Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008

HDD Servo Control Technologies
- What we have done and where we should go -

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Abstract: Since 1986 when digital servo was introduced in mass production HDDs, various kinds of control theories have been applied to HDD servo system, and the performances had improved dramatically. A 3.5-inch HDD capacity, which was 251 MB in 1988, has 1,000,000 MB (1 TB) capacity in 2007. The Fast access (high quality motion control), very precise positioning (nanometer scale), and robustness against environment changes, various kinds of disturbances, and plant dynamics fluctuations, have been required in HDD servo system to achieve the capacity increase. In this paper, some of key topics of recent HDD servo research are described and an integrated servo mechanical design based on detail understanding of plant dynamics and disturbance is described in detail as one of recent outstanding activities to improve positioning accuracy.

1. INTRODUCTION

HDD servo system is a typical mechatronics servo, requiring fast access (high quality motion control), very precise positioning (nanometer scale), and robustness against environment changes, various kinds of disturbances, and plant dynamics fluctuations. Over the past ten years, required positioning accuracy is one-twentieth smaller, and seek time is also reduced into half. Many aggressive challenges in servo control have been made to support this drastic HDD progress. [Yamaguchi et al. (2000), Yamaguchi (2001)]

Areal density has been increased to supply larger capacity HDDs. In terms of technology development subjects for this areal density increase, one of big differences from previous years is that we no longer count on “down sizing.” More efforts are necessary to increase track density, in other words, to improve positioning accuracy on the same form factor HDD. One of major efforts is to increase servo bandwidth and finally dual-stage servo using PZT actuators has been commercially applied to HDD. Although this technology has not widely been applied to today’s HDD yet due to cost issue, the dual stage actuator will be a key servo-mechanical technology in near future.

There are two major activities to suppress disturbances. One is to reduce internal disturbance such as flow induced vibration like disk flutter. The other is to reduce external vibration transmitted HDD through the PC chassis. One low cost solution to cope with the internal disturbance is to optimize the sensitivity function by taking sophisticated integrated servo mechanical design based on detail plant dynamic modeling. An interesting topic in this design is how to stabilize mechanical resonances and to reduce disturbance above the Nyquist frequency. For external vibration, feedforward control using sensor is commonly used.

In this paper, a brief overview of HDD servo control technology since mid 80s when digital servo has been applied to HDD is shown, and an integrated design of servo-mechanics is identified as one of most important design methods to support today’s HDD capacity increase.

2. HARD DISK DRIVE

2.1 Hard disk drive (HDD)

Figure 2.1 shows a photo of HDD. Several disks are stacked on the spindle motor shaft and rotate at 15,000 rpm in high end 3.5-inch drives and 5,400 – 7,200 rpm in 2.5-inch drives. On the surface of a disk, several hundred thousand data tracks are magnetically recorded, and the latest track pitch is about 100 nm. A slider is supported by a suspension and a carriage, and it is suspended less than half micro-inch above the disk surface. An actuator, called a voice coil motor (VCM), actuates the carriage and moves the slider on a desired track. The mechanical part of the plant, that is, the controlled object, consists of the VCM, the carriage, the suspension, and the sliders. On the back of the head-disk assembly mechanism (HDA), there is a circuit board on which a microprocessor or a digital signal processor (DSP) is mounted. The position signals are recorded magnetically on each disk using servo-writing equipment (servo track writer). The position signals are recorded in a certain time interval on each track. Consequently, the position error between the head and the track can be detected directly by reading the position signal.

2.2 Modeling

In HDD modeling, not only plant dynamics modeling but also disturbance modeling are important. Fig. 2.2 shows an example of disturbance model used in HDD benchmarking

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10.3182/20080706-5-KR-1001.2191

program [http://mizugaki.iis.u-tokyo.ac.jp/nss/]. In lower frequency range, RRO and torque disturbance are dominant, and in higher frequency range over about 1 kHz, disk flutter and arm / suspension flutter are observed. High servo bandwidth does not always bring us better positioning accuracy due to widely existing disturbances. As shown in Section 4, sophisticated servo loop shaping such as disturbance reduction using mechanical resonance has been applied.

Fig. 2.1. Hard Disk Drive

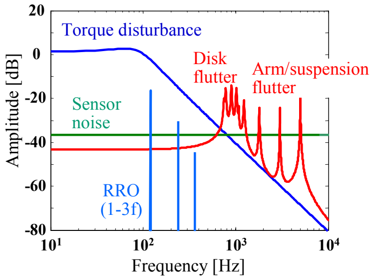


Fig. 2.2. Disturbance model

3. HDD SERVO CONTROL

3.1 Overview

Table 1 shows trends of HDD servo technologies for past three decades. Some of main topics presented in major conferences and journals are shown in this table. In the 80s and early 90s, applications of control theory such as H-inf control have been attempted. In the later 90s and this century, sophisticated servo control designs based on HDD specific features have been proposed. Popular classification of these designs would be (1) fast access servo, (2) settling servo, and (3) precise positioning and disturbance suppression servo. The table also uses this classification. However, there may be another viewpoint which describes HDD servo control development trend more clearly, clarifying features of HDD servo system. Based on the features, future directions of HDD servo research could be discussed and other industries could also understand HDD servo system.

3.2 Features of HDD Servo

Four features of HDD servo are identified in this paper, which have been well-developed in HDD servo design. They are (a) Two-degrees-of-freedom (TDOF) feedforward controller design, (b) Sampled servo design including multi-rate sampling, (c) Disturbance suppression control including high servo bandwidth design, and (d) Transient response control such as mode switching control (MSC).

(a) TDOF feedforward controller design: Fast access has been required for years, and TDOF control has been applied since early 90s. Important research topic was the design of feedforward controller to achieve an accurate inverse dynamics of the plant. The zero phase error tracking control (ZPETC) was applied and the perfect tracking control (PTC) was proposed using advantage of multi-rate sampling (Table 1.) PTC has been later modified to take into account of mechanical resonance. TDOF control is quite common in HDD servo system now.

(b) Sampled servo design including multi-rate sampling: Since position information is detected at a certain sampling period due to limited numbers of servo patterns on disk, HDD servo is necessarily a sampled servo and its sampling rate is limited by data format efficiency. One of the key designs to solve this issue is a multi-rate sampling. The sampling rate of control input can be n-times higher than the one of position detection. This idea has brought us stability for mechanical resonance peaks above the Nyquist frequency. Several designs using sampled control theory have been applied to handle dynamics occurred in inter-sampling (Table 1.) Multi-rate sampling and sampled control theory are also common technology in HDD servo system.

(c) Disturbance suppression control including high servo bandwidth design: Main stream of HDD servo development has been the increase of servo bandwidth. Typical ideas are dual stage actuator servo and multi-sensing (Table 1.) Since these ideas require new components with additional cost, other approaches to suppress disturbances have been studied. One typical research is phase stabilized design. By taking this approach, mechanical resonance peak can be utilized as disturbance rejection due to its high gain. But this requires detailed plant and disturbance modeling. Since this area has been one of latest hot topics to increase HDD track density, detail descriptions are shown in Section 4. Another approach is to design specific controller to reduce known disturbance like repeatable runout (RRO). Very narrow bandpass filter which has very high gain at a certain frequency (disk rotational frequency and its harmonics) are common to reduce RRO (Table 1.)

(d) Transient response control such as mode switching control (MSC): Initial value compensation (IVC) has been developed to improve transient response, and various modifications have been proposed. Several studies have achieved reducing mechanical residual vibration by designing seek trajectory considering mechanical model (Table 1.) MSC is quite common in HDD servo and trajectory design considering mechanical resonance have been widely studied in HDD servo system.

4. INTEGRATED DESIGN OF SERVO-MECHANICS

4.1 Phase Stable Design for Mechanical Resonance Modes

In the HDD servo system, there are various mechanical resonance modes. Such resonance modes have been generally regarded as unmodeled uncertainty and the notch filters have been used for decreasing the gain at these resonance frequencies. Although such control systems can avoid the instability caused by the mechanical resonances, they cannot suppress their vibrations, and such notch filters reduce the phase margin and increase the H_∞ norm of the sensitivity function.

In response to the above-mentioned problems, an integrated design method for a controller and a mechanical structure based on vector locus has been developed [Atsumi et al. (2003)]. Firstly, the mechanical system is designed so that all major resonances are in-phase. The frequency response of the mechanical system is shown in Fig. 4.1, and vector loci of these mechanical systems are shown in Fig. 4.2. In both figures, solid line indicates the result whose temperature is 25 degree and dashed line indicates 55 degree. These figures show that characteristics of resonance modes are changed by temperature conditions. However, these figures also indicate that these resonance modes could keep the in-phase condition in spite of the temperature fluctuation.

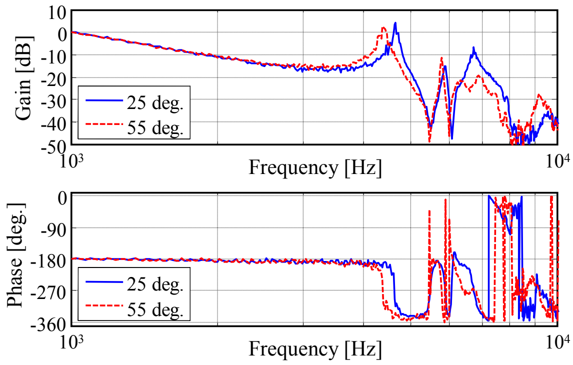


Fig. 4.1. Frequency response of in-phase mechanical system

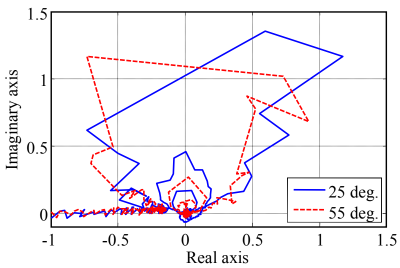
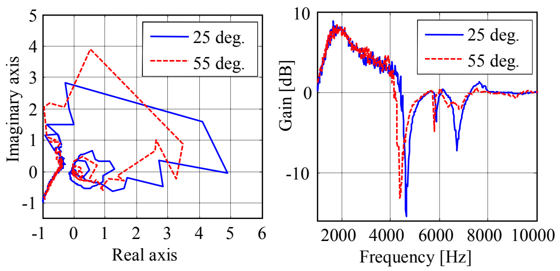


Fig. 4.2. Vector locus of in-phase mechanical system

If all resonances are in-phase, the control system can stabilize all resonance modes by phase condition. As a result, the feedback controller can be just an integrator and a phase lead filter. The controller is given by

$$C[z] = \frac{2.5445(z-0.9691)^2}{(z-1)(z-0.4361)} \tag{4.1}$$

The measured vector loci of open-loop characteristics are shown in Fig. 4.3 (a). This figure indicates that all of resonance modes are stabilized by phase condition. Fig. 4.3 (b) shows frequency response of the sensitivity function. The gain of the sensitivity function is below 0 dB at all high-order resonance frequencies. It is clear that the simple 2nd-order controller shown as (4.1) can suppress vibrations caused by all mechanical resonances because of its phase condition.



(a) Nyquist diagram (b) Sensitivity function

Fig. 4.3. Experimental results

4.2 Integrated Design for High Servo Bandwidth

Increasing the servo bandwidth is often effective to achieve better positioning accuracy. When the Nyquist frequency of the servo system is much higher than the primary resonance frequency, the servo bandwidth mainly depends on the effects of the primary resonance mode. The major issue due to the primary resonance mode is decreasing the gain margin of the servo system, and the system stability depends on the mechanical system gain at -180 degree phase crossing frequency of open loop. Therefore, the servo bandwidth is limited not only by the frequency of a primary resonance but also by its residue. Frequency of a primary resonance mode may not be able to be increased dramatically without reducing the size of HDD or using a dual stage actuator. Consequently, decreasing the residue in a primary resonance mode has been pursued to increase servo bandwidth [Atsumi, Shimizu et al. (2006)].

A general mechanical system of HDD head-positioning system (called the "high stiffness design" hereafter) is illustrated in Fig. 4.4 (a) and the modal shape of the primary resonance mode calculated by an FEM simulation is shown in Fig 4.4 (b). The shape of this primary resonance mode is arched, so the movement at the observed point is larger than other parts. As a result, the amount of the residue in the primary resonance becomes significant.

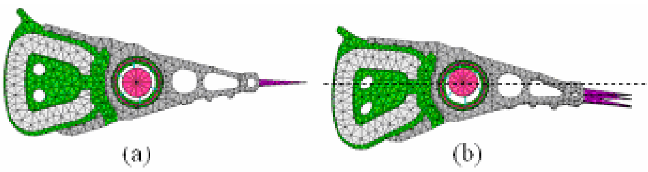


Fig. 4.4. Mechanical structure based on high stiffness design

In order to decrease the residue in a primary resonance, the mechanical system by using a four-bar linkage mechanism (called the "low residue design" hereafter) has been developed. This mechanical system is illustrated in Fig. 4.5 (a), and the modal shape of the primary resonance is shown in Fig 4.5 (b). These figures show the modal shape has little movement at the observed point. The result is that only a small amount of residue remains in the primary resonance.

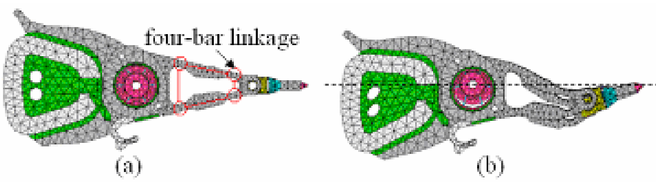


Fig. 4.5. Mechanical structure based on low residue design

The frequency responses of these mechanical systems are shown in Fig. 4.6 (solid line: low residue design, dashed line: high stiffness design). This figure indicates that low residue design enables us to improve the mechanical system gain at -180 degree phase crossing frequency of open loop even though the primary resonance frequency is lower (high stiffness: 5700 Hz, and low residue: 4800 Hz).

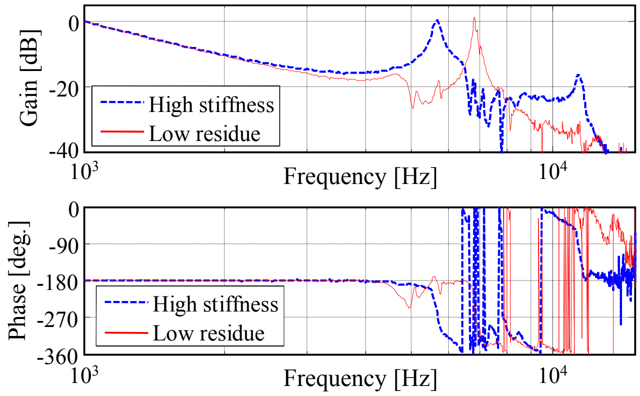


Fig. 4.6: Frequency response of mechanical system

The control system is independently designed for each mechanical system shown in Fig. 4.6. Table 4.1 shows servo bandwidth, gain margin, phase margin, and H_∞ norm of the sensitivity function.

Table 4.1. Experimental results

Design method	High stiffness	Low residue
Servo bandwidth	1400 Hz	1700 Hz
Gain margin	4.5 dB	4.6 dB
Phase margin	30 deg.	31 deg.
$\|S\|_\infty$	9.2 dB	9.2 dB

Fig. 4.7 shows the measured frequency responses of sensitivity functions and the low residue design method decreases sensitivity function's gain below 1 kHz by about 2.5 dB.

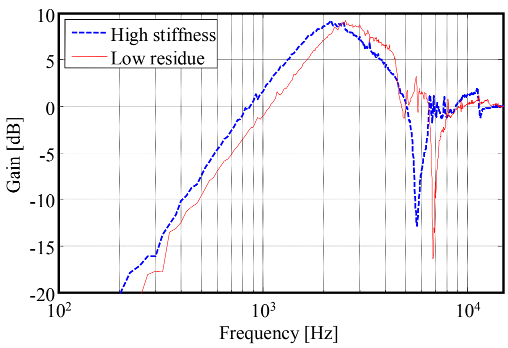


Fig 4.7. Experimental result of sensitivity function

4.3 Vibration Control above the Nyquist Frequency

The servo system has often mechanical vibrations whose frequencies are above the Nyquist frequency of the servo system. In this case, the servo system is required to suppress mechanical vibrations above the Nyquist frequency. To avoid instability, a control system in conventional HDDs compensates for these vibrations by using multi-rate notch filters. Although, such control systems can only avoid the instability caused by mechanical resonances, they cannot suppress their vibrations.

In response to the above mentioned problems, we proposed a design method for a sampled-data control system that suppresses vibrations whose frequencies are higher than the Nyquist frequency [Atsumi et al. (2006)]. The block diagram of the servo system is shown in Fig. 4.8. Where P_c is a controlled object in continuous time, C is a digital controller, H is a hold, and S is a sampler, d_c is a disturbance signal, y_c is a head position in continuous time, and y_d is a measurement signal of head position in discrete time.

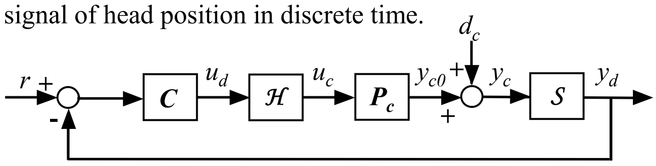


Fig. 4.8. Block diagram of servo system

It is obvious that a servo system has to control the maximum displacement of the control variable. This means that the infinity norms of signals are used for defining the norm of the sensitivity function for the HDD servo control. Therefore, the gain of the sensitivity function in sampled-data system at ω_0 is defined as follows.

$$|S_{sd}(j\omega_0)| = \frac{\| y_c(t) \|_\infty}{\| d_c(t,\omega_0) \|_\infty} = \sup_t |y_c(t)| \qquad (4.2)$$
$$= |\Gamma(\omega_0)| + \sum_{l=-\infty,\neq 0}^{\infty} |\Lambda(l,\omega_0)|$$

where

$$d_c(t,\omega) = e^{j\omega t}, \qquad (4.3)$$
$$\Gamma(\omega) = 1 - \frac{C[e^{j\omega T_s}]H(j\omega)P_c(j\omega)}{T_s + \sum_{k=-\infty}^{\infty} H(j\omega + j\omega_s k)P_c(j\omega + j\omega_s k)C[e^{j\omega T_s}]},$$
$$\Lambda(l,\omega) = -\frac{C[e^{j\omega T_s}]H(j\omega + j\omega_s l)P_c(j\omega + j\omega_s l)}{T_s + \sum_{k=-\infty}^{\infty} H(j\omega + j\omega_s k)P_c(j\omega + j\omega_s k)C[e^{j\omega T_s}]},$$

T_s is the sampling time [s], ω_0 is the disturbance frequency and ω_s is the sampling frequency [rad/s]. The above results indicate that the control system should have a certain level of $|HP_c|$ at ω_0 so that $|\Gamma|$ becomes small, in order to suppress vibration whose frequency is ω_0. And it should have small amount of $|HP_c|$ at the aliasing frequencies so that

$$\sum_{l=-\infty,\neq 0}^{\infty} |\Lambda(l,\omega_0)| \qquad (4.4)$$

becomes small.

On the other hand, a controlled object in a discrete-time system P_d and the sensitivity function in the discrete-time system S_d at ω_0 are given as follows.

$$P_d(j\omega_0) = \frac{1}{T_s} \sum_{k=-\infty}^{\infty} H(j\omega_0 + j\omega_s k)P_c(j\omega_0 + j\omega_s k) \qquad (4.5)$$

$$S_d(j\omega_0) = \frac{1}{1 + P_d(j\omega_0)C[e^{j\omega_0 T_s}]} \qquad (4.6)$$

The following is an example. The sampling time T_s was 153.46 μs (ω_s =2π×6516 [rad/s] and the Nyquist frequency was 3258 Hz). The frequency response of the mechanical system P_m is indicated in Fig 4.9 (solid line: mathematical model, dashed line: measurement data). From this frequency response, it assumes that $|HP_c|$ is vanishingly small when $\omega > 2\omega_s$. Thus, to save the calculation time, the frequency response of P_c is defined as

$$P_c(j\omega) = \begin{cases} P_m(j\omega) : |\omega| < 2\omega_s, \\ 0 : |\omega| \geq 2\omega_s \end{cases} \tag{4.7}$$

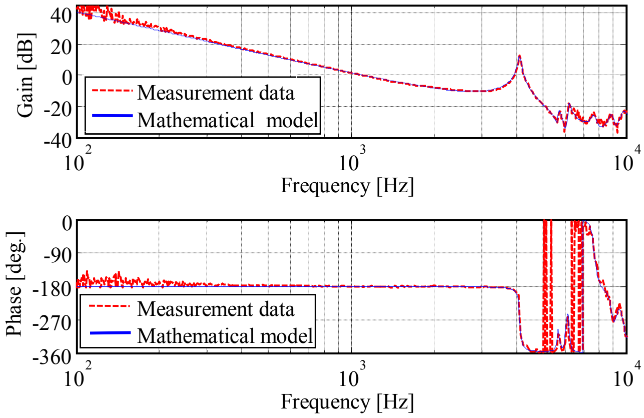


Fig. 4.9. Example of mechanical system

In this study, we focus on the vibration caused by the primary mechanical resonance at 4100 Hz. The discrete-time system is periodic with ω_s. Thus, stabilizing the phase of the resonance mode of P_d at 2400 Hz enables $|S_{sd}|$ <1 at 4100 Hz. To stabilize the phase of the resonance mode in P_d, C is set based on the following equation.

$$C[z] = 0.5597\frac{(z+1)(z-0.9529)(z-0.908)}{(z+0.5882)(z-1)(z-0.01812)} \tag{4.8}$$

The sensitivity function in the sampled-data system and the discrete-time system are calculated. Fig. 4.10 shows these results (solid line: sampled-data system, dashed line: discrete-time system) and $|S_{sd}|$ is below 0 dB at 4100 Hz. This means that the control system can suppress the vibrations above the Nyquist frequency.

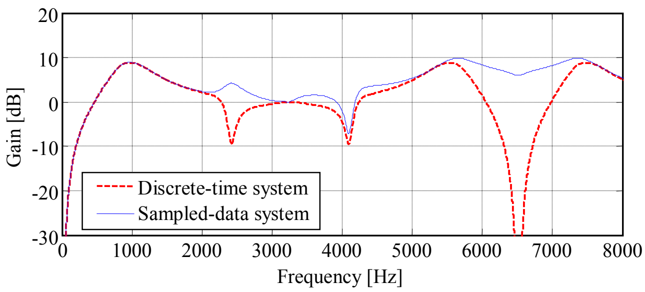
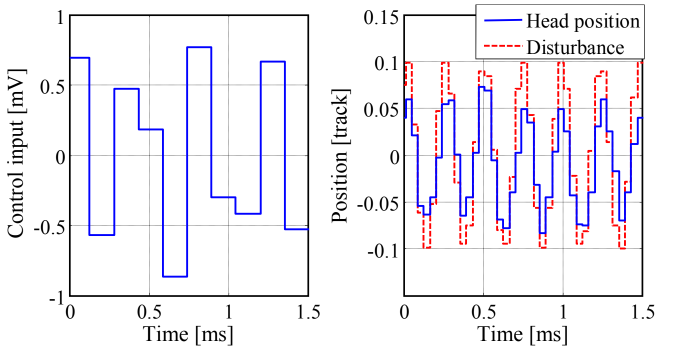


Fig. 4.10. Sensitivity function

Experiment has been done using this control system, in which the frequency of disturbance signal d_c is 4100 [Hz], and the vibration amplitude is 0.1 [track]. To measure the inter-sampling behavior, the sampling time of d_c and y_c is set as 38.4 μs (1/4 of T_s). The time response of the control input is shown in Fig. 4.11 (a), and the time responses of the d_c and y_c are shown in Fig. 4.11 (b). The experimental results also indicate that the control system can decrease vibrations whose the frequency is above the Nyquist frequency.



(a) Control input (b) Head position

Fig. 4.11. Experimental result

5. OTHER TOPICS

5.1 Dual stage actuator and multi-sensing

Although many useful servo design methods for driving dual-stage actuator have been developed in early and mid 90s, the industry still hesitates to use the dual-stage actuator system as a standard method. The major reason is cost, and the downsizing provides the opportunity to take an evolutional approach to achieve higher areal density. Since early 90s, we defined three generations of the dual-stage actuator, (1) suspension driven, (2) slider driven, and (3) head element driven. The second generation dual-stage actuator was already once applied to commercial HDD. At the same time, multi-sensing approach using additional sensor has been studied. This approach is quite reasonable, but has not been widely accepted due to cost. In the future HDDs will have to keep using 3.5-inch size form factor to achieve terabytes HDDs. Under this business situation, the dual-stage actuator as well as multi-sensing will be likely applied to HDDs.

5.2 Servowriter

In 2000s, servowriter has become very important device to achieve higher TPI. Originally push-pin type servowriter has been used. Recently media servowriter has been widely used. However, it becomes not easy to achieve required track pitch accuracy by taking semi-closed loop servo control using motor encoder system. Self servowriter is an idea which has been known since 90s. The self servowriter may take closed loop control just as HDD servo. Key design point is how to write next track servo pattern based on previous servo pattern trajectory with keeping appropriate track pitch accuracy. When writing servo signal for next track by tracking the previous servo pattern trajectory, the head motion is controlled by the closed loop characteristics. The closed loop gain at a portion of frequency range exceeds 1.0 which may cause the increase of propagation error. Several algorithms have been studied to compensate for this instability issue.

6. CONCLUSIONS

A couple of subjects should be pointed out for discussing our future directions. (1) Required positioning accuracy. Within five years, the areal density will be 1 Tbit/in^2, and 500 kTPI would be necessary, which requires 5 nm positioning accuracy (3 sigma). Under various disturbances from both internal and external, realizing 5 nm positioning would be challenging. We will have to take into account of best match between sensitivity function and disturbance frequency spectrum. (2) Quality of servo signal. The quality may include signal to noise ratio, resolution, linearity, and sampling rate. It is important to make clear better relationship between sampling rate and signal to noise ratio. Finally mutual communications with other technology area would become more important to find our future direction. It is valuable for this communication to clarify "feature of our servo control" to give overall understanding and to propose "benchmarking program" to give detail description.

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http://mizugaki.iis.u-tokyo.ac.jp/nss/

Table 1. Trend of HDD servo control technology

	1981 → 1990 → 2000 → 2005
Drive	5.25 inch; 3.5 inch; 1st HDD using digital servo; 2.5 inch; 1.8 inch 1.3 inch; 1.0 inch; 0.8 inch
Areal density (Mbit/in^2)	10; 10^2; 10^3; 10^4; 10^5
Track pitch (μm)	38.5; 15.0; 12.7; 8.5; 3.0; 1.3; 0.51; 0.32; 0.25; 0.17
Seek time (ms) (3.5 inch)	15; 11; 9; 8; 6; 4; 3.5
Control technologies — Seek servo	Velocity control (1974); Proximate time optimal control (PTOS); Finite state control; Jerk minimized; Sliding mode; TDOF for short span seek; TDOF of model following; N-delay Multi-rate feedforward; Multi-rate perfect tracking control (PTC); Final state control; Vibration minimized PTC; IVC one-degree-of-freedom servo
Trajectory design	Vibration minimized (input shaping); Deadbeat control; Vibration minimized trajectory
Settling servo	Start of mode switching (1977); Initial value compensation (IVC); Vibration minimized; Feedforward type IVC; Residual vibration analysis using shock response spectrum; Mode switching control with input restriction
Track-following servo	Lead lag (1974); Kalman filter (1976); Estimator state feedback; LQG/LTR; Multi-rate control; H∞ control; Dual stage servo; H2 control; Multi-rate notch filter; Various type of dual stage servo; Multi-sensing; Mechanical resonance phase stabilized; Sampled H∞; Active damping; Actuator-sensor collocation; PQ control (decoupling); Multi-rate RRO control; Phase stabilized servo with peak filter; Product application of dual stage servo
Disturbance control / Adaptive control	Disturbance observer; Repetitive control; Adaptive runout correction; Feedforward control with sensor; Self-tuning control; Learning control; Friction control; Adaptive seek control; RRO compensation (peak filter, feedforward, non-tracking); MRACS; RRO feedforward With inverse model; Hybrid identification