 100N and the optimum force ratio is 0.0064, the corresponding optimal total preload force is 942 N. By harmonic response analysis, the hysteresis curves under different preloaded forces (when the forces are 906N, 942 N and 978N) are obtained respectively as shown in Figure 14, and the loss factor and amplitude are shown in table 3. When the preload force is 942 N, the loss factor and amplitude drop rate are maximum and the effect of vibration control is excellent.

Figure 12. Model of deployment component with damping structure and its section diagram.

Figure 13. First three modes of deployment component with damping structure.

Figure 14. Hysteresis curves at 905.96N, 942.07N and 978.18N preload forces.
Table 3. Loss factors and amplitudes under different preload forces.

| Preload (N) | 906  | 942  | 978  |
|------------|------|------|------|
| Loss factor| 0.13725 | 0.26851 | 0.1376 |
| Amplitude (mm) | 1.2486 | 0.78434 | 1.2693 |
| Amplitude drop rate | 49.02% | 67.97% | 48.17% |

4. Conclusion
The paper introduces a preloaded multi-petal friction damping structure designed for deployment mechanism of large-scale flexible deployable satellite antenna. The friction damping structure is proposed to promise stability and accuracy of deployment process. Different from the complex structure of tuned mass dampers that most satellites use to restrain vibration, the proposed damping structure has eminent vibration suppression effect and simple structure. Theoretical analysis of the simplified model shows the loss factor can reach 0.3326. In finite element simulation of vibration dynamics, the loss factor of the deployment component with the damping structure can reach 0.26851 and the amplitude drop rate can reach 67.97%, which are obvious for the deployment mechanisms of large flexible deployable antenna with low-order natural frequency. Preloaded multi-petal friction damping structure provides a simple structure and obvious vibration suppression method for satellite antenna, which has a profound influence on the development and application of large flexible deployable antenna satellite in the future.

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