Part Design of Giant Magnetostrictive Actuator

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Abstract

The key parts of giant magnetostrictive actuator, flexure hinge and pre-stress disc spring, were designed and analyzed. Rotation stiffness and strength characteristics of flexure hinge were analyzed, calculation equations for rotation stiffness and strength were established as well. Fatigue characteristic was also analyzed as flexure hinge usually worked under high frequency situation. In order to improve output efficiency of the giant magnetostrictive actuator and reduce energy loss, an ideal spring force-deformation curve, whose shape was bilinear broken line, of the pre-stress disc spring was put forward, and a disc spring was designed by configuring its geometric parameters to make its spring force-deformation curve was approximate to the ideal spring force-deformation curve.

Keywords: giant magnetostrictive actuator; flexure hinge; rotation stiffness; strength; fatigue; disc spring

1. Introduction

In recent years, giant magnetostrictive material (GMM) has increasingly gained attention as a kind of high efficient electric (magnetic) energy to mechanical energy conversion materials. Giant Magnetostrictive Actuator (GMA) is a new kind of precise actuator based on GMM, with fast response, large output force, high resolution, large output displacement range, strong load capability. Therefore it is very useful in ultra-precision positioning, hydraulic servo valve, high speed On/Off valve, ultra-precision machining, active vibration control system [1-4].

Key parts of GMA, flexure hinge of micro-displacement amplifier and pre-press spring, would absorb part of output energy of GMM rod, causing energy loss and affecting output efficiency, so it is necessary...
to do detail design for flexure hinge and pre-press spring. In this paper, design methods for flexure hinge and pre-press spring are put forward, performance requirement for pre-press spring is introduced.

2. Design of single axis flexure hinge

Although magnetostrictive strain of GMM is large, up to 1600ppm, but from a macro point of view, its expansion is still relatively small, in many applications the micro-displacement of the giant magnetostrictive material needs to be amplified by amplification mechanism. The amplification mechanism is usually designed based on flexure hinges, such as shown in Fig.1 which is a kind of micro-displacement amplification mechanism. Flexure hinge is manufactured in a single block of material, using the weak region of the elastic deformation of the structure to accomplish a similar hinge rotation motion. It has advantages of small volume, no friction, small motion gap, and higher displacement resolution. As the output displacement of magnetostrictive material rod is in the micron level, in order to prevent the flexure hinges absorbing the small deformation of GMM in the process of passing micro-displacement, it’s necessary to do a precise design calculation for flexure hinge.

![Fig.1 Micro-displacement amplification mechanism using flexure hinge](image)

There are many forms of flexure hinge, the most common form is single-axis hinge with notch in the form of arc-type (Fig.2). In Fig.2, geometric parameters of flexure hinge are marked.

![Fig. 2 (a) geometric of flexure hinge; (b) parameters of flexure hinge](image)

Rotational deformation of flexure hinge is actually composed by many micro-bending deformation of the micro segments, and each micro segment can be seen as uniform beam of length $dx$, the moment acting on the micro-segment is about the same on both ends, based on this assumption, can get flexure hinge rotation angle equation [5]:

$$\alpha = \int_{\alpha}^{\alpha_f} \frac{12MR\sin\alpha}{Eb(2R + t - 2R\sin\alpha)^2} d\alpha$$

(1)
where, $E$ is flexure hinge material elastic modulus, $b$ is cross-section width of the hinge, $t$ is the flexible hinge thickness at the thinnest section, $R$ is the arc radius of the notch, as Fig.2 shows.

Based on the above equation, rotation stiffness around the Z-axis of the flexure hinge can be obtained:

$$K_z = \frac{EbR^2}{12} \cdot \Phi(t/R)$$

(2)

where

$$\Phi(t/R) = \left( \int_0^\theta \frac{\sin \alpha}{(2 + \frac{t}{R} - 2 \sin \alpha)^3} d\alpha \right)^{-1}$$

The values of $\Phi(t/R)$ can be get when $R/t$ take various values by using math software, and the fitting cubic polynomial of $\Phi(t/R)$ also can be get: $\Phi(t/R)=0.5239(t/R)^3+0.5271(t/R)2-0.0901(t/R)+0.0072$, which can be used to do design calculations of the flexure hinge easily.

From equation (2), three methods can be got to reduce rotation stiffness $K_z$ of the flexure hinge:

1) Reduce the width $b$ of flexure hinge;
2) When $t$ is constant, considering the effect of $R^2$ and $\Phi(t/R)$ to $K_z$ overall, determine to increase or decrease $R$ value;
3) When $r$ is constant, to reduce $K_z$ by reducing the minimum thickness $t$ of the flexure hinge.

In the design of flexure hinge, it requires flexure hinge response fast, also requires structural reaction force is small, and no fatigue damage occur as well, then it is necessary to consider the rotation stiffness and strength overall to meet the requirements of the flexure hinge geometric parameters.

Flexure hinge in the process of rotation deformation, the minimum thickness region has the maximum deformation, then the stress is greatest at this region, the strength of this region needs to be verified.

The relationship between maximum stress $\sigma_{max}$, rotation angle $\theta$, hinge rotation stiffness $E$ and geometric parameters, $t$, $b$ and $R$, of the hinge is as follows [6]:

$$\alpha_{max} = \frac{6EbR^2}{12bt^2} \cdot \frac{\varphi(t/R)\theta}{2\psi(t/R)}$$

(3)

where:

$$\psi(t/R) = \varphi(t/R) \cdot \left( \frac{R}{t} \right)^3$$

(4)

From equation (3) and (4), it can be seen, when rotation angle $\theta$ is constant, the maximum stress $\sigma_{max}$ only have relationship with $t/R$ and flexure hinge material properties, and the flexure hinge width $b$ has nothing to do with it.

When the giant magnetostrictive actuator driving signal frequency is relatively high, due to the stress value of flexure hinge at minimum thickness region is relatively large, it is easy to form a fatigue crack in this region, so it is necessary to verify the fatigue strength for the flexure hinge to ensure that the actuator work reliability.

The maximum tensile stress of flexure hinge is at the outer edge of the location of the minimum thickness. Assuming the driving signal is pulse width modulation (PWM) signal, the corresponding changes in magnetostrictive force waveform is similar to PWM signal, so the flexure hinge maximum tensile stress curve is as Fig.4 shows.

By equation (3) the flexure hinge maximum stress value is:

$$\sigma_{max} = \frac{E\theta}{2\psi(t/R)}$$

(5)
The minimum stress value is: \( \sigma_{\min} = 0 \) (when \( \theta = 0 \)). The corresponding \( \sigma-t \) curve at the maximum stress region of the hinge is as follows:

![The maximum tensile stress curve](image)

Fig.3 The maximum tensile stress curve

Obviously, this is a pulsation cycle, the stress ratio is:

\[
\sigma = \frac{\sigma_{\min}}{\sigma_{\max}} = 0
\]  

(6)

The average stress is:

\[
\sigma_{\bar{m}} = \frac{\sigma_{\max}}{2}
\]  

(7)

The stress amplitude is:

\[
\sigma_a = \sigma_{\max}
\]  

(8)

When \( r = 0 \), for general structures made of plastic material, fatigue damage usually occurs early to plastic damage, equation (9) can be used to calculate the fatigue strength [7]:

\[
n_{\sigma} = \frac{\sigma_{\sigma}}{\sigma_{\sigma} + \psi_{\sigma} \sigma_{m}} \geq n
\]  

(9)

Where: \( K_\sigma \) is effective stress concentration factor of the structure, because the \( R/t \) is relative large, the shape transition at flexure hinge notch is relatively smooth, so take \( K_\sigma = 1 \); \( \sigma_{\sigma} \) is the material fatigue strength, \( e_a \) is the size factor for the hinge structure, \( \beta \) is the surface quality factor of the component, \( \sigma_a \) is the stress amplitude, \( \psi_{\sigma} \) is the sensitivity factor of the material for asymmetry of the stress cycle, \( \sigma_{m} \) is the average stress, \( n \) is the safety factor.

3. Design of Pre-stress Disc Spring

When apply a certain axial pre-compression stress to the giant magnetostrictive material rod, larger displacement output can be gotten. The mechanism is: axial pressure force the internal magnetic domains undertaking zero magnetic field to arrange along with the perpendicular direction of the axis of the rod as much as possible; When the external electromagnetic excitation, most of the magnetic domain suddenly shifted in the direction of the external magnetic field, achieving greater magnetostriction coefficient, thereby increasing the displacement of micro-actuator output. In addition, giant magnetostrictive material is brittle material, can withstand the pressure of about 700MPa, but the tensile strength is only about 28MPa, easily broken, so GMM should not work under tensile stress or shear stress. To avoid the giant
magnetostrictive material working under tensile stress or shear stress, pre-pressure stress should be applied to the GMM rod, to avoid the giant magnetostrictive materials bearing tensile stress.

From the perspective of meeting the pre-stress requirement, we should choose a large spring stiffness of the pre-press spring, because in practice, pre-stress is large, for example, when pre-stress is 10Mpa, GMM rod diameter is 10mm in the case, the pre-pressure is 785N. So currently mostly used spring to provide pre-stress is disc spring, because disc spring has large stiffness and smaller volume advantages compared with other type spring, and deformation energy per unit volume of material is greater. But from the perspective of improving the work ability of the actuator, we do not want stiffness of the spring too large because during the actuator working process the spring will absorb part of the GMM rod deformation energy, larger stiffness of spring means more energy loss. Therefore the requirement for spring stiffness between applying pre-stress stage and magnetostrictive elongation stage is contradiction. So an ideal spring characteristic curve should be similar to the bilinear broken line, as Fig.4 shows, before pre-pressure rises to \( F_0 \), the disc spring stiffness is \( K_1 \), and after that, stiffness of the disc spring decrease to a smaller value \( K_2 \).

![Fig.4 The ideal spring characteristic curve](image)

Disc spring can be designed to achieve the approximate \( f-x \) broken line characteristic curve through selecting proper geometric parameters. Widely used single disc spring load and deformation basic equation is as follows [8]:

\[
F = \frac{xs^3}{ad^2}\left[\left(\frac{h}{s} - \frac{x}{s}\right)^2 + 1\right]
\]  

(10)

where, as Fig.6 shows, \( D \) is the disc spring outer diameter, \( d \) is the disc spring inner diameter, \( h \) is the inner height, \( s \) is the disc thickness, \( x \) is the disc spring vertical displacement under load, \( \alpha \) is the constant determined by \( D/d \).

![Fig.5 Disc spring geometric parameters](image)
The disc spring characteristic curve $f-x$ can be plotted with different parameters ($D$, $d$, $h$, $s$) values by mathematic software. By matching different $D$, $d$, $h$, $s$ values an ideal characteristic curve of disc springs that match the ideal bilinear broken line can be designed.

After determining the geometric parameters of the spring, the amount of spring pre-pressure can be calculated. Suppose the pre-pressure is $F$, the stiffness of the spring pre-stress stage is $K_i$, the amount of pre-pressure equation can be obtained as follows:

$$x_0 = \frac{F}{K_i}$$  \hspace{1cm} (11)

In this study, the design parameters of the disc spring are as follows:
- Outer diameter $D=40$(mm), inner diameter $d=20.4$(mm), disc thickness $s=0.8$(mm), inner height $h=1.1$(mm), $\alpha=0.762$TPa$^{-1}$.

Fig.7 shows the disc spring F-x curve drawn by using mathematic software.

Assuming the supposed pre-pressure applied to $F = 785$N, the pre-press deformation of the spring can be obtained through equation (11): $x_0 = 0.57$mm. It can be seen from Fig.7 that the pre-stress point, $F_0$, locates at an ideal location on the $f-x$ curve, the point locates at the region where the slope of the curve changes relatively fast.

4. Conclusions

In this paper, designs of the key parts, single-axis flexure hinge and disc spring, of giant magnetostrictive actuator are discussed and analyzed.

The single axis flexure hinge stiffness and strength characteristics are studied; calculation equations for rotation stiffness and strength for weak region are established as well. Fatigue characteristic is also analyzed as flexure hinge usually works under high frequency situation.

Requirements of GMA for pre-stress disc spring are analyzed, the spring should meet the pre-stress requirement while don’t cause too much output energy loss of GMM rod, so the bilinear broken line shape $f-x$ curve of disc spring are proposed, a disc spring is designed by configuring its geometric
parameters to make its spring force-deformation curve is approximate to the ideal spring force-deformation curve.

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