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Movement dynamics analysis for an optical pickup head lens actuator in a digital versatile disc system

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With the rapid development of CD/DVD (compact disc/digital versatile disc) technology, the disc rotation speed and the information density have increased greatly to fulfil the demand of high information bit rate transfer. As one of the most important elements in a DVD pickup head, objective lens actuators are required to provide a more precise means to accurately follow the disc deviation. The design of the lens actuator becomes all the more important and critical. In order to give complete suggestion to the design of the lens actuator, we put forward the movement differential equations as a new theoretical means to analyse the characteristics of the lens actuator reliably and rigorously. Based on the theoretical analyses, many important and effective methods are proposed. A typical experiment verifies that the performance of our lens actuator head, which adopts these methods, can satisfy the high speed and high density requirement of DVD systems.

Keywords: optical storage, DVD, lens actuator, focusing
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1. Introduction

In digital versatile disc (DVD) pickup heads, providing the precise means to accurately follow the disc deviation is the function of the objective lens actuators. With the rapid development of information technology, large capacity storage systems with fast data transfer have come into use, thus requiring higher density recording and high rotation speed recorders/players. These improvements demand that optical head lens actuators have excellent performance, such as faster response time, wider frequency band, higher action agility, smaller lens tilt angle and better tracking precision. The design of the lens actuator has become more critical and competent.

In this paper, we present a theoretical means that can be adopted to analyse the characteristics of the lens actuator more reliably and rigorously. We put forward the movement differential equations to give complete suggestion to the design of the lens actuator. From the results of the theoretical analysis, one can easily attain the necessary and effective methods to enhance the performance of the actuator. A typical experiment verifies that lens actuator heads adopting those methods can satisfy the requirements of high speed and high density of DVD systems.

2. Design targets of lens actuators for DVD pickup heads

In DVD systems, with higher physical density and faster rotation speed, the following are most importnat for the lens actuator:

1) increase of the acceleration versus driving current;

2) stability under external stochastic vibration;

3) enhancement of the frequency band;

4) smaller objective lens tilt angle;

5) suppression of cross-coupled vibration;

6) suppression of vibration in any unnecessary directions;

7) smaller linearity limitation in the focusing and tracking directions;

8) lower head heat dissipation.

With the increase of the disc rotation speed, the acceleration of the disc becomes higher. High acceleration of the rotating disc makes it increasingly difficult for the smaller laser beam spot to focus on the information track; so higher actuator driving acceleration must come into consideration. However, in order to decrease the relative acceleration between the adjusting lens and the rotating disc, the increase of actuator

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acceleration for the lens is also important. An objective lens actuator needs a wider frequency band as the rotation frequency of the disc improves, thus the second parasitic-resonance frequency has to be kept as high as possible. In general, the mechanical resonance frequency of DVD pickup actuators should be over 10 kHz. Small lens tilt is also very critical for its significant effect on the readout signal from the disc. If the lens tilt angle is large, the quality of the beam reflected from the rotational disc will greatly degrade and the servo performance of the DVD system greatly degenerate. For DVD pickup heads, the maximum tilt angle acceptable should be less than ± 1.5 mrad when the actuator is at its maximal displacement.

The lens actuator has two working directions: one is tracking parallel to the disc radial direction and the other focusing perpendicular to the disc surface. In order to assure excellent performance in both directions, suppression of vibration in any unnecessary direction must be taken into account, such as lens tangential tilt vibration, lens radial tilt vibration, and especially the cross-coupled vibration. Smaller linearity limitation in the focusing and tracking directions of the lens actuator is also important for the readout system. If the nonlinearity responses in the focusing and tracking directions are too large, the servo performance of the DVD system based on the action responses will greatly degrade. One cannot imagine any pickup head without any linearity response. In fact, the precision of focusing and tracking is decided greatly by the limitation of the action linearity. In the long-time working DVD system, the power consumption of the pickup head should be low enough to extend the running time, so lower heat dissipation should not be neglected.

3. Theoretical analyses on the lens actuator and its application

3.1. Analysis model

The lens actuator can be divided into the sliding rotary type and the suspension cantilever type according to different structures (Fig.1).[1,3] The cantilever type shown in Fig.1(b) has many unavoidable defects, such as difficulties in fixing the four-cantilever to the stable base, strict requirement of the cantilever material, and unreliable small lens tilt management. One of the advantages of the sliding rotary type is that the lens holder is dynamically balanced to be a stable moving structure. The tilt angle is also easily managed because the movable part moves along and around the guide shaft. However, friction between the fixed shaft and the moving bush will result in bad linearity characteristics.

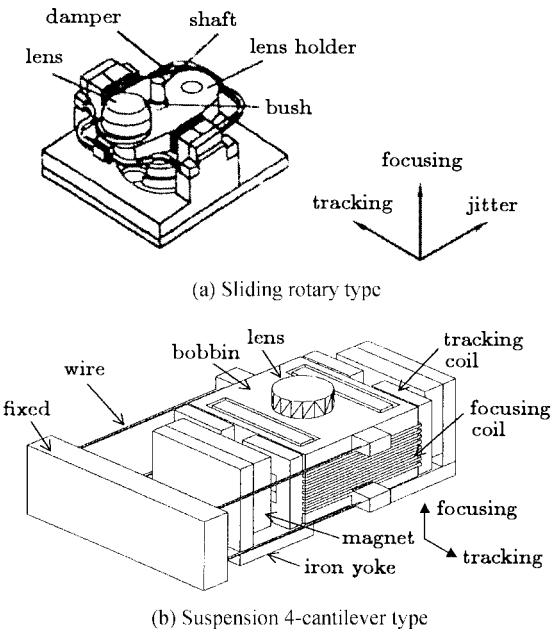


Fig.1. Typical lens actuator structure.

We choose the sliding rotary type actuator as the research object. So far, many articles have studied the characteristics of lens actuators. For example, the rotation equation is discussed to analyse tracking action,[1] and the moment balance equation is used to describe vibration performance.[2] Regrettably, almost none of these completely analyses the total action property. In order to realize the exact factors mainly determining the performance of the actuator, we have to consider all the possible movements in every direction fully. We propose the movement differential equations on the sliding rotary actuator as follows. Based on the differential equations, we can entirely analyse the characteristics of the lens actuator.

A theoretical model and force diagram of the sliding rotary actuator are shown in Fig.2. The object of our research is the movable part of the lens actuator, which slides along the centre shaft to attain focusing, and rotates around the shaft to realize tracking.

3.2. Theoretical analysis

The location of the lens in the movable part is shown as Fig.2. Because the lens is fixed tightly to the holder, the relative movement between the lens

and the holder can be left out of consideration at any time. Thus, in this paper, we consider the lens as one of the elements of the movable part, and the whole movable part is the object of our research.

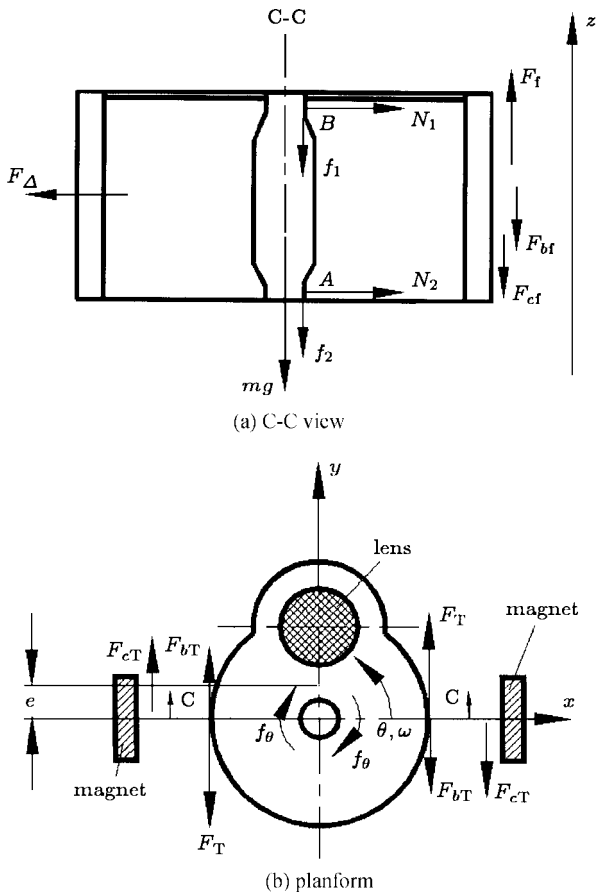


Fig.2. Sliding rotary actuator theoretical model and the force diagram.

Obviously, we can regard the movable part as a rigid body. Since the gyro-effect cannot be ignored, the initial position and angle deflection should also be considered, for the existing gap width between the fixed shaft and the bush of the movable part. However, in order to maintain a small tilt angle, the gap should be kept less than 10μm.

Orthogonal coordinates are defined as shown in Fig.2. While the z-axis is parallel to the axis of the fixed shaft, the x-axis is parallel to the radial direction of the disc and the y-axis along the tangential direction. The origin point of the coordinates is located at the centroid of the movable part. According to these definitions, the focusing direction is along the z-axis and the tracking direction along the x-axis. In the process of working, the position of the movable part is decided by the six coordinates of the centroid, which contain the rotation angles around the x-axis (α), y-axis (γ) and z-axis (θ), and the displacement along each coordinate axis. The middle part of the movable hollow bush is concave in view of reducing manufacturing difficulty and decreasing the friction caused by the contact of the shaft and the bush. Thus points A and B can be regarded as the force action points. The force diagram for the sliding rotary actuator is shown in Fig.2.

Based on mechanics principles, the common movement of the rigid body can be considered as the combination of two kinds of movements: the shifting of the centroid and the rotation around the centroid. The shifting of the centroid can be described by the centroid movement theory, and the rotation around the centroid can be defined by Eular dynamic theory. Because the gap between the fixed shaft and the movable bush is small enough, the direction of sliding friction can be considered as parallel to the axis of the shaft, i.e. either in the positive or in the negative direction of the z-axis, and the direction of rotary friction can be considered around the axis of the shaft. Thus the shifting equations and the rotation equations of the movable part are as follows:

z-axis direction (focusing direction)

$$m\ddot{z}_0 = F_{\rm f} - F_{bf} - f_1 - f_2 - mg - F_{cz}; \quad (1)$$

y-axis direction (undesired direction)

$$m\ddot{y}_0 = F_1\cos\omega t - F_{bt} + N_2\cos\omega t - N_1\cos\omega t - F_{cy}; \quad (2)$$

x-axis direction (undesired direction)

$$m\ddot{x}_0 = F_\Delta - F_1\sin\omega t + N_2\sin\omega t - N_1\sin\omega t - F_{cx}; \quad (3)$$

around z-axis direction (tracking direction)

$$J_{\rm P}\ddot{\theta} + C_{tz}\dot{\theta} + F_{b\theta} = F_{\rm T} - f_\theta; \quad (4)$$

around y-axis direction (undesired direction)

$$J_{\rm T}\ddot{\gamma} - J_{\rm P}\omega\dot{\alpha} + C_{ty}\dot{\gamma} = mge\sin\omega t; \quad (5)$$

around x-axis direction (undesired direction)

$$J_{\rm T}\ddot{\alpha} + J_{\rm P}\omega\dot{\gamma} + C_{tx}\dot{\alpha} = (mge + N_1 l_1 - N_2 l_2)\cos\omega t; \quad (6)$$

Here m is the mass of the movable part; ω is the rotational angle velocity around the z-axis; x_0, y_0 and z_0 are the coordinates of the centroid for the movable part in static state; F_Δ is the invariable magnetic force applied by the permanent magnet; N_1 and N_2 are the supporting forces at points A and B; F_1 is the deflection force if eccentricity exists; $J_{\rm P}$ is the pole moment of inertia of the movable part; $J_{\rm T}$ is the equator moment of inertia of the movable part; α, γ and θ are rotational angles around the x-, y- and z-axes; and l_1 and l_2 are the distances between centroid and points A and B.

It is easy to see that the focusing drive force $F_{\rm f} = N_{\rm f}B_{\rm f}L_{\rm f}I_{\rm f}$, the focusing restoration force $F_{bf} = K_z z_0$, the frictional force $f_1 = \mu N_1$, $f_2 = \mu N_2$, the damping force $F_{cz} = C_{tz}\dot{z}_0$, $F_{cx} = C_{tx}\dot{x}_0$, $F_{cy} = C_{ty}\dot{y}_0$, the deflection force $F_1 = me\omega^2$, the tracking drive force $F_{\rm T} = N_{\rm T}B_{\rm T}L_{\rm T}I_{\rm T}$, the restoration force $F_{b\theta} = K_\theta\theta$, and the frictional force $f_\theta = \mu(N_1 + N_2)$. $N_{\rm f}$ is the turn number of focusing coils; $B_{\rm f}$ is the intensity of the magnets for focusing; $L_{\rm f}$ is the valid length of focusing coils; $I_{\rm f}$ is the current in the focusing coils; $N_{\rm T}$ is the turn number of tracking coils; $B_{\rm T}$ is the intensity of the magnets for tracking; $L_{\rm T}$ is the valid length of tracking coils; $I_{\rm T}$ is the current in the tracking coils; K_z is the magnetic spring constant in the focusing direction; K_θ is the magnetic twisting spring constant in the tracking direction; C_{tx}, C_{ty} and C_{tz} are the damping coefficients; μ is the coefficient of sliding friction; and e is the distance between the deflective centroid and the axis of the fixed shaft.

Finally, we obtain the common movement differential equations for the sliding rotary actuator as follows:

$$m\ddot{z}_0 + K_z z_0 + C_{tz}\dot{z}_0 = N_{\rm f}B_{\rm f}L_{\rm f}I_{\rm f} - (\mu N_1 + \mu N_2 + mg), \quad (7)$$

$$m\ddot{x}_0 + C_{tx}\dot{x}_0 = -me\omega^2 \sin\omega t + (N_{2y} - N_{1y} + F_\Delta), \quad (8)$$

$$m\ddot{y}_0 + K_y y_0 + C_{ty}\dot{y}_0 = me\omega^2 \cos\omega t + (N_{2x} - N_{1x}), \quad (9)$$

$$J_{\rm T}\ddot{\alpha} + J_{\rm P}\omega\dot{\gamma} + C_{tx}\dot{\alpha} = (mge + N_1 l_1 - N_2 l_2)\cos\omega t, \quad (10)$$

$$J_{\rm T}\ddot{\gamma} - J_{\rm P}\omega\dot{\alpha} + C_{ty}\dot{\gamma} = mge\sin\omega t, \quad (11)$$

$$J_{\rm P}\ddot{\theta} + C_{tz}\dot{\theta} + K_\theta\theta = N_{\rm T}B_{\rm T}L_{\rm T}I_{\rm T} - \mu(N_1 + N_2). \quad (12)$$

We can analyse the movement characteristics of the lens actuator entirely from the above equations. Because the lens can be taken as one of the elements of the movable part in the actuator, the above equations also describe the motion characteristics of the lens. For example, equations (7) and (12) describe the action at any rational working position in the focusing and tracking directions. The two equations are the main objects of research in many articles.[4–9] Remarkably, equations (8)–(11) describe unwanted actions in the other four directions. If we want to fully investigate the vibrations in any unnecessary direction and the cross-coupled vibration, these equations will be the focus of the research.

4. Application of theoretical analysis to the focusing characteristics

A dissected schematic view of a typical sliding rotary lens actuator is shown in Fig.3. We study the movement dynamics characteristics of this type of actuator in the focusing and tracking directions by applying the theoretical equations (7) and (12).

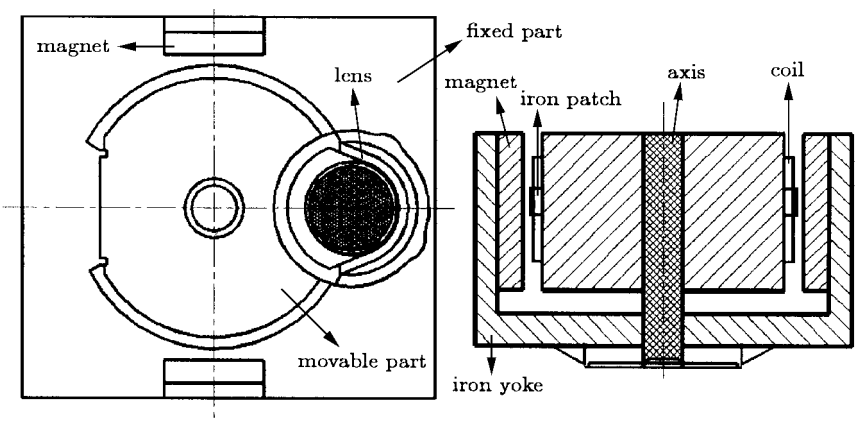


Fig.3. Schematic view of a typical sliding rotary type lens actuator.

When the movable part is in a static state, the force balance equation is expressed as

$$K_z z' = mg + f_{\rm s1} + f_{\rm s2}, \quad (13)$$

where z' is the displacement along the z-axis when the movable part is in static balance, and $f_{\rm s1}$ and $f_{\rm s2}$ are static frictions between the fixed shaft and the movable part.

If the movable part is in good force balance, any offset of the centroid can be ignored, and thus the deflection force derived from eccentricity should be neglected. The good force balance determines that the supporting force N_1 is equal to N_2, so

$$f_{\rm s1} = f_{\rm s2} = \mu_0 N_1/2 = \mu_0 N_2/2, \quad (14)$$

where μ_0 is the state friction coefficient.

Considering that the friction can be reduced to be small enough, we can obtain the equation as follows

$$f_{s1} + f_{s2} - f = 2\mu_0 N_1 - 2\mu N_1 = 0. \tag{15}$$

From equations (7), (13), (14) and (15), the dynamic equation in the focusing direction can be expressed as

$$m\ddot{z} + C_{tz}\dot{z} + K_z z = K_I I, \tag{16}$$

where K_I is the current gain coefficient, and $K_I = N_f B_f L_f$.

From equation (16), we can obtain the transfer function of focusing as follows

$$\begin{aligned}\frac{Z(s)}{I(s)} =& \frac{K_I}{ms^2 + C_{tz}s + K_z} \\ =& \frac{K_I/m}{s^2 + 2\dfrac{C_{tz}}{2m}s + \dfrac{K_z}{m}} \\ =& \frac{K\omega_{fn}^2}{s^2 + 2\zeta\omega_{fn}s + \omega_{fn}^2}.\end{aligned} \tag{17}$$

Equation (17) shows that the lens actuator is a quadratic oscillation system. It is easy to obtain the natural frequency in the focusing direction as

$$\omega_{fn} = \sqrt{\frac{K_z}{m}}. \tag{18}$$

Employing a similar method, we can also obtain the natural frequency in the tracking direction as

$$\omega_{tn} = \sqrt{\frac{K_\theta}{J_P}}. \tag{19}$$

From the above equations, we analyse the characteristics in the focusing direction for the application. The damping frequency is represented as

$$\omega_d = \omega_n\sqrt{1-\xi^2}, \tag{20}$$

where ξ is the equivalent damping coefficient in the focusing direction

$$\xi = C_{tz}/2\sqrt{mK_z}. \tag{21}$$

The response time of focusing is in proportion to the damping frequency. The relationships between the response time, the damping frequency and the equivalent damping coefficient in focusing are shown in Fig.4. From the curves, we can achieve the critical methods to efficiently reduce the focusing response time of the lens actuator. These methods are enhancing the focusing natural frequency and decreasing the focusing equivalent damping coefficient. The increase of the magnetic spring constant, and suppression of the damping coefficient and mass of the movable part will be most available. These methods are verified to satisfy the fast access requirement of high density and high speed DVD system.

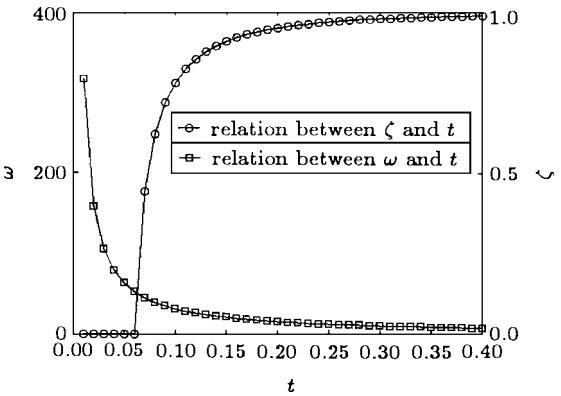


Fig.4. Damping frequency (ω) and equivalent damping coefficient (ξ) as functions of the response time (t_r) for a sliding rotary lens actuator in the focusing direction in a dynamic state.

In order to study the stability under external stochastic vibration, the static state response of the actuator must be considered. There are no currents in the focusing driving coils when the movable part is in a static state. The dynamic equation for a static actuator can be given as

$$m\ddot{z} + C_{tz}\dot{z} + K_z z = 0. \tag{22}$$

From equation (22), we can give the movement law for the static actuator as

$$\begin{aligned}z(t) =& \left[z_0\left(\cos\omega_d t + \frac{\zeta}{\sqrt{1-\zeta^2}}\sin\omega_d t\right)\right. \\ &\left. + \frac{\dot{z}_0}{\omega_d}\sin\omega_d t\right] e^{-\zeta\omega_{fn}t}.\end{aligned} \tag{23}$$

Equation (23) describes that the existence of free damping oscillation with the vibration frequency equal to the dynamic damping frequency. The vibration extent fluctuates according to the regulation

$$\frac{z_t}{z_{t+\tau}} = e^{\xi\omega_n\tau}, \tag{24}$$

where τ is the vibration period $\tau = \dfrac{2\pi}{\omega_n\sqrt{1-\xi^2}}$, $\xi\omega_n\tau = \xi\omega_n\dfrac{2\pi}{\omega_n\sqrt{1-\xi^2}} = \dfrac{2\pi\xi}{\sqrt{1-\xi^2}}$.

The relationships between the exponential power, the vibration period and the equivalent damping coefficient in the static focusing direction are shown in Fig.5. From the curves in the figure one can see that an enhancement of the focusing equivalent damping

coefficient can be adopted to increase the static focusing stability for the lens actuator. From the analyses above, the damping coefficient cannot be very low. In fact, a lens actuator with a damping coefficient of 0.1 will have good static focusing stability. The free damping oscillation is shown in Fig.6. However, suppressing the vibration period is also a useful method, which demands the increase of natural frequency by enhancing the restoration rigidity or reducing the mass of the movable part.

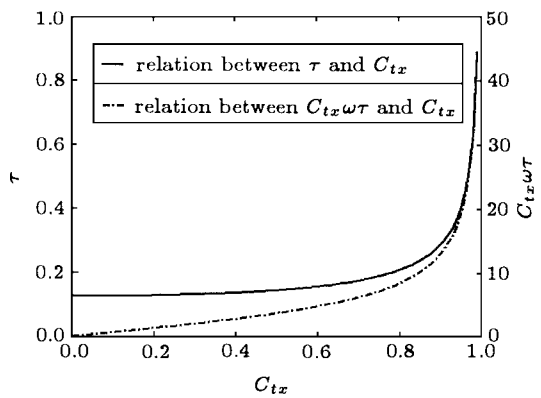


Fig.5. Relationships between the exponential power ($\xi\omega_n\tau$) and the vibration period (τ), and the equivalent damping coefficients (ξ) for a sliding rotary lens actuator in the focusing direction in a static state.

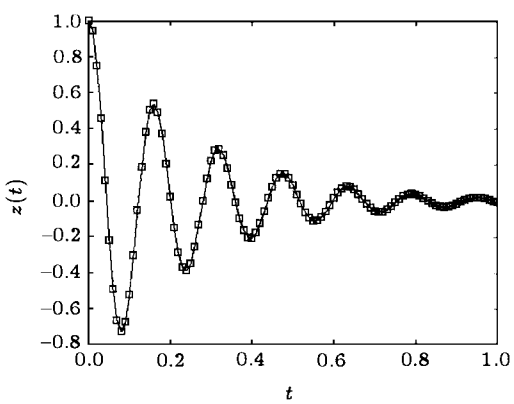


Fig.6. Free damping vibration for a sliding rotary lens actuator in a static state.

The moving vibration along the x-axis must be considered for influencing the characteristics of the lens actuator in the tracking direction. Equation (8) illustrates the movement features along the x-axis. Because the magnetic force F_Δ along the x-axis applied by the magnet can be taken as constant, the force balance equation along the x-axis is

$$F_\Delta + N_{2y} - N_{1y} = 0. \tag{25}$$

From equations (8) and (25), the vibration equation along the x-axis can be expressed as

$$m\ddot{x}_0 + C_{tx}\dot{x}_0 = -me\omega^2 \sin\omega t. \tag{26}$$

Thus the movement characteristics equation along the x-axis is

$$x_0 = X\cos(\omega t - \varphi), \tag{27}$$

where

$$X = \frac{me\omega}{C_{tx}^2 + m^2\omega}\sqrt{C_{tx}^2 + m^2\omega^2}, \tag{28}$$

$$\operatorname{tg}\varphi = \frac{m\omega}{C_{tx}}. \tag{29}$$

Equations (27) and (29) show that the moving vibration along the x-axis can be ignored if no mass deviation exists (i.e. e=0); otherwise, it cannot be ignored. From equation (28), reducing the mass of the movable part, enhancing the x-axis damping coefficient and decreasing the working frequency can be adopted to depress the moving vibration along the x-axis for the lens actuator.

5. Typical experimental results

Two samples of actuator are adopted in the experiment to test our theory. Table 1 shows the properties for each actuator. We apply smaller and lighter coils to reduce the mass and depress the heat dissipation, and install stronger and wider magnets to enhance the drive force efficiency and increase the displacement linearity for sample 2. However, the gaps of the two samples between the holder and the shaft are both less than 10μm. Therefore the characteristics of lens tilt are good in both cases.

Table 1. Properties of the two sliding rotary actuators.

Items	Sample 1	Sample 2
Mass of movable part	800 mg	600 mg
Damping coefficient in focusing direction	0.2	0.1
Restoration rigidity in focusing direction	1.04	1.06
Damping coefficient in tracking direction	0.2	0.1
Restoration rigidity in tracking direction	2.08	1.62

Figure 7 shows the frequency characteristics (Bode diagram) of two samples of sliding rotary actuator. Figures 7(a) and 7(b) show the frequency characteristics for sample 1. Figures 7(c) and 7(d) show the frequency characteristics for sample 2. In the focusing direction, Fig. 7(a) indicates that the first resonant frequency of sample 1 is about 36 Hz. Figure 7(c) shows that the first resonant frequency of sample 2 will be shifted to about 42 Hz. In the tracking direction, we can obtain similar results. The response time of sample 2 is much shorter than sample 1 and the driving sensitivity of sample 2 is much higher (Table 2). The figures also show that the second parasitic resonant frequencies are both over 20 kHz and the peaks are small enough to be ignored. The damping time of external vibration in static stability are enough short for the two samples, on account of adequate damping coefficients. Table 2 shows the comparison of experimental results for the two samples.

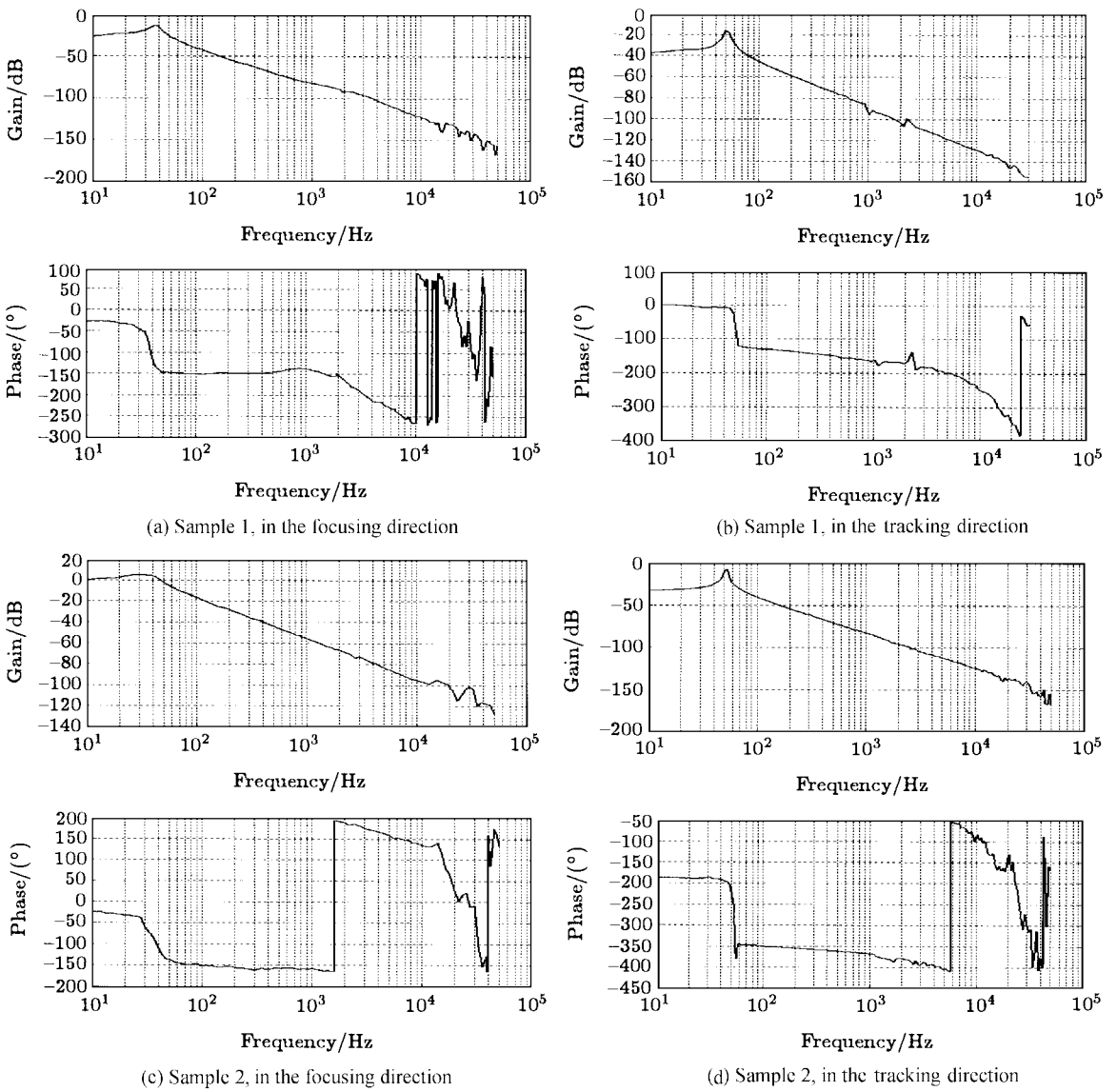


Fig.7. Frequency response characteristics for the two sliding rotary actuators.

Table 2. Comparison of experimental results.

Items		Sample 1	Sample 2
Natural frequency in focusing direction		36 Hz	42 Hz
Natural frequency in tracking direction		51 Hz	52 Hz
Driving sensitivity in	ac (5 Hz)	0.91 mm/V	1.22 mm/V
focusing direction	dc (200 Hz)	38.51 μm/V	48.96 μm/V
Driving sensitivity in	ac (5 Hz)	0.32 mm/V	0.58 mm/V
tracking direction	dc (200 Hz)	27.89 μm/V	47.89 μm/V

6. Conclusion

We have successfully developed a useful theoretical means that can be adopted to analyse the characteristics of the lens actuator more reliably and rigorously. We put forward the common movement differential equations to give complete suggestion to the design of the sliding rotary lens actuator. The static and dynamic analytic models for the actuator are achieved, and improvement methods are proposed. Comparing two sample actuators, the typical experiment verifies that lens actuator heads adopting those methods can satisfy the requirements of high speed and high density of DVD systems, with higher driving sensitivity and more stability under external vibration.

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