Compensation of mechanical resonances by adaptive
feed-forward cancellation for head positioning control system
in hard disk drives

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Abstract
To increase recording density, a head positioning control system of a hard disk drive (HDD) has to compensate for vibration due to mechanical resonances of the system. In this study, a novel scheme of an adaptive feedforward cancellation (AFC) is proposed to compensate for the vibration due to the mechanical resonances. Compared to a notch filter, which is commonly used to compensate for the vibration, the proposed AFC has the advantage of estimating the amplitude of the vibration from the output signal. In addition, although the proposed AFC can be implemented as a discrete system, the design parameters: frequency, damping ratio, and gain can be set directly in a digital signal processor. The proposed control system was applied to a dual-stage actuator system, which is the mainstream head positioning control system of the current HDDs. The results indicate that the proposed system could compensate for the vibration due to the mechanical resonance.

Keywords : Hard disk drive, Positioning control, Vibration compensation, Adaptive feed-forward cancellation

1. Introduction

With the advent of the advanced information society, internet companies have begun constructing large data servers for storage of the large amount of digital data (Digital Data Storage Outlook. 2017). The main components of the data server are hard disk drives (HDDs) that can store a large amount of digital data (Yamato Y. 2016). The recording capacity of the HDDs should be further increased for expansion of the digital data stores. To increase the recording capacity, it is necessary to improve the accuracy of the positioning control system of the magnetic head to read/write data on the disk (Yamaguchi T et al. 2012). Diameter of the disk size is standardized as 2.5 or 3.5 inch, therefore the recording density on the disk should be increased. The recording density of the HDD is decided by track misregistration (TMR) (Abrahamson, S. and Huang, F. 2015; Hong F. and Du C. 2010). The TMR represents the misalignments between the magnetic head and the target positions on the disk. If the pitch of the data track on the HDD is smaller than the scale of the TMR, then the magnetic head might record on the wrong data track. The pitch of the data track should thus be larger than the scale of the TMR. For high recording density, the pitch has to be reduced, that is, the TMR has to be decreased. The head positioning system must sufficiently compensate for vibration, which can increase the scale of the TMR. Hence, the required head positioning accuracies is nanoscale (Kemao P et al. 2004; Mohammadeza C. et al. 2014; Atsumi T. 2017).

In previous studies, various positioning control methods were developed to improve the head positioning accuracy of HDDs (Atsumi T. and Messner W. C. 2012; Bagherieh O. and Horowitz R. 2018; Hirata M. et al. 2003; Hu X et al. 1991; Hui L et al. 2009; Ito J. and Atsumi T. 2018). One of these control methods is the adaptive feedforward cancellation (AFC) scheme (Sacks A et al. 1993; Bodson M et al. 1994). The AFC is a disturbance compensation method based on adaptive algorithm. It is mainly used to compensate for vibrations synchronized with the rotation of the spindle motor.
(Zhang J et al. 2000; Yabui S et al. 2012; Yabui S et al. 2013(a); Yabui S et al. 2013(b)). The AFC estimates the amplitude of the vibration and generates a control input to compensate for the position error signal. The position error signal is a relative deviation between the magnetic head and target position; therefore, the vibration itself is not compensated for by the conventional AFC. The magnetic heads of HDDs comprise various mechanical parts, and any external forces can cause vibrations due to mechanical resonance. The vibration worsens the head positioning accuracy, and the head positioning control system has to compensate for the vibration.

In this study, a control system using AFC that can compensate for the vibration due to the mechanical resonance is proposed. The proposed control system has advantages with comparison to a notch filter which is generally used to compensate for the vibration (Yamaguchi T et al. 2012). The current control system is usually implemented on a digital signal processor (DSP), and it is necessary to discretize the controller using z-conversion (Alonso, Concheir A. 1983). In the parameters of discretize the controller, the physical parameters such as frequency and damping ratio, cannot be directly set to the DSP. On the other hand, the parameters in the proposed AFC can be set directly in the DSP. Moreover, the advantage of the proposed control system is able to estimate the amplitude of the vibration caused by mechanical resonance. The proposed system can compensate for the vibration and estimate its amplitude simultaneously.

Real-time simulations were conducted to verify the effectiveness of the proposed AFC. The current head positioning control systems of the HDD employs dual-stage actuators comprising a voice coil motor (VCMs) and a piezo actuator. The proposed AFC is designed to compensate for the mechanical resonances of the VCM and the piezo actuator. From the results, the proposed system can compensate for the vibration due to the mechanical resonances in both actuators. In addition, the amplitude of the vibration can be estimated from the AFC output.

2. The proposed control system employment of the AFC for mechanical resonance compensation

The head positioning control system with the AFC is described in this section. First, the block diagram of the control system with the conventional AFC (Sacks A et al. 1993; Bodson M et al. 1994) is shown in Fig.1.

![Fig. 1 Block diagram of control system with conventional AFC](image)

where $C(z)$ is the main feedback controller, $P_0(s)$ is the plant, $H$ is the zero-order hold, and $K$ is the sensor. The symbol $k$ is the sample number, $z^{-1}$ represents 1 sample delay operator, and $t$ is the time. The symbol $d(t)$ is the disturbance, $e(k)$ is the position error signal, which is the difference between the target position $r(k)$ and current position $y(t)$ sampled by $K$. The conventional AFC is arranged in parallel with the main controller, as shown in Fig.1. The AFC compensates for $d(t)$ by updating parameters in real-time using an adaptive algorithm. The adaptive algorithm is expressed by the following equation.

$$u_{afc}(k) = p(k) \cos(\omega_{afc}Tk) + q(k) \sin(\omega_{afc}Tk)$$

$$p(k) = p(k - 1) + \lambda e(k) \cos(\omega_{afc}Tk + \theta)$$

$$q(k) = q(k - 1) + \lambda e(k) \sin(\omega_{afc}Tk + \theta)$$

where $\omega_{afc}$ is the angular frequency of the AFC output $u_{afc}(k)$, $\lambda$ is the step size parameter, $\theta$ is the phase and $T$ is the sampling time. The AFC generates $u(k)$ calculated on the basis of the learning parameters $p(k)$ and $q(k)$. The learning parameters $p(k)$ and $q(k)$ are updated to compensate for the position error signal caused by the disturbance $d(k)$ at the angular frequency $\omega_{afc}$ (Sacks A et al. 1993). In this control system, although the AFC cancels the relative displacement, the AFC doesn’t compensate for vibration due to the mechanical resonances.
The authors proposed a novel control scheme of the AFC which can compensate for the vibration due to the mechanical resonances. A block diagram of the proposed control system is shown in Fig.2. The difference between the proposed and conventional control system is that the AFC is arranged in parallel with the plant \( P(q) \) in the proposed system. Here, the plant is considered as a discrete time transfer function from the head position \( y(k) \) to the AFC control output \( u(k) \) to the head position \( q(k) \) sampled by \( K \). The plant of the head positioning control system can be rewritten as the discrete control system shown in Fig.3. The adaptive algorithm of the AFC employed in the proposed system is expressed by the following equation.

\[
    u_{a fc}(k) = p(k) \cos(\omega_{a fc}Tk) + q(k) \sin(\omega_{a fc}Tk) \tag{4}
\]

\[
    p(k) = \sum_{a=1}^{k} e^{-\zeta_{a fc}a \omega_{a fc}T} \lambda \ u(a) \ \cos(\omega_{a fc}Ta + \theta) \tag{5}
\]

\[
    q(k) = \sum_{a=1}^{k} e^{-\zeta_{a fc}a \omega_{a fc}T} \lambda \ u(a) \ \sin(\omega_{a fc}Ta + \theta) \tag{6}
\]

where \( \epsilon = e^{-\zeta_{a fc}a \omega_{a fc}T} \) is the forgetting factor. The purpose of the proposed control system is to prevent feedback of the vibration included of the actual magnetic head position \( y(k) \) to the control system. The learning parameters \( p(k) \) and \( q(k) \) are updated to compensate the vibration due to mechanical resonances \( P(e^{j\omega_{a fc}T})d(k) \) at \( \omega_{a fc} \). That is, although \( y(k) = P(e^{j\omega_{a fc}T})C(e^{j\omega_{a fc}T})e(k) + P(e^{j\omega_{a fc}T})d(k), \) \( y_{a fc}(k) = y_{a fc}(k) = P(e^{j\omega_{a fc}T})C(e^{j\omega_{a fc}T})e(k) \).

The adaptive algorithm can be transformed into the discrete time transfer function from \( u(k) \) to \( u_{a fc}(k) \). The transfer function is expressed by the following equation.

\[
    F(z) = \lambda \frac{z^2 \cos(\theta) - ze^{-\zeta_{a fc}a \omega_{a fc}T} \cos(\omega_{a fc}T + \theta)}{z^2 - ze^{-\zeta_{a fc}a \omega_{a fc}T} \cos(\omega_{a fc}T) + e^{-2\zeta_{a fc}a \omega_{a fc}T}} \tag{7}
\]

Moreover, eq.(7) can be transformed to the continuous time transfer function by the following equation.

\[
    F(s) = \lambda \frac{\cos(\theta) s + \omega_{a fc} \sin(\theta) + \omega_{a fc} \zeta_{a fc} \cos(\theta)}{s^2 + 2\zeta_{a fc} \omega_{a fc} s + \omega_{a fc}^2} \tag{8}
\]

where \( s \) is the Laplace operator. From eq.(8), the design parameters are considered as follows: \( \lambda \) is the gain, \( \zeta_{a fc} \) is the damping ratio, \( \omega_{a fc} \) is the natural angular frequency, and \( \theta \) is equal to zero in the transfer function. Although the AFC can be implemented as a digital control system, these parameters can be set directly in the DSP.
### 3. Compensation for mechanical resonances of head positioning systems in HDDs

#### 3.1. Control object

The proposed AFC was applied to compensate for the vibration due to the mechanical resonances in the head positioning control systems of HDDs. The current HDD comprises the VCM, several piezo actuators, several magnetic heads, several disks, and the spindle motor, as shown in Fig. 4. The head positioning control system employs dual-stage actuator consisting of the VCMs and the piezo actuators (HGST. 2017). The diameter of the disk of the mainstream HDD is either 2.5 or 3.5 inch, these are called 2.5-inch HDD or 3.5-inch HDD, respectively. In this paper, the study target is the 3.5-inch HDD.

![Picture of the HDDs](image)

**Fig. 4** Picture of the HDDs

Figure 5 indicates an example image of the head position of the 3.5-inch HDD. A length of the magnetic head is 52mm, a movable angle of magnetic head is 30deg, and magnetic-head radius of rotation is 27.22mm.

![Example image of the head position of 3.5-inch HDD](image)

**Fig. 5** Example image of the head position of 3.5-inch HDD

Figure 6 presents an overview of the proposed head positioning control system. The magnetic heads read the position signals on the disks, and the signals are input to the servo controller written in the DSP. The DSP calculates the control signal for the VCM and piezo actuator. In this paper, the plant is defined by $P_{VCM0}(s)$ and $P_{PZT0}(s)$ in continuous time. The symbol $P_{VCM0}(s)$ is the transfer function from $u_{vcm}(k)$ to $y(t)$, and $P_{PZT0}(s)$ is the transfer function from $u_{pzt}(k)$ to $y(t)$. The transfer function $P_{VCM0}(s)$ and $P_{PZT0}(s)$ are model of the 3.5-inch HDD. These transfer functions are expressed by the following equations:

$$P_{VCM0}(s) = \kappa_{vcm} \sum_{i=1}^{16} \frac{\alpha_{vcm-i}}{s^2 + 2\zeta_{vcm-i}\omega_{vcm-i}s + \omega_{vcm-i}^2}$$

$$P_{PZT0}(s) = \kappa_{pzt} \sum_{i=1}^{9} \frac{\alpha_{pzt-i}}{s^2 + 2\zeta_{pzt-i}\omega_{pzt-i}s + \omega_{pzt-i}^2}$$

In eqs.(9) and (10), $\kappa_{vcm}$ is 647.8 and $\kappa_{pzt}$ is $6.83e^9$; the other parameters are summarized in Tables 1 and 2, based on a previous study (Atsumi. 2018). The frequency responses of $P_{VCM0}(s)$ and $P_{PZT0}(s)$ are shown in Fig.7. The peaks due to
mechanical resonance are above 5 - 6 kHz for both $P_{VCM}(s)$ and $P_{PZT}(s)$. The control system must compensate for the mechanical resonance. In the controller design, the transfer functions $P_{VCM}(s)$ and $P_{PZT}(s)$ are discretized as $P_{VCM}(z)$ and $P_{PZT}(z)$ using bilinear transformation.

![Schematic of an HDD head positioning control system](image)

**Fig. 6** Schematic of an HDD head positioning control system

**Table 1** Parameters of $P_{VCM}(s)$

| $i$ | $\omega_{\text{VCM}-i}$ | $\alpha_{\text{VCM}-i}$ | $\zeta_{\text{VCM}-i}$ |
|-----|--------------------------|--------------------------|--------------------------|
| 1   | 0.50                     | 0.00                     | 0.00                     |
| 2   | 6100 x $2\pi$            | -1.00                    | 0.02                     |
| 3   | 9600 x $2\pi$            | 0.10                     | 0.04                     |
| 4   | 6500 x $2\pi$            | -0.10                    | 0.02                     |
| 5   | 8050 x $2\pi$            | 0.04                     | 0.01                     |
| 6   | 9600 x $2\pi$            | -0.70                    | 0.03                     |
| 7   | 14800 x $2\pi$           | -0.20                    | 0.01                     |
| 8   | 17400 x $2\pi$           | -1.00                    | 0.02                     |
| 9   | 21000 x $2\pi$           | 3.00                     | 0.02                     |
| 10  | 26000 x $2\pi$           | -3.20                    | 0.012                    |
| 11  | 26000 x $2\pi$           | 2.10                     | 0.007                    |
| 12  | 29000 x $2\pi$           | -1.50                    | 0.01                     |
| 13  | 32000 x $2\pi$           | 2.00                     | 0.03                     |
| 14  | 38000 x $2\pi$           | -0.20                    | 0.01                     |
| 15  | 43000 x $2\pi$           | 0.30                     | 0.01                     |
| 16  | 44800 x $2\pi$           | 0.50                     | 0.01                     |

**Table 2** Parameters of $P_{PZT}(s)$

| $i$ | $\omega_{\text{PZT}-i}$ | $\alpha_{\text{PZT}-i}$ | $\zeta_{\text{PZT}-i}$ |
|-----|--------------------------|--------------------------|--------------------------|
| 1   | 6560 x $2\pi$            | 0.007                    | 0.02                     |
| 2   | 8050 x $2\pi$            | 0.012                    | 0.02                     |
| 3   | 9200 x $2\pi$            | 0.015                    | 0.025                    |
| 4   | 12900 x $2\pi$           | 0.01                     | 0.007                    |
| 5   | 15500 x $2\pi$           | 0.10                     | 0.01                     |
| 6   | 17200 x $2\pi$           | 0.20                     | 0.01                     |
| 7   | 18050 x $2\pi$           | 0.40                     | 0.009                    |
| 8   | 27100 x $2\pi$           | 1.50                     | 0.007                    |
| 9   | 39500 x $2\pi$           | 1.30                     | 0.04                     |

![Frequency responses of plant](image)

**Fig. 7** Frequency responses of plant
3.2. The proposed control system in the head positioning control system

A block diagram of the basic control system of the head positioning system is shown in Fig. 8. The decoupled controller design, which is a representative control system, is employed in the control system of the current HDD (Kobayashi M et al. 2004; E. Hong et al. 2013; Huang D et al. 2014 Bashash S. and Shariat S. 2019). The sampling time $T$ for $K$ is $11\mu s$. The symbol $C_{VCM}(z)$ is the main feedback controller for the VCM and mainly controls the rigid body mode of $P_{VCM}(z)$. The symbol $C_{PZT}(z)$ is the main feedback controller for the piezo actuator and mainly controls the rigid body mode of $P_{PZT}(z)$. The symbol $D(z)$ is a decoupled filter that eliminates the mutual interference between the VCM and piezo actuator. The controllers $C_{VCM}(z)$ and $C_{PZT}(z)$ are designed according to the previous study (Atsumi. 2018). The frequency responses of $C_{VCM}(z)$, $D(z)$, and $C_{PZT}(z)$ are shown in Fig. 9. The symbol $d_{vcm}(k)$ and $d_{pzt}(k)$ are the external forces in each corresponding control loop. The proposed control system in the head positioning control system is shown in Fig. 10. The nine proposed AFCs were implemented to compensate for the mechanical resonance in each actuator. The design parameters of the proposed AFCs: $\omega_{afc_{i}}$, damping ratio $\zeta_{afc_{i}}$, and gain $\lambda_{i}$ are shown in Table 3. The control system was discretized with sampling time $T$.

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**Fig. 8** Block diagram of basic control system of head positioning control system of HDDs

**Fig. 9** Frequency responses of controller
3.3. Real-time simulation

To verify the effectiveness of the proposed control system, real-time simulation was performed. The control system is established in MATLAB and Simulink to achieve real-time simulation. The transfer functions are defined as the state-space model, and the dynamics and adaptive algorithms of the AFCs are calculated for each sampling time. The representative benchmark model of the HDD (Yamaguchi T et al. 2012) is also established using MATLAB and Simulink. In this paper, the simulation model is called as the ”MATLAB/Simulink” model. In the simulation, white noise defined as the HDD benchmark (Yamaguchi T et al. 2012) was input as the disturbance $d_{\text{VCM}}(k)$ and $d_{\text{PZT}}(k)$ to confirm the frequency response of the control system. However, $d_{\text{VCM}}(k)$ and $d_{\text{PZT}}(k)$ do not include the signal whose frequency is higher than the highest frequency of the mechanical resonance $\omega_{\text{VCM}}_{16}$. The time responses of the output signals $y_{\text{VCM}}(k) : w/o AFC$, $y_{\text{VCM-afc}}(k) : w / AFC$ in the VCM control loop, and $y_{\text{PZT}}(k) : w/o AFC$, $y_{\text{PZT-afc}}(k) : w / AFC$ in the piezo actuator control loop are shown in Fig.11. The amplitude spectra of these signals are shown in Fig.12. The spectra were calculated using discrete Fourier transform (DFT), and the amplitudes were derived from absolute values of the series of the DFT using Parseval’s theorem.
Table 3 Parameters of the proposed AFC

| i  | $\omega_{jfc-i}$     | $\Delta_i$ | $\zeta_{jfc-i}$ | $\theta_i$ |
|----|----------------------|------------|----------------|-----------|
| 1  | 5300 x 2\pi          | 0.20      | 0.02           | 60        |
| 2  | 9500 x 2\pi          | 0.03      | 0.02           | -40       |
| 3  | 14800 x 2\pi         | 0.02      | 0.04           | 45        |
| 4  | 17500 x 2\pi         | 0.05      | 0.02           | 90        |
| 5  | 21000 x 2\pi         | 0.15      | 0.02           | -120      |
| 6  | 26200 x 2\pi         | 0.15      | 0.014          | 80        |
| 7  | 26800 x 2\pi         | 0.12      | 0.014          | -100      |
| 8  | 29000 x 2\pi         | 0.10      | 0.04           | 40        |
| 9  | 38200 x 2\pi         | 0.01      | 0.02           | 0         |
| 10 | 6450 x 2\pi          | 0.05      | 0.05           | -130      |
| 11 | 7900 x 2\pi          | 0.04      | 0.05           | -160      |
| 12 | 9200 x 2\pi          | 0.03      | 0.05           | -180      |
| 13 | 12700 x 2\pi         | 0.20      | 0.10           | -160      |
| 14 | 15500 x 2\pi         | 0.15      | 0.008          | -120      |
| 15 | 17200 x 2\pi         | 0.15      | 0.01           | -130      |
| 16 | 18050 x 2\pi         | 0.90      | 0.01           | -105      |
| 17 | 27100 x 2\pi         | 0.60      | 0.007          | -145      |
| 18 | 39500 x 2\pi         | 0.10      | 0.02           | -190      |

Fig. 11 Time responses of the plant output

Fig. 12 Amplitude spectra of the plant output
Table 4  Results of the standard deviation $\sigma$ of the output signals

|                      | for VCM       | for piezo actuator |
|----------------------|--------------|-------------------|
| $\sigma$ of $y_{\text{cm}}(k)$ | 1.38 nm | 1.44 nm |
| $\sigma$ of $y_{\text{cm-afc}}(k)$ | 1.44 nm | 1.30 nm |
| Improvement ratio    | 27%          | 43%               |

In Figs. 11 and 12, the amplitudes of the output signals are compensated by using the proposed AFC. The standard deviation $\sigma$ of the output signals are shown in Table 4. The frequency responses from $u_{\text{cm}}(k)$ to $y_{\text{cm-afc}}(k)$ and from $u_{\text{pzt}}(k)$ to $y_{\text{pzt-afc}}(k)$ are shown in Fig.13(a) and (b), respectively. These frequency responses are considered as the characteristics of the identified plant in each control loop. The frequency responses of the original plant $P_{\text{cm}}(z)$, $P_{\text{pzt}}(z)$, which are calculated using the transfer function, are also indicated in Fig.13(a) and (b). The proposed AFC can compensate for the peak gain of the mechanical resonances, in comparison with the original frequency responses of $P_{\text{cm}}(z)$, $P_{\text{pzt}}(z)$. Value of decreasing rates of the peaks are shown in Table 5.

![Amplitude spectrum and phase response graphs for VCM and piezo actuator](image)

Fig. 13  Frequency responses of the plant for both actuators

Table 5  Value of the decreasing rates of the peaks due to the mechanical frequency at the AFC’s frequency $\omega_{\text{afc-i}}$

| $i$     | $\omega_{\text{afc-i}}$ | Decreasing rates |
|---------|----------------------|-----------------|
|         |                      | for VCM         | for piezo actuator |
| 1       | $5300 \times 2\pi$  | 10.56dB         | 5.32dB          |
| 2       | $9500 \times 2\pi$  | 6.86dB          | 5.27dB          |
| 3       | $14800 \times 2\pi$ | 6.92dB          | 1.82dB          |
| 4       | $17500 \times 2\pi$ | 12.19dB         | 7.58dB          |
| 5       | $21000 \times 2\pi$ | 12.37dB         | 17.37dB         |
| 6       | $26200 \times 2\pi$ | 16.17dB         | 20.64dB         |
| 7       | $26800 \times 2\pi$ | 6.82dB          | 21.03dB         |
| 8       | $29000 \times 2\pi$ | 8.65dB          | 14.09dB         |
| 9       | $38200 \times 2\pi$ | 3.08dB          |                 |
| 10      | $6450 \times 2\pi$  | 5.32dB          |                 |
| 11      | $7900 \times 2\pi$  | 5.27dB          |                 |
| 12      | $9200 \times 2\pi$  | 1.82dB          |                 |
| 13      | $12700 \times 2\pi$ | 7.58dB          |                 |
| 14      | $15500 \times 2\pi$ | 17.94dB         |                 |
| 15      | $17200 \times 2\pi$ | 17.37dB         |                 |
| 16      | $18050 \times 2\pi$ | 20.64dB         |                 |
| 17      | $27100 \times 2\pi$ | 21.03dB         |                 |
| 18      | $39500 \times 2\pi$ | 14.09dB         |                 |

Figure 14 shows the frequency response from $e(k)$ to $y_{\text{cm}}(k)$, which represents the open loop characteristics of the VCM control loop, and from $e(k)$ to $y_{\text{pzt}}(k)$, which represents the open loop characteristics of the piezo actuator control loop. Figure 15 presents the vector loci of these open loop characteristics. The open loop characteristics calculated by the
transfer function are also shown in Figs.14 and 15. From these frequency responses, the proposed AFC can compensate for the mechanical resonances. Particularly, the VCM control loop can be stabilized by using the proposed AFC. The gain and phase margins for each control system are shown in Table 6.

Table 6 The stability performance of each control loop

| System                  | Gain margin w/o AFC | AFC Gain margin | Phase margin w/o AFC | AFC Phase margin |
|-------------------------|---------------------|----------------|----------------------|-----------------|
| VCM loop                | N/A (unstable)      | 7.53dB         | 45.67deg             | 39.60deg        |
| Piezo actuator loop     | 13.98dB             | 12.77dB        | 96.84deg             | 98.84deg        |
| Dual-stage loop         | N/A (unstable)      | 7.74dB         | 43.54deg             | 43.32deg        |

Fig. 14 Frequency responses of the open loop in the control system
In the proposed control system, the frequency responses of the sensitivity function for the VCM control loop, the piezo actuator control loop, and the dual-stage control loop were derived from \( r(k) \) and \( e(k) \), as shown in Fig.16. The maximum gain of the sensitivity function is less than 6dB and there is no remarkable peak. The proposed AFC used can compensate for mechanical resonance and stabilize the control system.

![Fig. 15 Vector locus of the open loop in the control system](image)

![Fig. 16 Frequency responses of sensitivity function in the proposed controller design](image)

### 3.4. Comparison to the notch filter

The notch filter is a representative method that can compensate for the mechanical resonance. In the control system, it is common to connect multiple the notch filters in series between the controller output and plant input, as shown in Fig.17. The symbols \( N_{VCM}(z) \) and \( N_{PZT}(z) \) are the notch filters for the VCM and piezo actuator control loops, respectively. In the frequency responses, the notch filters suppress the gains around the resonance frequencies as shown in Fig.18. The notch filters \( N_{VCM}(z) \) and \( N_{PZT}(z) \) are designed with reference to a previous study (Atsumi, 2018). The notch filters decrease amplitude of the output to prevent excitation of the resonances. The difference of the proposed AFC from the notch filter is that it can be implemented in parallel with the plant, and the proposed AFC generates an output that can cancel the vibration for each resonant mode. The amplitude spectra of the proposed AFC and the notch filter output are shown in Fig.19. The amplitude spectra of \( u_{ntc-vcm}(k) \) and \( u_{ntc-pzt}(k) \) are decreased by the notch filter to prevent excitation of the resonances. On the other hand, the amplitude spectra of \( u_{afc-vcm}(k) \) and \( u_{afc-pzt}(k) \) are generated by the AFC to cancel the vibrations due to the resonances. The differences of the amplitude spectra at the central frequency of the notch filters are shown in Tables 7 and 8.
Table 7  The amplitude spectra of $u_{ntc}$ and $u_{afc}$ at the central frequency of the notch filters

| Central freq of notch | $u_{ntc}$ | $u_{afc}$ |
|-----------------------|----------|----------|
| 1                     | 5500 x 2π | 0.01     | 0.98     |
| 2                     | 8000 x 2π | 0.04     | 0.05     |
| 3                     | 14800 x 2π| 0.02     | 0.09     |
| 4                     | 17500 x 2π| 0.01     | 0.15     |
| 5                     | 21000 x 2π| 0.01     | 0.55     |
| 6                     | 26300 x 2π| 0.00     | 0.33     |
| 7                     | 29000 x 2π| 0.00     | 0.10     |
| 8                     | 32000 x 2π| 0.00     | 0.01     |
| 9                     | 38200 x 2π| 0.00     | 0.01     |

Table 8  The amplitude spectra of $u_{ntc}$ and $u_{afc}$ at the central frequency of the notch filters

| Central freq of notch | $u_{ntc}$ | $u_{afc}$ |
|-----------------------|----------|----------|
| 1                     | 6560 x 2π | 0.02     | 0.14     |
| 2                     | 8050 x 2π | 0.06     | 0.13     |
| 3                     | 12900 x 2π| 0.01     | 0.16     |
| 4                     | 15500 x 2π| 0.00     | 0.25     |
| 5                     | 18050 x 2π| 0.00     | 0.28     |
| 6                     | 27100 x 2π| 0.02     | 1.00     |
| 7                     | 39500 x 2π| 0.00     | 0.07     |

Furthermore, the amplitude of the vibration due to the mechanical resonances can be observed from the AFC output. Figure 20 shows the time response of the proposed AFC output $u_{afc}$ at the central frequency of the notch filters. The time responses are shown with respect to two conditions: the original plant model and the plant model with a 20% gain ($v_{cm}$ and $pzt$). It can be seen that the gain in resonance increases, the AFC output also increases by about 20%. By transmitting the magnitude of this signal to a self-diagnostic function called S.M.A.R.T, which is mounted on the HDD (Lucas P. Q et al. 2016), the amplitude estimation is expected to be widely applicable such as failure detection.
4. Conclusion

In this paper, the authors proposed the novel scheme of the AFC that can compensate for the vibration due to the mechanical resonance in the head positioning control system of the HDD. Although a conventional AFC is used to compensate for the position error signal, the proposed AFC can compensate for the vibration due to the mechanical resonance. The design procedure of the proposed AFC is manageable because the design parameters: frequency, damping ratio, and gain can be set directly in the discrete system. The effectiveness of the proposed control system was confirmed from the real-time simulations based on MATLAB/Simulink. In the case of the dual-stage actuator consisting of the VCM and the piezo actuator, the proposed control AFC could compensate for the vibration due to the mechanical resonance. Moreover, the amplitude of the vibration could be estimated from the output signal of the proposed AFC. It is expected that the estimation results can be used to predict the failure or the aging deterioration.

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