Design, Simulation and Failure Influence of Tape Spring Hinges in a Large Deployable Antenna

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Abstract. The design, simulation and failure influence of tape spring hinges in a large deployable antenna is studied. By sensitivity analysis between modal frequencies of the large deployable antenna and stiffness coefficients of the tape spring hinge, it is found that compression stiffness along longitudinal axis and bending stiffness about transverse axis are the key points of tape spring design. Stiffness coefficients, buckling loads and bending moments of the tape spring hinge are obtained by mechanical simulation. By in orbit disturbance analysis, disturbance loads applied on the tape spring hinge result from satellite attitude control and scanning mechanism operation are presented. Finally, failure influences of the tape spring hinges on both antenna modal frequencies and pointing accuracy are studied.

Keywords: Tape spring hinge; Large deployable antenna; Pointing accuracy.

1. Introduction

Large deployable antennas are widely used in space missions for advantages of low mass, small volume and high gain [1,2]. Large deployable antennas are stowed for launching and deploys to working configuration in orbit by deployable mechanisms [3-5]. Different kinds of deployable mechanisms have been explored and applied to spacecrafts, such as spring-driven mechanism for the ETS-VIII satellite module reflector [6,7], cable-actuated mechanism for the Thuraya satellite large ring reflector [8], air-driven mechanism for inflatable reflectors [9] and SMA (shape memory alloy)-driven mechanism for membrane mirrors of space telescopes [10]. Compared with the presented deployable mechanisms, a tape spring could provide linear moments for small rotations and constant moments for large rotations, and because peak moment of the tape spring occurs at the end of the linear range, the tape spring acts as a self-locked mechanism in structures, which is very beneficial for stability of the antennas, and have been studied in both theory and practice [11-15]. Walker et al. [16] focused on deployment dynamics of three dimensional tape-spring folds. Kim et al. [17] proposed a systematic approach for designing tape spring hinges for any number of solar panels. Soykasap et al. [18] presented a study of large tape springs made of carbon/epoxy composite to be used as ribs of an ultra-thin shell space deployable reflector. However, much of the previous work only focused on tape spring itself, the study of influences of tape spring failure on antenna structures has not been reported yet.

As shown in Fig. 1, a large deployable antenna mainly consists of the flexible mesh reflector, the electronics module structure and the support structure. The support structure is in triangle configuration for high stiffness consideration, and includes the main arm and the sub arm. There are three revolution bearing hinges and one tape spring hinge for each side of the arm. Before being launched in orbit, the deployable arm is locked on the satellite and will be released to deployment when in orbit. Also a scanning mechanism is allocated between the electronics module structure and the deployable arm for scanning requirement. The high flexibility and low damping of the antenna has proposed critical
requirement not only for stability control of the antenna itself, but also for attitude control of the satellite. According to the requirement of satellite control subsystem, the first resonant frequency of the large deployable antenna should be no less than 0.90 Hz.

Figure 1. Configuration of the large deployable antenna.

In this study, the design, simulation and failure influence of tape spring hinges in a large deployable antenna is discussed. The remainder of this paper is organized as follows. The second section illustrates configuration and parameters of the tape spring hinge which is used in the large deployable antenna. The third section studies sensitivity relationship between modal frequencies of the antenna and stiffness coefficients of the tape spring hinge. The fourth section presents mechanical simulation results of the tape spring hinge, which includes stiffness analysis, buckling analysis and moment analysis. The fifth studies in orbit disturbance loads applied on the tape spring hinge result from satellite attitude control and scanning mechanism operation. The subsequent section illustrates failure influences of the tape spring hinges on both antenna modal frequencies and pointing accuracy. And the final section summarizes the study and presents the conclusions.

2. Overview of the Tape Spring Hinge

A tape spring hinge is composed of two tape springs mounted symmetrically on hinge brackets with bolts, as presented in Fig. 2. Cover plates are applied to fix the tape springs to the hinge brackets in order to ensure sufficient bonding strength.

Figure 2. The tape spring hinge configuration.

To prevent material yield, dimension and material parameters of the tape spring should meet the following relationship [19].

$$\frac{R}{t} \geq \frac{\sqrt{3}E}{2(1+\nu)\sigma_s}$$

(1)

Where R is initial transverse radius of the tape spring; t is the thickness; E is the elastic modulus; \(\nu\) is the Poisson ratio; \(\sigma_s\) is yield strength of the material.

Dimension and material parameters of the tape spring used in the large deployable antenna are listed in Table 1.

| E (GPa) | \(\nu\) | \(\sigma_s\) (MPa) | R (mm) | t (mm) | Subtended angle \(\theta\) (°) | Length L (mm) |
|--------|--------|-------------------|--------|--------|-----------------------------|---------------|
| 206    | 0.3    | 1120              | 40     | 0.2    | 60                          | 183.4         |

3. Modal Frequencies Sensitivity Analysis

Modal vectors and modal frequencies of the antenna can be obtained by solve the differential equation below:
\[ \mathbf{Mx} + \mathbf{Kx} = 0 \]  \hspace{1cm} (2)

The responding kth modal frequency and modal vector are \( \omega_k \) and \( \Phi_k \) respectively, and

\[ (\mathbf{K} - \omega_k^2 \mathbf{M}) \Phi_k = 0 \]  \hspace{1cm} (3)

Suppose the design variable is \( X \), then

\[ (\frac{\partial\mathbf{K}}{\partial X} - 2\omega_k \frac{\partial \mathbf{M}}{\partial X} - \omega_k^2 \frac{\partial \mathbf{M}}{\partial X}) \Phi_k = 0 \]  \hspace{1cm} (4)

While \( \Phi_k^T \mathbf{M} \Phi = I \)

Sensitivity coefficient of the kth modal frequency to the design variable \( X \) is defined as

\[ \frac{\partial \omega_k}{\partial X} = \frac{\Phi_k^T}{2\omega_k} \left( \frac{\partial \mathbf{K}}{\partial X} - \omega_k^2 \frac{\partial \mathbf{M}}{\partial X} \right) \Phi_k \]  \hspace{1cm} (5)

Modal frequencies sensitivity analysis is conducted on the above large deployable antenna, of which relationship between the first six modal frequencies (f1~f6) and the six stiffness coefficients (KTX, KTY and KTZ refer to extensile stiffness along X, Y and Z axis, KRX, KRY and KRZ refer to bending stiffness about X, Y and Z axis) of the tape spring hinge is studied, and the analysis results are shown in Fig. 3. The first six modal frequencies are largely determined by the stiffness coefficient KTX and KRZ. Fig. 4 shows the 1st modal frequencies versus stiffness coefficients KTX and KRZ, and for tape spring hinge with KTX over than 3000N/mm and KRZ over than 40Nm/rad, the first modal frequency of the large deployable is more than 0.90Hz.

![Figure 3. Sensitivity analysis results of the first six modal frequencies.](image)

![Figure 4. The first modal frequencies versus stiffness coefficients KTX and KRZ.](image)

### 4. Mechanical Simulation of the Tape Spring Hinge

#### 4.1. Stiffness Analysis

A refined finite element model of the tape spring hinge is build as shown in Fig. 5, the tape spring sheets and the connect parts are simplified with solid elements and share the same nodes at the connection area. All degrees of freedoms of the nodes at one end of the tape spring are constrained, and loads are applied at the free end of the hinge.
Figure 5. Finite element model of the tape spring hinge.

By the former sensitivity simulation results, stiffness coefficients which have significant influence on the antenna are KTX, KRX, KRY and KRZ, as defined in Fig. 6.

(a) KTX (Extensile stiffness along X axis)                           (b) KRX (Bending stiffness about X axis)

(c) KRY (Bending stiffness about Y axis)                           (d) KRZ (Bending stiffness about Z axis)

Figure 6. Definitions of the stiffness coefficients

Stiffness analysis results are illustrated in Fig. 7, the hinge performs different deformation types under the given loads, and the final stiffness coefficients are summarized in Table 2.

(a) KTX                           (b) KRX

(c) KRY                           (d) KRZ

Figure 7. Deformation of the hinge under different loads.

Table 2. Stiffness coefficients of the hinge.

| KTX (N/mm) | KRX (Nm/rad) | KRY (Nm/rad) | KRZ (Nm/rad) |
|------------|--------------|--------------|--------------|
| 3042       | 52           | 3403         | 243          |

4.2. Buckling Analysis

Structure buckling can be determined by the following eigenvalue equation:

\[
(K + \lambda D)\delta u = 0
\]  

(6)
Where $K$ refers to the initial stiffness matrix, $KD$ and $u$ are the load stiffness matrix and the displacement vector.

By solving Eq. 6, the buckling loads and buckling shapes could be obtained, as the buckling loads are determined by the eigenvalues and the buckling shapes are determined by the eigenvectors.

Buckling shapes of the tape spring hinge under different loads are illustrated in Fig. 8. Fig. 8(a) shows buckling shape under a compression load along the $X$ axis, and the tape spring deforms at the center edge of the tape. Fig. 8(b) presents buckling shape under a torsion load about the $X$ axis, and the tape spring deforms at one edge of the tape. Figs. 8(c) and 8(d) are buckling shapes under bending loads about $Y$ and $Z$ axes respectively, and the buckling areas are also located at edges of the tapes. The corresponding buckling loads of different load types are listed in Table 3.

![Figure 8. Buckling shapes of the tape spring hinge.](image)

### Table 3. Buckling loads of the tape spring hinge-simulation results.

| Load types                        | Simulation Results |
|-----------------------------------|--------------------|
| Compression along the $X$ axis (N) | 1376               |
| Torsion about the $X$ axis (Nm)    | 3.5                |
| Bending about the $Y$ axis (Nm)    | 16.8               |
| Bending about the $Z$ axis (Nm)    | 6.7                |

#### 4.3. Moment Analysis

Tape springs have two bending modes, the opposite sense bending and the equal sense bending. For the opposite sense bending, only moment in longitudinal direction arises in the spring. And for the equal sense bending, moments occur in both longitudinal and transverse directions besides torsion about the longitudinal axis. For a tape spring as shown in Fig. 1, static bending moments for equal sense bending and opposite sense bending are $M_e = D\theta (\nu - 1)$ and $M_o = D\theta (\nu + 1)$ respectively. Where $D = Et^3/(12(1-\nu^2))$ is bending stiffness of the tape spring, in which $E$ is the elastic modulus; $\nu$ is the Poisson ratio; $t$ is the thickness; $\theta$ is initial subtended angle of the tape spring.

For opposite sense bending, moment-curvature relationship is determined by Eq. 7 listed below [20].

\[
M = 2R\sin\left(\frac{\theta}{2}\right)D\left\{\kappa_i + \frac{\nu}{R} - \nu\left(\frac{1}{R} + \nu\kappa_i\right)\left[\frac{2}{\lambda} \cosh\lambda - \cos\lambda \right] \right. \\
\left. + \frac{1}{\kappa_i} \frac{1}{R} + \nu\kappa_i \right\}^2 \left[\frac{1}{4} \frac{2}{\lambda^2} \cosh\lambda - \cos\lambda \right] - \frac{\sinh\lambda \sin\lambda}{\left(\sinh\lambda + \sin\lambda\right)^2}\right\} \\
(7)
\]

Where $\lambda = \sqrt{3(1-\nu^2)}s/\sqrt{f/\kappa_i}$, $\kappa_i$ is curvature in longitudinal direction.
Folding bend moment versus bend angle of a tape spring with parameters listed in Table 1 is shown in Fig. 9, the bend moment reaches its peak value at the bend angle of 9°, and then buckling occurs at the tape spring, the bend moment drops sharply to a constant value. Before buckling, the moment-angle relationship is approximately to be linear. Folding of the tape spring hinge is illustrated in Fig. 10 with four discrete statuses.

![Figure 9. Folding bend moment versus bend angle of a tape spring.](image)

![Figure 10. Folding of the tape spring hinge.](image)

5. In Orbit Disturbance Analysis
For satellite antennas, there are two major sources of disturbances, of which the first one comes from attitude control of the satellite, as the satellite engine works for station keeping or modification, a vibration disturbance will arise at the satellite-antenna interface. Also, when the large deployable antenna scans to catch and track desired objects, with rotations of the scanning mechanism, the antenna will move toward different positions, which will induce vibration of the antenna. Fig. 11 presents the mentioned two kinds of disturbances applied on the deployable antenna. As shown in Fig. 11(a), the maximal transient acceleration along the rolling axis, pitch axis and yaw axis is about 0.4°/s². Also, Fig. 11(b) illustrates disturbance angular accelerations about the scanning mechanism axes when the motor speeds up and down with a maximum acceleration of 0.005°/s².
Simulation results of the disturbance loads at the tape spring hinges interface are listed in Table 4. The in orbit disturbance loads is much less than the buckling loads presented in Table 3, which indicates that the tape spring hinges have no risk of bucking in orbit.

Table 4. In orbit disturbance loads of the tape spring hinges.

| Load types                        | Simulation Results |
|-----------------------------------|--------------------|
| Compression along the X axis (N)  | 16                 |
| Torsion about the X axis (Nm)     | 0.10               |
| Bending about the Y axis (Nm)     | 0.82               |
| Bending about the Z axis (Nm)     | 0.02               |

6. Failure Influence Analysis

6.1. On Antenna Modal Frequencies

Failure of the tape spring hinges will largely affect the structural stiffness, and further reduces modal frequencies of the antenna, which will make it difficult for the satellite attitude control. For failures of one tape spring hinge and two tape spring hinges, the final modal frequencies of the antenna together with the normal status are listed in Table 5.

Table 5. Mode Frequencies for different failure modes.

| Mode Frequencies/Hz | Normal Status | Failure of one hinge | Failure of two hinges |
|---------------------|---------------|----------------------|-----------------------|
| X axis              | 1.03          | 0.96                 | 0.93                  |
| Y axis              | 1.01          | 0.79                 | 0.78                  |
| Z axis              | 5.80          | 3.16                 | 0.10                  |

Modal shapes of the antenna with two tape spring failures are illustrated in Fig. 12, and the lowest modal frequency of 0.10Hz corresponds to modal shape along Z axis of the satellite, which results from stiffness reduction of the support structure for tape spring failures.
6.2. On Pointing Accuracy

As structural stiffness of the antenna changes, when the scanning mechanism operates to drive the reflector to desire position, the antenna may vibrate and further worsen pointing accuracy of the system. Figs. 13-15 illustrate pointing curves of the antenna with different tape spring hinge failure modes. In Fig. 13, the antenna scans about the pitch axis from 0° to 45°, and if one hinge is failed, the maximal point error is about 0.0025°, but for failure of both two hinges, the maximal pointing error is about 0.50°, which reveals that pointing error of the antenna is largely determined by stiffness of the structure. In Fig. 14, the antenna scans about the azimuth axis from 0° to 20°, and the maximal pointing errors of one hinge failure and two hinges failures are 0.0002° and 0.005°. Pointing error considering combination scanning about both the pitch and the azimuth axes is shown in Fig. 15 and the maximal pointing error is over than 1.30°, which indicates that the antenna could not meet the operation requirement any more.

![Figure 13. Scanning about the pitch axis.](image)

![Figure 14. Scanning about the azimuth axis.](image)

![Figure 15. Scanning about both the pitch and azimuth axes.](image)

7. Summary

In this study, the design, simulation and failure influence of tape spring hinges in a large deployable antenna is discussed, and the main results can be summarized as follows:

a) Modal frequencies of the mentioned large deployable antenna are largely determined by the extensile stiffness along X axis and bending stiffness about Z axis, which provide directions for structure optimization of the tape spring hinge.

b) The in orbit disturbance loads is much less than the buckling loads, which indicates that the tape spring hinges have no risk of bucking in orbit.

c) For the worst case of two tape spring hinges failures, the lowest modal frequency of the antenna drops to be 0.10Hz, and the maximal pointing error is over than 1.30°, the antenna could not meet the operation requirement any more.

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