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Design and Control of Tracking Actuator for Optical Disk

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Abstract

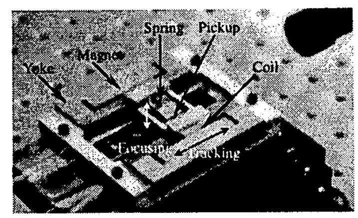
In optical disk system, a tracking system consists of a coarse actuator and a fine tracking actuator. Double actuality system requires many devices in order to obtain large stroke and precise displacement simultaneously. Moreover, the moving parts become sophisticated because of the complicated configuration. In this paper, one-dimensional new tracking actuator combining a fine tracking actuator and a coarse actuator is designed. The Voice Coil Motor is used as a basic drive mechanism. The permeance method is used to analyze and to design VCM for large driving force and its characteristic is tested through experiments. The focus spring is designed in terms of the stability and characteristic of system. Since friction is dominant error source, friction identification is carried out. The designed actuator shows its ability to resolve a motion less than 0.1μm.

1. Introduction

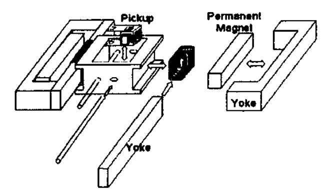
Optical disk consists of media part, mechanism part and signal processing part etc. The mechanism device is divided into spindle mechanism for rotating disk, coarse mechanism for gross movement and fine mechanism for precision movement. Also, fine mechanism is divided into fine tracking and fine focusing mechanism.

The tracking actuator moves the radial direction of track. It consists of 2-stage actuator as a fine actuator and a coarse actuator, in order to have wide moving range over 33mm and high resolution under ±0.1μm. A coarse actuator uses gears or ball screw for a gross movement. It has low resolution by backlash etc., but moving range is very wide relatively. A fine actuator uses wire spring and voice coil motor. It can obtain precision motion, but spring gives the limit of moving range.

There were a few of studies on combining a coarse actuator with a fine tracking actuator. But it was difficult to decrease the mass of pickup because of the complexity of optical system construction. So, the large friction force by and the inertia force by the mass of pickup make them difficult. One of conventional studies had used the separated optical system in order to decrease the mass of pickup. But, since it needed a lot of devices, optical system had become very complex. In these days, the pickup has small size and mass because simple optical system is possible.



(a) The assembled mechanical system



(b) The disassembled mechanical system

Fig. 1 Designed tracking system

In general, it is difficult to obtain the moving range over 33mm and the resolution under 0.1μm with one sensor. But, multi-sensing method makes above specifications possible. Coarse motion is detected by counting the number of track that pickup has passed over and fine motion is sensed by the deviation of beam from track. When a coarse actuator and a fine tracking actuator are combined, various problems appear such as large driving force, stability and friction problem. In this paper, these problems are considered in actuator design. After design, tracking actuator is manufactured and controlled with high resolution under 0.1μm.

2. System design

The actuator is driven by VCM(Voice Coil Motor) with high resolution and large moving range. Electromagnetic actuator has dual path since the leakage of flux is rare and it has good linearity as shown in Fig.1[1].

When system is designed, what was considered are following.

First, the permeance method had been used in order to design and analyze VCM for large driving force. To increase driving force, many parameters had been considered as variables.

Second, when the leaf spring had been designed, the sensitivity of focusing and the stability of tracking was considered together. The focusing actuator was loaded to track bobbin. The driving force is applied to track bobbin and position sensing is carried out in pickup. Since the applied point of driving force and the focus beam position that is to be controlled are different, the stability problem is occurred. So, for stable control, the loading direction of focusing actuator is very important. If spring direction is consistent with the radial direction, since the stiffness of tracking direction of leaf spring becomes small in the state of maximum focusing actuation, the stability problem is occurred, too. In order to remove this stability problem, pickup had been loaded to track bobbin to be perpendicular with tracking direction as shown in Fig.1. Through the shape change of leaf spring, the stiffness of tracking direction must be maintained to be high in the state of low focusing stiffness.

Third, the friction and the inertia are dominant error causes. The mass increase of tracking actuator by combining a coarse and a fine actuator cause friction and inertia problem. The effect of friction between holes and axis had been examined. Also, in order to decrease friction force, the plastic coating substance had been used.

2.1 VCM Design

The permeance method was used to design VCM. The designed mechanism had been verified by electromagnetic analysis program and by gauss meter, the measurement device of magnetic flux density.

Permeance method is analogous to electric circuit analysis. This method is possible to approach analytically in analyzing magnetic circuit. The laws that are analogous to Ohm's law is used in permeance method. This similar law can be used as the current and voltage law applied to electric circuit analysis. Eq.(1) is the basic law in permeance method.

$$\Im = \Re\phi \tag{1}$$

where $\Im$ is magnetomotive force, $\Re$ is reluctance and ϕ is magnetic flux.

Magnetomotive force is corresponded to voltage, reluctance to resistance and magnetic flux to current.

The reciprocal of $\Re$ is permeance, $\wp = \mu A / l$. μ is

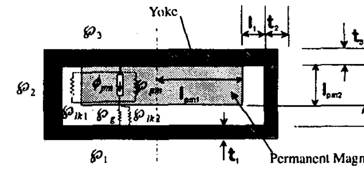


Fig. 2 The equivalent magnetic field circuit of designed tracking system

permeability, A is cross section area and l is the path length. Permeance is determined to the form of magnetic mechanism by Empirical formula[2].

Fig.2 shows the equivalent circuit of magnetic mechanism. Each permeance by empirical formula is following.

$$\wp_1 = \frac{\mu_0 \mu_R t_1 h}{l_1 + \frac{t_3}{2} + \frac{l_{pm1}}{2}} \tag{2}$$

$$\wp_2 = \frac{\mu_0 \mu_R t_2 h}{l_2 + l_{pm2} + \frac{t_1}{2} + \frac{t_2}{2}} \tag{3}$$

$$\wp_3 = \frac{\mu_0 \mu_R t_3 h}{l_1 + \frac{t_3}{2} + \frac{l_{pm1}}{2}} \tag{4}$$

$$\wp_{lk1} = \frac{\mu_0 (2 l_{pm1} + h)}{\pi} \tag{5}$$

$$\wp_{lk2} = 0.52 \mu_0 l_{pm1} \tag{6}$$

$$\wp_g = \frac{\mu_0 l_{pm1} h}{l_2} \tag{7}$$

$$\wp_{pm} = \frac{\phi_0}{|\Im_c|} = \frac{B_r h l_{pm1}}{|H_c l_{pm2}|} \tag{8}$$

where t_1, t_2, t_3 is each thickness of yoke, l_1, l_2 is each gap between permanent magnet and yoke, l_{pm1} is the length of magnet, l_{pm2} is the thickness of magnet and h is the height of magnet and yoke.

The dimensions of magnet and yoke were determined by considering the overall size of actuator. The magnetic flux density in the gap was obtained by using magnetic circuit analysis.

Each dimension had been designed to increase magnetic flux density passed to $\wp_g$.

Eq.(9) is the magnetic flux in the gap.

$$\phi_g = \frac{\wp_g \wp_{eq}}{\wp(\wp_{pm} + \wp + \wp_{lk1}) - \wp^2} \phi_{pm} \tag{9}$$

where $\wp = \wp_g + \wp_{lk2}$, $\wp_{eq}^{-1} = \wp_1^{-1} + \wp_2^{-1} + \wp_3^{-1}$, ϕ_g is the flux in the gap and ϕ_{pm} is the flux of permanent magnet.

Also, the magnetic flux density is obtained from eq.(9).

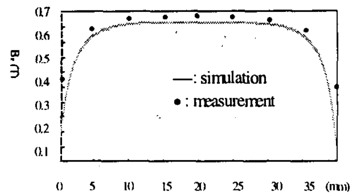


Fig. 3 The simulation and measurement results of magnetic flux density

The magnetic flux density is determined to various parameters in Fig.2. The dominant parameters, which affect to magnetic flux density value, are the thickness of magnet(l_{pm2}), the height of magnet(h) and the gap between magnet and yoke(l_2). As the thickness of and the height of magnet increase, magnetic flux density increases. Also, as the gap between magnet and yoke decreases, magnetic flux density increases. But effects of other parameters to magnetic flux density are negligible. Just, the yoke must be thick not to be saturated. Three variables l_{pm2}, h, l_2 were assigned to 5mm, 8mm, 3mm for considering overall actuator size. In this case, magnetic flux density in the gap, B_g became 0.4587T. If l_{pm2}, h, l_2 were assigned to each 8mm, 10mm, 2mm, B_g would be 0.6274T. B_g increased about 37% by only small change of dimensions of yoke and magnet.

To verify results from permeance method analysis, the electromagnetic analysis program and the gauss meter instrument had been used. Fig.3 shows the simulation and measurement results of magnetic flux density in the gap.

Fig.3 shows that the average of magnetic flux density in the gap is about 0.65T. It is a little higher than the value obtained by permeance method since 3-dimensional flux leakage is not considered in electromagnetic analysis program.

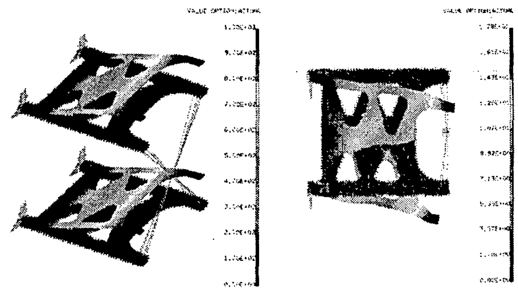
2.2 Focusing Spring Design

The stiffness of focusing must be low even though the stiffness of tracking direction is very high. So, Spring must be designed that the first resonance of focusing is under a few tens Hz. On the other hand, the first resonance of tracking direction must be over about 2kHz for the stability of tracking actuation.

In case of using general leaf spring, the ratio of two stiffness and two resonant frequencies are as following eq.(10) with negligible Poisson's ratio.

$$\frac{K_f}{K_t}=\frac{t_p^2}{b_p^2} \qquad \frac{\omega_f}{\omega_t}=\frac{t_p}{b_p} \tag{10}$$

where b_p , t_p are each the width of and the thickness of leaf spring.



(a) ω_f = 121 H z (b) ω_t = 5050 Hz

Fig. 4 The simulation result by IDEAS

The resonance of tracking direction reduces the phase margin and the gain margin when the actuator is controlled. So, the resonant frequency of tracking direction must be high. Although the damping substance is used to increase gain margin and to obstruct sudden phase drop by reducing the amount of resonance, the effect of damper is limited. It is reasonable to increase the first resonant frequency of tracking direction through the change of structure.

It is difficult to obtain over the value of a few hundreds as eq.(10) if general leaf springs are used. It is hard to acquire the suitable characteristics by the only change of the spring thickness and width. So, the shape of leaf spring is changed to obtain the desirable characteristics in this paper.

Dynamic analysis program is used in order to obtain the dynamic characteristics of leaf spring. Through this analysis, the resonant frequency of focusing direction and of tracking direction are obtained each ω_f = 481.8 Hz, ω_t = 1561.7 Hz for the rectangular leaf spring. This result is anticipated in eq.(10). For improving characteristics, the H-shaped spring is applied. Through the same analysis, each first resonant frequencies are ω_f = 385.5 Hz, ω_t = 3381.3 H. This is the improved result notably.

In H-shaped leaf spring, for focusing mode, the large deformed part was weakened by reducing of spring area in order to improve focusing characteristic. On the other hand, for tracking mode, the large deformed part was reinforced by extending spring area. Fig.4 shows the simulation result of adopted leaf spring by trial and error.

The resonance of focusing direction becomes a little high if tracking direction is stiffer, but this problem can be solved through increasing of driving force of focusing actuator. The worthless dynamic mode is removed by using damper.

2.3 Friction Characteristics

Since the tracking actuator is the sliding type, friction is the one of the dominant errors. To reduce the friction force between axis and hole, plastic-coating substance was used. The effect of plastic-coating substance had been verified by Johnson's method for friction measurement. The overall mass of actuator was a little heavy, about 10g [3].

Craig Johnson proposed the friction estimation method through the measurement of control error. If system is linear, when feedback control and feed-forward control are applied suitably, the nonlinear terms of system appear in the form of feedback force error. If friction is the dominant error of actuator, this nonlinear term can be measured from feedback force error. The relation between velocity and feedback force error is the friction characteristic of actuator[4].

Proposed actuator can be assumed a second-order linear system under 1kHz. So, the actuator can be applied to Johnson's method. It is assumed that the system has only friction as nonlinear characteristics. To obtain friction characteristic, after PD controller and feed-forward controller were designed and manufactured in the base of Johnson's proposition, friction experiment had been achieved. In this experiment, the feedback force error was obtained for sinusoidal input, 1Hz.

Fig.5 shows the relation between the error and the velocity of actuator. Friction is independent of the amount of velocity and just dependent on direction as in Fig.5 Friction force can be obtained from the relation between force and voltage. Also, Fig.5 shows that feedback force error has 180° phase difference with velocity. This shows that the error by inertia is negligible. In this experiment, feed-forward controller compensates for the error by inertia. But, there is no inertial effect for the low frequency input, 1Hz. Although the feed-forward controller is removed, friction force can be obtained accurately.

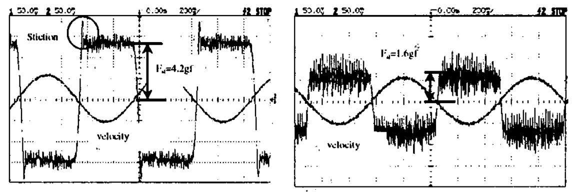
Fig.5(a) shows that there exists stiction and that dynamic friction force is large in the case of no coating. Fig.5(b) shows that dynamic friction coefficient is under 0.16 in the case of using plastic-coating substance. Plastic-coating substance is being used widely in order to reduce friction force in micro actuating system. It has the excellent durability as well as the insensitiveness to temperature variation.

3. Control and Experiment

3.1 Dynamic characteristic experiment

Eq.(11) is obtained through Laplace Transform of the motion equation of focusing actuator

$$\frac{X_f(s)}{I(s)}=\frac{Bl}{m_p s^2+cs+k_f} \qquad (11)$$



(a) no coating (b)plastic coating(DM-804)

Fig. 5 The identification of friction

where m_p is the mass of pickup, c is damping coefficient, k_f is spring constant, x_f is the displacement of pickup focus, I is input current, B is magnetic flux density and l is the effective length of coil.

Dynamic analyzer was used in order to obtain the transfer function of focusing actuator. Fig.6 shows the result of experiment for the characteristic response of focusing actuator. It is open loop transfer function of focusing actuator. This result can be used for design of focusing controller.

By the effect by the inductance of coil and the back emf, there exists the phase drop at cutoff frequency between 1kHz and 2kHz. For blocking of sudden phase drop, current back driver was used.

Fig.6 shows that there is no phase drop with the current back driver.

Similarly, eq.(12) are obtained through Laplace Transform of the motion equation of tracking actuator.

$$\frac{X(s)}{I(s)}=\frac{Bl(cs+k_t)\big/(m_p+m_b)}{\{\frac{m_p m_b}{m_p+m_b}s^4+(cs+k_t)s^2\}}-\frac{(cs+k_t)\big/(m_p+m_b)}{\{\frac{m_p m_b}{m_p+m_b}s^4+(cs+k_t)s^2\}}\frac{F_c(s)}{I(s)} \qquad (12)$$

where m_b is the mass of track bobbin, k_t is the spring constant of trcaking direction, x is the displacement of pickup and F_c is the friction.

In this paper, the displacement of pickup, x must be controlled. But, the diving point of force is different with x. Since the pickup and the track bobbin are connected by leaf spring, the tracking directional stiffness of leaf spring, k_t must be very high. If k_t is infinite, transfer function will become inertia system.

If no damping material, there would be needless resonance such as first spring mode, the first tracking resonance and the coil resonance etc. If above unnecessary resonant modes are occurred at low frequency, there exists limit of increasing the gain of controller because of small phase and gain margin.

Fig.7 shows the result of the response experiment from 5Hz to 30kHz with damping material. It is open loop

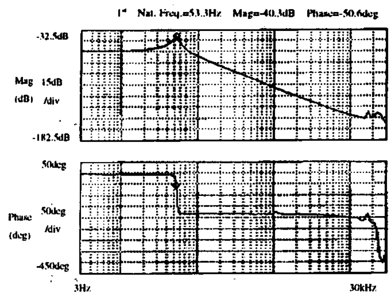


Fig. 6 The transfer function of focus actuator with current back driver

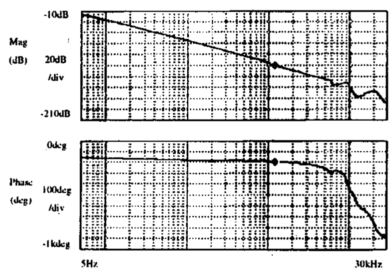


Fig. 7 The transfer function of tracking actuator with damping material

transfer function and used for design of tracking controller. Unnecessary modes disappear with damping material but the resonance by coil at 6kHz does not disappear. This mode can be removed by using UV hardening substance.

These designs need to obtain phase margin required for the stable control of actuator. The actuator requires over 45 degree as gain margin. Since actuator is designed that 2nd resonance was occurred over 2kHz as above result, the required phase margin could be obtained. Also, the required gain margin could be obtained easily because of high sensor gain of photodiode. The bandwidth of actuator was over 100Hz, too.

3.2 System control

Fig.8 shows the block diagram of system. Lead compensation or PD control is used as controller.

There are a lot of researches on removing friction, but they use the absolute velocity information of actuator for removing friction effect. The absolute velocity of pickup cannot be detected in optical disk since the just position error is obtained from the beam deviation from track[5].

Eq.(13) is the transfer function of tracking actuator..

$$G(s)=\frac{\dfrac{cs+k_t}{(m_p+m_b)}}{\dfrac{m_p m_b}{m_p+m_b}s^2+cs+k_t} \tag{13}$$

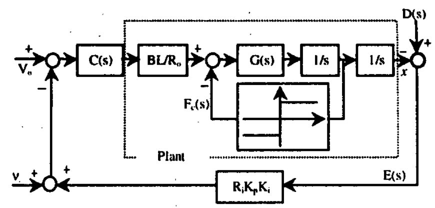


Fig. 8 The block diagram of tracking system

where D(s) is the eccentricity, E(s) is the beam deviation from track, C(s) is the controller, R_o is the resistance of amplifier, V_o is input voltage, R_I is the resistance of conversion, K_p is the sensitivity of photo detection(A/W), K_I is the conversion constant(W/mm) and $F_c(s)$ is friction force.

Eq.(14) is the arrangment of above block diagram with D(s) as input.

$$\begin{aligned} X(s) &= \frac{K_s C(s)G'(s)}{1+K_s C(s)G'(s)}D(s) \\ &-\frac{G'(s)}{1+K_s C(s)G'(s)}F_c'(s) \\ &+\frac{C(s)G'(s)}{1+K_s C(s)G'(s)}v \end{aligned} \tag{14}$$

where are $K_s = R_i K_p K_i$, $G'(s)=\frac{Bl/R_o\, G(s)}{s^2}$, $F_c'(s)=\frac{F_c(s)}{Bl/R_o}$.

Eq.(14) shows that if $|K_s C(s)|$ is infinite, beam follows the track very well and the effect of friction and of noise disappear. Since the sensitivity of photodiode is excellent, the good chacteristic of actuator is obtained despite of the low control gain. Since it is to difficult to constrcut optical system, voltage V_o as input is used in the experiment. In case of using V_o, eq.(14) is changed to eq.(15).

$$\begin{aligned} X(s) &= \\ &\frac{C(s)G'(s)}{1+K_s C(s)G'(s)}G'(s)\{V_o(s)+v\} \\ &-\frac{G'(s)}{1+K_s C(s)G'(s)}F_c'(s) \end{aligned} \tag{15}$$

Eq.(15) shows that the noise affects to the resolution of system. Also, since the displacement sensor used in this experiment has low gain, the control gain must be high.

3.3 Control Experiment

Fig.9 shows step response of tracking actuator. The settling time is about 4.5msec.

Fig.10 shows the result of actuator characteristic experiment in case of using PD controller.

For 50μm, 20Hz input, output is same to input voltage. The error is about 0.08μm. This is the desirable result because of under 0.1μm, system requirement.

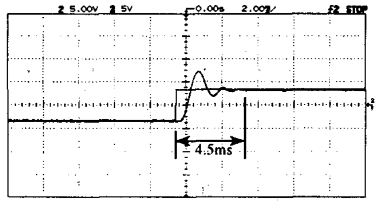


Fig. 9 Step response of tracking actuator

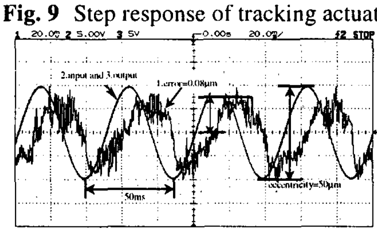


Fig. 10 The error, input and output for 20Hz sinusoidal input, 50μm when sensor gain = 2.5μm /V

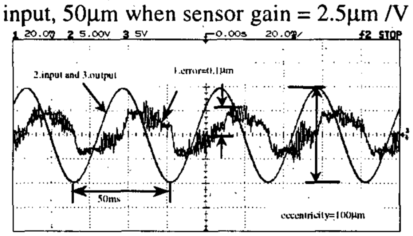


Fig. 11 The error, input and output for 20Hz sinusoidal input, 100μm when sensor gain = 5.0μm/V

Similarly, The results for each 10Hz and 40Hz inputs was about 0.75μm, 0.125μm.

Since the error is caused by friction and inertia force dominantly. The effects of friction and of inertia force are estimated from this result. The error for 10Hz input is not much different with the error for 20Hz inputs.

Although inertia effect is four times, there is no variation of errors. This is because Cloulomb's friction is the dominant error cause in low frequency. But the variation of errors for 20Hz and 40Hz input are more or less large.

The error caused by inertia force is 5% at 10Hz, 17% at 20Hz, 46% at 40Hz of the whole error. Two error is almost same in the neighborhood of 40Hz.

Fig.11 shows the output and error for 100μm, 20Hz input. The sensor gain is 5μm/V. The system resolution is about 0.1μm. Since the actuator has Coulomb's friction, even though 20Hz input amplitude becomes double when compared with 50μm input, the error caused by frictoin does not increases. Since only the effect of inertia force becomes double and the error by inertia is small at 20Hz, the whole error increases a little as shown in Fig.11

Since the sensor gain has limit for the stability, fricion must be reduced to under a certain amount to obtain the desirable resolution of system when designed. If only friction is considered as the error cause, friction force must be reduced to under 2gf.

4. Conclusions

In this paper, one-dimensional tracking actuator with the performance of 2-stage tracking has been designed and controlled in optical disk. Three design categories are needed as following.

First, driving force must be large to overcome friction and inertia force. VCM was designed by permeance method. The result had been verified by electromagnetic analysis program and by experiment. Second, when leaf spring for focusing was designed, the stability of and the sensitivity of tracking were considered. Problems about dynamic motion are proposed and solved by using dynamic analysis program. For control, open loop transfer functions were obtained and controller was designed based on the results. Third, friction that affects to system resolution was analyzed and removed by plastic-coating substance. The effects of friction and of inertia were verified by experiment. Friction is the more dominant cause of error than inertia force.

The characteristic responses of actuator were obtained by experiment to verify the system design.

PD control is applied to system and the desirable resolution, under ±0.1μm is obtained.

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