RESEARCH ON GIANT MAGNETOSTRICTIVE ACTUATOR FOR LOW FREQUENCY ADAPTIVE VIBRATION CONTROL

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Abstract

Using elastic and isolating objects with scheme of passive control can efficiently control the high frequency vibration, however it generally invalidates in control of low frequency vibration especially under 5 Hz. Vibration control using smart-material actuators has utilized in structural vibration control. The ability to deliver large forces, wide-band responses, precise and relatively large displacement has giant magnetostrictive actuator the great potentials in vibration, especially in the low frequency vibration control. In this research, a giant magnetostrictive actuator for low frequency adaptive vibration control has been realized and investigated. In order to avoid the quadric frequency effect on magnetostriction, and to realize control using linear harmonic signal, a permanent-biased giant magnetostrictive actuator is designed and constructed. In the meantime, a magnetostrictive adaptive control algorithm and a control system are realized. A self-tuning adaptive control method for the vibration control is proposed in comparison with the inverse feedforward-PID control which is fit to step response of GMA in previous work. The research into the actuator for low frequency adaptive vibration control is finally experimentally verified. It proves that giant magnetostrictive actuators may secure the applications in adaptive control of low frequency vibration.

Keywords: Giant magnetostrictive actuator, Adaptive vibration control, Permanent-biased actuator design, PID control

1 Introduction

Giant magnetostrictive materials (GMM) has been widely used in many applications due to its unique features of high magneto-mechanical coupling, such as high force, giant strain, high energy density, broad operating bandwidth. This renders it an obvious choice of material in developing new generation transducers and mini/micro actuators for precision positioning, vibration control and linear motor. Such applications have been introduced in [1-4]. However, GMM generally plays nonlinear quadric magnetostrictive effect [5], which means a general GMM actuator can always generate elongation when excited either positive or negative magnetic field.

Using elastic and isolating objects with scheme of passive control can efficiently control the high frequency vibration, however it generally invalidates in control of low frequency vibration especially under 5 Hz [6].
Vibration control using GMM actuator has utilized in heavy structural vibration control, micro-vibration control and adaptive control as well [7-9]. And it emerges the great potentials both on high and low frequency control.

In this research, a giant magnetostrictive actuator for adaptive control in vibration is realized. In order to avoid the quadric frequency effect of magnetostriction, and to perform a linear harmonic signal control, a permanent magnet biased giant magnetostrictive actuator is designed and constructed. In the meantime, a magnetostrictive adaptive control algorithm and a control system are realized. And the research on giant magnetostrictive actuator for low frequency adaptive vibration control is experimentally verified. The results prove that giant magnetostrictive actuators may secure the applications in control of low frequency adaptive vibration.

2 Design of the giant magnetostrictive actuator

2.1 Design principle of the quasi linear driving actuator

Magnetostriction can be described generally as deformation of a body in response to a change in its magnetization. It is characterized by an even function of the magnetic field, which implies that magnetostriction is proportional to strength of the externally activated magnetic fields and the strain of magnetostrictive rod is independent of the sign of the applied longitudinal magnetization. However, both expansion and contraction can be realized by magnetizing a GMM rod in sequence with two magnetic fields, H1 and H2, having different strengths and in opposite directions (Fig. 1).

Precisely owing to this effect, a giant magnetostrictive actuator may perform both positive and negative displacement following an alternative electromagnetic excitation under a permanent magnet biased circumstance. Accordingly, an active or an adaptive control strategy for vibrating attenuation may be realized by the actuator that enables following harmonic driving signal.

By this regime the sinusoidal current excitation can make the GMA not only perform elongation and contraction, but also realize a quasi linear harmonic oscillated output if a minor current signal is used to generate displacement within the linear region of magnetostriction of the GMM.
2.2 Giant magnetostrictive actuator design

A permanent-magnet-biased Giant magnetostrictive actuator (GMA) is designed and implemented in the research at first. In particular, the design of the GMA is based on the fact that both expansion and contraction can be accomplished by magnetizing a GMM rod consecutively with two magnetic fields with different magnitudes and opposite directions. By this conception we designed the actuator as shown in Fig. 2. The actuators’ design specifications: general size with ~ $\phi \times 32 \times 130$ mm, driven by a $\phi \times 10 \times 100$ mm TbDyFe rod, electromagnetic excited by solenoid (~670 turns, 100 mm length, 1 mm wire diameter, 10 mm inner-diameter, 30 mm outer diameter, resistance ~1.9 $\Omega$), 8 MPa pre-stress applied by two Belleville washers (stiffness: 3.0037×106 N/m) inserted on both sides between the covers (Ni-Fe alloy, 3 mm thickness, permeability ~10000 $\mu_0$). Additionally, for the actuator, the pre-biased permanent magnet barrels (NdFeB, each barrel: 2 mm wall thickness, 50 mm length, 32 mm outer diameter) is capable of providing magnetic intensity around 15 kA/m for obtaining approximately half saturated magnetostriction.

Fig. 3(a) displays the prototype and (b) the deformation with the applied magnetic field variation from -53.6 to 53.6 kA/m (corresponding to an AC current varying from -8 A to 8 A). The maximum negative displacement reaches -53 $\mu$m under 13.4 kA/m (-2A current) electromagnetic fields and the maximum positive displacement reaches 48 $\mu$m under 53.6 kA/m fields, so a total of ~ 100 $\mu$m peak to peak varied displacement or amplitude for vibration can be reached. This capacity may be adequate for active or adaptive vibration control, especially for precise or micro/nano vibration control. Fig. 3 (c) and (d) present the measured GMA displacement versus a minor sinusoidal current excitation $I = 0.1 \sin(2\pi \times 10t)$. Fig. 3(c) shows the linear relation between the input sinusoidal current and the output sinusoidal displacement. Fig. 3 (d) presents a quasi linear relation between the GMA displacement and the input current in magnitude within one period.
Fig. 3 (a) Experimental setup for measuring the prototype displacement, (b) the generated displacement of the prototype by applying scanning magnetic fields, (c) the input sinusoidal current and the output sinusoidal displacement of GMA, (d) the quasi linear relation between the GMA displacement and the magnitude of input current in one period.

2.3 Dynamic modeling of displacement of the GMA

Many dynamic modeling methods have been proposed in the recent years [10]. In the paper, the research in this section focuses on the dynamic modeling and analysis on the designed GMA. By the work, the dynamic displacement behavior of the designed GMA versus input current may be well understood. According to the Jiles-Atherton theory [5], the total magnetization $M$ of GMM and the exciting magnetic field strength, $H$ are constrained by the following differential equation:
The Jiles–Atherton model is mainly described by physical parameters and calculated by the internal magnetic-field intensity of ferromagnetic material. This model is mainly based on the magnetic hysteresis curve with differential equations consisting of five parameters $M_s, k, c, \alpha, \tilde{\alpha}$. Some parameters in equation (1) can be expressed as follows:

$$z = \frac{H + \tilde{\alpha}M}{a}$$  \hspace{1cm} (2)

$$M_{an} = M_s [\coth(z) - \frac{1}{z}]$$  \hspace{1cm} (3)

$$\delta_M = \begin{cases} 0, & \dot{H} < 0, M_{an} - M > 0 \\ 0, & \dot{H} > 0, M_{an} - M < 0 \\ 1, & \dot{H} (M_{an} - M) \geq 0 \end{cases}$$  \hspace{1cm} (4)

$$H = f_g n I$$  \hspace{1cm} (5)

With $\frac{dM}{dH} = f(M, H)$, integration of Equation (1) gives:

$$M = \int f(M, H) dH$$  \hspace{1cm} (6)

$$\lambda = \frac{3}{2} \frac{\lambda_s}{M_s^2} M^2$$  \hspace{1cm} (7)

In Eqs. (1-7), $k$ is the irreversible loss coefficient, $M_s$ is the saturation magnetization, $f_g$ is the shape coefficient of the solenoid, $n$ is the number of turns of the coil per meter, $I$ is the input current strength. $\delta_M$ is a coefficient used to avoid negative value of magnetic susceptibility when the magnetization reaches to saturation. $\delta$ gets either 1 or -1 as $H$ increases or decreases. $\lambda$ is the magnetostriction coefficient (the GMA rod strain), $\lambda_s$ is the saturation magnetostriction coefficient, $M_{an}$ is the magnetization without magnetic hysteresis, $\alpha$ is the magnetic domain coefficient ($\tilde{\alpha} = \alpha + \frac{9\lambda_s \sigma_0}{2\mu_0 M_s^2}$), $\alpha$ the magnetic coupling parameter for quantifying the moments interactions), $c$ is the reversible coefficient, $a$ is the shape coefficient of magnetization without hysteresis. The values of these coefficients used in this work are: $k = 3509, M_s = 5.21 \times 10^5 A/m, n = 670, \lambda_s = 1057.8 \times 10^{-6}, K_r = 2.96 \times 10^7 N/m, c = 0.3886$ and $a = 7296.7$. 

$$\frac{dM}{dH} = \left\{ \frac{\delta_M [M_{an} - M]}{k\delta - \frac{\tilde{\alpha}}{1-c} [M_{an} - M]} + \frac{1}{\tilde{\alpha}} \right\} - \frac{1}{\tilde{\alpha}}.$$  \hspace{1cm} (1)
Consequently, the general output displacement of the GMA varied with the input exciting current i.e. the input magnetic field may be carried out by the GMA rod’s strain, $\lambda$ from equation (7) under an equivalent dynamic model of the GMA (Fig. 4).

![Equivalent dynamic model of a giant magnetostrictive actuator](image)

In Fig.4, $L$ represents the length of the GMM rod. $K_r$ is the equivalent stiffness of the GMM rod. $C_r$ is the equivalent damping coefficient of the GMM rod. $C_t$ is the equivalent damping coefficient of the disc spring. $K_t$ is the equivalent stiffness of disc spring. $M_r$ represents the equivalent mass of the GMM rod. $M_t$ is the mass of sleeve plus coil. $M_i$ is the mass of output shaft and load.

From Fig.4, the output displacement $d$ of GMA is as follows:

$$d(s) = \frac{K_r L \lambda(s)}{(M_r + M_i)s^2 + (C_r + C_t)s + K_r + K_t - \frac{(C_r s + K_t)^2}{M_i s^2 + 2C_t s + 2K_t}}$$

Considering the linear elastic law of the GMA rod and the influence from mass and damping of GMA rod, the transfer function of the strain of GMM and its applied current is as follows:

$$\frac{\lambda(s)}{I(s)} = \frac{Nd_{33} / L_s}{\rho L_i^2 \frac{C_s}{E_H} s^2 + \frac{C_D}{E_H} s + 1}$$

Where $E_H$ is Young’s modulus, $\rho$ is density, $C_D$ is internal damping, $d_{33}$ is piezomagnetic coefficient, $L_s$ and $N$ are length and number of turn of coil respectively.

From Eqs. (8) and (9), the transfer function of the output displacement of GMA and its applied current is as follows:

$$\frac{d(s)}{I(s)} = \frac{d(s) \lambda(s)}{\lambda(s) I(s)} = \frac{K_r L}{(M_r + M_i)s^2 + (C_r + C_j)s + K_r + K_b - \frac{(C_r s + K_b)^2}{M_b s^2 + 2 C_b s + 2 K_b}} \frac{Nd_{33} / L_s}{\rho L_i^2 \frac{C_s}{E_H} s^2 + \frac{C_D}{E_H} s + 1}$$

The corresponding parameters are given in Table 1. From equation (10), the generated displacement of the GMA is calculated and presented in Fig. 5 when applying a current of 8A with a frequency varied from 0~2000Hz.
Table 1. Value of the corresponding parameters

<table>
<thead>
<tr>
<th>Corresponding Parameters</th>
<th></th>
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<tbody>
<tr>
<td>$M_r = 0.045,\text{kg}$</td>
<td>$M_h = 0.5,\text{kg}$</td>
</tr>
<tr>
<td>$L = 100,\text{mm}$</td>
<td>$N = 670$</td>
</tr>
<tr>
<td>$K_h = 5.4 \times 10^6 ,\text{N/m}$</td>
<td>$\rho = 9250,\text{kg/m}^3$</td>
</tr>
<tr>
<td>$d_{33} = 4.915 \times 10^{-9},\text{m/A}$</td>
<td>$C_r = 3.3 \times 10^3 ,\text{Ns/m}$</td>
</tr>
<tr>
<td></td>
<td>$C_b = 1 \times 10^3 ,\text{Ns/m}$</td>
</tr>
<tr>
<td></td>
<td>$E^H = 3 \times 10^{10} ,\text{N/m}^2$</td>
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Fig. 5 Displacement of the GMA versus frequency

Understood by the results shown in Fig. 5, the designed GMA can fully deliver out the displacement of the GMM rod at frequency less than 100 Hz, and reach to the saturation displacement at around 350 Hz. But the displacement decreases more and more sharply as the exciting current applies with frequency more than 500 Hz. Evidently, the designed GMA is quite competent to be the actuator in vibration control at low frequency less than 100Hz.

3 Control Algorithm and Simulation

3.1 Identification Method

A robust identification method depends on its rapidity and stability of identification. And a good identification method can decrease oscillating in the beginning of identification process, which makes GMA tracking the
reference signal early and accurately. In this section, the recursive extended least square method (RELSM) is adopted in the minimum variance self-tuning control (MVSTC), the RELSM law can be realized in MATLAB. It is important that convergence of identification parameters is faster than other identification method. Assume that system is adopted with the CARMA model as follows

\[ A(z^{-1}) y(k) = z^{-d} B(z^{-1}) u(k) + C(z^{-1}) \xi(k) \]  

(11)

Where, \( u(k) \) and \( y(k) \) are the input and the output of system, respectively, \( d \) is pure delay, \( k \) is sample point and

\[
\begin{align*}
A(z^{-1}) &= 1 + a_1 z^{-1} + a_2 z^{-2} + \cdots + a_n z^{-n} \\
B(z^{-1}) &= 1 + b_1 z^{-1} + b_2 z^{-2} + \cdots + b_n z^{-n} \\
C(z^{-1}) &= 1 + c_1 z^{-1} + c_2 z^{-2} + \cdots + c_n z^{-n}
\end{align*}
\]

The CARMA model in equation (11) is written in least square form, namely

\[ y(k) = \{\phi^T(k)\} \{\theta\} + \hat{\xi}(k) \]  

(12)

Where

\[
\begin{align*}
\{\phi(k)\} &= [-y(k-1), \cdots, -y(k-n_u), u(k-d), \cdots, u(k-d-n_y), \xi(k-1), \cdots, \xi(k-n_z)]^T \in \mathbb{R}^{(n_u+n_y+n_z) \times 1} \\
\{\theta\} &= [a_1, \cdots, a_n, b_1, \cdots, b_n, c_1, \cdots, c_n]^T \in \mathbb{R}^{(n_u+n_y+n_z) \times 1}
\end{align*}
\]

The \( \hat{\xi}(k) \) in \( \{\phi(k)\} \) is not observable, therefore the \( \xi(k) \) is replaced by its estimated value \( \hat{\xi}(k) \) that is as follow,

\[ \hat{\xi}(k) = y(k) - \hat{y}(k) = y(k) - \{\phi^T(k)\} \{\hat{\theta}\} \]  

(13)

Where

\[
\begin{align*}
\{\phi(k)\} &= [-y(k-1), \cdots, -y(k-n_u), u(k-d), \cdots, u(k-d-n_y), \hat{\xi}(k-1), \cdots, \hat{\xi}(k-n_z)]^T \in \mathbb{R}^{(n_u+n_y+n_z) \times 1} \\
\{\hat{\theta}\} &= [\hat{a}_1, \cdots, \hat{a}_n, \hat{b}_1, \cdots, \hat{b}_n, \hat{c}_1, \cdots, \hat{c}_n]^T \in \mathbb{R}^{(n_u+n_y+n_z) \times 1}
\end{align*}
\]

And \( \{\hat{\theta}\} \) may be equal to \( \{\hat{\theta}(k)\} \) or \( \{\hat{\theta}(k-1)\} \). \( \phi(k) \) is replaced by \( \hat{\phi}(k) \), then using analogous derivation of recursive least square method, the estimated equations of RELSM are attained. They are as follows:

\[
\begin{align*}
\{\hat{\theta}(k)\} &= \{\hat{\theta}(k-1)\} + \{K(k)\} \{y(k) - \{\hat{\phi}^T(k)\} \{\hat{\theta}(k-1)\} \} \\
\{K(k)\} &= \frac{\{P(k-1)\} \{\hat{\phi}(k)\}}{1 + \{\hat{\phi}^T(k)\} \{P(k-1)\} \{\hat{\phi}(k)\}} \\
\{P(k)\} &= \{I\} - \{K(k)\} \{\hat{\phi}^T(k)\} \{P(k-1)\}
\end{align*}
\]

(14)

3.2 Control Laws and Design

The block diagram of Self-tuning adaptive control (STAC) is consisted of parameters estimator and control algorithms shown in Fig.6, which forms a real-time control system by tuning controller’s parameters automatically. The parameters’ estimator achieves model identification for GMA on line by RELSM which is
classical identification method. The controller is realized with minimum variance self-tuning regulator (MVSR). The reference signal may be either square wave or sine wave when the GMA is used to drive mechanism, or zero when the GMA is used to protect mechanism from interference. Moreover, the feedback from output displacement of GMA may strengthen the robust property of controller and make GMA track the reference signal actively.

The MVSR law is derived from equations (15) to (18), which are given below. First, the mathematics model of GMA is identified by the parameters estimator with input $u(k)$ and output $y(k)$ of system in equation (11). And then, by an assumption that the expectation of output is $yr(k+d)$, and real output is $y(k+d)$, the performance indicator can be given in equation (15), which must be least by designing control law. Finally, the output displacement is close infinitely to reference signal if equation (16) is established.

$$J = E\{[y(k+d) - yr(k+d)]^2\}$$  \hspace{1cm} (15)

$$C(z^{-d})yr(k+d) = G(z^{-d})y(k) + F(z^{-d})u(k)$$  \hspace{1cm} (16)

Where $F(z^{-d})$ and $G(z^{-d})$ are from Diophantine equations as

$$\begin{cases} C(z^{-1}) = A(z^{-1})E(z^{-1}) + z^{-d}G(z^{-d}) \\ F(z^{-d}) = B(z^{-1})E(z^{-1}) \end{cases}$$  \hspace{1cm} (17)

Where

$$E(z^{-1}) = 1 + e_1z^{-1} + \cdots + e_nz^{-nc}$$
$$F(z^{-1}) = g_0 + f_1z^{-1} + \cdots + f_nfz^{-nf}.$$  
$$G(z^{-1}) = f_0 + g_1z^{-1} + \cdots + g_ngz^{-ng}$$

Then, the closed loop system may be easily obtained, refer to Fig. 4

$$y(k) = \frac{C z^{-d}B}{1 + z^{-d}BG \frac{A}{F}} yr(k+d) + \frac{C}{1 + z^{-d}BG \frac{A}{F}} \xi(k) = yr(k) + E \xi(k)$$  \hspace{1cm} (18)

Where $A$ and $B$ are from equation (11), $G$ is from equation (17).

![Fig. 6 The block diagram of adaptive self-tuning control system](image)
To emphasize the better control effect of STAC in comparison with other control methods for vibration control, in the paper, the Inverse Feed-forward-PID controller is studied. The block diagram of Inverse Feed-forward-PID control (IFPIDC) system is shown in Fig. 7. The inverse Jiles-Atherton model, which compensates the hysteresis loop from GMM, is based on Jiles-Atherton model and its related parameters which are identified off line. The three parameters ($K_p$, $K_i$ and $K_d$) of PID control are determined by Ziegler-Nichols (Z-N) method. The Jiles-Atherton model has been successfully used in IFPIDC to improve GMA’s response speed [11].

The two control results of track step signal for GMA are shown in Fig. 8. It is understood that the tracking result with STAC is smoother than that with IFPIDC by simulations.

**Fig. 7** The block diagram of inverse Feedforward-PID control system [8]

**Fig. 8** The tracking result with STAC and with IFPIDC

### 3.3 Simulation Results of Active Control
The simulation presents both a free vibration control and a chirp signal disturbance suppression for GMA by using the above two controllers. Result of free vibration control is shown in Fig. 9, which proves the amplitude of free vibration of GMA can be decreased to the extent by STAC better than that by IFPIDC. And result of the chirp signal (frequency 0-5 Hz) disturbance suppression is shown in Fig. 10, which can prove that the disturbed amplitude is controlled much better by using STAC than by IFPIDC as GMA receives the chirp signal.

4 Experimental Test of Active Control

The results derived from simulation have been further validated by experimental tests. The experimental setup is shown in Fig. 11. It mainly includes: a GMA, a laser displacement sensor, a computer, a tailored programmable current supply and a serial ports acquisition gadget. First, the laser displacement sensor acquires the displacement signal to computer by the serial ports acquisition gadget. Then, the controller which is realized with MATLAB script of LABVIEW programming software receives and processes the displacement signal. Finally, the computer sends the control signal from controller to the current supply to excite the GMA. The experimental tracking results of step signal and chirp disturbance (frequency 0-5 Hz) control are shown in Fig. 12 (a) and (b), respectively. The results shown in Fig. 12 indicate that both the displacement tracking and active
control are effective although they are not smoother than simulations results. This validates the displacement tracking and the disturbance suppression (up to about 72% attenuation) property of GMA.

Fig. 11 The experimental setup

Fig. 12 Experimental Test (a) tracking result with STAC (b) chirp disturbance suppression with STAC

5 Conclusions and Perspectives
In this research, a permanent-magnet-biased giant magnetostrictive actuator for low frequency adaptive vibration control has been designed and investigated. The realized actuator may generate displacement avoiding the inherent quadric frequency effect on magnetostriction, and enable the linear harmonic signal control realized. The magnetostrictive adaptive control algorithm and a control system, i.e. the self-tuning adaptive control method proves efficient control results especially in comparison with the inverse feed-forward-PID control in previous work. The experimental results verify that permanent magnet biased giant magnetostrictive actuator may provide adequate behaviors in responding to harmonic control signal to realize active/adaptive control.

The research into the actuator for low frequency adaptive vibration control is finally experimentally verified. The chirp signal disturbance of 0-5Hz can be suppressed to about 72%. It proves that giant magnetostrictive actuators may secure the applications in adaptive control of low frequency vibration.

Acknowledgements

The work was supported by NSFC Fund (11172169, 10972137, 11072148), Research Project of State Key Laboratory of Mechanical System and Vibration( MSVZD201102), and the Shanghai Astronautical Research Funding (HTJ10-08), for which the authors are most grateful.

Reference