An Optimal Approach to Rotational Vibration Suppression using Disturbance Observer in Disk Drive Systems

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Abstract

This paper investigates the design of disturbance observer for rotational vibration suppression in disk drive systems. The design aims to provide an optimal controller which satisfies both vibration performance and robust stability. It consists of an inversion method, a special filter, and optimization scheme. Firstly a new inversion method is introduced, which provides more accurate inversion compared to conventional zero phase error method. The inversion is to deal with unstable zeros in the plant model. Secondly a special filter for disturbance selection is given, which features adjustable gain and band pass characteristics so that it enables flexible shaping of the loop considering the trade-off between performance and stability margins. And finally the parameters of disturbance observer are optimized in conjunction with external disturbance model. Simulation and experiment on commercial hard disk drives confirm that the design is very effective to such disturbance which is hard to be handled by conventional approach.

Key Words : Vibration, Frequency selection, Disturbance observer, Disk drive, Optimization

1. Introduction

Recently HDD is being used not only for conventional desktop or notebook PC but also for CE products such as DVR (digital video recorder), set-top box, HDTV, and so on. This implies that the external vibration condition, which affects to HDD's operating performance, becomes diversified. For example, in case of NAS (Network attached storage) or RAID (redundant array of independent disks) application, numbers of HDDs are stacked as an array wherein the vibration generated by a drive affects to others through the fixture. In that case, the frequency spectrum of the vibration is widely spread from hundreds to thousands Hz. Meanwhile, in the case of general PC or DVR application, the HDD is excited by optical disk drives or cooling fans and thus the frequency spectrum of the external disturbance is limited within hundreds of Hz. Moreover as the recording density increases, the amount of the equivalent external disturbance becomes much bigger than before. As a result, drive performance under external vibration has become one of the most important factors in HDD performance.

In order to suppress the external vibrations in HDD, several approaches have been proposed such as active feed-forward compensation, anti-vibration mounting, gain switching, loop shaping using disturbance observer, and so on. The active feed-forward scheme compensates disturbance by sensing acceleration applied to HDD and anti-vibration mounting design reduces vibrations transmitted to HDD itself [1][2]. On the other hand, a control scheme, so called DOB(disturbance observer) has been widely used in control engineering fields [1][5][6]. It mainly focuses on slowly varying disturbances and has proven to be very effective to vibration ranges of less than a few hundreds of Hz. However these are inappropriate for our development purposes for the following reasons. One is the cost. In highly competitive market of HDD, the cost is one of the most important factors. Unfortunately the active feed-forward scheme requires additional accelerometers and the anti-vibration mounting requires hardware modification. Another is the target disturbance. Since we are to suppress vibration above hundreds of Hz, it is of concern to secure sufficient stability margin. However the gain switching or conventional DOB severely degrades the stability margin and also it was often neglected.

In this paper, a new optimal design method for DOB

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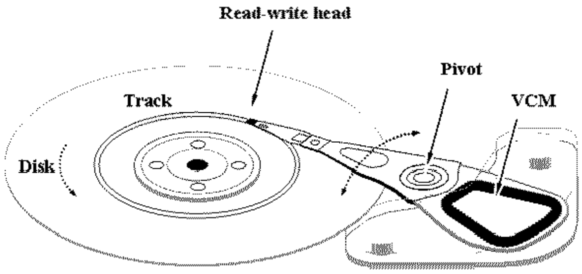

Fig. 1 A typical servomechanism of hard disk drive

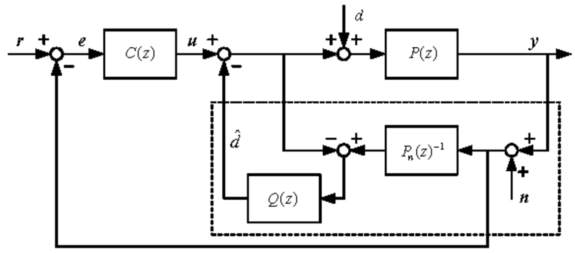

Fig. 2 Block diagram of servo system with add-on disturbance observer

considering both performance and stability margin is proposed to compensate the external vibrations in HDD. First, the plant model inversion is addressed. The biggest challenge in the plant inversion is unstable zeros from non-minimum phase system and non-causality due to physical dynamic constraint. In [1], ZPET (zero phase error tracking) type inversion in discrete time domain has been proposed but has a limited inversion error. We introduce a new method in terms of zero gain error, which provides more correct inversion in the practical implementation point. Secondly, we propose a new filter, often denoted by Q filter, for DOB. The filter features adjustable gain and band pass characteristics which enables flexible shaping of the loop considering the trade-off between performance and stability margins. Third, we present a simple analysis framework to predict the change of loop shape by add-on type DOB. The prediction is approximated by only the Q filter and thus provides useful insight for DOB design. Finally, a controller optimization is addressed. It is based on the PES, which is estimated by position mode equivalent disturbance and sensitivity function. Since it accounts for stability margin and bound constraints for the worst case model, it gives practically optimal controller which guarantees robustness against plant uncertainty.

This paper is organized as follows. Chapter 2 introduces HDD's mechanical structure and the role of servo. Chapter 3 presents the proposed DOB design and compares it with conventional methods. Chapter 4 describes the optimal design method of the DOB with previously designed structure accounting for stability margin. Chapter 5 demonstrates the effectiveness of the proposed design via extensive simulation and experiments using commercial HDD's. Finally, chapter 6 summarizes propositions and prospects further application possibility.

2. System description

Typical structure of HDD is shown in Fig. 1. User data is written on the surface of the magnetic disk. The head magnetizes target position on the disk with encoded information. The read/write head is attached on the edge of the slider in the actuator which rotates with respect to the pivot axis. The actuator is controlled by the current applied to the VCM and the disk rotates with fixed speed by spindle motor. A special pattern, so-called servo pattern, is written on the disk to find out the relative position of head on the disk. Servo pattern includes preamble, gray code, burst and etc and is written in evenly divided sectors along with disk's circumferential direction. In the pattern, gray code gives absolute radial position information and burst gives relative position information in a track. As mentioned, because disk rotates with fixed speed the servo system of HDD inherently becomes discrete time system. The sampling frequency is determined by the disk's rotational frequency and the number of servo sectors.

The main role of the servo control in HDD is divided into two classes, track following and track seeking. The purpose of track following is to maintain the head on a track with minimum deviation, which can be regarded as a regulation problem. The purpose of track seeking is to move the head to the target track with minimum time. In any case, the purpose of the servo is minimizing radial position error at the destination so that reliable read/write operation is guaranteed for all cases. Thus, if there is any external disturbance which makes the PES high, read/write operation cannot be performed in a stable manner as in a normal operation condition. Consequently, high PES by external vibration can results in read/write retry and thus degrades the read/write performance.

In this regard, as a performance index for the drive against external vibration, TP (throughput) under vibration is defined as the ratio of data rate between with and without external vibration. Thus, it indicates the performance degradation under external vibration compared to no vibration condition. Therefore, various specifications for TP are proposed for each operation condition and application. That is, some application requires high TP for on-track read/write operation under the vibration of thousands Hz while the other one requires high TP for random read/write. Generally, the specification is defined for random write and sequential write operation. Assuming the same vibration level, the sequential write operation requires higher performance for it takes at least one disk revolution time at each retry.

In what follows, we investigate a method to improve drive performance against external vibration in light of read/write on random location. Since the most important performance in disk drive operation is a read/write in an adjacent track, we will focus on improving track following performance under vibration.

3. Controller design

In dynamic systems like disk drives which include rotating mechanics of spindle and actuator, mechanical disturbance of low frequency ranges is dominant. A design rule of thumb for this kind of servo system is to increase the loop gain at low frequency regions. Frequently it is denoted by DC gain. For that purpose, the simplest way is to increase the representative gain of controllers. However this inevitably degrades whole stability margin of the servo system and as total loop gain is increased, it easily becomes susceptible to the noise at high frequency ranges.

Meanwhile, we can design a compensator with add-on fashion to the main control loop. It is the well-known scheme of DOB [1]. The key considerations of DOB design are nominal plant modeling with related inversion and a filter at the end of the DOB output, so called Q filter in many literatures. The filter should be designed accounting for non-causality due to the relative degree of nominal plant model and noise reduction.

2.1. Basic Concept

General structure of the DOB of position feedback is shown in Fig. 2. The fundamental concept begins with that if we have the information of the input and the output of a plant and a precise model for the plant, we can estimate the disturbance by the plant inversion [1]. Though its concept is simple, its direct implementation is infeasible due to some physical constraints. That is to say, the actual plant cannot be modeled perfectly and measurement noise cannot be removed completely so that it is impossible to estimate the disturbance applied to the system without any errors. Moreover the nominal model of general dynamic system is usually strictly proper. This makes the inversion of the model non-causal and thus related calculation requires additional filter which have low pass characteristics with relative degrees at least larger than the plant model's one. In Fig. 2 ignoring main servo loop and considering only DOB blocks the output PES is given by

$$
\begin{aligned}
y &= G_{uy}u + G_{dy}d + G_{ny}n \\
&= \frac{PP_n}{P_n + Q(P-P_n)}u + \frac{(1-Q)PP_n}{P_n + Q(P-P_n)}d + \frac{-PQ}{P_n + Q(P-P_n)}n
\end{aligned} \tag{1}
$$

where P_n is nominal plant, P is real plant and Q is a filter for DOB realization.

Since in general measurement noise is concentrated in high frequency regions and mechanical disturbance is in low frequency regions, both measurement noise reduction and low frequency disturbance estimation can be achieved by designing the Q filter in (1) as low pass type. What makes this more viable is the fact that in many cases the plant at low frequency regions can be modeled with high accuracy. Though the measurement noise at high frequency is not the actual problem in the HDD because the main source of the HDD noise is position error signal (PES) demodulation noise [3][4] which is broadband, the model mismatch can be regarded as another noise source. Note that the model mismatch exists mainly in the high frequency region. Hence, in this regard previous researches for DOB have tended to center around proper design of Q filter [1][5][6]. More specifically it is the filter of low pass type and unit magnitude in the pass region.

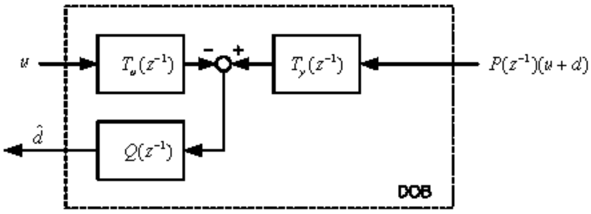

Fig. 3 Simplified block of disturbance observer

However the DOB is usually used as an add-on compensator to the main servo loop which often already has sufficiently high gain at low frequency regions, so that conventional design would yield unnecessary increase of the loop gain in the low frequency regions. The HDD also belongs to this class. Also the redundant compensation inevitably causes severe degradation of stability margin. This has been limited the capability of DOB to merely a solution for low frequency disturbance. To cope with this, we suggest using the Q filter of band pass type. Detailed is described in what follows.

2.2. Controller Structure

This chapter addresses the method for plant inversion in discrete time domain. With rigid body assumption, HDD's plant of current input and PES output can be modeled in continuous time domain by

$$P(s) = \frac{K_a}{S^2}e^{-T_d s} \tag{2}$$

where, K_a is acceleration constant and T_d is calculation time delay. By using zero order hold for DA output, its equivalent in discrete time is given by

$$P(z) = K\frac{(z+b_1)(z+b_2)}{z(z-1)^2} \tag{3}$$

Note, in discrete time model two zeros which never existed in continuous time model are appeared. This is due to sampling effect. What makes this of concern is the fact one of the zeros is located outside of unit circle (unstable zero). In the inversion of plant model, it becomes unstable pole and causes divergence of internal variable. Now consider following generalized plant where z-1 operator is used for notational convenience.

$$P(z^{-1}) = \frac{z^{-\gamma}B^{+}(z^{-1})B^{-}(z^{-1})}{A(z^{-1})} \tag{4}$$

where, $B^{+}(z^{-1})$ is part for stable zero, $B^{-}(z^{-1})$ is part for unstable zero.

In Fig. 2, the part of DOB can be rearranged as in Fig. 3, which is appropriate for plant inversion comparison. With the rearrangement, ZPEI (zero phase error inversion) is given by

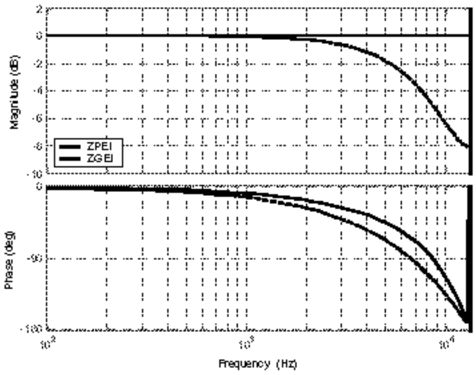

Fig. 4 Bode plot of two inversion methods to deal with unstable zeros.

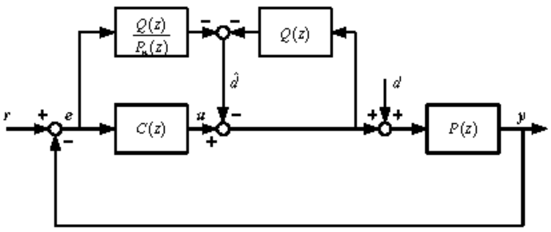

Fig. 5 Equivalent block diagram to analyze disturbance observer

$$T_y(z^{-1})=\frac{A(z^{-1})B_n^-(z)}{z^{-r}B^+(z^{-1})} \tag{5}$$

$$T_u(z^{-1})=B^-(z^{-1})B_n^-(z) \tag{6}$$

where,

$$B^-(z^{-1})=b_0+b_1z^{-1}+\cdots+b_pz^{-p},\ B^-(z)=b_0+b_1z+\cdots+b_pz$$

$$B_n^-(z)=\frac{B^-(z)}{[B^-(1)]^2}$$

In each block of $T_y(z^{-1})$ and $T_u(z^{-1})$ whose relative degree is $-(r+p)$ and $-p$, respectively. p denotes the number of unstable zeros. Thus the required relative degree on Q filter is more than the negative degree of $T_y(z^{-1})$. In this case, estimated disturbance becomes

$$\hat{d}=Q(z^{-1})B^-(z^{-1})B_n^-(z)d \tag{7}$$

In [1], r step delay was reflected to $T_u(z^{-1})$ side and relaxed order constraints of Q filter.

$$T_y(z^{-1})=\frac{A(z^{-1})B_n^-(z)}{B^+(z^{-1})} \tag{8}$$

$$T_u(z^{-1})=z^{-r}B^-(z^{-1})B_n^-(z) \tag{9}$$

$$\hat{d}=z^{-d}Q(z^{-1})B^-(z^{-1})B_n^-(z)d \tag{10}$$

Looking over (8) and (9), now the required order of Q filter is more than p. However in the final result of (10), the r step delay is included to the estimated disturbance. This means that the phase error due to this delay still cannot be cancelled.

As an alternative, we propose a new inversion of ZGEI (zero gain error inversion). The structure for ZGEI is given as follows.

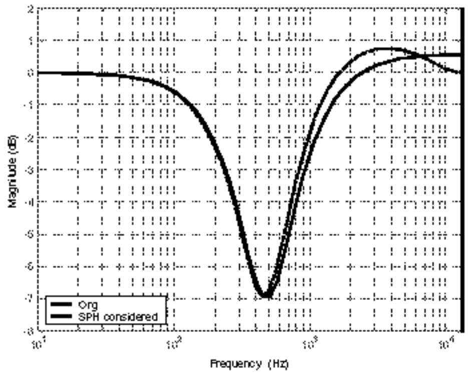

Fig. 6. Bode magnitude plot of (1-Q)

$$T_y(z^{-1})=\frac{A(z^{-1})}{z^{-r}B^+(z^{-1})B_m^-(z^{-1})} \tag{11}$$

$$T_u(z^{-1})=\frac{B^-(z^{-1})}{B_m^-(z^{-1})} \tag{12}$$

where,

$$B^-(z^{-1})=b_0+b_1z^{-1}+\cdots+b_pz^{-p}$$

$$B_m^-(z^{-1})=b_0z^{-p}+\cdots+b_{p-1}z^{-1}+b_p$$

The relative degree of $T_y(z^{-1})$ and $T_u(z^{-1})$ is $-r$ and 0, respectively. Thus the required relative degree of Q filter is more than r step delay. Compared to ZPEI in case of r step delay is less than the number of unstable zero, this enables more flexibility on the design of Q filter. The estimated disturbance is as follows.

$$\hat{d}=Q(z^{-1})\frac{B^-(z^{-1})}{B_m^-(z^{-1})}d \tag{13}$$

(Proof is given in the appendix)

To compare in depth, now consider following example. For comparison convenience, Q is assumed to be 1 and design target is the nominal model of the form (2) as shown in Fig. 10.

$B^-(z)=z+b_u$, $Q(z)=1$, $b=4.35$, $r=1$, $T_s=37.9\mu$sec

For the plant model, ZPEI and ZGEI are as follows whose FRFs are shown in Fig. 4.

$$\hat{d}_{ZPEI}=z^{-r}\frac{(z+b_u)(z^{-1}+b_u)}{(1+b_u)^2}d$$

$$\hat{d}_{ZGEI}=\frac{z+b_u}{b_uz+1}d$$

As shown it the figure, ZPEI produces non-ideal phase drop due to r step delay, which yields larger phase distortion than ZGEI. Unlike this, ZGEI gives no gain error as desired and moreover its phase distortion is less than that of ZPEI. Hence we used ZGEI for the plant inversion.

The DOB of Fig. 2 can be converted into equivalent form of Fig. 5 which is more appropriate form for real implementation because the part of plant inversion and Q filter is added to the main loop with feedback nature. Here the transfer function of equivalent controller is

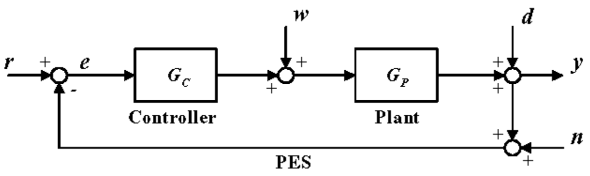

Fig. 7 General block diagram of HDD servo system

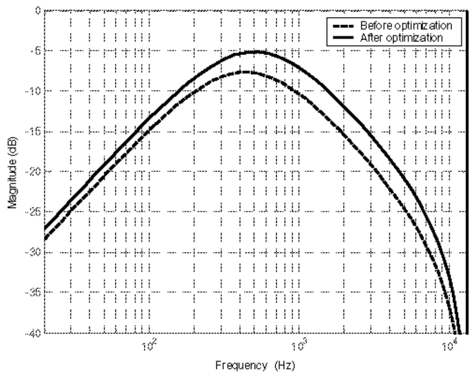

Fig. 11 Bode magnitude plot of filter Q

given by

$$C_{DOB}(z) = \frac{P_n(z)C(z) + Q(z)}{(1 - Q(z))P_n(z)} \tag{14}$$

If we design the Q filter as general low pass type, it is $Q(j\omega) = 1 + 0i$ at 0 Hz. This means infinity gain at zero frequency in (14) and hence it increases the servo system's disturbance rejection capability especially for low frequency regions. However, in view of the shaping of ESF (error sensitivity function) it is inevitable to compromise the rejection capability at high frequency by enhancing that of low frequency. It is so called "water bed effect" indicated by Bode theorem. Thus as shown in Fig. 9 (main loop without DOB), if existing controller is already compensate sufficiently at low frequency regions this kind of DOB design results in excessive suppression at the regions and unnecessarily deteriorates high frequency characteristic and stability margin. To resolve this, we propose following filter of band pass type.

$$Q(z) = g\frac{(z-1)}{(z-p_1)(z-p_2)} \tag{15}$$

where g, p_1, p_2 is design parameter determined by disturbance characteristic. This enables the DOB to be an efficient loop shaping tool and increases the possible compensation frequency. Especially, the adjustable gain on the filter including band-pass frequency enables more flexibility on the shaping of the loop considering trade-off between performance and stability margin. Though higher order filter may enable more flexible degree of freedom on the loop shaping, on one hand it will complicate controller implementation and parameter tuning. Also considering the type of add-on fashion, the simplest form is desirable if possible. As a compromise,

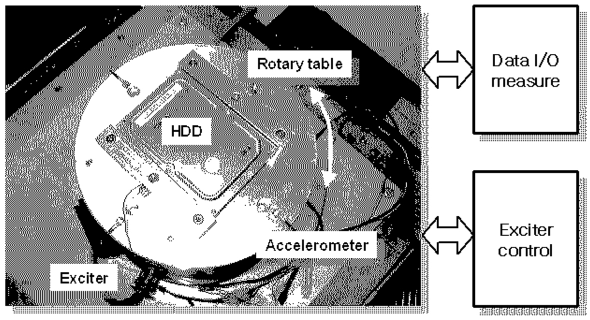

Fig. 8 Experimental setup

this paper uses the structure of (15).

Now take a look at how the add-on DOB affects to the main loop. This will give us useful insight on how to properly design the DOB. The DSF (disturbance sensitivity function) without DOB is given by

$$G_{DSF}(z) = \frac{P(z)}{1 + P(z)C(z)} \tag{16}$$

If DOB is included, DSF with the equivalent controller of (14) becomes

$$G_{DSF}(z) = \frac{P(z)}{1 + P(z)C_{DOB}(z)} \tag{17}$$

Thus, the changes on main loop by add-on DOB can be estimated by

$$\left|\frac{G_{DSF\_DOB}(z)}{G_{DSF}(z)}\right| = \left|\frac{\{1 + P(z)C(z)\}P_n(z)\{1 - Q(z)\}}{\{1 + P(z)C(z)\}P_n(z) + \{P(z) - P_n(z)\}Q(z)}\right| \tag{18}$$

Now, if we assume $P(z) = P_n(z)$, it is simplified as

$$\left|\frac{G_{DSF\_DOB}(z)}{G_{DSF}(z)}\right| = |1 - Q(z)| \tag{19}$$

Note this approximation contains only filter dynamics. As shown in Fig. 10, the nominal plant and real plant matches at most of the regions except high frequency ranges where most of resonances reside in. Thus above approximation is quite useful to estimate the loop changes due to add-on DOB. In Fig. 6, the $|1 - Q(j\omega)|$ used in the experiment is shown. Where the magnitude is less than 1 represent the area enhanced by add-on DOB. The result indicates that it is desirable to place the dip of $|1 - Q(j\omega)|$ at the frequency of dominant external disturbance. In Fig. 12, the comparison between the conventional design and the proposed is shown where the proposed design provides more stability margin and at the same time higher rejection performance for the target disturbance (Table 1).

4. Controller Optimization

This chapter describes how to optimally design the proposed DOB. In what follows, we denote it by FS-DOB highlighting its frequency selectivity. As the plant inversion is determined by the nominal model of the target, remaining objects for optimization are the

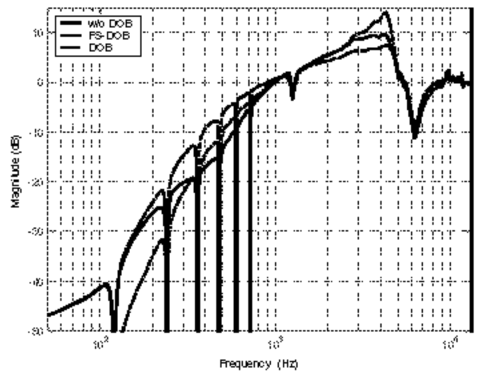

Fig. 9 Bode magnitude plot of error sensitivity function

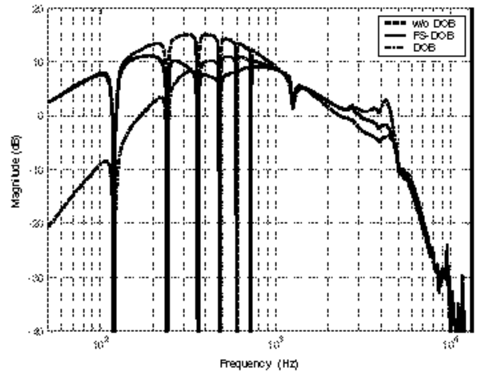

Fig. 12. Bode magnitude plot of disturbance sensitivity function

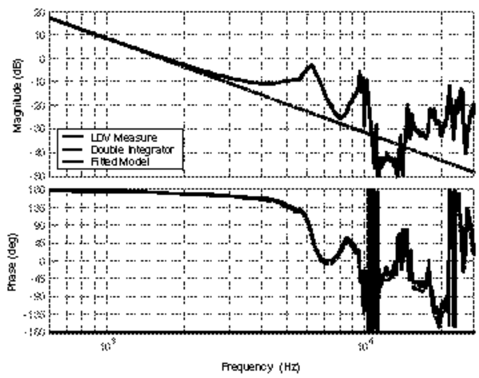

Fig. 10 Bode plot of the plant model

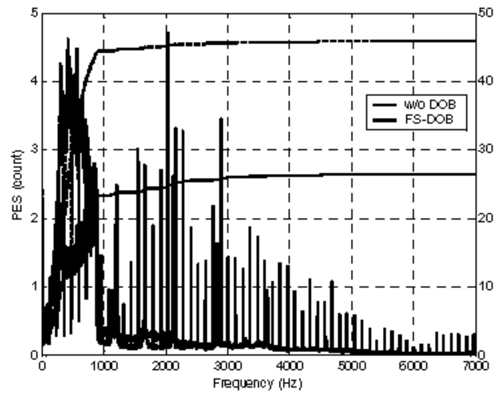

Fig. 13 PES and cumulative PES under rotational vibration of 6.5 rad/s^2 rms @ 100 ~ 900 Hz

parameters of (15). The design purpose is to minimize the effect of external disturbance on the PES while satisfying the required stability margin.

Typical servo block diagram of HDD can be represented by Fig. 7. The inputs to the system are input disturbance(w), output disturbance(d) and measurement noise(n). If we have the knowledge of the FRF of PES, plant and controller, the equivalent disturbance can be calculated from following relations [3][4].

$$PES(k) = G_s(z)\{G_p(z)w(k) + d(k) + n(k)\} \quad (20)$$

where, $G_s(z) = \dfrac{1}{1 + G_p(z)G_c(z)}$

Also by using the calculated disturbance, the PES according to the changes of controller design can be easily estimated from (20). However the PES from (20) is not appropriate for an index of optimization because it is a data vector of frequency domain. Fortunately, the PES of HDD system has the characteristic of stationary random. Hence, by using the Parseval's theorem the variance of time domain can be estimated. That is, if arbitrary process $x(k)$ in discrete time domain is stationary random in $0 \le k < N$ and its mean is zero, following is satisfied.

$$PES(k) = G_s(z)\{G_p(z)w(k) + d(k) + n(k)\} \quad (20)$$

$$\sigma^2 = E[x^2(k)] = \frac{1}{N}\sum_{k=0}^{N-1} x^2(k) = \frac{1}{N^2}\sum_{k=0}^{N-1} \left|X_m\right|^2 \quad (21)$$

where, N is "# of data sample" and X_m is the magnitude of FFT. For the accuracy of the estimation, it is important the accuracy of plant model used in (20). This paper adopted 60th order LMS fitted model based on the LDV measurement data. Finally the optimization problem is formulated as follows.

Definition

i) x : controller parameter vector $\in R^3$

ii) B_G : the set of all x which satisfy bound constraint

iii) B_S : the set of all x which satisfies "M_s stability"

iv) M_s stability : the status that the system satisfies GM, PM and SPH constraints.

v) SPH : sensitivity peak above first main resonance (butterfly mode), which is used to secure stability robustness against model mismatch at high frequency regions.

$$\mathrm{SPH}(G, \omega_H) \equiv \sup_{\omega > \omega_H} \left\| \frac{1}{1 + G(j\omega)} \right\|$$

Problem

With above definition, the optimization problem is to

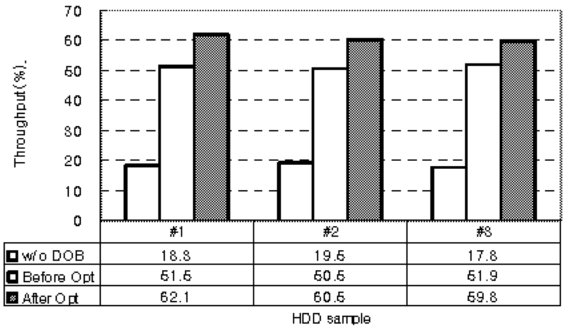

Fig. 14 Throughput under rotational vibration of 6.5 rad/s^2 rms @ 100 ~ 900 Hz

find out x which satisfies

$F(x^*) \le F(x)$, $\forall x \in (B_G \cap B_S)$

where $F(x) \equiv \sup_{d \in D} \left\{ \sqrt{E\left[y(k,d)^2 \right]} \right\}$ ($0 \le k < N$) and $y(k,d)$ denotes the sequence of PES with the equivalent disturbance model $d \in D(j\omega)$ obtained by (20) for a number of drives.

This is constrained nonlinear optimization problem. With commercial optimization tool of Matlab®, the "fmincon" is capable of solving this kind of problem. However it gives only local minima thus, the selection of proper initial value is the key for successful optimization. By using (19) and equivalent disturbance from (20), the value can be easily decided.

Another factor is that the performance should be satisfied for a number of drives manufactured. Though all the drives go through the same manufacturing process, quality deviation of the product is inevitable due to the mechanical variations stemmed from actuator assembly, HDA (head disk assembly) status and so on. And this results in different performance even if the same controller is used. To cover this uncertainty, a number of drives randomly sampled are used for the generation of the worst case disturbance model.

5. Experimental results

Experimental setup is shown in Fig. 8. The drive is 3.5" form factor with 160 Gb/platter, 2 head and 140kTPI. The number of servo sector is 220 and spindle rotational speed is 7219 rpm. This is equivalent to the sampling rate of 26.5 kHz. As mentioned in chapter 2, HDD's actuator generates rotary movement with respect to pivot center and it is most susceptible to the RV (rotational vibration) of drive in-plane direction. Thus, generally RV performance is measured as a representative index of the HDD's performance for external vibration.

In the experiment, excitation condition is random signal of average 6.5 rad/s2 over 100 ~ 900 Hz. Performance target is TP degrade less than 50%. Data transfer rate is measured by commercial software, Iometer®.

Table 1 Comparison of major servo index between conventional and proposed design

	DSF @400Hz (dB)	F_{gx} (Hz)	PM (deg)	GM (dB)	SPL (dB)	SPH (dB)
w/o DOB	15.1	1338.5	44.3	5.2	7.5	1.5
FS-DOB	7.6	1675.8	35.3	3.5	9.8	2.0
DOB	10.1	1940.4	31.6	2.5	14.4	2.5

(where F_{gx} denotes gain cross over frequency, SPL denotes sensitivity peak below first resonance frequency and DOB denotes conventional design.)

Meanwhile, as show in Fig. 6, the filter of (15) boosts up ESF magnitude around Nyquist frequency, which is undesirable in view of SPH stability. To resolve this, a low pass filter of 1st order is added to the Q filter. The filters before and after optimization are shown in Fig. 11.

By using FS-DOB, we have attained 51.3% TP with initial design before optimization. After optimization additional 9.5% improve is observed. In consequence, total 42.3 % improve have been achieved by the proposed design compared to the design with only conventional main loop (Fig. 14). The stability constraint used in the optimization is M_s =[3.5dB 35deg 3.0dB] and it is confirmed that the stability is satisfied with the FS-DOB (Table 1). In Fig. 12 and Fig. 13, it is verified that the shape of DSF is adjusted according to the dominant disturbance shape.

6. Concluding remarks

This paper suggested a frequency selective and optimal design method for DOB as a way to minimize the effect of external disturbance on the PES. The proposed is summarized as follows. First, a solution to deal with unstable zero in the plant model was presented. The ZGEI in the discrete time domain is shown to be more accurate plant inversion. Second a filter structure for the Q filter was suggested to accommodate the frequency characteristic of external disturbance and the stability margin. Third, a new stability margin which is appropriate for HDD servo system was defined and the optimal controller which satisfies the stability and minimizes the PES was designed.

Previous researches regarding DOB have been focused on the enhancement of the disturbance compensation capability by estimating the disturbance as correct as possible. Hence the works were confined to something like plant model's accuracy improvement or the bandwidth increase of the Q filter as low pass filter. Also in the researches, quantitative consideration of the stability margin was often ignored.

Compared to this, the proposed is valuable in the aspect of what follows. First, stability margin was considered quantitatively which enabled that the design of robust controller which is capable of coping with the variation of the mass manufactured devices like HDD. Second by adopting frequency selective Q filter, not only was practical disturbance rejection capability improved, the limit of conventional DOB as low frequency disturbance solution was overcome. Third, DOB was refined in the point of loop shaping. Actually this implies the possibility of DOB as a synthesis mean for higher order controller other than conventional use of disturbance compensation.

The method proposed here can be easily applied to other servo systems with only minor modification such as plant and disturbance modeling. Also with more sophisticated filter, more flexible loop shaping can be easily achieved. In that case the trade-off should be accounted between the performance and the implementation complexity with related calculation delay. In highly competitive HDD market, the cost is the most important aspect which should be considered in the development. In this regard, the proposed offers a practical solution for performance enhancement, for it does not require any additional sensors or hardware modification.

Appendix

Consider following equation regarding estimated disturbance

$$\hat{d}(k) = Q(z^{-1})\frac{B^-(z^{-1})}{B_m^-(z^{-1})}d(k) \quad \text{(A1)}$$

By using the identity of $z = e^{i\omega T_s}$, frequency response function neglecting filter Q is given by

$$\frac{B^-(z^{-1})}{B_m^-(z^{-1})} = \frac{b_0 + b_1 e^{-j\omega} + \cdots + b_p e^{-jp\omega}}{b_0 e^{-jp\omega} + \cdots + b_{p-1}e^{-j\omega} + b_p} \quad \text{(A2)}$$

For simplicity $T_s = 1$ is assumed.

Also using Euler's formula, the equation can be rewritten by

$$\frac{B^-(z^{-1})}{B_m^-(z^{-1})} = \frac{(b_0 + b_1 c_\omega + \cdots + b_p c_{p\omega})}{(b_0 c_{p\omega} + \cdots + b_{p-1}c_\omega + b_p)} \cdot \frac{-j(b_1 s_\omega + b_2 s_{2\omega} + \cdots + b_p s_{p\omega})}{-j(b_0 s_{p\omega} + \cdots + b_{p-2}s_{2\omega} + b_{p-1}s_\omega)} \quad \text{(A3)}$$

where, $c_\omega = \cos(\omega)$, $s_\omega = \sin(\omega)$

Finally with basic trigonometric property, the magnitude is simplified as

$$\left|\frac{B^-(z^{-1})}{B_m^-(z^{-1})}\right| = \sqrt{\frac{b_0^2 + b_1^2 + \cdots + b_p^2 + f(c_\omega, \cdots, c_{p\omega})}{b_p^2 + \cdots + b_1^2 + b_0^2 + f(c_\omega, \cdots, c_{p\omega})}} = 1 \quad \text{(A4)}$$

This completes the proof of ZGEI of (A1).

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