Design and analysis of active vibration damper for telescope by linear

motor

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ABSTRACT

The imaging quality and resolution of telescopes are deeply affected by the vibration that caused by electromechanical system and wind shake. Normally, vibration is caused by insufficient damping of the structure. In this paper, an active damper system based on linear motor is proposed to suppress vibration. The model of the whole control system is established at the beginning of this paper. LQR (Linear Quadratic Regulator) algorithm is proposed and the simulation is performed based on actual parameters of system. The results show that the system has a higher stability with higher Q value. The dynamic characteristics of the structure was obtained by analyzing the modal test data from accelerators. The experiments have been carried out to test the performance of the system. The results indicated that the active vibration damper can reduce the structure vibration 93.8% at 5.5Hz and increase the stiffness of structure. **Keywords:** Active vibration damper; modal analysis; linear motor; LQR; telescope

1. INTRODUCTION

More and more large and complex astronomical telescopes have been built in recent years in order to improve the ability of light-gathering and resolution. But the large telescopes are sensitive to vibration. Vibrations can be caused by many different sources like wind shaking, mechanical components (e.g. fans, pumps and motors), or even telescope tracking errors [1]. Wind shaking will impact the large telescope significantly because they have large area exposed to wind. Moreover, the natural frequency of the first mode of the large telescopes deceases with the increase of the aperture. Vibrations of the telescope's mechanical structure distort the optical path between the telescope's first optical element and the imaging [2]. Even minute vibrations can deteriorate significantly the image quality [3]. Normally, the passive vibration isolation is used to mitigate the vibration. But the low frequency vibration cannot be effectively reduced by means of passive vibration isolation. So more efficient methods must be proposed to suppress vibration in order to improve image stability and image quality.

A variety of researches about structural control system have been done in recent years. The major methods include base isolation, damper and mass damper. The mass damper is the system that equipped with additional mass, spring and damper to change the characteristics of structure [4]. The system with mass damper is very effective for eliminating low frequency vibration. Normally, mass damper is classified as TMD (Tuned Mass Damper) and AMD (Active Mass Damper). The advantage of TMD is that it doesn't need any power and electronics. Currently, TMD is applied in SMT [5], FAST [6] and GPi[7] successfully. But the disadvantage of TMD is that it only can mitigate target mode. Multiple

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target modes must employ multiple TMDs. The AMD was proposed first time by professor J.T.P.Yao in 1972. The AMD has smaller size and higher damping ratios than TMD. Furthermore, single set of AMD can be used to efficiently damp multiple target modes. So the AMD is widely used in SOFIA [8] and FAST [6] telescopes, as well in tall building and bridge [9].

An AMD system normally comprises actuator, controller, a mass block, sensor and power supply. The actuator is the key component in AMD system among all the parts. Currently, the commonly used types of actuators are hydraulic actuator, PZT, MR (magnetorheological fluid) and motor-type actuator. The motor-type AMD system is free from oil leakage, long distance, high dynamic response compared to hydraulic AMD systems. So the motor-type AMD systems are widely used in many fields. The voice-coil actuators are used in SOFIA to eliminate image jitter on Primary Mirror Assembly [8]. Guenfaf et al [10] applied the induction motor and LQR control algorithm to control vibration of buildings. Qing et al [11] applied AMD system based on PMSM to solar arrays of satellite.

The primary aim of this research is to apply AMD system based on linear motor to mitigate the vibration of structure. In order to achieve the AMD performance, the mathematics and control algorithms was studied in detail. The experimental test was performed with three-story structure. The efficiency of the AMD system was proved by estimating its control performance with uncontrolled system.

2. MECHANICAL MODEL OF THE STRUCTURE

The mathematic of structure is the fundamental to apply control algorithm. The three-story structure is adopted in this research. The AMD system is installed on the top of structure because the displacement of top floor has maximum displacement comparing with other floor. The AMD system will dissipate energy from host structure and mitigate the vibration of host structure. Figure 1 shows dynamic system of structure and the lumped mass model of AMD system.

For the three-story structure, the dynamic characteristic can be written in eq. (1) if damping is not taken into consideration.

$$[M]{\ddot{x}} + [K]{x} = 0$$
(1)

Where [M] is mass matrix. $[M] = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix}$. [K] is stiff matrix. $[K] = \begin{bmatrix} k_1 + k_2 & -k_2 & 0 \\ -k_2 & k_2 + k_3 & -k_3 \\ 0 & -k_3 & k_3 \end{bmatrix}$. Where

 k_1, k_2 and k_3 are lateral stiffness of each layer.



Figure.1. Structural dynamic system implemented with an AMD system

For AMD system, the dynamic characteristic can be written from eq. (2) to eq. (6) if the three-story structure is considered as lumped mass system.

$$\ddot{X}_1 = \frac{1}{M} \{ -KX_1 - C\dot{X}_1 + k(x - X_1) + f_c \}$$
(2)

$$\ddot{x} = \frac{1}{m} \{ -k(x - X_1) - c(\dot{x} - \dot{X}_1) - f_c \}$$
(3)

$$\dot{I} = \frac{1}{L}(-RI + u - e_b) \tag{4}$$

$$f_c = K_c I \tag{5}$$

$$e_b = K_e(\dot{x} - X_1) \tag{6}$$

Where f_c is motor torque, e_b is back EMF, K_e is back EMF coefficient, L and R are inductance and resistance of motor, M, K, C is mass, stiff, damping of host structure, m, k, c is mass, stiff, damping of AMD system, X₁ is the displacement of host structure, x is the displacement of AMD system. The eq. (2) to eq. (6) can be rewritten in the form

$$x_{d} = x - X_{1}$$

$$\ddot{X}_{1} = -\frac{C}{M} \dot{X}_{1} - \frac{K}{M} X_{1} + \frac{k}{M} x_{d} + \frac{K_{c}}{M} I$$
(8)

$$\ddot{x_d} = \frac{c}{M} \dot{X_1} + \frac{K}{M} X_1 - \left(1 + \frac{m}{M}\right) \frac{k}{m} x_d - \left(\frac{1}{M} + \frac{1}{m}\right) K_c I \quad (9)$$
$$\dot{I} = -\frac{K_d}{L} \dot{x_d} - \frac{R}{L} I + \frac{1}{L} u \qquad (10)$$

So the state space representation of whole system can be rewritten in the form

$$\dot{X} = AX + Bu \tag{11}$$
$$y = CX \tag{12}$$

Where X is state vector,
$$X = \{\dot{X}_1 \ \dot{x}_d \ X_1 \ x_d \ I\}^T = \{x_1 \ x_2 \ x_3 \ x_4 \ x_5\}^T$$
, y is displacement of host structure, $B = [0 \ 0 \ 0 \ 0 \ \frac{1}{L}]^T$, $C = [0 \ 0 \ 1 \ 0 \ 0]$.

$$A = \begin{bmatrix} -\frac{C}{M} & 0 & -\frac{K}{M} & \frac{k}{M} & \frac{K_c}{M} \\ \frac{C}{M} & 0 & \frac{K}{M} & -(1+\frac{m}{M})\frac{k}{m} & -(\frac{1}{M}+\frac{1}{m})K_c \\ 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & \frac{K_e}{L} & 0 & 0 & -\frac{R}{L} \end{bmatrix}$$

The state feedback can be deduced from above equations. Where u is the control input of motor.

$$u = -KX \tag{13}$$

3. OPTIMAL CONTROL LAW

The control capacity of AMD system depends on the kind of algorithm that is employed. Many control algorithms are well developed and studied in AMD system. Heo et al [4] proposed the unified Lyapunov control algorithm to reduce the vibration of two-storied building structure. Cao et al [12] applied the H ∞ control strategy to structure-ATMD to improve the stability. Metered et al [13] applied fuzzy logic control algorithm to MR damper for seat suspension system. Compared with other control strategy, linear quadratic (LQ) control can achieve the optimality and stability together.

Furthermore, LQ control can achieve the optimal solution under the environmental of uncertainty. So LQ control strategy is widely used in AMD system [10] [14].

An AMD system can be described by mathematical model in eq. (11) and eq. (12). The cost function can be defined as

$$J = \int_0^\infty [X^T Q X + U^T R U] dt$$
(15)

Where Q is a weighting matrix for the structural response and R is a weighting matrix for the control force. The feedback law can be constructed for the plant as

$$\mathbf{U} = -\mathbf{R}^{-1}\mathbf{B}^{\mathrm{T}}\mathbf{P}\mathbf{X} = -\mathbf{K}\mathbf{X} \tag{16}$$

Where K is Kalman gain matrix, the positive matrix P can be got from the Riccati equation (17). Then the closed loop system is asymptotically stable and it minimizes the cost function.

$$\mathbf{P}\mathbf{A} + \mathbf{A}^{\mathrm{T}}\mathbf{P} - \mathbf{P}\mathbf{B}\mathbf{R}^{-1}\mathbf{B}^{\mathrm{T}}\mathbf{P} + \mathbf{Q} = \mathbf{0}$$
(17)

In order to validate the effectiveness of LQR algorithm, the simulation was performed according to the mathematics of AMD system that described in previous chapter. The parameters of AMD system are listed in Table 1.

Symbol	Value(Unit)	Symbol	Value(Unit)
R	10.2(^Ω)	L	2.63(mH)
Ke	25.15(V/m/s)	Кс	25.15(N/A)
К	127*10 ³ (N/m)	k	8*10 ³ (N/m)
М	31(Kg)	m	2(Kg)
С	200(N/m/s)	с	6.3(N/m/s)

Table 1. AMD	system	parameters
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The time response to impulse input is shown in Figure 2. The simulation parameter are Q=q*diag[1,1,1,1,1] and R=1. It can be seen that large Q leads to an increase in stability of the system. In principle, the large Q means that the eigenvalues of the matrix of the closed-loop system are farther to the left of the S plane. The results confirmed the theoretical analysis.



Figure.2. output curve with different matrix Q

4. EXPERIMENTAL TEST

4.1 Test Structure

The AMD system and test structure employed in this research are shown in Figure 3. The one degree-of-freedom AMD system uses linear servo motor of the type Parker LXR series, model 404LXR. The motor is integrated with linear encoder. The travel distance is 100mm and the continuous force is 50N. The base of the motor is mounted on the top of structure. The mass is attached to the moving carriage of the motor. The spring and damping are installed between moving carriage and top floor.

The adopted structure is three stories, steel frame. The total height of the structure is 524mm. The weight of entire structure is 31kg, and the lumped mass of each floor is 9.36kg. The stiffness of top floor is 127×10^3 N/m. The displacement sensor (KAMAN KD-2306) is eddy current sensor that measure the displacement of top floor. Three accelerator sensors (PCB 393B04) are mounted on each floor to measure the acceleration.

In order to apply the AMD more effectively and to see the limitation of the control, the experimental approach is used to identify the modal parameter of the structure. The exciter is electrodynamic shaker (STI DC-600-6) and the swept sinusoidal signal is used as excitation signal. The frequency range of swept signal is from 5Hz to 50Hz which can cover all modal frequencies of the structure. The result is shown in Figure 4. It indicates that the modal frequency of structure are 5.5Hz, 9.8Hz, 20.78Hz and 21.45Hz.



Figure.3. Test structure



Vibration suppression demands high bandwidth of actuator. In order to evaluate the bandwidth of the motor, the dynamic response characteristic of motor is measured by FFT analyzer (Onosokki DS3200). The result is depicted in Figure 5. It can be seen that the bandwidth of about 12Hz can be achieved when a cascaded current and velocity loop is selected for drive system. The bandwidth of motor is lower than the modal frequency of structure at 20.78Hz and 21.45Hz. So the high order modal frequency cannot be mitigated efficiently.

4.2 Estimation of Experimental Results

The experiments are conducted to study the effectiveness of the vibration suppression. Sine sweep signal is used as excitation signal and the frequency range is from 5Hz to 50Hz. The displacement on the top floor with/without AMD is plotted in Figure 6 in order to compare the control performance of the AMD system. In this research, the vibration reduction is evaluated by two criteria. The first criterion is a measure of the normalized maximum floor displacement relative to the ground, as follows:

$$J_1 = \max(\frac{|x(t)|}{x_{max}}) \tag{18}$$

where x(t) is the relative displacement of the top floor, and x_{max} is the uncontrolled maximum displacement.

The other criterion is displacement transmissibility as shown in eq. (19). where x_g is the displacement of base floor. The x_q is 0.7mm in this research.

$$J_2 = \frac{|x(t)|}{x_g} \tag{19}$$

The experimental result at each modal frequency is shown in Table 2. From the table 2, the displacement reduction with AMD are evaluated to 93.8%, 39.3%, 15.3% and -25.5% at each modal frequency of structure. So we can draw the following conclusion.

case	5.5Hz	9.8Hz	20.78Hz	21.45Hz
Uncontrolled	3.06	0.28	0.85	0.55
Controlled	0.19	0.17	0.72	0.69
J_1	0.062	0.607	0.847	1.255
J_2	0.27	0.4	1.03	0.99

Table 2. Response of displacement (mm)

(1). The AMD system can mitigate the vibration dramatically, especially at low frequency band.

(2). The displacement reduction is decreased with increase the frequency. The displacement at modal frequency 21.45Hz increases when the AMD system is added on. Moreover, the controlled displacement lag behind the uncontrolled displacement at high modal frequency. The reason for this is that bandwidth of linear motor is too low for the high modal frequency. So the vibration at high modal frequencies cannot be mitigate effectively.

(3). The AMD system changes the mechanical characteristic of structure. The low modal frequency of 5.5Hz and 9.8Hz do not exist when the AMD system is applied. This is very useful for drive system of telescope because it enlarge the bandwidth of control system. Therefore the servo control system can achieve highly dynamic and highly accuracy.



Figure.5. Velocity frequency responses motor



Figure.6. Sine sweep test result

5. CONCLUSION

This paper proposes an AMD system based on linear motor that effectively suppresses the vibrations of one-degreeof-freedom structural system. The lumped mass model of AMD system is proposed according to the adopted three-story structure. The LQ algorithm is put forward in order to mitigate vibration effectively. The simulation result shows that LQ algorithm can get the good stability of the system by selecting suitable Q. Experimental tests were carried out to validate the control plan. The displacement reduction is evaluated to 93.8%, 39.3%, 15.3% and -25.5% at each modal frequency of structure. The result indicates that the motor-type AMD system can mitigate the vibration efficiency, especially at low frequency band.

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