Nano-contact, contact detection and prevention in high density hard disk drives

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**Nano-contact, Contact Detection and Prevention in High Density
Hard Disk Drives**

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2012

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Abstract

This thesis presented theoretical and experimental studies of the characteristics of the head/disk interface at very low flying height for the next generation high density hard disk drives.

The study started with a discussion of the tribological background of the head/disk interface and presented a review of the literature related to studies of the head/disk interface. Then the thermal protrusion of a thermal flying height control (TFC) slider and the patterned media were chosen to perform a contact analysis at the head-disk interface (HDI). Results obtained from finite element analysis for the contact pressure at elastic-plastic asperity micro contacts and subsurface stress distribution were presented.

Subsequently, an active slider based on piezoelectric shear-mode actuation was proposed and investigated for flying height control. A finite element model of a PZT slider using shear-mode deformation was developed. The electromechanical simulation and air-bearing simulation were performed to investigate the shear-mode deformation and the static flying attitude of the PZT slider. The effects of location of the PZT sheet and the dimension of air bearing surface on the static flying attitude of the PZT slider were studied.

This was followed by a novel structure design of an active- slider with a piezoelectric actuator at the top of the slider body and a soft layer between the substrate and head elements. A parametric simulation was performed to obtain an optimized dimension of the PZT actuator and the most effective soft material. Then a slider structure with large actuation efficiency was achieved.

Thereafter, a ZnO piezoelectric contact sensor integrated with thermal flying height

control (TFC) slider for contact detection was proposed and investigated. The effects of the contact force and thermal-stress on voltage output of the integrated contact sensor in TFC slider were investigated using finite element method. The simulation results showed that the shear-mode ZnO sensor is more sensitive to contact force than the transverse-mode ZnO sensor. Parametric optimization was performed to achieve high sensitivity for the contact sensor.

Finally, piezoelectric micro cantilever with high quality ZnO films were fabricated and characterized to demonstrate the feasibility of an integrated piezoelectric contact sensor and possible active slider. The dynamic performance of the piezoelectric cantilever was characterized by impedance analyzer and LDV. A silicon cantilever with a small ZnO sensor was fabricated. The dynamic performance of this cantilever excited by a mechanical impact was investigated. The experimental results showed that the response from the sensor and the LDV measurements are very similar to each other, which indicated that the thin film ZnO sensor is of good sensitivity.

Keywords: magnetic data storage, magnetic disk drive, head-disk interface, tribology.

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4. Shengnan Shen, Hui Li, Kensuke Amemiya, Bo Liu and Hejun Du, “Parametric Simulation of Piezoelectric Flying Height Control Slider using Shear-mode Deformation”, 21th ASME Annual Conference on Information Storage & Processing Systems, 2011.
5. Shengnan Shen, Peihong Wang, Bo Liu and Hejun, “Simulation of Thermal Flying Height Control Slider with Built-in Contact Sensor”, 21th ASME Annual Conference on Information Storage & Processing Systems, 2011.
6. Peihong Wang, Shengnan Shen, Hejun Du, Mingsheng Zhang and Bo Liu, “Fabrication and Performance of ZnO Piezoelectric Cantilever for Micro Transducers”, 21th ASME Annual Conference on Information Storage & Processing Systems , 2011.
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Chapter 1 Introduction

1.1 Magnetic Recording Hard Disk Drives

Magnetic recording is the most common technology used to store information. Magnetic recording media have the following advantages: inexpensive media, stable storage, and relatively high data rate, high volumetric density, erasable and rewritable. Magnetic storage devices used for audio, video, and data-processing (computer) applications are tape, floppy disk, and hard disk drives. Magnetic recording and playback are accomplished by the relative motion between the magnetic medium and a stationary or rotating read-write magnetic head.

In 1956, International Business Machines (IBM) introduced the 305 RAMAC (Random Access Method of Accounting and Control) as the first magnetic hard disk for data storage [1]. This device would allow information to be recorded and retrieved in any order. It was originally called “fixed disk” or “Winchesters” (a code name used for a popular IBM product). They later became known as “hard disks” to distinguish them from “floppy disks”. The development of hard disk drive technology since then has been no less than amazing. Currently, hard disk drives with storage capacity on the order of 1000 GB are available and are used in modern computer systems. Recently (2011), Hitachi Global Storage Technologies [2] shipped a so-called “Ultrastar 7K3000” [3] with storage capacity up to 3 terabytes (or 3000 GB).

Fig. 1.1 shows a typical magnetic disk drive (top cover removed) and key components of the drive. A stack of disks is mounted on a sealed, grease-lubricated ball bearing spindle. The suspension/arm is mounted on the side of the disk and driven by the voice coil motor. The sliders, each equipped with integrated thin film read/write heads, are mounted on the backside of the suspension (Fig. 1.2). The disks are rotated

by a DC motor at speeds ranging from a few thousand to about 10,000 RPM. When the hard disk drive is powered on, the disks start to rotate until they reach a constant velocity. The voice coil motor moves the suspension and the slider over the disk surface. A converging channel between the slider and the disk generates a so-called “air bearing”. As a result of the air bearing, the slider flies over the disk, enabling the read/write heads to record information on circumferential tracks of the disk, and to read information from the disk.

During the operation of a disk drive, the slider flies over the disk at a fixed head-to-disk spacing (flying height). To increase the recording density, the head-to-disk spacing of the slider must be minimized. In today’s disk drives, the head-to-disk spacing is currently maintained to be less than 2 nm with the TFC technology.

Fig. 1.1 Components of a typical hard disk drive (HDD).

Fig. 1.2 Typical Pico and Femto sliders attached to their suspensions

(<http://www.hitachigst.com>).

1.2 Head-Disk Interface

A schematic of the head-disk interface is shown in Fig. 1.3. Reliable operation of a modern magnetic disk drive depends critically on maintaining a controlled spacing between the magnetic recording head and the recording medium. The distance between the bottom surface of the read/write element and the top surface of the magnetic layer on the disk is defined as head media spacing (HMS). A nearly constant spacing between the slider and the disk is called flying height (FH) as shown in Fig. 1.3.

The magnetic layer of the disk consists of a flat rigid substrate, a recording layer, an under-layer, a protective layer and a lubricant layer. There are two kinds of substrate for practical use. One is an aluminum-magnesium alloy disk on which a nickel-phosphorus film is plated. Another is a glass disk which is surface-strengthened chemically or crystallized to improve the mechanical strength.

Fig. 1.3 Definition of head/disk interface spacing parameters.

Fig. 1.4 Slider form factor historical overview

(<https://www1.hgst.com/hdd/hddpdf/tech/chart10.pdf>).

Slider size and mass have changed from the full size slider ($4.1 \times 3.2 \times 0.85$ mm) to the current Femto slider ($0.85 \times 0.70 \times 0.23$ mm) as illustrated in Fig. 1.4. The air bearing surface is a patterned surface. The flying attitude, pitch and roll angles of the slider are determined by the force balance of the generated pressure and the applied force from the suspension. The slider surface (ABS) is covered by a diamond-like-

carbon (DLC) overcoat to protect the surface from wear. The recording transducers are mounted at the end of a small ceramic slider with a carefully designed air bearings that is self pressurized by the airflow of the rapidly spinning disk.

Optimized design of this air bearing and the head disk interface (HDI) is needed to reliably provide a reliable physical spacing between the head's magnetic sensors and the disk. High density drives of the future will require that mechanical spacing, or flying height, approach zero. Similar reductions will be required in disk and head protective overcoat layers, since these layers add to the magnetic spacing.

1.3 Areal Storage Density

The areal storage density is measured in bits per unit area of disk surface. The areal storage density is the product of track density and linear density. The track density is measured in the number of tracks per unit length in the radial direction of the disk, while the linear density is measured in the number of bits per unit length in the circumferential direction. Over the past several decades, the size of disk drives has shrunk from early bulk refrigerator sized units to the current battery powered units that can fit into a shirt pocket. Area density meanwhile has increased from the order of 5 Kb/in² (kilo-bit per square inch) in 1960 to about 150 Gb/in² (giga-bit per square inch) in current drives. The flying height has been reduced significantly from about 20 μ m in 1956 to about 8~10 nm today. It is believed that using perpendicular magnetic technology and low flying height technology, 1 Tb/in² areal density with corresponding physical spacing of around 2~3 nm will be achievable in the future [4]. Fig. 1.5 shows the progress of areal density of Hitachi hard disk drives [5].

Fig. 1.5 History of areal density progress.

(<https://www1.hitachigst.com/hdd/hddpdf/tech/chart19.pdf>)

1.4 Current and Future Technologies for Areal Storage Density Increase

In recent years, the rate of growth of areal density in hard disk storage has slowed down due to a physical limit in magnetic recording [4]. The magnetic material on the disk is composed of small grains. Each bit on a conventional magnetic medium covers approximately 100 such grains to ensure reliable storage of information. However, there is a lower limit for the grain size below which the magnetization may switch randomly due to thermal excitation. This limit is referred to as the “superparamagnetic limit” [6-9]. Since the conventional longitudinal recording is approaching this limit, alternative recording technologies to further increase the areal storage density are being researched extensively in industry as well as academia. They are perpendicular magnetic recording (PMR), heat assisted magnetic recording (HAMR), patterned

media and active slider technologies.

Perpendicular magnetic recording is currently being introduced in commercially available computer hard disk drives. Heat assisted magnetic recording and patterned media technologies are still in development. The active slider technology for HDDs is one of the most important technologies to decrease the flying height. There have been proposed various kinds of actuators to drive an active slider, e.g., thermal, electromagnetic, electrostatic, and piezoelectric actuators. A thermal flying height control (TFC) slider is actuated by a thermal actuator and commercially applied in current hard disk drivers. The piezoelectric flying-height control (PFC) slider is still in investigation.

Patterned media recording technologies, thermal flying height control slider and piezoelectric flying height control slider will be investigated in this thesis.

1.4.1 Patterned Media Technology

Use of patterned media is a new approach of magnetic recording that has the potential of increasing the areal density beyond 1 Tb/in² [4, 10]. There are two approaches in patterned media recording (see Figure 1.6). One is the discrete track recording (DTR) and the other is bit patterned media (BPM). In DTR, the bits are stored on discrete circumferential tracks on the disk. Since the information is stored on discrete tracks, cross talk between bits in the radial direction on the disk can be eliminated. The width and depth of the so-called “grooves” between two such tracks are typically on the order of 40 to 100 nm.

In bit patterned media, each bit is stored on a single magnetic domain or “island” as shown in Fig. 1.6. Each island on BPM may be composed of one grain or several exchange coupled grains. Magnetic transitions no longer occur between

random grains, since distinct boundaries between islands are formed both in the circumferential and the radial directions on the disk. Since each island is a single magnetic domain, BPM is thermally stable at densities higher than can be achieved with the conventional media.

Fig. 1.6 Schematic for conventional media, discrete media and bit patterned media

(source: Hitachi GST).

1.4.2 Thermal Flying Height Control Slider

The thermal flying height control slider concept involves imbedding a separate heating element into the head structure. Fig. 1.7 shows a cross-section of the head identifying the heater location relative to the read/write elements. This separate heater structure enables the control of the read/write element protrusion independently from the effect generated by the read/write elements during read or write operations.

To supply the necessary current to the separate heating element, additional circuitry was added to the arm electronics (AE) pre-amplifier and two additional

traces were added to the head wiring. The heater current is controlled by a separate control function. There will be a heat generation inside the slider body when current is applied to the heater element. Applying heat to the head results in an increased protrusion of the read/write elements thus reducing the spacing to the disk.

Since there is some time constant before the head reaches a steady state temperature, the heater is turned on a certain time before the read or write operation is to be executed.

Once the spacing has reached the stable target value, the read or write operation can be executed. Note that during the write operation, the current to the heater is reduced when the write current is applied. This is done because the application of current to the write element causes additional localized heating which produces additional protrusion; this effect is compensated by reducing the current to the heating element. The end result is a consistent spacing of the read/write elements to the disk throughout the write operation.

Fig. 1.7 Cross-section of head structure of TFC slider.

Using a thermal actuator, the thermal flying height control (TFC) slider enables the control of the temperature distribution inside the slider thereby resulting in a larger flying height reduction [11-17]. In a recent investigation, a thermal insulator was proposed to control the temperature distribution inside the slider thereby resulting in a larger flying height reduction than that obtained with a TFC slider without insulators [18]. Moreover, the simulation results showed that two separate TFC heaters, each with an individual thermal insulator element, can further reduce the flying height at the read/write element, compared to that of a single TFC heater [19]. Dual TFC heaters and thermal insulators can provide a flexible control of flying height and thermal response time [20]. However, the incorporation of thermal insulators and dual TFC heaters can't reduce the time required for establishing thermal equilibrium. That is, the dynamic response of TFC slider is slower and it is difficult for TFC slider to catch up the dynamic variation caused by micro waviness and disk vibration.

1.4.3 Piezoelectric Flying Height Control Slider

Some active sliders with a piezoelectric actuator have been proposed. A novel concept for an active slider was proposed and experimentally successful in controlling the head/disk spacing [21], as shown in Fig. 1.8. The active material was inserted in a channel that runs across the full width of the slider at its top rear. Two deep slots defined a central bender portion that was isolated mechanically and electrically from the outer air bearing rails. When a voltage was applied to the central active material, shown here as a multi-layered insert with vertical electrodes, the material expanded in the direction of the electric field. The slider ceramic under the active material constrained this expansion and the resulting bending moment caused

the rear of this portion of the slider to move downwards. Note that the material inserted above the air bearing rails (at the sides of the slider) was not driven electrically, so the slider flying attitude remained unchanged.

Fig. 1.8 'Taildragger' slider in which the active material is laminated into a 3380 thin film head row, making a unimorph bender. The cross-section along the length of the slider in the bending section is also shown.

But it is larger than the practical sliders. After that, a series of slider designs with different PZT thin films were proposed [22-25]. However, these PZT sliders are difficult for fabrication and their air bearing stiffness becomes weak due to their structure design. Recently, a shear-mode deformation of PZT sheet was used to enable dynamic flying height control [26]. The PZT sheet is deformed in the shear mode and the flying height of the magnetic head is changed when voltage is applied between the lower and upper electrodes. The experiment results show its response time is significantly faster than that of a conventional thermal flying height control

slider.

1.5 Objectives and Scope of This Research

The purpose of this research is to study the feasibility of achieving high areal recording densities towards 10 Tbit/in² by investigating critical tribological issues of head-disk contact and utilizing flying height control sliders with piezoelectric actuator and built-in sensors.

Under very tiny magnetic spacing for high areal density magnetic recording, intermittent contact cannot be avoided. Tribological issues related to the head disk interface and pattern media technology are investigated by contact modeling analysis. The piezoelectric flying height control slider has recently been proposed in magnetic recording disk drives to reduce the flying height. In this work, the piezoelectric flying height control slider using shear-mode deformation of piezoelectric transducer (PZT) is investigated. The effects of the shear-mode deformation on the static flying attitude of the PZT slider are investigated. After that, a new novel structure design of active slider with piezoelectric actuator is proposed and studied with an optimized dimension. The active slider can effectively reduce the flying height.

On the other hand, a slider with a built-in sensor to detect contact between slider and disk is also proposed and investigated. Both numerical simulation and experimental studies are presented.

The results and insight obtained from this research will be useful in helping the data storage industry improve the performance of hard disk drive products and gain useful insight in developing higher areal recording densities.

1.6 Dissertation Structure

This dissertation is organized into eight chapters.

Chapter 1 gives an introduction to the history and structure of hard/disk drives, the head disk interface, and air bearing surface. The main current and future technologies for high areal density magnetic recording are explained and illustrated. The research objectives and dissertation outline are presented.

Chapter 2 presents a review on head-disk interface tribology in the nanometer regime.

Chapter 3 investigates the contact of the thermal flying height slider and patterned media by considering the effects of major surface condition changes of bit patterned media after surface planarization, bit pitch, and different surface patterning of bit patterned media.

Chapter 4 examines the piezoelectric flying height control slider using shear-mode deformation of piezoelectric transducer.

Chapter 5 proposes a novel structure design of an active slider with piezoelectric actuator and an optimized dimension.

Chapter 6 investigates thermal flying height control slider with a built-in contact sensor for detecting contact.

Chapter 7 presents an inexpensive and low-temperature approach for integrating piezoelectric sensor in the fabrication of current sliders. The fabricated prototype sliders have been tested experimentally.

Chapter 8 presents the conclusions and future work of this dissertation.

CHAPTER 2 Literature Review on Head/Disk Interface Tribology in the Nanometer Regime

This chapter reviews the generalized Reynolds lubrication equation which is the governing equation for the flying behavior of a slider air bearing in a hard disk drive. The contact models for single asperity are reviewed and the validity of commercial software ANSYS for head-disk contact simulation is verified. Also, piezoelectric transducers used in slider for controlling the flying height are reviewed.

2.1 The Reynolds Equation and Slider Equilibrium Equations for Head-Disk Interface

The classical Reynolds equation has been applied to predict the performance of the air bearing film operation between slider and disk surfaces. It is based on the assumptions of negligible inertial and body forces, laminar flow, Newtonian viscosity, no-slip boundary condition at the wall, and small film thickness. However, when the film thickness is much smaller than the molecular mean free path of the gas, which is very common in today's magnetic recording applications, the no-slip boundary condition at the wall is no longer satisfied. Then, the classical Reynolds equation should be modified to accommodate the slip flow. Various slip correction models have been proposed by Burgdorfer [27], Hsia and Domoto [28], Gans [29], and Mitsuya [30]. Fukui and Kaneko [31] derived a molecular gas lubrication model (FK model) based on the linearized Boltzmann equation. It is valid for arbitrary Knudsen numbers, defined as the ratio of the molecular mean free path to the film thickness, by semi-numerically calculating the flow rates of fundamental flows in the lubricant film: Poiseuille flow, Couette flow and thermal creep flow. A recent particle-based simulation study [32] confirmed the validity of the FK model for the slider-disk

spacing as low as 10 nm. In the previous derivations of the first order and the second order slip models of the generalized Reynolds equation, a length scale equal to the mean free path of the gas molecules was used in a Taylor expansion of the mean velocity field. The first order slip model comes from a first order Taylor expansion and the second order slip model comes from a second order Taylor expansion. The coefficients of the derived lubrication equation depend on that length scale. Wu [33] re-derived the first order and the second order slip models by using a somewhat more physical approach, in which the requirement of the expansion length scale to be the mean free path is relaxed. The newly derived lubrication equations have forms similar to the old first order and second order slip models, but the new second order slip model predicts results closer to the FK model when the modified inverse Knudsen number is small. The new second order slip model equation does not suffer from the pressure singularity of the FK model at contact.

2.1.1 The Generalized Reynolds Equation

The pressure distribution between the slider and the disk, p , is governed by the compressible Reynolds equation,

$$\frac{\partial}{\partial x}[ph^3 \frac{\partial p}{\partial x}] + \frac{\partial}{\partial y}[ph^3 \frac{\partial p}{\partial y}] = 6U\mu \frac{\partial}{\partial x}[ph] + 6V\mu \frac{\partial}{\partial y}[ph] + 12\mu \frac{\partial}{\partial t}[ph] \quad (2.1)$$

where h is the local slider-disk spacing; μ is the viscosity of the air; U and V are the slider velocities in the x (along slider's length direction) and y (along slider's width direction) directions, respectively. This equation is derived under the assumptions of negligible inertial and body force, laminar flow, Newtonian viscosity, no-slip boundary condition at the wall, and small film thickness.

To accommodate the slip flow, the Reynolds equation is modified based on the

linearized Boltzmann equation. By defining the following non-dimensionalized variables:

$$X = \frac{x}{L}, Y = \frac{y}{L}, H = \frac{h}{h_{0sta}}, P = \frac{p}{p_a}, T = \omega t \quad (2.2a)$$

where L, h_{0sta}, p_a, ω are the slider's length, nominal flying height, ambient pressure and the appropriate angular frequency. Then, all of above generalized forms of the Reynolds equation can be written as

$$\frac{\partial}{\partial X} [\hat{Q}PH^3 \frac{\partial P}{\partial X} - \Lambda_x PH] + \frac{\partial}{\partial Y} [\hat{Q}PH^3 \frac{\partial P}{\partial Y} - \Lambda_y PH] = \sigma \frac{\partial}{\partial T} [PH] \quad (2.2b)$$

where Λ_x and Λ_y are the bearing numbers in the x and y directions. σ is the squeeze number. The Λ_x , Λ_y , and σ are defined by

$$\Lambda_x = 6\mu UL / p_a h_{0sta}^2, \Lambda_y = 6\mu VL / p_a h_{0sta}^2, \sigma = 12\mu\omega L^2 / p_a h_{0sta}^2 \quad (2.2c)$$

$\hat{Q}$ in Eq. (2.2b) is the Poiseuille flow factor, which reflects the type of slip-flow modification used, i.e.,

$$\hat{Q} = 1, \text{ Continuum model for } H > 6400 \text{ nm}$$

$$\hat{Q} = 1 + 6a \frac{K_a}{PH}, \text{ First order slip model for } H > 400 \text{ nm}$$

$$\hat{Q} = 1 + 4a \frac{K_a}{PH}, \text{ New first order slip model}$$

$$\hat{Q} = 1 + 6a \frac{K_a}{PH} + \frac{8}{3} \left[\frac{K_a}{PH} \right]^2, \text{ 1.5 order slip model for } H > 100 \text{ nm}$$

$$\hat{Q} = 1 + 6a \frac{K_a}{PH} + 6a \left[\frac{K_a}{PH} \right]^2, \text{ Second order slip model for } H > 35 \text{ nm}$$

$$\hat{Q} = 1 + 4a \frac{K_a}{PH} + 3\left[\frac{K_a}{PH}\right]^2, \text{ New second order slip model}$$

$$\hat{Q} = 1 + f\left(\frac{K_a}{PH}\right), \text{ Fukui-Kaneko model for } H > 7 \text{ nm}$$

$$\hat{Q} = 1 + 6a \frac{K_a}{PH} + 12a\left[\frac{K_a}{PH}\right]^2, \text{ Pressure gradient slip model for } H < 7 \text{ nm} \quad (2.3a)$$

where $K_a = \lambda / h_{0,sta}$ is the Knudsen number, λ is the mean free path of the gas molecules, and $a = (2 - \alpha) / \alpha$ in which α is the surface accommodation factor.

In the new simulator, $\hat{Q}$ is simplified as

$$\hat{Q} = 1 + C_1 \frac{K_a}{PH} + C_2 \left[\frac{K_a}{PH}\right]^2 \quad (2.3b)$$

The user can define the Poiseuille flow factor by stating the parameters C_1 and C_2 .

The boundary condition for Eq. (2.2b) is $P|_{\Gamma} = 1$.

2.1.2 Flying Model of Slider and Its Dynamics

Fig. 2.1 shows the flying model of a slider on a disk. The slider suspension system is simplified into a vertical spring, which has three degrees of freedom, namely pitch angle, roll angle and flying height. The roll angle is positive when the outside rail is higher than the inside rail, and the skew angle is positive when the airflow blows from the outside rail to the inside rail.

Fig. 2.1 The flying model of a slider on a disk.

The 2-D equations of motion of the air bearing slider flying in partial contact over a rotating disk are,

$$\begin{cases} M\ddot{z} = -K_z z + \sum \iint (p - p_a) dx dy - D_z \dot{z} - F_{o-z}(z, \theta, \gamma) \\ I_\theta \ddot{\theta} = -K_\theta \theta + \sum \iint (p - p_a)(x_g - x) dx dy - D_\theta \dot{\theta} - M_{o-\theta}(z, \theta, \gamma) + \iint \tau_{wx} z_g dx dy \\ I_\gamma \ddot{\gamma} = -K_\gamma \gamma + \sum \iint (p - p_a)(y_g - y) dx dy - D_\gamma \dot{\gamma} - M_{o-\gamma}(z, \theta, \gamma) + \iint \tau_{wy} z_g dx dy \end{cases} \quad (2.4)$$

where $z = Z / h_{0sta}$, Z is absolute slider-disk spacing at position (x, y) ; h_{0sta} is the nominal slider-disk spacing; θ is the pitch angle, γ is the roll angle; M_h is the slider mass; I_θ and I_γ are the slider moments of inertia along the pitch and roll directions, respectively; K_z , K_θ and K_γ are the flexure stiffness coefficients along the z , θ and γ directions, respectively; D_z , D_θ and D_γ are the flexure damping coefficients along the z , θ and γ directions, respectively; p is the pressure distribution between the slider and the disk; and p_a is the ambient pressure. τ_{wx}, τ_{wy} are the wall shear, they are expressed as,

$$\begin{aligned} \bar{\tau}_{wx} &= w_p \frac{h}{2} \frac{\partial p}{\partial x} - w_c \frac{\mu}{h} U_x \\ \bar{\tau}_{wy} &= w_c \frac{\mu}{h} U_y - w_p \frac{h}{2} \frac{\partial p}{\partial y} \end{aligned} \quad (2.5)$$

where w_p and w_c are defined shear stress coefficients introduced to incorporate rarefaction effects [34]. The $F_{o-z}(z, \theta, \gamma)$, $M_{o-\theta}(z, \theta, \gamma)$ and $M_{o-\gamma}(z, \theta, \gamma)$ are the force or force moment caused by contact force or intermolecular surface force or lubricant along the z , θ and γ directions, respectively.

By solving the generalized Reynolds equation together with the equation of motion of the slider, the steady state flying attitude for a given slider ABS design and disk surface topology can be obtained. The solution will be used in the chapters 4 and 5 to calculate the effects of actuated deformation of slider body on the flying attitude of slider.

2.2 Asperity Contact Mechanics

Increasing recording density of magnetic disk drives requires further reduction of the spacing between the magnetic head and magnetic media (magnetic spacing). It is expected that the magnetic spacing between magnetic head and magnetic media will only be 2~3 nanometers when technology moves to 10 Tb/in². With such small a magnetic spacing, the contacts between disks and sliders will be inevitable.

Solid surfaces, irrespective of their method of formation, contain irregularities or deviations from a prescribed geometrical form. The high points on the surfaces are referred to as asperities, peaks, summits, or hills and the low points are referred to as valleys. When two nominally flat surfaces are placed in contact, surface roughness causes contact to occur at discrete contact spots (junctions). Deformation occurs in the region of the contact spots, establishing stresses that oppose the applied load. An asperity relevant for contact mechanics is defined as that which makes a contact in a particular application (contacting asperity). As the two surfaces are brought in contact, nanoscale asperities are the first to come into contact, instantly deforming plastically,

and merging to form micro- to macro contacts, which upon further application of load may deform by elastic or elastic-plastic deformation [35]. Thus, size and number of contacting asperities are dependent on the interface conditions. Both friction and wear are strongly influenced by the degree of plastic interaction of the contacting asperities.

2.2.1 Contact Models for a Single Asperity

Related to contact modeling is contact mechanics in which the two contact bodies are topographically smooth and the emphasis is on determining the relationship between the applied load, contact area, and contact stress. However, in problems involving topographically smooth surfaces the real area of contact is the same as the apparent area of contact. Real surfaces, however, always possess some degree of roughness. Thus contact between two bodies always occurs at or near the peaks of contacting asperities and so the real area of contact will generally be much less than the apparent contact area.

A single asperity contact reduces to a problem of deformation of two curved bodies in contact. For the analysis of a single asperity contact, it is convenient to model an asperity as a small spherically shaped protuberance. For surfaces with anisotropic roughness distribution asperities may be modeled with curved bodies of specific geometries.

Consider two rough solid bodies brought into physical contact through the action of applied forces. Contact between the two bodies occurs over many small areas, each of which constitutes a single asperity contact. It is necessary to relate the force acting on a single asperity to its deformation and contact area. The well-known solution of this problem was developed in the late nineteenth century by Hertz [36].

The hertz solution provides closed-form expressions to deformations and stresses of

two spheres in a purely elastic contact with perfectly smooth surfaces. The two spheres may have different radii and different elastic properties. The Hertz assumptions are:

- (1) The contact area is elliptical;
- (2) Each body is approximated by an elastic half-space loaded over the plane elliptical contact area;
- (3) The dimensions of the contact area must be small compared to the dimensions of each body and to the radii of curvature of the surfaces;
- (4) The strains are sufficiently small for linear elasticity to be valid; however, caution must be used in applying the results of the theory to low modulus materials like rubber where it is easy to produce deformations which exceed the restriction to small strains.
- (5) The contact is frictionless, so that only a normal pressure is transmitted.

Fig. 2.2 Contact between two elastic spherical objects.

Assume that the spheres are squeezed together by the force W . The deformation field in the solids can be determined by minimizing the elastic deformation energy. Due to the spherical shape of the asperity, the contact pressure is not uniform over the contact area but reaches the maximum Hertzian contact pressure (p_0) at the center of the contact area and falls to zero at the edge of the contact area. The radii of the spherical asperity of each body are denoted by R_1 and R_2 , respectively. The radius a of the circular contact region (see Fig. 2.2) is given by

$$a^3 = \frac{3WR}{4E^*} \quad (2.6)$$

where

$$\frac{1}{E^*} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \quad (2.7)$$

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \quad (2.8)$$

$$p(r) = p_0 \sqrt{1-r^2/a^2} \quad (2.9)$$

$$W = \int_0^a 2\pi r p(r) dr = \frac{2}{3} p_0 \pi a^2 \quad (2.10)$$

The mean contact pressure is

$$p_m = \frac{2}{3} p_0 = W / \pi a^2 \quad (2.11)$$

where E_1 and E_2 are the elastic moduli of the solid and ν_1 and ν_2 are corresponding Poisson ratios. The distance s the two solids approach each other is given by

$$s = \frac{a^2}{R} = \left(\frac{9W^2}{16RE^{*2}} \right)^{1/3} \quad (2.12)$$

For the special case (Fig. 2.3), the closed-form solutions render an equivalent case where a single elastic sphere, having an equivalent elastic modulus, E^* , and an equivalent radius, R , is in contact with a rigid flat. The interference, ω , can be described as the distance the sphere is displaced normally into the rigid flat. The area A_E of the contact zone can be calculated as

$$A_E = \pi a^2 = \pi R \omega \quad (2.13)$$

The normal load F_E between a single asperity and a flat surface can then be obtained as

$$F_E = \frac{4}{3} E^* \sqrt{R} (\omega)^{3/2} \quad (2.14)$$

$$\frac{1}{E^*} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \quad (2.15)$$

Fig. 2.3 Schematic view of a spherical asperity in contact with a flat surface.

Hertz [37] only considered the elastic deformation of a spherical asperity and a flat surface. As the normal load between a spherical asperity and a plane increases, plastic deformation can occur at the contact interface. Greenwood and Williamson [38] introduced a contact model which accounts for the plastic deformation of the contact area. They assumed that the maximum contact pressure p_0 is proportional to the

surface hardness H . Defining K as the ratio of the maximum contact pressure and the surface hardness, the critical value w_c (critical interference) for plastic deformation to occur is given by

$$\omega_c = \left(\frac{\pi}{2}\right)^2 \left(\frac{KH}{E}\right)^2 R \quad (2.16)$$

The normal load needed for the critical value ω_c to be reached is [32]

$$W = \frac{4}{3} ER^{\frac{1}{2}} \omega_c^{\frac{3}{2}} \quad (2.17)$$

Tabor [39] showed experimentally that the onset of plastic flow occurs when the maximum contact pressure p_0 between a sphere and a plane reaches a value of $0.6H$, where H is the hardness. Greenwood and Williamson [38] used the information from Tabor's experiment and obtained the critical value of displacement for which plastic deformation occurs at the contact between a spherical asperity and a flat surface as

$$\omega_c = \left(\frac{\pi}{2}\right)^2 \left(\frac{0.6H}{E}\right)^2 R \approx 0.89 \left(\frac{H}{E}\right)^2 R \quad (2.18)$$

Here, the contact area and contact load is calculated in the same way as in Hertz's model.

Chang et al. [40-41] reinterpreted the concept from Greenwood and Williamson's model and proposed a volume-conservation asperity-based model (CEB model), which was an attempt to overcome the shortcoming that contact is purely plastic or purely elastic by combining the two above-mentioned extreme approaches (Fig. 2.4). In their model the asperity deformation law is purely elastic or purely plastic and the model "switches" between these two modes. The "switch" is the critical interference value, which is estimated from equalization of plastic flow pressure and maximum hertzian

pressure. This model is based on volume conservation of an assumed plastically deformed region of the asperity. It is assumed that asperity deformation is localized mainly in the vicinity of the contact. For interferences smaller than the critical interference ω_c , the contact is elastic and follows a Hertz solution. For $\omega > \omega_c$, the contact area of a plastically deformed asperity is

$$A = \pi R \omega \left(2 - \frac{\omega_c}{\omega}\right) \quad (2.19)$$

Thus, the contact load for such an asperity is

$$W = \pi R \omega \left(2 - \frac{\omega_c}{\omega}\right) K H \quad (2.20)$$

Fig. 2.4 Deformation of an asperity using a volume conservation model [40].

Kogut and Etsion [42] further improved the elastic-plastic contact model of a spherical asperity contact with a flat surface. They separated the contact in four regions ($\omega'/\omega_c < 1$; $1 \leq \omega'/\omega_c < 6$; $6 \leq \omega'/\omega_c < 110$; $110 \leq \omega'/\omega_c$) and used curve fitting of the contact pressure, contact area and contact load as a function of the dimensionless interference ω'/ω_c to predict elastic contact, elastic-plastic contact and plastic contact conditions.

2.2.2 Finite Element Model by ANSYS

As reviewed above, the Hertz model [37] only considered the elastic deformation of a spherical asperity. The CEB model proposed by Chang et al. [40-41] was based on volume conservation of an assumed plastically deformed region of the asperity. They have the limitation to investigate the head-disk interface in the nanometer spacing with the elastic-plastic deformation due to the head-disk contact. Actually, the commercial software ANSYS was commonly used to model the elastic-plastic material response for the single asperity model.

In this part, a finite element method by ANSYS was used to investigate the case of an elastic-perfectly plastic sphere in frictionless contact with a rigid flat (shown in Fig. 2.5). The von Mises criterion defined the yielding of the material.

To improve upon the efficiency of computation, an axisymmetric 2-D model was used. A commercial package ANSYS 11 was used to solve the non-linear contact problem. First, two-dimensional axis symmetric model was developed to form the single asperity contact. The hemisphere was discretized by PLANE 82 elements; the region of most interest was adjacent to the contact interface and had the greatest concentration of elements for lower interferences. Away from the contact region, the mesh became coarser to minimize the computational effort. To detect contact between the hemisphere and the rigid flat, two-dimensional surface-to-surface contact elements, CONTA 172 and TARGET 169 were employed. The bilinear isotropic hardening `_BISO_` option in the ANSYS program was chosen to account for the elastic-plastic material response for the single asperity model. The rate-independent plasticity algorithm incorporated the von Mises criterion and isotropic strain hardening behavior was used for the analysis. The von Mises criterion defined the yielding of the material.

The material properties assigned were Young's modulus $E=207000 \text{ N/mm}^2$, Poisson's ratio $\nu=0.3$, and yield strength $Y=400 \text{ N/mm}^2$. Tangent modulus is assumed as zero.

The finite element model was validated for mesh and step convergence by varying the number of elements and step sizes. The resulting mesh consisted of 16849 elements in total. The mesh consisted of approximately 21140 nodes in total. The nodes at the bottom boundary of the mesh were constrained against displacement in the y-direction and the nodes at the centre of bottom are fixed against displacement in the x-direction. This boundary condition may be valid for the modeling of asperity contacts for two reasons: (1) The asperities are actually connected to a much larger bulk material at the base and will be significantly restrained there, and (2) since the high stress region occurs near the contact, the boundary condition at the base of the hemisphere will not greatly affect the solution because of Saint Venant's principle. The contact force acting on the hemisphere was added on the centre of rigid body. The radius of the contact area was determined by finding the edge of the contact, or the location of the last activated contact element. The finite element mesh is shown in Fig. 2.5. There are two ways to simulate the contact problem. The first applies a force to the rigid body and then computes the resulting displacement. The second applies a displacement and then computes the resulting contact force. In both methods, the displacement, stress, and strain in the elastic body can be determined, as well as the contact pressure. In this model the first approach was used. The contact problem and the elasto-plastic material property made the analysis highly nonlinear and difficult to converge. An iterative scheme was used to solve for the solution, and several load steps were used to enhance solution convergence. Initially, a small load was set of the total load and then it was incremented after the load step converges. ANSYS internally controlled the load stepping to obtain a converged solution by using the bisection method. This continued

until a converged solution was found for the desired load.

Fig. 2.5 Elasto-plastic hemispherical contact with a rigid flat.

Fig. 2.6 Finite element mesh of elasto-plastic hemispherical contact with a rigid flat by ANSYS.

2.2.3 Comparison of Different Solutions

As the load applied over the asperities increases, a plastic region is developed at the subsurface and occurs beneath the surface, which is in an elastic state. On further loading or as the interference increases, the plastic region first reaches the surface of

the sphere. This state of transition or regime is termed as elastic-plastic state: yielding will initiate at that point and an elastic core forms at the junction of the contacting region, which is enveloped by the plastic region. Further increasing the load, the plastic region grows gradually shrinking the elastic core at the surface. At some point the elastic core is completely eliminated and the fully plastic region forms on the surface, which is the elastic-plastic to full plastic transition. The resulting stress distribution behavior is explained in Fig. 2.7. Initially the plastic region is developed below the subsurface and occurs beneath the surface; it is shown in Fig. 2.7(a). Then the elastic core developed on the contact interface region is enveloped by the plastic region; at that point yielding initiates and the resulting plot is shown in Fig. 2.7(b). As the load increases, the plastic region also increases. The inner elastic core disappears and the fully plastic region prevails over the contact region, as shown in Fig. 2.7(c).

(a)

(b)

Fig. 2.7 Von Mises stress at different interference: (a) plastic region develops in the subsurface; (b) plastic region initially touched the surface; (c) elastic core completely disappears; (d) plastic region dominates the asperity.

The normalized maximum von Mises stress is plotted in Fig. 2.8 as function of load force F . For relatively shallow indentation, the maximum von Mises stress increases monotonically with load. For $F > 5$ N, the maximum von Mises stress reaches the yield strength of the material and a small plastic zone develops adjacent to the contact region.

Fig. 2.8 Maximum von Mises stress/yield strength versus load force.

In order to validate the model, mesh convergence must be satisfied. Besides, the model solution was compared with the Hertz elastic solution. Fig. 2.8 shows that when $F < 5$ N, it is an elastic deformation. The comparison of the relationship of indenting depths and load between finite element results and Hertz solution is shown in Fig. 2.9. It can be found that there is no significant difference between these two results. So in the elastic regime the results of these two models are close.

Fig. 2.10 shows indenting depths at different load in the plastic regime, it can be found that the finite element results and Hertz solution close to each other in the elastic regime. After that, they depart from each other.

Fig. 2.9 Indenting depths at different loads for elastic regime.

Fig. 2.10 Indenting depths at different loads.

Fig. 2.11 Indenting depths at different loads of three models.

Fig. 2.11 shows that the indenting depths of plastic CEB model are larger than the FEM model. It results from the false volume-conservation mechanism assumed in the CEB model in which asperity deformation is localized mainly in the vicinity of the contact. In contrast, a purely elastic Hertz solution underestimates values obtained from FEM analysis.

The Hertz model only considered the elastic deformation of a spherical asperity which underestimated the results in some degree. The CEB model assumed plastically deformed region of the asperity which overestimated the results. The commercial software ANSYS was validated and chosen to investigate the elastic-plastic deformation due to head-disk contact in next chapter.

2.3 Piezoelectric Transducers Used in Slider for Flying Height Control

In 1880, Pierre Curie and his brother Paul-Jaques discovered that an external force applied to certain crystals can generate a charge on the surface of the crystal

found to be roughly proportional to the applied mechanical stress [43]. One year later, they discovered the opposite also holds true; an applied voltage generates a deformation of the crystal. This interaction between electrical and mechanical systems is termed “piezoelectricity”. The prefix “piezo-“ is derived from the Greek word “press”. The direct piezoelectric effect refers to the generation of an electric polarization due to a mechanical stress. Conversely, the indirect piezoelectric effect refers to the generation of a strain in a material due to the application of an electric field.

A simplified model of piezoelectricity involves the concept of anions (-) and cations (+) moving in opposite directions under the influence of an electric field or mechanical force. The forces generated by this motion cause lattice deformations for ionic crystals lacking a center of symmetry.

Fig. 2.12 shows what happens when a piezoelectric crystal is deformed. As a mechanical deformation (stress) is applied to the crystal, electric dipoles are generated due to the shifting of anions and cations and a potential difference develops. It should be noted that the potential difference in piezoelectric materials is a function of the changing stress; therefore, piezoelectric materials are typically used in dynamic situations.

Fig. 2.12 Piezoelectricity in an ionic crystal.

Several ceramic materials have been described as exhibiting a piezoelectric effect. These include lead-zirconate-titanate (PZT), lead-titanate (PbTiO_2), lead-zirconate (PbZrO_3), and barium-titanate (BaTiO_3). These ceramics are not actually piezoelectric but rather exhibit a polarized electrostrictive effect. A material must be formed as a single crystal to be truly piezoelectric. Ceramic is a multi crystalline structure made up of large numbers of random orientated crystal grains. The random orientation of the grains results in a net cancelation of the effect. The ceramic must be polarized to align a majority of the individual grain effects. The term piezoelectric has become interchangeable with polarized electrostrictive effect in most literature.

The piezoelectric effect, relates mechanical effects to electrical effects. These effects, as shown above, are highly dependent upon their orientation to the poled axis. It is, therefore, essential to maintain a constant axis numbering scheme.

Fig. 2.13 Axis nomenclature

The piezoelectric constants relating the mechanical strain produced by an applied electric field are termed the strain constants, or the "*d*" coefficients. The units may then be expressed as meters per meter, per volts per meter (meters per volt).

$$d_{ab} = \frac{\text{strain development}}{\text{applied electric field}} \quad (2.21)$$

where *a* is electrical direction, *b* is mechanical direction. The large *d_{ij}* constants relate to large mechanical displacements which are usually sought in motional transducer devices. Conversely, the coefficient may be viewed as relating the charge collected on the electrodes, to the applied mechanical stress. *d₃₃* induced polarization in direction 3 (parallel to direction in which ceramic element is polarized) per unit stress applied in direction 3 or induced strain in direction 3 per unit electric field applied in direction 3 (Fig. 2.14(a)). *d₃₁* induced polarization in direction 3 (parallel to direction in which ceramic element is polarized) per unit stress applied in direction 1 (perpendicular to direction in which ceramic element is polarized) or induced strain in direction 1 per unit electric field applied in direction 3 (Fig. 2.14(b)). *d₁₅* induced polarization in direction 1 (perpendicular to direction in which ceramic element is polarized) per unit shear stress applied about direction 2 (direction 2 perpendicular to direction in which

ceramic element is polarized) or induced shear strain about direction 2 per unit electric field applied in direction 1 (Fig. 2.14(a)).

Fig. 2.14 Basic piezoelectric modes (<http://piezo.com/catalog8.pdf%20files/Cat8.59-61.pdf>).

Although the magnitudes of piezoelectric voltages, movements, or forces are small, and often require amplification (a typical disc of piezoelectric ceramic will increase or decrease in thickness by only a small fraction of a millimeter, for example) piezoelectric materials have been adapted to an impressive range of applications. The piezoelectric effect is used in sensing applications, such as in force or displacement sensors. The inverse piezoelectric effect is used in actuation applications, such as in motors and devices that precisely control positioning, and in generating sonic and ultrasonic signals.

Some active sliders with a piezoelectric actuator have been proposed. Yeack-Scranton et al. [21] proposed a novel concept for an active slider and succeeded in experimentally controlling the head/disk spacing. Their active slider had bulk stacked lead–zirconate–titanate (PZT) in the grooves fabricated on the surface opposite the ABS. However, it is not available in commercial HDDs, because it is larger than the

current sliders and it is difficult to process the top surface of the slider. Some active sliders with a PZT unimorph actuator have been proposed. Tagawa et al. [22] proposed a slider using double-layered composite PZT thin film. Kurita et al. [23] proposed a slider using a PZT sheet. Suzuki and Kurita [24] proposed a slider using PZT sol-gel thin film. Although these active sliders with PZT/silicon unimorph actuator were as large as the current sliders, they could not be adapted to the conventional slider process, because they required processing on the surface opposite the ABS. The PZT thin-film actuator also requires processing at over 600°C which is around 400°C higher than the upper temperature limit in the conventional process. Their air bearing stiffness between the ABS and disk are also weak because they have deep grooves. Chapter 4 to Chapter 6 will investigate piezoelectric flying height control slider and build-in contact sensor in the thermal flying height control slider.

2.4 Summary

The modified Reynolds lubrication equation is presented with various slip correction model and dynamics model. By solving the generalized Reynolds equation together with the equation of motion of the slider, the steady state flying attitude for a given slider ABS design and disk surface topology can be obtained. The finite element method by commercial software ANSYS is used to simulate an elastic-plastic contact analysis for single asperity. In order to verify its validity, the results are compared with Hertz solution and CEB solution. In addition, active sliders with a piezoelectric actuator in the literature are reviewed.

CHAPTER 3

Finite Element Analysis of Contact Performance of Patterned Media Based Head-Disk Systems

Bit patterned media is a promising approach for further increasing the recording density of future magnetic disk drives. Another important factor for future high density recording is that the magnetic spacing between the head-slider and disk-media will be further reduced. As a result, the chance of slider-disk contact will be increased and the stress caused by such contact becomes an important issue for future disk drives. This chapter reports understanding of stress distribution due to asperity micro contacts. A two dimensional finite element model between head and patterned media was developed. The thermal protrusion of a thermal flying height control (TFC) slider and the wave surface of an elastic-plastic thin film were chosen to perform a contact analysis at the head-disk interface (HDI). It investigated the effects of major surface condition changes of bit patterned media after surface planarization, bit pitch, and different surface patterning of bit patterned media. Simulation results for the contact pressure at asperity micro contacts and subsurface von Mises equivalent stress were presented.

3.1 Introduction

Discrete track recording (DTR) and bit patterned media Recording (BPM) are promising new approaches for future high density magnetic data recording [44-47]. Increasing recording density of magnetic disk drives requires further reduction of the spacing between the magnetic head and magnetic media (magnetic spacing). It is expected that the magnetic spacing between magnetic head and magnetic media will only be 2~3 nanometers when technology moves to 10 Tb/in². With such small a

magnetic spacing, the contacts between disks and sliders will be inevitable.

It is well known that the contact induced interface failure is one of the leading causes of the hard disk drive failure. Bit-pattern media tends to increase the surface roughness/waviness. Such increased roughness and waviness increases the chance of undesirable head-disk contact, head crash, surface wear, and wear induced-interface failure [48]. Also, such contacts/crash induce stress-strain inside the patterned media which can result in interface failure and data loss. Hence, detailed knowledge of the contact stress at asperity micro contacts of such pattern media based head-disk interface is essential.

Contact between engineering surfaces has been a subject of considerable interest and intensive research. Early analyses were focused on contact problems dealing with idealized smooth surfaces, such as cylinders and spheres, and isotropic elastic half-spaces, the emphasis being on analytical methods yielding solutions for the contact pressure distribution [49-50]. Progress in computer technology and numerical techniques have enabled the study of contacting solid possessing more realistic material behaviors. Kral et al [51] analyzed repeated indentation of an elastic-plastic half-space by a rigid sphere and presented finite element results showing the variation of the contact pressure and plastic deformation in the subsurface with the number of loading cycles (the contact cycles) and the likelihood for the material to reach a steady-state elastic behavior. King [52] analyzed indentation of elastic layered media and obtained the relationship between the effective elastic modulus in terms of the layer thickness and the elastic properties of the layers and substrate materials. Komvopoulos [53-54] performed finite element analysis of normal contact on elastic and elastic-plastic layered media and analyzed the contact pressure distribution of multi-layered media. The influence of interactions between neighboring asperities on the resulting

contact deformation has been demonstrated by Berthe Vergne [55] and Komvopoulos et al [56]. Z. Q. Gong and Komvopoulos [57] performed a plane-strain finite element analyses to study the effect of surface geometry on contact deformation of elastic-plastic layered media and contact pressure distribution. Yeo and Polycarpou also used finite element analysis to study elastic contact behavior of patterned media [58]. Besides, Yeo and Polycarpou have performed parametric study to improve contact and wear performance of discrete track recording media [59].

3.2 Mechanical Performance Study of Patterned Media Based Head-Disk Systems

The previous studies have provided useful insight into the contact stress and strain fields of elastic-plastic layered media and the role of surface geometry on contact deformation of homogeneous media. However, the magnetic media is of multi-layer structure. The elastic-plastic deformation of such multi-layered media, especially multi-layered bit patterned media, has not been analyzed yet.

An elastic-plastic contact analysis for the multi-layered bit patterned media is presented in this part. The numerical analysis is based on the finite element analysis method. The major surface condition of bit patterned media after surface planarization is classified into two cases, the bit protrusion case and the bit recession case. The relationship between contact pressure and indenting depths are obtained for the above mentioned two typical patterned media cases.

3.2.1 Modeling of Patterned Media Based Head-Disk Systems

A schematic for the slider/patterned media interface is shown in Fig. 3.1. Disk is modeled as elastic-plastic multi-layered structure. The overall mechanical properties of different layers are the same as what were used by other researchers for continuous

media [60-61], except for the properties of the fill-in material. The continuous magnetic layer is assumed to be etched to the magnetic recording layer only. The etched out area is filled with material such as SiO₂, planarized and covered by carbon overcoat as a surface protection layer. The etch depth is assumed to be at magnetic layer only and this is for the convenience of analysis. The thickness of carbon overcoat and magnetic layer is 1 nm and 10 nm, respectively [57][62]. Material properties used in this modeling work are listed in Table 3.1.

The mechanical properties of the magnetic bit area and the fill-in area are different. As a result, the two areas may not be of the same height. Such situation is modeled as “bit protrusion case” and “bit recession case”, respectively, as illustrated in Fig. 3.1. For the bit protrusion case, the magnetic bit area (bit area) is the protruded area and the filled non-magnetic area (fill-in area) is the recess area. The bit recession case is just the opposite: the filled non-magnetic area is the protruded area and the bit area is the recession area. As a first order approximation, such surface protrusion-recession is approximated by a sinusoidal curve.

The thermally actuated flying height control (TFC) slider is used for this analysis. The contact area on slider surface is limited to the thermal protruded area only. It is assumed that such a protruded area of the slider body moves towards disk surface and causes stress distribution inside the patterned media.

A commercial package ANSYS 11.0 was used to solve the non-linear contact problem. Fig. 3.2 shows the two-dimensional finite element models of patterned media head-disk system. The possible contact area of the thermal protruded slider is approximated by a sphere surface. Due to symmetry, only one-half of such spherical contact area and layered medium were modeled, in order to reduce the computation time. The slider body is of significantly higher hardness and stiffness. Therefore, the

slider is assumed to be rigid. Each layer of the finite element mesh is modeled as elastic-plastic material having elastic modulus and yield strength equal to those of its bulk value. Based on the usual plane strain assumption, the patterned media were discretized by PLANE 82 elements. The region adjacent to the contact interface is of the greatest concentration of elements. Away from the contact region, the mesh becomes coarser to minimize the computational time. The contact elements used in this modeling work are those two-dimensional three node surface-to-surface contact elements (CONTA 172 and TARGET 169 in ANSYS). The bilinear isotropic hardening option in the ANSYS program is chosen to account for the elastic-plastic material response for this modeling. The rate-independent plasticity algorithm incorporates the von Mises criterion and isotropic strain hardening behavior is used for the analysis. The von Mises criterion defines the yielding of the material. Totally, 21140 elements and 67090 nodes are used in the analysis.

The nodes at the bottom boundary of the mesh are constrained against displacement in the y-direction and the nodes at the vertical boundaries of the mesh are constrained against displacement in the x-direction. The finite element model is validated for mesh and step convergence by varying the number of elements and step sizes. The maximum number of sub-steps in computation is 1000 for very large interference. Force is applied to the rigid slider body and simulation is performed by calculating the resulting displacement.

(a)

(b)

Fig. 3.1 Schematic illustration of slider-pattern media interface: (a) bit-protrusion case (the protruded area is the magnetic bit area) and (b) bit recession case (the protruded area is the filled nonmagnetic area).

Fig. 3.2 Finite element mesh of layered patterned medium where the surface protrusion/ recession is approximated by a sinusoidal surface.

Table 3.1 Mechanical properties of various layers.

Materials	Elastic modulus (MPa)	Yield strength (MPa)	Poisson's ratio
Carbon	168000	13000	0.3
Magnetic	130000	2670	0.3
Fill-in	73000	8400	0.17

3.2.2 Results and Discussion

The relationship between contact force (load) and indenting depths (displacement) of the two typical pattern media cases is shown in Fig. 3.3. It can be observed that the bit protrusion case is of larger stiffness than the bit recession case, if the indentation depth is not too high. This is because the Young's modulus of the bit material (the magnetic material) is larger than that of the fill-in material and, therefore, the indenting depths of the bit-protrusion area is smaller than that of the fill-in protrusion (bit recession) case.

Fig. 3.3 Load versus indenting depths of the two typical patterned media cases.

(a)

(b)

Fig. 3.4 Relationship between maximum von Mises stress and indenting depths for two typical pattern media cases: (a) the bit protrusion case and (b) the bit recession case.

Fig. 3.4 (a) and (b) shows the maximum von Mises stress at different indenting depth of two typical patterned media cases. It is indicated that, there is no significant difference between these two cases for carbon layer. However, for magnetic layer, the maximum von Mises stress of the bit protrusion case reached the yield strength earlier than the bit recession case. This is because the yield strength of the magnetic media material (bit material) is smaller than that of the fill-in material.

(a)

(b)

Fig. 3.5 Contact pressure distribution for the bit protrusion case: (a) shallow indentation and (b) deep indentation.

Fig. 3.5 shows contact pressure distribution of the bit protrusion case. It is indicated that for very shallow indentation ($d=0.14$), the contact pressure distribution is similar to the profile predicted by the Hertz theory. However, increasing the

indentation depth causes the maximum contact pressure to shift toward the edge of the contact area. Further increase of the indentation depth produces pressure spikes at the edges of the contact area because of the development of a plastic zone in the second soft layer.

(a)

(b)

Fig. 3.6 Contact pressure distribution for the bit recession case: (a) shallow indentation and (b) deep indentation.

Contact pressure distribution of the bit recession case is shown in Fig. 3.6. It is indicated that for a shallow indentation, increasing the indentation depth causes the maximum contact pressure grow bigger, because in the second soft layer, the deformation is still elastic. However, further increase of the indentation depth produces other two small pressure peaks at the two edges of the contact area.

In conclusion, it's necessary to develop contact models that account for the actual surface topographies and materials used in high-performance disk drives for pursuing higher magnetic recording densities. In this part, a contact analysis of the slider-pattern media interface was performed using a surface topography description and mechanical properties resembling those of magnetic recording heads and thin-film rigid disks. The pattern media surface was approximated by a sinusoidal curve. Finite element results for the contact pressure at elastic-plastic asperity micro contacts and subsurface stress were presented. Two typical patterned media cases, the bit protrusion case and the bit recession case were studied. Based on the presented results and discussion, the following main conclusions can be drawn.

(1) The bit protrusion case is of larger stiffness than the bit recession case, if the indentation depth is not too high. The corresponding indenting depths of the bit-protrusion area is smaller than that of the fill-in protrusion (bit recession) case.

(2) For carbon layer, there is no significant difference between these two cases. However, for magnetic layer, the maximum von Mises stress of the bit protrusion case reached the yield strength earlier than that of the bit recession case.

(3) For the bit protrusion case, the contact pressure distribution of the bit protrusion case is similar to the profile predicted by the Hertz theory, if the indentation depth is small. However, increasing the indentation depth causes the maximum contact pressure

to shift toward the edge of the contact area. Further increase of the indentation depth produces pressure spikes at the edges of the contact area.

(4) For the bit recession case, increasing the indentation depth causes the maximum contact pressure grow bigger and further increase of the indentation depth produces other two small pressure peak at the edges of the contact area.

3.3 Effects of Bit Pitch on Mechanical Performance of Patterned Media

An elastic-plastic contact analysis for the multi-layered bit-patterned media is presented in this part. The numerical analysis is based on the finite element analysis method. The major surface condition changes of bit-patterned media after surface planarization can be classified into two cases, the large density case (the small pitch case) and the small density case (the large pitch case). The relationship between the contact pressure and indenting depths are obtained for the above mentioned two pitch cases.

3.3.1 Modeling of Layered Patterned Medium with Different Pitch

A schematic for the slider/patterned media interface is shown in Fig. 3.7. The patterned disk is modeled as elastic-plastic multi-layered structure. The overall mechanical properties of different layers are the same as what were used by other researchers for continuous media [58], except the properties of the fill-in material. The continuous magnetic layer is etched to magnetic bits and the etched out area is filled with the material such as SiO_2 , planarized and covered by carbon overcoat. The etch depth is assumed to be at magnetic layer only, for the convenience of analysis. The thickness of carbon overcoat and magnetic layer is chosen to be 1 nm and 16 nm, respectively [61-62]. The material properties used in this modeling work are listed in

Table 3.2 [63].

Different areal recording densities correspond different pitch of bit-patterned media. As a result, two cases of recording densities are considered, “0.5 Tb/in²” and “2 Tb/in²”. Such situation is modeled as “the large pitch case” and “the small pitch case”, respectively. For the large pitch case, the size of pitch is 36 nm, and the dot size is 18 nm. For the small pitch case, the size of pitch is 18 nm, and the dot size is 9 nm. Besides, the mechanical properties of the magnetic bit area and the fill-in area are different. Therefore, the two areas may not be of the exact same height. In this work, only the bit protrusion case is considered [64]. The magnetic bit area is the protruded area and the filled non-magnetic area is the recess area. As a first order approximation, such a surface of protrusion-recession is approximated by a sinusoidal curve.

The thermally actuated flying height control (TFC) slider is used for this analysis. The contact area on slider surface is limited to the thermally protruded area only. It is assumed that such a protruded area of the slider body moves towards disk surface and causes a stress distribution inside the patterned media. The thermally protruded area of the slider is assumed to be a rigid cylinder of 10 μm .

The commercial software ANSYS 11.0 was used to solve the non-linear contact problem. Fig. 3.8 shows the two-dimensional finite element models of patterned media head-disk system. The possible contact area of the thermally protruded slider is approximated by a sphere surface. Due to symmetry, only one-half of such spherical contact area and layered medium were modeled, in order to reduce the computation time. The slider body is of significantly higher hardness and stiffness. Therefore, the slider is assumed to be rigid. Each layer of the finite element mesh is modeled as elastic-plastic material having elastic modulus and yield strength equal to those of its bulky value. Based on the usual plane strain assumption, patterned media were discrete

by PLANE 82 elements. The region adjacent to the contact interface is of the greatest concentration of elements. Away from the contact region, the mesh becomes coarser to minimize the computational time. The contact elements used in this modeling work were those two-dimensional three node surface-to-surface contact elements (CONTA 172 and TARGET 169 in ANSYS). The bilinear isotropic hardening option in the ANSYS program is chosen to account for the elastic-plastic material response for this modeling. The rate-independent plasticity algorithm that incorporates the von Mises criterion and the isotropic strain hardening behavior is used for the analysis. The von Mises criterion defines the yielding of the material. Totally, 24286 and 33095 elements and 72240 and 98281 nodes are used in the analysis.

The nodes at the bottom boundary of the mesh are constrained against the displacement in the y-direction and the nodes at the vertical boundaries of the mesh are constrained against the displacement in the x-direction. The finite element model is validated for mesh and step convergence by varying the number of elements and step sizes. The maximum number of sub-steps in computation is 1000 for very large interference. The load force is applied to the rigid slider body and the simulation is performed by calculating the resulting displacement.

Fig. 3.7 Schematic illustration of slider-patterned media interface.

Fig. 3.8 Finite element mesh of layered patterned medium with the protrusion/ recession approximated by a sinusoidal surface (a) the small pitch case and (b) the large pitch case.

Table 3.2 Mechanical properties of various layers.

Materials	Elastic modulus (MPa)	Yield strength (MPa)	Poisson's ratio
Carbon	168000	13000	0.24
Magnetic	150000	3400	0.3
Fill-in	83000	4300	0.23
IL	135000	3900	0.3

3.3.2 Results and Discussion

In this work, only results of the worst contact between the protruded asperity and the slider are presented. The relationship between the contact force (load) and indenting depths (displacement) of the two pitch cases is shown in Fig. 3.9. It can be observed that the small pitch case is of smaller stiffness than the large pitch case. This is because contact area is small for the small pitch case and, therefore, the indenting depth of the large pitch case is smaller than that of the small pitch case.

Fig. 3.9 Relationship of load versus indenting depths with the two typical pitch cases.

(a)

(b)

Fig. 3.10 Relationship between maximum von Mises stress and contact force for the two typical patterned media cases, (a) the small pitch case and (b) the large pitch case.

Fig. 3.10(a) and 3.10(b) shows the maximum von Mises stress under different contact force of two pitch cases. It is indicated that for the carbon layer, there is no significant difference between these two cases. However, for the magnetic layer, the

maximum von Mises stress of the small pitch case reached the yield strength earlier than that of the large pitch case. This is because the contact area of the small pitch case is smaller than that of the large pitch case. Therefore, the small pitch case subjects a larger stress.

(a)

(b)

Fig. 3.11 Contact pressure distribution at the protruded asperity for the small pitch case:

(a) shallow indentation and (b) deep indentation.

Fig. 3.11 shows contact pressure distribution in the middle of asperity of the small pitch case. It is indicated that for a shallow indentation ($d=0.25$ nm), the contact pressure distribution is similar to the profile predicted by the Hertz theory. However, increasing the indentation depth causes the maximum contact pressure to shift toward the edge of the contact area. Further increase of the indentation depth produces pressure spikes at the edges of the contact area because of the development of a plastic zone in the second soft layer.

(a)

(b)

Fig. 3.12 Contact pressure distribution at the second asperity for the bit recession case

(a) shallow indentation, and (b) deep indentation.

Contact pressure distribution at the middle asperity of the large pitch case is shown in Fig. 3.12. It is indicated that for the shallow indentation, increasing the indentation depth causes the maximum contact pressure growing bigger, similar to the small pitch case. Further increase of the indentation depth produces two small pressure spikes at the edges of the contact area because of the development of a plastic zone in the second soft layer.

In this part, a contact analysis of the HDI was performed. The patterned media surface was approximated by a sinusoidal curve. Two typical pitch cases, the small pitch case and the large pitch case were studied. Based on the presented results and discussion, the following main conclusions can be drawn.

(1) The small pitch case is of larger stiffness than the large pitch case.

(2) For carbon layer, there is no significant difference of the maximum von Mises stress distribution between these two cases. However, for magnetic layer, the maximum von Mises stress of the small pitch case reached the yield strength earlier than the large pitch case. This is because the contact area of the small pitch case is smaller than that of the large pitch case.

(3) For the contact pressure distribution in the middle of asperity, the two typical pitch cases are similar. In the shallow indentation ($d=0.14$ nm) stage, the contact pressure distribution of the two typical pitch case is similar to the profile predicted by the Hertz theory. However, increasing the indentation depth causes the maximum contact pressure to shift toward the edge of the contact area. Further increase of the indentation depth produces pressure spikes at the edges of the contact area.

3.4 Effect of Surface Patterning on Mechanical Performance of Patterned Media

An elastic-plastic contact analysis for the multi-layered bit-patterned media is presented in this part. The numerical analysis is based on the finite element analysis method. Here, the formed surface condition of single bit-patterned media after surface planarization can be classified into two cases, single protrusion case and double protrusion case. The relationship between contact pressure and indenting depths are obtained for the above mentioned two cases.

3.4.1 Modeling Procedure

A schematic for the slider/patterned media interface is shown in Fig. 3.13. Disk is modeled as elastic-plastic multi-layered structure. The overall mechanical properties of different layers are the same as what were used by other researchers for continuous media [58], except the properties of the fill-in material. The continuous magnetic layer is etched to magnetic bits and the etched out area is filled with material such as SiO₂, planarized and covered by carbon overcoat. The etch depth is assumed to be at magnetic layer only, for the convenience of analysis. The thickness of carbon overcoat and magnetic layer is 2 nm and 16 nm, respectively [61-62]. Material properties used in this modeling work are listed in Table 3.3 [63]. Besides, the mechanical properties of the magnetic bit area and the fill-in area are different. Therefore, the two areas may not be of the exact same height. In this work, only bit protrusion case is considered [64]. The magnetic bit area is the protruded area and the filled non-magnetic area is the recess area.

For a given areal recording densities, after planarization process of magnetic layer, different surface patterning of bit protrusion will be formed. As a result, two typical

surface patterning cases are considered, “single protrusion case” and “double protrusion case”, respectively, as illustrated in Fig. 3.13. For such condition, the areal recording density is 2 Tb/in², the size of pitch is 18 nm, the dot size is 9 nm.

The thermally actuated flying height control (TFC) slider is used for this analysis. The contact area on slider surface is limited to the thermal protruded area only. It is assumed that such a protruded area of the slider body moves towards disk surface and causes stress distribution inside the patterned media.

A commercial package ANSYS 11.0 was used to solve the non-linear contact problem. Fig. 3.14 shows the two-dimensional finite element models of patterned media head-disk system. The possible contact area of the thermal protruded slider is approximated by a sphere surface. Due to symmetry, only one-half of such spherical contact area and layered medium were modeled, in order to reduce the computation time. The slider body is of significantly higher hardness and stiffness. Therefore, the slider is assumed to be rigid. Each layer of the finite element mesh is modeled as elastic-plastic material having elastic modulus and yield strength equal to those of its bulk value. Based on the usual plane strain assumption, patterned media were discretized by PLANE 82 elements. The region adjacent to the contact interface is of the greatest concentration of elements. Away from the contact region, the mesh becomes coarser to minimize the computational time. The contact elements used in this modeling work were of those two-dimensional three node surface-to-surface contact elements (CONTA 172 and TARGET 169 in ANSYS). The bilinear isotropic hardening option in the ANSYS program is chosen to account for the elastic-plastic material response for this modeling. The rate-independent plasticity algorithm that incorporates the von Mises criterion and isotropic strain hardening behavior is used for the analysis. The von Mises criterion defines the yielding of the material. Totally, 24286 and 17569

elements and 72240 and 23672 nodes are used in the analysis.

The nodes at the bottom boundary of the mesh are constrained against displacement in the y-direction and the nodes at the vertical boundaries of the mesh are constrained against displacement in the x-direction. The finite element model is validated for mesh and step convergence by varying the number of elements and step sizes. The maximum number of sub-steps in computation is 1000 for very large interference. Force is applied to the rigid slider body and simulation is performed by calculating the resulting displacement.

Fig. 3.13 Schematic illustration of slider-pattern media interface: (a) single protrusion case and (b) double protrusion case.

(a)

(b)

Fig. 3.14 Finite element mesh of layered pattern medium where the surface protrusion/recession is approximated by a sinusoidal surface: (a) single protrusion case and (b) double protrusion case.

Table 3.3 Mechanical properties of various layers.

Materials	Elastic modulus (MPa)	Yield strength (MPa)	Poisson's ratio
Carbon	168000	13000	0.24
Magnetic	150000	3400	0.3
Fill-in	83000	4300	0.23
IL	135000	3900	0.3

3.4.2 Results and Discussion

The relationship between contact force (load) and indenting depths (displacement) of the two patterning cases is shown in Fig. 3.15. It can be observed that there is no big difference of single protrusion case and double protrusion case, because the pitch of that two cases are the same.

Fig. 3.15 Relationship of load versus indenting depths with the two typical surface patterning cases.

(a)

(b)

Fig. 3. 16 Relationship of maximum von Mises stress and different contact forces with the two typical pattern media cases: (a) single protrusion case and (b) double protrusion case.

Fig. 3.16(a) and Fig. 3.16(b) shows the maximum von Mises stress at different contact forces of two surface patterning cases. It is indicated that for magnetic layer,

there is no significant difference between these two cases. However, for carbon layer, the maximum von Mises stress of double protrusion case reached the yield strength earlier than the single protrusion case. This is because the contact area of the double protrusion case is smaller than that of single protrusion case. Therefore, the double protrusion case subjected larger load per square nanometers than single protrusion case.

Fig. 3.17 Contact pressure distribution at the middle asperity for the single protrusion case: (a) shallow indentation and (b) deep indentation.

Fig. 3.17 shows contact pressure distribution at the middle asperity of the single protrusion case. It is indicated that for very shallow indentation ($d=0.25$ nm), the contact pressure distribution is similar to the profile predicted by the Hertz theory. However, increasing the indentation depth causes the maximum contact pressure to shift toward the edge of the contact area. Further increase of the indentation depth produces pressure spikes at the edges of the contact area because of the development of a plastic zone in the second soft layer.

(a)

(b)

Fig. 3.18 Contact pressure distribution at the second asperity for double protrusion case: (a) shallow indentation and (b) deep indentation.

Contact pressure distribution at the middle asperity of the double protrusion case is shown in Fig. 3.18. It is indicated that for shallow indentation, increasing the indentation depth causes the maximum contact pressure grow bigger, similar to the single protrusion case. Further increase of the indentation depth produces two small pressure spikes at the edges of the contact area because of the development of a plastic zone in the second soft layer.

In conclusion, two typical surface patterning cases, single protrusion case and double protrusion case were studied. Based on the presented results and discussion, the following main conclusions can be drawn.

(1) There is no big difference of single protrusion case and double protrusion case.

(2) For magnetic layer, there is no significant difference between these two cases.

However, for carbon layer, the maximum von Mises stress of double protrusion case

reached the yield strength earlier than the single protrusion case. This is because the contact area of the double protrusion case is smaller than that of single protrusion case.

(3) For the contact pressure distribution at the middle asperity, the two typical pitch cases are similar. In the shallow indentation ($d=0.25$ nm) stage, the contact pressure distribution of the two typical pitch case is similar to the profile predicted by the Hertz theory. However, increasing the indentation depth causes the maximum contact pressure to shift toward the edge of the contact area. Further increase of the indentation depth produces pressure spikes at the edges of the contact area.

3.5 Summary

A contact analysis of the slider-patterned media interface is performed using a surface topography description and mechanical properties resembling those of magnetic recording heads and thin-film rigid disks. The patterned media surface is approximated by a sinusoidal curve. Typical patterned media cases, the bit protrusion case, the bit recession case, the small pitch case, the large pitch case, single protrusion case and double protrusion case are studied. Finite element results for the contact pressure at elastic-plastic asperity micro contacts and subsurface stress distribution are presented.

CHAPTER 4

Piezoelectric Flying Height Control Slider Using Shear-Mode Deformation

The piezoelectric flying height control slider has recently been implemented in magnetic recording disk drives to reduce the flying height. This chapter examines the piezoelectric flying height control slider using shear-mode deformation of piezoelectric transducer (PZT). A finite element model of the PZT slider using shear-mode deformation is built, and the electromechanical simulation and air-bearing simulation is performed to investigate the effects of the shear-mode deformation on the static flying attitude of the PZT slider. Parametric simulation of the location of PZT sheet and air bearing surface (ABS) of slider is performed to achieve a low flying height and robust head-disk interface.

4.1 Introduction

The active slider technology has recently been implemented in magnetic recording disk drives to reduce the flying height at the read/write element. The two typical actuators to drive an active slider are thermal actuator and piezoelectric actuator.

Using a thermal actuator, the thermal flying height control (TFC) slider enables the control of the temperature distribution inside the slider, thereby resulting in a greater flying height reduction [11–17]. In a recent investigation, a thermal insulator was proposed to control the temperature distribution inside the slider, thereby resulting in a greater flying height reduction than that obtained with a TFC slider without insulators [18]. Moreover, the simulation results showed that two separate TFC heaters, each with an individual thermal insulator element, can further reduce the flying height at the read/write element, when compared with that of a single TFC heater [19]. Dual TFC

heaters and thermal insulators can provide a flexible control of flying height and thermal response time [20]. However, the incorporation of thermal insulators and dual TFC heaters do not reduce the time for establishing thermal equilibrium. In other words, the dynamic response of TFC slider is slower, and it is difficult for the TFC slider to catch up the dynamic variation caused by micro waviness and disk vibration.

Some active sliders with a piezoelectric actuator have already been proposed. A novel concept for an active slider was proposed earlier and was successful in experimentally controlling the head/disk spacing [21]; however, it was larger than the practical sliders. Subsequently, a series of slider designs with different PZT thin films were proposed [22–25]. Nevertheless, these PZT sliders are difficult for fabrication and their air-bearing stiffness becomes weak owing to their structure design. Recently, a shear-mode deformation of PZT sheet was used to enable dynamic flying height control [26]. The PZT sheet was deformed in the shear mode and the flying height of the magnetic head was changed when the voltage was applied between the lower and upper electrodes. The experiment results showed that its response is significantly faster than that of a conventional TFC slider.

4.2 Simulation of Piezoelectric Flying Height Control Slider Using Shear-Mode Deformation

In this part, we examined the piezoelectric flying height control slider using shear-mode deformation of piezoelectric transducer (PZT). We built a finite element model of a piezoelectric flying height control slider using shear-mode deformation. By applying a drive voltage, we simulated the shear-mode deformation of PZT slider and the corresponding flying attitude of the slider.

4.2.1 Finite Element Model of PZT Slider Using Shear-Mode Deformation

A schematic model of a PZT slider is shown in Fig. 4.1. A three-pad conventional slider of a standard "Pico-slider" size ($1.25 \times 1 \times 0.3 \text{ mm}^3$) was chosen. A PZT sheet (NEC/Tokin N-21) [65] was designed between the substrate and basecoat in the conventional slider, and its distance to the trailing edge of slider (D) was $42.8 \text{ }\mu\text{m}$. The thickness of the PZT sheet (L) was chosen to be $200 \text{ }\mu\text{m}$, and its width and height were designed as the same as those of the slider.

Fig. 4.1 Schematic model of PZT slider using shear-mode deformation.

When applying a drive voltage on the PZT sheet, the PZT sheet was found to have a shear-mode deformation Δx [65–66] towards the disk, which resulted in the change of flying attitude of the slider.

$$\Delta x = d_{15} \times V \quad (4.1)$$

where V is the drive voltage and d_{15} is the piezoelectric charge constant of the shear-mode deformation. In the simulation, we chose substrate side as the upper electrode and basecoat side as the lower electrode.

Figure 4.2 shows the cross-sectional view of the finite element model of the PZT

slider using the commercial software ANSYS. It comprised the head elements of pole tips, shields, writer coil, and PZT sheet. The ANSYS element types of SOLID98 and SOLID226 were chosen for the slider and PZT sheet, respectively. Tables 4.1 and 4.2 list the material properties of PZT slider components used in the finite element simulation [13, 15, 18].

Fig. 4.2 Finite element model of PZT slider (half slider plotted).

Table 4.1 Material properties of the slider used in ANSYS.

Materials	Young's modulus (GPa)		Poisson's ratio	Density (kg/m ³)
Al ₂ O ₃ TiC (Substrate)	390		0.22	2200
Al ₂ O ₃ (Basecoat)	138		0.25	2200
NiFe (pole, shield)	200		0.3	700
Cu (writer coil)	130		0.34	8960
PZT (NEC Tokin N-21)	Y ₁₁	61	0.34	7820
	Y ₃₃	50		

Table 4.2 Electrical properties of PZT sheet used in ANSYS.

PZT	Piezoelectric charge constant (e-12 m/V)		Piezoelectric voltage constant (e-3 V·m/N)		Dielectric constants	
NEC Tokin N-21	d ₃₁	-198	g ₃₁	-12.1	ϵ_{33}/ϵ_0	1800
	d ₃₃	417	g ₃₃	25.4		
	d ₁₅	711	g ₁₅	41.0	ϵ_{11}/ϵ_0	2000

4.2.2 Simulation Results

We first performed the electromechanical simulation of PZT slider under a given drive voltage to obtain the piezoelectric shear-mode deformation of the slider body. The deformed air-bearing surface of the PZT slider can be used to calculate the steady-state flying characteristics of the PZT slider using air-bearing simulation [67]. Subsequently, the effects of the drive voltage on the static flying attitude of the slider were investigated.

4.2.2.1 Electromechanical Simulation of PZT Slider

Figure 4.3 shows the finite element simulation results of the slider deformation under a drive voltage of 80 V. It can be seen that the shear-mode deformation of the slider body is gradually increased from the substrate side to basecoat side. Furthermore, it can be observed that the deformation of the trailing part is maximum at 72.9 nm towards the disk.

Fig. 4.4 shows the shear-mode deformation profile of the PZT slider under the

drive voltage of 80 V. The profile is linear along the thickness direction of the PZT sheet. Moreover, from the trailing edge of the slider to the location of the PZT sheet (42.8 μm to the trailing edge), the deformation is constant with a value of 72.9 nm.

(a) Drive voltage (V)

(b) Structural deformation of the whole slider (μm)

(c) Cross-sectional view of structural deformation (μm)

Fig. 4.3 Structural deformation of PZT slider under the voltage of 80 V.

Fig. 4.4 Deformation profile of PZT slider under the voltage of 80 V.

4.2.2.2 Static Air Bearing Simulation of PZT Slider under a Voltage of 80 V

Fig. 4.5 compares the air-bearing pressure distribution on the air-bearing surfaces (ABS) of the conventional slider and PZT slider. The maximum pressure is 1.65×10^6 Pa on the ABS of the conventional slider, and is located at the trailing edge of the rear pad. On the ABS of the PZT slider, the distribution of the peak pressure obviously has

a wide shape pattern, and the maximum pressure is increased to 3.74×10^6 Pa, which is about 2.3 times greater than that of the conventional slider. This is because the trailing part of the slider is vertically shifted towards the disk owing to the shear-mode deformation of the PZT sheet. Correspondingly, the air-bearing pressure profiles have been compared in Fig. 4.6. It can be observed that the PZT slider has a narrower but sharper pattern of pressure profile, when compared with the conventional slider.

Fig.4.7 compares the flying height profiles of the conventional slider and PZT slider. The lowest flying height of the conventional slider is 8.40 nm and the pitch angle is $97.66 \mu\text{rad}$ in the steady state. By applying a voltage of 80 V on the PZT sheet of the PZT slider, the lowest flying height is reduced to 4.24 nm and the pitch angle is significantly reduced from $97.66 \mu\text{rad}$ to $5.29 \mu\text{rad}$.

Fig. 4.5 Air-bearing pressure distribution on air-bearing surface.

Fig. 4.6 Comparison of air-bearing pressure profiles.

(a)

Fig. 4.7 Comparison of flying height profiles.

4.2.2.3 Shear-Mode Deformation and Variation of Flying Attitude at Different Drive Voltages

Fig. 4.8 shows the shear-mode deformation profiles of PZT slider under the voltages from 30 to 120 V. It can be seen that all deformation profiles show a similar pattern. Furthermore, the maximum deformation is observed to be proportional to the drive voltage, as indicated by Eqn. (4.1).

With an increase in the drive voltage, the pattern of the air-bearing pressure on the trailing part of the slider becomes narrower and the peak value of the air-bearing pressure is gradually increased (Fig. 4.9). It can be found that the pressure at the leading side of the trailing part is significantly increased with the increase in the drive voltage. Especially, in the cases of 100 and 120 V, a pressure peak appears at the leading side of the trailing part of the slider, as well as at the lowest flying height (shown in Fig. 4.10).

Fig. 4.10 shows that when the drive voltage is beyond 80 V, the flying height of the

PZT slider is not significantly reduced like that in the cases of 30, 50, and 80 V. The slider is rotated to fly at a reduced pitch angle or even at a negative pitch angle, rather than a further reduced flying height. For example, in the cases of drive voltages of 100 and 120 V, the flying height at the leading side of the trailing part of the slider is smaller than that at the trailing side. This indicates that the slider has a rotation to be balanced at a negative pitch angle. The results of negative pitch angles at the drive voltages of 100 and 120 V, shown in Fig. 4.11, confirm this point. This may be owing to the fact that the very strong air-bearing stiffness at low flying height resists the trailing part of the slider further dropping-down towards disk.

Fig. 4.11 also indicates that the PZT slider is gradually rotated from a positive to a negative pitch angle with an increase in the drive voltage on the PZT sheet. This phenomenon can be explained in view of the balance of the air-bearing force on the slider, as shown in Fig. 4.12. For a steady flying of the slider, the forces acting on the slider should be in the balance state, that is,

$$\begin{aligned}
 F_{suspension} &= F_{left} + F_{right} \\
 F_{left} > F_{right} &: \text{positive pitch angle} \\
 F_{left} < F_{right} &: \text{negative pitch angle}
 \end{aligned} \tag{4.2}$$

At a high drive voltage, the shear-mode deformation of the trailing part of the slider is large and uniquely approached to the disk. The air-bearing force on the trailing part ($F_{trailing}$) takes a primary role to balance the push-down force from the suspension ($F_{suspension}$). On the other hand, the rest of the slider (side pads, leading pads, etc.) move away to the disk owing to a large shear-mode deformation of the slider, that is, $F_{leading}$ is gradually reduced. The variation of $F_{trailing}$ and $F_{leading}$ will cause the change in the weight factor of F_{left} and F_{right} , which determine the pitch angle of the flying slider. When the drive voltage reaches a certain value (about 80 V for present design), the

slider is rotated counterclockwise and balanced at a small negative pitch angle (cases of 100 V and 120 V) owing to $F_{left} < F_{right}$.

The reason for the successfully flying of the slider with a small negative pitch angle should be the fact that the large and wide shear-mode deformation of the trailing part of the slider can still hold sufficient air-bearing force, in association with the weak supporting from the rest of the parts of the slider (side pads, leading pads, etc.). However, this kind of negative pitch angle flying is not applicable to the present hard disk drive.

Hence, a high drive voltage may cause the rotation of the PZT slider and slider flying at small/negative pitch angle. A parametric study (such as the thickness of PZT sheet (L), the distance of PZT sheet to the trailing edge of slider (D), the ABS design, etc.) needs to be performed in future works to achieve a robust head-disk interface and optimize the power efficient, that is, achieving a large flying height reduction at a low power cost.

Fig. 4.8 Comparison of deformation profiles under different drive voltages.

Fig. 4.9 Comparison of air-bearing pressure profiles under different drive voltages.

Fig. 4.10 Comparison of flying height profiles under different drive voltages.

Fig. 4.11 Pitch angle of the PZT slider under different drive voltages.

Fig. 4.12 Mechanism underlying variation of flying attitude of the PZT slider.

4.3 Parametric Simulation of Piezoelectric Flying Height Control Slider using Shear-Mode Deformation

In this part, electromechanical and air-bearing simulations were performed to investigate the effects of shear-mode deformation on the static flying attitude of a PZT slider. The locations of the PZT sheet and the air bearing surface (ABS) of the slider were investigated to achieve a low flying height and robust head-disk interface.

4.3.1 Finite Element Model of PZT Slider and Air Bearing Surface

A schematic model of a PZT slider is shown in Fig. 4.13. A three-pad conventional slider of a standard "Pico-slider" size ($1.25 \times 1 \times 0.3 \text{ mm}^3$) was chosen. A PZT sheet (NEC/Tokin N-21) [66] was designed between the substrate and basecoat in the conventional slider, and its distance to the trailing edge of slider was denoted as ' D '. The thickness of the PZT sheet (L) was chosen to be $200 \text{ }\mu\text{m}$, and its width and height were designed same as those of the slider.

Fig. 4.13 Schematic model of PZT slider using shear-mode deformation.

When applying a drive voltage on the PZT sheet, the PZT sheet was found to have a shear-mode deformation [65-66] towards the disk, which resulted in the change of flying attitude of the slider (Eqn. (4.1)).

Fig. 4.14 shows a cross-sectional view of the finite element model of PZT slider using the commercial software ANSYS. It comprises the head elements of pole tips, shields, writer coil and PZT sheet. The ANSYS element types of SOLID98 and SOLID226 were chosen for the slider and the PZT sheet respectively. The material properties of the PZT slider components used in the finite element simulation are given in [13, 15, 18]. The reference design of the ABS and its parameters used in the

simulation are shown in Fig. 4.15. To study the effects of the ABS design on the flying attitude of PZT sliders, four sliders were designed by modifying the geometric dimensions of the trailing pad of the reference design, as listed in Table 4.3.

Fig. 4.14 Finite element model of PZT slider (half slider plotted).

Dimension (mm)	1.25×1×0.3
Minimal FH (nm)	9.1
Pitch (rad)	108.7e-6
Roll (rad)	-0.7e-6
First Etch Depth (μm)	0.15
Second Etch Depth (μm)	23.5

Fig. 4.15 Reference air bearing surface design (Slider 0).

Table 4.3 The slider designs investigated in this work.

	Length ($L1$) (μm)	Length ($L2$) (μm)	Width (W) (μm)	Depth (h) (μm)
Slider 0	250	190	180	0.15
Slider 1	200	140	155	0.15
Slider 2	150	90	130	0.15
Slider 3	250	190	180	0.25
Slider 4	250	190	180	0.35

4.3.2 Simulation Results

We first performed the electromechanical simulation of PZT slider under a given drive voltage to obtain the piezoelectric shear-mode deformation of slider body. The deformed air-bearing surface of the PZT slider was used to calculate the steady-state flying characteristics of the PZT slider using air-bearing simulation [67]. Subsequently, the effects of drive voltage on the static flying attitude of the slider were investigated.

4.3.2.1 Distance of PZT Sheet from Trailing Edge

The effects of the location of PZT sheet inside the sliders on the flying performance of the sliders were investigated. Three slider designs were simulated and the distance of PZT sheet to the trailing edge of the sliders (D) was chosen to be 42.8 μm , 142.8 μm and 242.8 μm , respectively.

Fig.4.16 shows the voltage actuated deformation profiles of the sliders with different locations of PZT sheet. The maximum deformation of sliders towards disk is the same (27.34 nm) for all the cases due to same drive voltage of 30 V at PZT sheet.

Fig. 4.17 compares the air bearing pressure profiles of three sliders. It is found that a ‘shoulder’ pattern appears when the drive voltage is applied at the PZT sheet since the slider body from its trailing edge to the PZT sheet is actuated and deformed towards the disk. Fig. 4.18 shows the corresponding flying height profiles of the sliders. The case of ‘ $D=42.8 \mu\text{m}$ ’ achieves the lowest flying height. In the cases of ‘ $D=142.8 \mu\text{m}$ ’ and ‘ $D=242.8 \mu\text{m}$ ’, the minimal flying height is even little increased compared to the conventional slider without the PZT application. This is because the voltage actuated deformation of the slider body towards disk compresses the air flow and results in an increased air bearing pressure/force under it. The short distance of the PZT sheet to the trailing edge of the slider makes it easy to drop the actuated trailing

part of the slider, that is, achieve a low flying height and vice versa.

Fig. 4.19 shows the pitch angles of the sliders after applying a 30 V drive voltage at the PZT sheet. It is observed that the reduction in pitch angle is almost the same for the three slider designs.

The reduction in the pitch angle of PZT slider can be explained by the balance of the air bearing force acting on the air bearing surface of the slider. The air bearing pressure/force acting on the trailing part of the slider is increased since the trailing part is actuated towards disk when a drive voltage is applied at the PZT sheet. On the other hand, the air bearing force acting on the leading part of the slider is reduced since the voltage actuated deformation of the slider towards the disk increases the distance between the leading part of the slider and the disk. To achieve a force and force moment balance between the air bearing force and the suspension force, the slider is rotated by some degree depending on the drive voltage applied to the PZT sheet.

Fig. 4.16 Comparison of the deformation profiles of the slider body at different PZT locations.

Fig. 4.17 Comparison of the air bearing pressure profiles at different PZT locations.

Fig. 4.18 Comparison of the flying height profiles at different PZT locations.

Fig. 4.19 Comparison of the pitch angles at different PZT locations.

4.3.2.2 Dimensions of the Center-Trailing Pad

The effects of the dimensions of center-trailing pad on the flying performance of sliders were investigated. Three slider designs, namely, ‘Slider 0’, ‘Slider 1’ and ‘Slider 2’ with a distance of 42.8 μm from PZT sheet to the trailing edge of the sliders were simulated. The length and width of the center-trailing pad were reduced from ‘Slider 0’ to ‘Slider 2’ as listed in Table 4.3. But the locations of the trailing edge of the central-trailing pad as well as the remaining parts of the sliders were kept unchanged. The original flying attitude of the three sliders was designed to be similar by adjusting the flying conditions.

Fig. 4.20 compares the air bearing pressure profiles of the three slider designs at a 30 V drive voltage. It can be seen that the ‘Slider 2’ with the smallest center-trailing pad has the largest maximum air bearing pressure among the three slider designs. Fig. 4.21 compares the corresponding flying height profiles of the three slider designs. It can be seen that ‘Slider 2’ achieves the lowest flying height among the three slider designs. The reductions in pitch angles for ‘Slider 0’, ‘Slider 1’ and ‘Slider 2’ are 35.89 μrad, 36.95 μrad and 38.07 μrad respectively. The difference between them is small

compared to their original pitch angles. This means that a small center-trailing pad can help to achieve a low flying height for the piezoelectric flying height control slider, but cannot prevent the reduction in pitch angle.

Fig. 4.20 Comparison of the air bearing pressure profiles of different ABS designs
($D=42.8 \mu\text{m}$).

Fig. 4.21 Comparison of the flying height profiles of different ABS designs ($D=42.8 \mu\text{m}$).

Fig. 4.22 Comparison of the pitch angles of different ABS designs ($D=42.8 \mu\text{m}$).

4.3.2.3 Depth of the Center-Trailing Pad

The effects of the depth of the center-trailing pad on the flying performance of the sliders were investigated. The dimension of the central-trailing pad of the slider was kept unchanged, but its depth was increased from $0.15 \mu\text{m}$ of ‘Slider 0’ to $0.25 \mu\text{m}$ of

‘Slider 3’ and $0.35\ \mu\text{m}$ of ‘Slider 4’. The original flying attitude of the three sliders was designed to be similar by adjusting the flying conditions.

At a high drive voltage of 120 V, the peak of the air bearing pressure acting on the actuated part of the center-trailing pad of the slider shifts from the trailing side to the leading side as shown in Fig. 4.23. With an increase in drive voltage from 0 V to 120 V, the minimal flying height of ‘Slider 0’ is significantly reduced initially, and then the slider starts rotating (Fig. 4.24) owing to the high air-bearing stiffness at low flying height region.

Similarly, ‘Slider 3’ and ‘Slider 4’ are gradually rotated clockwise when the drive voltage at the PZT sheet is continuously increased (from Fig. 4.25 to Fig. 4.28). In contrast to the ‘Slider 0’, the peak of the air bearing pressure and the minimal flying height of ‘Slider 3’ and ‘Slider 4’ remain on the trailing side of the center-trailing pad even when the drive voltage is increased to 120 V.

Fig. 4.23 The air bearing pressure profiles of ‘Slider 0’ when the drive voltage is increased from 0 V to 120 V.

Fig. 4.24 The flying height profiles of ‘Slider 0’ when the drive voltage is increased from 0 V to 120 V.

Fig. 4.25 The air bearing pressure profiles of ‘Slider 3’ when the drive voltage is increased from 0 V to 120 V

Fig. 4.26 The flying height profiles of ‘Slider 3’ when the drive voltage is increased from 0 V to 120 V.

Fig. 4.27 The air bearing pressure profiles of ‘Slider 4’ when the drive voltage is increased from 0 V to 120 V.

Fig. 4.28 The flying height profiles of 'Slider 4' when the drive voltage is increased from 0 V to 120 V.

Fig. 4.29 Comparison of pitch angles when the drive voltage is increased from 0 V to 120 V.

Fig. 4.29 shows a linear reduction in the pitch angles of the sliders with an increase in drive voltage on the PZT sheet. Specially, the pitch angle of 'Slider 0' is reduced below zero (a negative pitch angle) at the drive voltage of 120 V, since its original

pitch angle is smallest among the three sliders. Hence, a large original pitch angle is needed to keep the slider flying with a positive pitch angle at the high drive voltage. Moreover, the reduction ratios for the three sliders are almost the same as shown in Fig. 4.29. This means that even a large depth of the center-trailing pad cannot prevent the reduction in pitch angle of the piezoelectric flying height control slider.

4.4 Summary

This chapter examined the piezoelectric flying height control slider using shear-mode deformation. A finite element model of a PZT slider using shear-mode deformation was built. The electromechanical simulation and air-bearing simulation were performed to investigate the shear-mode deformation and the static flying attitude of the PZT slider, and the effects of location of the PZT sheet and air bearing surface on the static flying attitude of the PZT slider.

The electromechanical simulation results show that the flying height and pitch angle of the PZT slider are significantly reduced with the increase in the drive voltage on the PZT sheet. However, beyond the drive voltage of 80 V for the proposed PZT slider, the reduction in the flying height of the PZT slider is found to be limited owing to the high air-bearing stiffness at low flying height region. Furthermore, the PZT slider can be rotated and balanced at a negative pitch angle.

The parametric simulation results show that a short distance of the PZT sheet to the trailing edge of the slider can help to achieve a low flying height. A small center-trailing pad of the slider can also help to achieve a low flying height, but cannot prevent the reduction in pitch angle. The depth of center-trailing pad does not change the reduction ratio of the pitch angle when increasing the drive voltage. A big pitch angle value is needed to avoid pitch angle falling below zero at a high drive voltage.

CHAPTER 5

Novel Active Slider with Piezoelectric Actuator Using Transverse-Mode Deformation

This chapter proposes a novel structure design of active slider with a piezoelectric actuator at the top of the slider body and a soft layer between the substrate and head elements. A parametric simulation was performed to obtain an optimum dimension of PZT actuator and the most efficient soft material. Then three types of slider structures were proposed and investigated with considering air bearing effects to achieve large actuation efficiency.

5.1 Introduction

The active slider technology has recently been implemented in magnetic recording disk drives to reduce the flying height at the read/write element. Some active sliders with a piezoelectric actuator have already been proposed. A novel concept for an active slider was proposed earlier and was successful in experimentally controlling the head/disk spacing [21]; however, it was larger than the practical sliders. Subsequently, a series of slider designs with different PZT thin films were proposed [22-25]. Nevertheless, these PZT sliders are difficult for fabrication and their air-bearing stiffness becomes weak owing to their structure design. Recently, a shear-mode deformation of PZT sheet was used to enable dynamic flying height control [26]. The PZT sheet was deformed in the shear mode and the flying height of the magnetic head was changed when the voltage was applied between the lower and upper electrodes. The experiment results showed that its response time is significantly faster than that of a conventional thermal flying height control slider. After that, a piezoelectric flying height control slider using shear-mode deformation of piezoelectric transducer (PZT)

was modeled [68-69]. An electromechanical simulation and air-bearing simulation was performed to investigate the effects of the shear-mode deformation on the static flying attitude of the PZT slider. The parametric simulation of the location of PZT sheet and air bearing surface (ABS) of slider were performed to achieve a low flying height and robust head-disk interface.

In this part, a novel structure design of active slider with piezoelectric actuator. A parametric simulation was performed to obtain an optimum dimension of PZT actuator and the most efficient soft material. Then three types of slider structures were proposed and investigated with considering air bearing effects to achieve large actuation efficiency.

5.2 Finite Element Model of PZT Actuated Slider

A Femto-slider ($0.85 \times 0.7 \times 0.23$ mm) was chosen in this investigation as shown in Fig. 5.1. A layer of soft material was built between the substrate and head elements. The thickness of soft material was denoted as t . A layer of thin film PZT was bonded at the top of the slider body. The length, width and thickness of PZT actuator were denoted as L , W and T , respectively. The bottom and top surfaces of PZT actuator were defined as lower and upper electrodes. The polar direction of PZT actuator was defined in Y direction. A driving voltage was added between two electrodes, thus the PZT actuator deflected as a cantilever beam. The leading (left) side of slider was fixed in all directions as the boundary conditions of the simulation.

Fig. 5.1 Schematic model of the PZT actuated slider.

Fig. 5.2 shows the finite element model of PZT slider using the commercial software ANSYS. It comprises the head elements of pole tips, shields, writer coil and PZT actuator. The ANSYS element types of SOLID98 and SOLID226 were chosen for the slider and the PZT actuator respectively. The material properties of the PZT slider components used in the finite element simulation are [13, 15, 65, 66]. The material properties for three types of soft materials used in the simulation are listed in Table 5.1.

We performed the electromechanical simulation of PZT slider under a given driving voltage to obtain the piezoelectric transverse-mode deformation of the slider body. Fig. 5.3 shows the deflection of the slider body under a driving voltage of 10 V. The maximum deflection of slider trailing edge towards disk is 18.113 nm. Such a deflection will compress the air flow between slider and disk and reduce the flying height.

Table 5.1 Material properties of candidate soft layers.

	Polymer	Young's Modulus (Gpa)	Density (kg/m ³)	Poisson's ratio	Fabrication
PTFE	Polytetrafluoroethylene	0.41	2150	0.46	RF sputtering
PMMA	Polymethylmethacrylate	1.8~3.1	1170	0.35~0.4	Casting
PVDF	PolyvinylideneFluoride	2	1780	0.225	Sputtering

Fig. 5.2 Finite element model of the PZT actuated slider by ANSYS.

Fig. 5.3 Structural deflection of slider under a driving voltage of 10 V.

5.3 Parametric Simulation of PZT Actuator and Soft Layer

The dimension of PZT actuator was optimized to obtain a large deflection of slider trailing edge. Fig. 5.4 shows the effect of PZT width on the deflection of slider trailing edge towards disk under a driving voltage of 10 V at the PZT actuator. The PTFE was used as soft material and its thickness was chosen to be 150 μm . The length of PZT actuator was chosen to be 250 μm and thickness was chosen to be 10 μm . The width of PZT actuator was increased from 100 μm to 700 μm . It can be seen that increasing the width of PZT actuator from 100 μm to 700 μm , the deflection of slider trailing edge will be increased from 6.7 nm to 15 nm. Therefore, a large width will be efficient to produce the deflection.

Fig. 5.4 The effect of PZT width on the deflection of slider trailing edge under 10 V driving voltage.

Fig. 5.5 The effect of PZT thickness on the deflection of slider trailing edge.

Fig. 5.5 shows the effect of PZT thickness on the deflection of slider trailing edge towards disk under a driving voltage of 10 V at the PZT actuator. In this case, the PTFE was used as soft material and its thickness was chosen to be 150 μm . The length of PZT actuator was chosen to be 250 μm and width was chosen to be 700 μm . The thickness of PZT actuator was increased from 10 μm to 60 μm . It can be seen that increasing the thickness of PZT actuator, the deflection of slider trailing edge will be decreased.

Fig. 5.6 The effect of PZT length on the deflection of slider trailing edge.

Fig. 5.6 shows the effect of PZT length on the deflection of slider trailing edge towards disk under a driving voltage of 10 V at the PZT actuator. In this case, the PTFE is used as soft material and its thickness was chosen to be 150 μm. The thickness of PZT actuator was chosen to be 10 μm and width was chosen to be 700 μm. The length of PZT actuator was increased from 250 μm to 850 μm. It can be seen that increasing the length of PZT actuator, the deflection of slider trailing edge will be increased. Therefore, the length of 850 μm is used in the following work.

Fig. 5.7 shows the effect of soft material thickness on the deflection of slider trailing edge under a driving voltage of 10 V. In this case, the PTFE was chosen as soft material. The dimension of PZT actuator (length*width*thickness) was chosen to be 850*700*10 μm. The thickness of PTFE was increased from 50 μm to 200 μm. It can be seen that increasing the thickness of PTFE, the deflection of slider trailing edge will be increased.

Fig. 5.7 The effect of soft material thickness on the deflection of slider trailing edge.

Fig. 5.8 compares the deflection of slider trailing edge with different types of soft materials. In this case, the soft materials including PVDF, PTFE, PMMA were used as the filling materials and its thickness was chosen to be 150 μm . And the case without filling soft material was also investigated. The dimension of PZT sheet (length*width*thickness) was chosen to be 850*700*10 μm . It can be seen that the filling material with small Young's modulus of material will obtain a large deflection of slider trailing edge.

Fig. 5.8 Comparison of deflection of slider trailing edge with different types of soft materials.

5.4 Optimization of Slider Structure for High Actuation Efficiency

Three types of slider structures were proposed to achieve a high actuation efficiency which was defined as the ratio of the reduced flying height to deflection of slider trailing edge (Fig. 5.9). A layer of soft material was built between the substrate and head elements. A PZT actuator was bonded at the top of slider body. The dimension of PZT actuator (length*width*thickness) was chosen to be 850*700*10 μm . The PTFE was used as soft material and its thickness was chosen to be 150 μm . The difference of three structures is the height and location of the soft material. In the structure 1, the height of soft material was designed as the same as the slider height. In the structure 2 and structure 3 the height of soft material was chosen to be 100 μm , and was located aligned with top and bottom surface of slider, respectively.

We first performed the electromechanical simulation of PZT slider under a given driving voltage to obtain the piezoelectric transverse-mode deformation of slider body. The deformed air-bearing surface of the PZT slider was used to calculate the steady-

state flying characteristics of the PZT slider using air-bearing simulation [70]. Subsequently, the effects of driving voltage on the static flying attitude of the slider were investigated.

Fig. 5.9 Three types of structure designs of PZT sliders.

Fig. 5.10 compares the deflection of three types of slider structure designs. It can be seen that the slider structure 3 can achieve the biggest deflection. Slider design of structure 2 has the smallest deflection. The deflection of the structure 3 is about 5 times larger than that of structure 2, and more than 2 times larger than that of structure 1 (Table 5.2).

Fig. 5.10 Deflection of slider trailing edge at different driving voltages.

Table 5.2 Deflection of slider trailing edge under 1V/5V/10V driving voltage.

	Design	1 V	5 V	10 V
Deflection of slider trailing edge (nm)	Structure 1	3.2762	16.381	32.762
	Structure 2	1.51	7.55	15.1
	Structure 3	7.4221	37.11	74.221

Fig. 5.11 shows the air bearing surfaces before and after slider deflection happening. It can be found the deflection of trailing edge obviously changes the surface profiles of slider. Correspondingly, the flying height of slider is reduced as shown in Table 5.3. The slider with structure 3 under driving voltage of 1 V obtains an actuation efficiency of 42.19% which is about three times larger than that of the slider with structure 1 under driving voltage of 5 V. It indicates the structure 3 can have a low power consumption and high actuation efficiency.

Fig. 5.12 and Fig. 5.13 show the five mode shapes of structure 1 and structure 3. The mode frequency analysis of slider with PZT actuator is listed in Table 5.4. It can

be found that structure 1 has larger first four mode frequency than that of structure 3.

(a) Original air bearing surface design of slider 'Panda 3'.

(b) New air bearing surface of slider 'Panda 3' with slider structure 1.

(c) New air bearing surface of slider 'Panda 3' with slider structure 3.

Fig. 5.11 ABS designs of slider 'Panda 3' before and after slider deflection.

Table 5.3 Flying height reduction of different slider structures.

	Deflection of Trailing Edge (nm)	Flying Height (nm)	Δ Flying Height (nm)	Efficiency
Structure 1 (5 V)	16.381	5.05822	1.67219	13.27%
Structure 3 (1 V)	7.4211	2.65872	4.07169	42.19%

Fig. 5.12 Mode shapes 1 to 5 of structure 1.

Fig. 5.13 Mode shapes 1 to 5 of structure 3.

Table 5.4 Mode frequency analysis of slider with PZT actuator.

		Mode 1	Mode 2	Mode 3	Mode 4	Mode 5
Natural frequency (Hz)	Structure 1	0.312×10^6	0.343×10^6	0.716×10^6	0.732×10^6	0.755×10^6
	Structure 3	0.164×10^6	0.346×10^6	0.541×10^6	0.635×10^6	0.100×10^6

5.5 Summary

A novel structure design of active slider with piezoelectric actuator in an optimized dimension is proposed. The PZT actuator is bonded on the top of the slider. A parametric simulation of dimension of PZT actuator is performed to obtain an optimized dimension. Three types of soft materials are compared and it is found that PTFE is most effective to achieve large deformation towards disk. By using the

optimized dimension of PZT actuator and the most effective soft material PTFE, three different fillings of soft material are studied and compared. The filling ‘structure 3’ has a low power consumption and high actuation efficiency.

CHAPTER 6

Simulation of Contact Sensor in Thermal Flying Height Control Slider

In this chapter, a piezoelectric contact sensor in the thermal flying height control (TFC) slider is proposed and investigated. A finite element model is built for the thermal flying height control slider with a piezoelectric contact sensor, which is used to detect the contact between the slider and disk. A constant force is applied at the maximum thermal protrusion point of air bearing surface. It simulates the forces induced electrical voltages between the two electrodes of the sensor using its transverse-mode and shear-mode, separately. Parametric simulations are performed to investigate a ‘proper design’ of the contact sensor with high sensitivity to contact force. It also simulates the thermal-stress effects of TFC heater on the sensitivity of contact sensor.

6.1 Introduction

In the pursuit of high areal recording density towards 10 Tb/in², the flying height should be as small as possible. However, under such small flying height, contact between slider and disk is inevitable. Severe or repeated contacts of the slider with the disk will cause the failure of the hard disk drive. Therefore, detection and possible avoidance of such head-disk contact becomes an extremely important issue for the reliability of future disk drives.

Conventionally, the contact between slider and disk is detected by acoustic emission (AE) sensor and laser Doppler vibrometer (LDV) [71-74]. AE sensor measures the elastic stress waves generated by contact and is generally effective to detect high intensity and catastrophic slider-disk contact; the use of an external AE

sensor for HDD reduces its sensitivity. The LDV measures the contact-induced vibration by analyzing frequency shift of laser beam reflected from the slider. LDV is typically large and relatively expensive.

In the thermal flying height control TFC sliders, because the reading element is near the lowest area of the air bearing surface when TFC heater is in operation [13], contact detection using reader signal becomes feasible. An experimental work indicated that reader signal had a quick response and high sensitivity to the continuous and intermittent contact [75].

In this work, a thermal flying height control (TFC) slider with a built-in piezoelectric contact sensor was investigated. Thin film ZnO was used as piezoelectric contact sensor because ZnO was more compatible with semiconductor industry than other piezoelectric material [76-77]. A finite element model of this kind of slider was built, which was used to detect the contact between the slider and the disk. A constant force was applied at the maximum thermal protrusion point of air bearing surface. It simulated the forces induced electrical voltages between the two electrodes of the sensor using its transverse-mode and shear-mode, separately. Parametric simulations were performed to investigate a ‘proper design’ of the contact sensor with high sensitivity to contact force. It also simulated the thermal-stress effects of TFC heater on the sensitivity of contact sensor.

6.2 Finite Element Model of TFC Slider with a Built-in Contact Sensor

A schematic model of TFC slider with built-in contact sensor is shown in Fig. 6.1. A Femto-slider was used in this work. The thickness of Al_2O_3 basecoat was 0.03 mm. The TFC heater was in a shape of plate with dimension (length×width×thickness) of $22 \times 14 \times 0.2 \text{ }\mu\text{m}$, and its distance to ABS was chosen to be $5 \text{ }\mu\text{m}$. Thin film ZnO was

used as a contact sensor, symmetrically deposited between substrate and Al_2O_3 basecoat in the conventional slider. The thickness of the ZnO sensor was chosen to be $2\text{ }\mu\text{m}$, and its length and width were chosen to be $60\text{ }\mu\text{m}$ and $40\text{ }\mu\text{m}$, respectively. The distance of the sensor to the air bearing surface (ABS) was denoted as ' d '. In the simulation, we chose the substrate side of the sensor as the upper electrode and the basecoat side as the lower electrode.

Fig. 6.1 Schematic model of TFC slider with contact sensor.

Fig. 6.2 shows the cross-sectional view of the finite element model of the TFC slider with built-in contact sensor using the commercial software ANSYS. It comprised the head elements of pole tips, shields, writer coil, TFC heater and ZnO sheet. The ANSYS element types of SOLID98 and SOLID226 were chosen for the slider and sensor sheet, respectively. Tables 6.1 and 6.2 list the material properties of PZT slider components used in the finite element simulation [68][78]. The leading surface ($x=0\text{ }\mu\text{m}$) of slider was fixed in x , y and z directions in the simulation as the displacement boundary conditions.

Fig. 6.2 Finite element model for the TFC slider used in this work.

Table 6.1 Material properties of the slider used in ANSYS.

Materials	Young's modulus (GPa)	Poisson's ratio	Density (kg/m ³)	Coefficient of thermal expansion×E-6 (K-1)	Thermal conductivity (W/mK)
Al ₂ O ₃ TiC (Substrate)	390	0.22	2200	7.1	20
Al ₂ O ₃ (Basecoat)	138	0.25	2200	7.1	1.8
NiFe (pole, shield)	200	0.3	700	12.8	35
Cu (writer coil)	130	0.34	8960	15.4	403
ZnO (sensor)		0.34	5750	0.651	110

Table 6.2 Electrical properties of ZnO sensor used in ANSYS.

ZnO	Dielectric constants (dimensionless)	ϵ_{33}/ϵ_0	12.64
		ϵ_{11}/ϵ_0	9.16
	Piezoelectric Matrix (C/m ²)	e_{31}	-0.573
		e_{33}	1.32
		e_{15}	-0.48
	Elastic Coefficient Matrix (10 ⁹ N/m ²)	c_{11}	209.7
		c_{12}	121.1
		c_{13}	105.1
		c_{33}	210.9
		c_{44}	42.47

6.3 Simulation Results

We investigated the contact force induced electrical voltage on the ZnO sensor by applying a 1 μN force at the maximum thermal protrusion point of air bearing surface. The reference dimension of the ZnO sensor was chosen to be $60 \times 40 \times 2 \mu\text{m}$ and the distance of sensor to ABS was chosen to be 2 μm . Two different types of designs of the ZnO sensor with transverse-mode and shear-mode were compared. A parametric simulation of the dimension and location of ZnO sensor was performed to obtain a 'proper design' of the contact sensor with high sensitivity to contact force. Also, thermal-stress effects on the ZnO sensor were investigated.

6.3.1 Contact Sensor based on Transverse-Mode and Shear-Mode Piezoelectricity

A constant force (1 μN) was applied at the maximum thermal protrusion point of air bearing surface. The induced electrical voltages between the two electrodes of the contact sensor are calculated using transverse-mode and shear-mode piezoelectricity, separately. The results in Fig. 6.3 show the ZnO sensor with shear-mode deformation (-0.87 mV) is more sensitive to contact force than that with transverse-mode deformation (0.05 mV). The contact sensor with shear-mode is chosen and investigated in the rest of the work.

(a)

(b)

Fig. 6.3 Contact force ($1 \mu\text{N}$) induced electrical voltage (V) in the contact sensor using (a) transverse-mode and (b) shear-mode.

6.3.2 Dimension and Location of Contact Sensor

Parametric simulations of dimension and location of contact sensor inside the TFC slider were performed with only considering contact force effects. In all simulations, a $1 \mu\text{N}$ force was applied at the maximum thermal protrusion point of air bearing surface.

Fig. 6.4 shows the contact force induced electrical voltage at different sensor dimension from $60 \times 40 \mu\text{m}$ to $120 \times 100 \mu\text{m}$. The thickness of sensor was kept as $2 \mu\text{m}$ and the distance of sensor to ABS was kept as $2 \mu\text{m}$. It can be observed that the voltage amplitude between the two electrodes of the contact sensor is reduced from 0.88 mV to 0.26 mV when the dimension of sensor is increased from $60 \times 40 \mu\text{m}$ to $120 \times 100 \mu\text{m}$. It indicates that the contact sensor with a small dimension will have a high sensitivity to contact force.

Fig. 6.4 Contact force ($1 \mu\text{N}$) induced electrical voltage at different sensor dimension.

Fig. 6.5 Contact force ($1 \mu\text{N}$) induced electrical voltage at different sensor thickness.

Fig. 6.6 Contact force ($1 \mu\text{N}$) induced electrical voltage at different distance of sensor to ABS.

Fig. 6.5 shows the contact force induced electrical voltage at different sensor thickness. The dimension of the ZnO sensor (length $\times$ width) was kept as $60\times 40 \mu\text{m}$ and the distance of sensor to ABS was kept as $2 \mu\text{m}$. It can be observed that the voltage amplitude is increased by more than two times when the sensor thickness is increased from $1.2 \mu\text{m}$ to $2.4 \mu\text{m}$. So the contact sensor with a large thickness will have a high sensitivity to the contact force.

Fig. 6.6 shows the contact force induced electrical voltage at different distance of sensor to ABS. The dimension of the ZnO sensor was kept as $60\times 40\times 2 \mu\text{m}$. It can be found that the electrical voltage is reduced with an increase in the distance of sensor to ABS. So the contact sensor with a short distance to ABS will have a high sensitivity to contact force.

Fig. 6.7 Transient output voltage of sensor under input impulse force.

A transient analysis was conducted when an impulse force was applied to the slider without considering the air bearing. The dimension of the ZnO sensor was chosen to be $60 \times 40 \times 2 \mu\text{m}$ and the distance of sensor to ABS was chosen to be $1 \mu\text{m}$. The loading location was kept same as before. The amplitude of the half-sine wave of impulse force was $1 \mu\text{N}$. The duration of the impulse was $0.5 \mu\text{s}$. The impulse started from $0.2 \mu\text{s}$ and ended at $0.7 \mu\text{s}$. The time step was $0.02 \mu\text{s}$. Fig. 6.7 shows the impulse input force and output voltage of sensor. It can be seen that the response of the sensor is like a quasi-static process. A half-sine wave of output voltage is obtained and no delay is observed. The amplitude of the output voltage is -0.86 mV , which is almost same as the static results (-0.87 mV in Fig. 6.6).

6.3.3 Effects of Thermal-Stress on Contact Sensor

It also simulated the thermal-stress effects of TFC heater on the sensitivity of contact sensor without considering the contact force. The dimension of the ZnO sensor was chosen to be $60 \times 40 \times 2 \mu\text{m}$. The heat transfer coefficient on ABS was set to be a

constant as $1 \times 10^5 \text{ W/m}^2 \cdot \text{K}$ as the boundary conditions.

Fig. 6.8(a) and Fig. 6.8(b) shows the temperature distribution and thermal protrusion towards disk at 30 mW power input when the distance of sensor to ABS is 1 μm . The maximum temperature rise of slider is 105 K, and maximum thermal deformation towards disk is about 19.4 nm. The output voltage between the two electrodes of the contact sensor is 24.19 V (Fig .6.8(c)).

(a)

(b)

(c)

Fig. 6.8 Distribution of (a) temperature rise (K), (b) thermal protrusion (μm) and (c) electrical voltage (V) of TFC slider at given 30 mW power input.

A parametric simulation of the distance of sensor to ABS was performed. The power input of TFC heater was chosen to be 30 mW and the distance of the sensor to ABS was changed from 1 μm to 35 μm . The results in Fig. 6.9 show the voltage amplitude is sharply reduced from 24.19 V to 0.94 V when the distance of sensor to ABS is increased from 1 μm to 15 μm . Then the sign symbol of voltage is changed from positive to negative at the distance around 15~20 μm . After that the voltage amplitude is increased and almost achieves a constant value of -5.35 V at the distance of 35 μm . Totally, the thermal-stress effects on contact sensor are significantly large and the amplitude of thermal-stress induced output voltage is orders larger than that induced by contact force (1 μN).

Fig. 6.10 compares the deformation profiles of four edges of contact sensor with 1 μm , 35 μm , and 15 μm distance of sensor to ABS, respectively. 'A1' and 'A2' are top and bottom edges of sensor at basecoat side, while 'B1' and 'B2' are top and bottom

edges at substrate side, as shown in Fig. 6.1. It can be found that in three cases, the deformation of top edges 'A1' and 'B1' are almost same. However, when the distance is 1 μm (Fig. 6.10(a)), the deformation of the edge 'A2' at basecoat side is significantly larger than that of the edge 'B2' at substrate side. When the distance is 35 μm (Fig. 6.10(c)), the situation is opposite, that is, the edge 'B2' at basecoat side protrudes more than the edge 'A2' at substrate side. It is the reason for the change of sign symbol of output voltage at different distance of sensor to ABS. Fig. 6.10(b) shows the deformation of four edges are almost same when the distance of sensor to ABS is 15 μm . It means that the difference of shear stress at two sides of the contact sensor is small which results in a small output voltage (Fig. 6.9).

Interestingly, the results in Fig. 6.9 also indicate that it is possible to eliminate the effects of thermal-stress from TFC heater on the contact sensor and obtain a 'zero' output voltage with a proper 'optimized' distance of sensor to ABS. For the present slider design, the optimized distance of sensor to ABS will be around 15~20 μm .

Fig. 6.9 Thermal-stress induced electrical voltage at different distance of sensor to ABS.

(c)

Fig. 6.10 Sensor deformation towards disk at different distance of sensor to ABS (a) 1 μm , (b) 15 μm , and 35 μm .

6.3.4 Time Response of Thermal-Stress Induced Electrical Voltage

We also performed the transient simulation to investigate the response time of the thermal-stress induced electrical voltage. The boundary condition of heat transfer coefficient on ABS was calculated from the static air bearing simulation and static thermal-structure simulation. Fig. 6.11 shows the output voltage versus time at a step power input of 30 mW when the distance of sensor to ABS is 1 μm and 35 μm , respectively. It can be found that the output voltages almost become saturated after 0.3 ms. The corresponding frequency is significantly smaller than contact frequency when the periodical head-disk contact happens.

Fig. 6.11 Time response of thermal-stress induced electrical voltage.

6.4 Summary

A finite element model was developed to investigate the effects of contact force and thermal-stress on voltage output of the built-in contact sensor in TFC slider . The simulation results show that the ZnO sensor with shear-mode is more sensitive to contact force than that with transverse-mode. The sensitivity of contact sensor can be increased by reducing the cross-sectional area of sensor, increasing the thickness of sensor, choosing a short distance of sensor to air bearing surface.

In addition, the thermal-stress effects from TFC heater on contact sensor are significantly large and the amplitude of thermal-stress inducing output voltage is orders larger than that induced by contact force. However, by optimizing the distance of contact sensor to air bearing surface, it is possible to eliminate the thermal-stress effects. Finally, the simulation results show that the response time of the thermal-stress inducing electrical voltage of contact sensor is about 0.3 ms.

CHAPTER 7

Fabrication and Performance of ZnO

Piezoelectric Cantilever for Micro Transducers

In this chapter, a Zinc oxide (ZnO) piezoelectric cantilever is fabricated and characterized and the characterization of the ZnO thin film by X-ray Diffraction (XRD), AFM (Atomic force microscopy) and field emission scanning electron microscope (FE-SEM) method are also presented. The ZnO film is deposited by Radio Frequency (RF) magnetron sputtering system under optimized deposition parameters at room temperature. The ZnO piezoelectric cantilever is fabricated using micromachining techniques and its dynamic performance is characterized by impedance analyzer and laser Doppler vibrometer (LDV). Finally, the comparison of transverse piezoelectric performance of the ZnO film between this work and others is presented and discussed.

7.1 Introduction

As the recording density of hard-disk drives continuously increases, the spacing between the slider and the disk decreases quickly. The 10 Tb/in² recording density requires a 2~3 nm or less of head-media spacing [79]. In this case, contact between slider and disk cannot be avoided completely. In order to reduce the damage from head disk interface caused by contact, it is necessary to detect the contact and then feedback such interaction to a spacing control system to adjust the gap between slider and disk quickly. Due to contact detection ability and quick response, piezoelectric sensor and actuator are more suitable than thermal ones for such application [80-81]. ZnO thin film has good piezoelectric performances and its fabrication process is compatible with thin film magnetic head processes. As a result, ZnO film based sensor/actuator will be

designed and used in the slider to detect the contact between head and disk and then adjust the flying height.

Although there are many literatures about fabrication and characterization of ZnO films [82-89], a few presented the detailed evaluation of the piezoelectric quality while depositing and characterizing the ZnO film [90-92]. The reports discussing the transverse piezoelectric constant of ZnO film are fewer. In this work, a ZnO piezoelectric cantilever was fabricated using micromachining techniques and its dynamic performance was measured in order to evaluate the transverse piezoelectric constant of the ZnO film. Before fabricating the ZnO cantilever, the ZnO thin film sputtered under optimized deposition conditions was characterized by XRD, AFM and FE-SEM method. Last, the comparison of transverse piezoelectric performance of ZnO film between this work and others was presented and discussed.

7.2 Deposition, Characterization and Optimization of ZnO Thin Film for Piezoelectric Cantilevers

7.2.1 Equipment Setup for ZnO Deposition

The dual target magnetron sputtering system used in this study is shown in Fig. 7.1. Various ZnO thin films under different deposition conditions were deposited on substrates using RF magnetron sputtering system in order to optimize the deposition parameters.

The substrate was (100) p-type silicon wafer on which a 1 μm of silicon oxide was grown and a 200 nm of Al layer was sputtered as electrode layer. The ZnO target was 50 mm in diameter with 99.99% purity and the distance between target and substrate was 70 mm. The chamber was evacuated to a pressure of less than 6×10^{-3} Pa before the generation of plasma, activated by RF power at 13.56 MHz. The concentrations of Ar

and O_2 were regulated by mass flow controllers. The substrate temperature was room temperature and the substrate platform was rotated at a speed of 30 R/min during the film deposition. To prevent contaminants from being deposited on the substrate and stabilize the plasma, the target was presputtered for 30 minutes before depositing ZnO film on substrate.

The influence of different RF power, $O_2/(O_2+A_r)$ gas ratio and the working pressure on the quality of ZnO thin film were investigated in our experiment. The deposited ZnO thin films were characterized by the X-ray Diffractometer (XRD), scanning electron microscopy (SEM) and atom force microscopy (AFM) methods.

Fig. 7.1 Dual target magnetron sputtering system.

7.2.2 Optimization of ZnO Thin Film

The growth rate of ZnO thin film under different RF power was investigated first. The lower growth rate of ZnO thin film was very important to achieve high quality piezoelectric ZnO thin film [93]. The relationship of ZnO deposition rate with different RF power is shown in Fig. 7.2. The deposition rate of ZnO thin film increases from 0.02 $\mu\text{m/h}$ to 0.5 $\mu\text{m/h}$ linearly when the RF power increases from 30 W to 110 W. The above measured growth rate is much lower than 1 $\mu\text{m/h}$, which is very helpful to obtain high piezoelectric quality of ZnO thin film. In the following experiments, the RF power is fixed as 70 W.

Fig. 7.2 ZnO deposition rate versus RF power.

Fig. 7.3 The deposition rate of ZnO film as a function of $O_2/(O_2+Ar)$ gas ratio and working pressure.

Fig. 7.3 shows the deposition rate of ZnO films as a function of $O_2/(O_2+Ar)$ gas ratio and working pressure. It can be seen that the higher $O_2/(O_2+Ar)$ gas ratio in the working gas results in lower deposition rate of ZnO film. The growth rate of ZnO film is dramatically decreased with increasing oxygen content from 0 to 20%, and then decreased slowly beyond 20% O_2 content. Similar results were also observed by other researchers [94-95]. The reason should be that O_2 has a lower sputtering yield than Ar . The net sputtering rate and the deposition rate of ZnO film decrease when the $O_2/(O_2+Ar)$ gas ratio increases in the working gas.

Fig. 7.3 also shows that the deposition rate of ZnO film varies between 4.52 nm/min and 5 nm/min when the working pressure increases from 0.3 Pa to 1.33 Pa. Different results can be found in other literatures. For example, the deposition rate

decreases when increasing working pressure in [96]; but in [95] the deposition rate increases with working pressure. Anyway, the deposition rate of ZnO film changes very slightly under various working pressures; namely the working pressure has little influence on the growth rate of ZnO film, which can be seen from other research results [94-96].

(a)

(b)

Fig. 7.4 (a) XRD patterns of ZnO thin film and (b) FWHM of ZnO (002) peak and grain size in ZnO thin film under different $O_2/(O_2+Ar)$ gas ratios.

Fig. 7.4(a) gives the XRD patterns of the ZnO thin film deposited at room temperature with various $O_2/(O_2+Ar)$ gas ratios. In these experiments, the $O_2/(O_2+Ar)$ gas ratios are 0%, 20%, 25%, 33.3% and 50% respectively; and the working pressured is fixed as 1.3 Pa. Firstly, Fig. 7.4(a) shows that all the deposited ZnO thin films have a diffraction peak located around 34.422° , which originates from the (002) plane of the ZnO films. So all the ZnO films have preferred c-axis orientation. Secondly, the intensity of the (002) peak increases with the $O_2/(O_2+Ar)$ gas ratio and reaches a maximum at an $O_2/(O_2+Ar)$ gas ratio of 33.3%. Thirdly, the diffraction angle of the (002) peak decreases from 34.52° to 34.32° and then increases, when the $O_2/(O_2+Ar)$ gas ratio increases from 0% to 50%. The shift of the ZnO (002) diffraction peak's

location results from the residual stress in the thin film [97-98].

The full width at half maximum (FWHM) of ZnO(002) peak and the grain size in ZnO film as a function of gas ratio were shown in Fig. 7.4(b). The grain size is calculated using Sherrer formula:

$$D = \frac{0.89\lambda}{B \cos \theta} \quad (7.1)$$

where λ , B and θ are the wavelength of the X-ray source (λ is equal to 0.154056 nm for Cu target), FWHM of ZnO(002) diffraction peak and the Bragg diffraction angle, respectively. Fig. 7.4(b) indicates that the FWHM varies between 0.285° and 0.326° and reaches its minimum of 0.285° when gas ratio is 25%. Correspondingly, the grain size changes between 25.23 nm and 28.85 nm and reaches its maximum of 28.85 nm when gas ratio is 25%. So the ZnO thin film deposited with O₂/(O₂+A_r) gas ratio of 25% has the best quality of crystallization quality. Although the optimized gas ratio in our experiments is different with others, our deposited ZnO thin film has smaller FWHM and larger grain size by comparing them with those values in other published literatures [99-100].

(a)

(b)

Fig. 7.5 (a) XRD patterns of ZnO thin film and (b) FWHM of ZnO (002) peak and grain size in ZnO thin film under different working pressures.

Fig. 7.5(a) gives the XRD patterns of the ZnO thin film deposited under various working pressures. In this case, the working pressures are 0.3 Pa, 0.5 Pa, 0.8 Pa, 1.1 Pa and 1.3 Pa respectively; and the $O_2/(O_2+A_r)$ gas ratio is fixed as 20%. Firstly, Fig. 7.5(a) shows that all the deposited ZnO thin films have a preferred c-axis ZnO(002) orientation. Secondly, the intensity of the (002) peak increases with the working pressure and reaches a maximum under working pressure of 0.8 Pa, and then decreases. Thirdly, the diffraction angle of the (002) peak increases from 34.1° to 34.34° and then decreases slightly, when the working pressure increases from 0.3 Pa to 1.3 Pa. As can be seen from Fig. 7.5(b), the FWHM of ZnO(002) diffraction peak decreases to its minimum of 0.243° from 0.354° when the working pressure increases from 0.3 Pa to 0.8 Pa, and then increases to 0.258° . Correspondingly, the grain size in the ZnO thin film increases to its maximum of 33.83 nm from 23.31 nm quickly and then decreases to 31.87 nm slowly during the working pressure increases from 0.3 Pa to 1.3 Pa.

By comparing Fig. 7.5(b) with Fig. 7.4(b), we can conclude that the working pressure has higher influence on the FWHM and grain size in the ZnO thin film than $O_2/(O_2+A_r)$ gas ratio. So the working pressure is also a very important parameter in the deposition of ZnO thin film and should be considered carefully.

Fig. 7.6 and Fig. 7.7 show the AFM images and surface roughness of the ZnO thin film deposited under different $O_2/(O_2+A_r)$ gas ratios, respectively. From Fig. 7.6, it can be seen that the surface of the ZnO film deposited with pure A_r gas is roughest, which is same to the conclusion in literatures [98-99]. Moreover, the grain size in the ZnO film deposited with pure A_r gas is biggest, which also can be seen from Fig. 7.6. When the O_2 gas is introduced into the gas flow, the surface of the ZnO thin film becomes more uniform. Fig. 7.7 shows that the biggest roughness of the ZnO thin film is 5.05 nm. The roughness of the ZnO thin film changes between 4.54 nm and 4.84 nm

slightly when the $O_2/(O_2+A_r)$ gas ratio increases from 20% to 50%. From the above analysis we can find that introducing some oxygen gas in the ZnO film deposition process is very helpful to flatten the surface of the ZnO thin film.

Fig. 7. 6 AFM images of the ZnO thin film deposited under different $O_2/(O_2+A_r)$ gas ratios.

Fig. 7.7 Surface roughness of ZnO thin film as a function of $O_2/(O_2+Ar)$ gas ratio.

Fig. 7.8 and Fig. 7.9 show the AFM images and surface roughness of the ZnO thin film deposited under different working pressures. It can be seen from Fig. 7.8 qualitatively that all the ZnO films have a dense and smooth surface under different working pressures. However, the 3-D surface morphology in Fig. 7.8 cannot tell the readers which roughness is larger or smaller clearly. Fig. 7.9 shows that the roughness value increases from 3.44 nm to 4.82 nm quickly, and then decreases to 4.68 nm slightly. It can be found from Fig. 7.7 to Fig. 7.9 that the surface roughness of ZnO film changes between 4.54 nm and 5.50 nm under different $O_2/(O_2+Ar)$ gas ratios; however the surface roughness changes between 3.44 nm and 4.82 nm under different working pressures. So the working pressure has higher influence on the surface roughness of the ZnO film than $O_2/(O_2+Ar)$ gas ratio.

Fig. 7.8 AFM images of the ZnO thin film deposited under different working pressures.

Fig. 7.9 Surface roughness of ZnO thin film as a function of working pressure.

Fig. 7.10 SEM cross-sectional morphology of the ZnO thin film deposited on Al/SiO₂/Si substrate.

The microstructure of a 1.16 μm thick ZnO thin film sputtered on Al/SiO₂/Si substrate was examined using SEM. The FE-SEM cross-sectional morphology of the ZnO thin film is given in Fig. 7.10. It clearly shows that the column texture of the ZnO thin film and the columnar growth is perpendicular to the surface, which indicates the ZnO film is highly c-axis-oriented. So the deposited ZnO thin film should have high piezoelectric quality.

From above analysis, the best deposition conditions are O₂/(O₂+Ar) gas ratio of 25% and working pressure of 0.8 Pa. The ZnO thin film deposited under optimized deposition parameters will have great crystallinity and very smooth surface. The optimized deposition parameters of ZnO thin film are summarized in Table. 7.1, and will be used in next section.

Table 7.1 Optimized deposition conditions of ZnO thin film.

Deposition parameters	Values
Target	ZnO (99.99%)
RF Power	70 W
Substrate	Al/SiO ₂ /Si
Substrate temperature	Room temperature
Working pressure	0.8 Pa
O ₂ /(O ₂ +Ar) gas ratio	25%
Target-substrate distance	70 mm
Deposition rate	0.27 $\mu\text{m/h}$
Deposition time	3h

7.3 Fabrication and Measurement of Piezoelectric Cantilever

The optimized deposition parameters which are summarized in section 7.2 will be used in the following fabrication of piezoelectric cantilever.

The micro fabrication sequence of the ZnO piezoelectric cantilever is illustrated in Fig. 7.11. Firstly, silicon dioxide (SiO₂) layer with thickness of about 1 μm was grown on both sides of silicon wafer. Then a layer of 120 nm Al was sputtered on the front side of Si wafer as the bottom electrode, followed by depositing 0.8 μm ZnO thin film on the bottom electrode, shown in Fig. 7.11(a). After that, a layer of 100 nm/20 nm Au/Ti was sputtered on the ZnO film and patterned by a lift-off process as the top electrode, as shown in Fig. 7.11(b). Subsequently, the ZnO film was wet etched by diluted HCl solution to expose the bottom electrode and the Al layer around the cantilever, as shown in Fig. 7.11(c). Then the Al layer, the SiO₂ layer and Si with a thickness around the cantilever were etched by wet etching, reactive ion etching (RIE) and deep reactive ion etching (DRIE), respectively, as shown in Fig. 7.11(d). The

thickness of Si etched can be any value according to the requirement and actually this thickness was just the silicon' thickness in the released cantilever. Finally the SiO_2 layer on the back side of the wafer was patterned and etched by RIE to expose the silicon and then the exposed silicon was etched through to release the cantilever by DRIE method, as shown in Fig. 7.11(e).

Fig. 7.11 Main fabrication process of ZnO piezoelectric cantilever: (a) SiO_2 growth and Al and ZnO deposition; (b) fabricating top electrode by a lift-off process; (c) wet etching ZnO film; (d) etching Al, SiO_2 and Si; (e) releasing cantilever by DRIE.

The SEM images of the fabricated ZnO cantilever from top and bottom views are

shown in Fig. 7.12(a) and 7.12(b), respectively. The length of the cantilever is 1000 μm , the width is 500 μm and the thickness is about 10 μm .

(a)

(b)

Fig. 7.12 SEM images of fabricated ZnO piezoelectric cantilever: (a) front-side view;
(b) back-side view.

The microstructure and crystalline qualities of the ZnO thin film were investigated by X-ray Diffractometer (XRD), field emission scanning electron microscopy (SEM) and atom force microscopy (AFM) before depositing top electrode on the ZnO film.

The XRD pattern of ZnO thin film in Fig. 7.13 indicates that the ZnO film has a very high peak with diffraction angle of 34.44° and this is just the ZnO (002) diffraction peak. So the deposited ZnO film has a preferred c-axis orientation, which is necessary to achieve ZnO film with high piezoelectric quality. Moreover, the full-width at half-maximum (FWHM) of the ZnO (002) diffraction peak calculated from the XRD data is very small only 0.187° . The grain size in the ZnO film can be calculated using XRD data and is 43.98 nm, which is larger than others in literatures [87][91]. The above analysis about the XRD pattern shows that the deposited ZnO thin film has very good crystalline quality.

Fig. 7.13 XRD pattern of ZnO thin film.

The surface morphology and RMS surface roughness of the ZnO film were measured by AFM. The 2-D and 3-D view of the film surface are shown in Fig. 7.14.

Fig. 7.14(a) shows that the grain size in the surface of ZnO film are very uniform and the film is very dense. The 3-D view shown in Fig. 7.14(b) indicates that the surface is very smooth and the RMS (root mean square) surface roughness of the ZnO film is 6.024 nm.

Fig. 7.14 AFM images of the ZnO thin film on Al/SiO₂/Si substrate: (a) 2-D view and (b) 3-D view.

The thickness and the cross-sectional image of the ZnO thin film were observed by FE-SEM and are given in Fig. 7.15. It clearly shows the column texture of the ZnO thin film and the columnar growth is perpendicular to the surface, which indicates the ZnO film is c-axis-oriented and so has hexagonal crystals. So the deposited ZnO thin film should have high piezoelectric quality. The layers and the interface between different layers also can be seen clearly from Fig. 7.15 and it shows that the thickness of the ZnO film is about 813 nm.

Fig. 7.15 SEM cross-sectional morphology of the ZnO thin film deposited on Al/SiO₂/Si substrate.

How to observe the piezoelectric effect or inverse piezoelectric effect of the deposited ZnO film and evaluate its piezoelectric performance is the most important. In our experiment, the impedance analysis of the ZnO piezoelectric cantilever was carried out first since this method can provide an effective and convenient method to evaluate the resonant behavior of piezoelectric micro cantilevers. Fig. 7.16 shows the impedance and phase spectra of the ZnO cantilever using the impedance analyzer. As seen from the phase angle curve in this figure, the resonant frequency of the first mode of the cantilever is 24988 Hz.

Fig. 7.16 Impedance and phase spectra of fabricated ZnO piezoelectric cantilever.

Laser Doppler vibrometer (LDV) was used to measure the dynamic response of the ZnO piezoelectric cantilever. Here the dynamic testing included measurement of resonant frequency of the cantilever and cantilever's deformation under different driving voltage. The testing process for measuring the resonant frequency of the cantilever was in the following: a frequency-sweeping AC voltage signal with sine waveform was generated by a signal generator and then applied on the top and bottom electrode of the ZnO cantilever. The amplitude of the AC voltage was 0.05 V and the frequency was from 10 kHz to 40 kHz. The spot of the laser was located on the tip of the cantilever. The velocity of the cantilever's tip was measured by LDV during the frequency-sweeping process.

The measured velocity curve of the cantilever tip in time domain is given in Fig. 7.17(a). It clearly shows that the cantilever resonates one time during the sweep time. The magnitude of velocity in frequency domain obtained by applying FFT on the

velocity in time domain is given in Fig. 7.17(b), which shows that the resonant frequency of the cantilever is 25031 Hz. This value is very close to 24988 Hz from impedance analysis, but this method is much complex.

(a)

(b)

Fig. 7.17 Tested results from LDV: (a) velocity curve in time domain, (b) magnitude of velocity curve in frequency domain.

As the piezoelectric cantilever deformation was very small, the velocity of the cantilever tip was measured by LDV first and then integrated to obtain the amplitude

of the cantilever tip under different driving voltages. The amplitude of the cantilever tip versus driving voltage is given in Fig. 7.18. As seen in Fig. 7.18, the amplitude of the cantilever tip increases from 16.8 nm to 52.5 nm when the driving voltage increases from 0.2 Vrms to 0.6 Vrms. The tip deflection is linear with the driving voltage.

Fig. 7.18 The amplitude of ZnO cantilever tip and the calculated d_{31} of ZnO film versus driving voltage.

If the influence of top and bottom electrode and silicon oxide layer on the deflection of cantilever is omitted (that is, the piezoelectric cantilever is a bimorph), the transverse piezoelectric constant d_{31} of the ZnO film can be calculated using the following equation [101]:

$$d_{31} = \frac{\sigma(L)[E_e^2 t_e^4 + E_e E_p (4t_e^3 t_p + 6t_e^2 t_p^2 + 4t_e t_p^3) + E_p^2 t_p^4]}{3L^2 t_e (t_e + t_p) E_e E_p U} \quad (7.2)$$

where L is the length of the cantilever beam. $\sigma(L)$ is the tip deflection of the cantilever. t_p and t_e are the thickness of ZnO and silicon, respectively. E_p and E_e are the Young's modulus of ZnO and silicon, respectively. U is the driving voltage. According to equation (7.2), the calculated piezoelectric constant d_{31} of the ZnO film under different driving voltages is listed in Table 7.2 and is also shown in Fig. 7.18. As can be seen from Fig. 7.18, the piezoelectric constant d_{31} of the deposited ZnO film is almost unchanged under different driving voltages and its average value is -3.21 pC/N. This value has the same order of magnitude as the value for bulk ZnO material, which is -5.43 pC/N [102].

Although the transverse piezoelectric constant d_{31} of our deposited ZnO film is a little smaller than that of bulk ZnO material, it is relatively high by comparing with others in literatures given in Table 7.3. Moreover, Table 7.3 shows that most of other ZnO films were deposited when the substrate was being heated or the ZnO film was post annealed at high temperature after deposition. But in our case, any heating or post annealing operation was not carried out on the ZnO thin film. As we all know, heating substrate during depositing ZnO film or post annealing ZnO film after deposition can increase the crystallization quality and even the piezoelectric quality of ZnO thin film. So the piezoelectric quality of the ZnO film must be increased highly if the heating operation on ZnO thin film is used in our experiment.

Table 7.2 Calculated d_{31} piezoelectric constant of the ZnO film under different drive voltage.

	Drive voltage (V)	Deflection (nm)	d_{31} (pC/N)
1	0.2828	16.7994	-3.08
2	0.4242	26.9108	-3.29
3	0.5656	34.9522	-3.21
4	0.707	44.1879	-3.24
5	0.8484	52.5478	-3.21
			-3.21(Average value)

Table 7.3 Comparison of piezoelectric constant d_{31} of the ZnO film among different research groups.

Author	Fabrication method of ZnO	Substrate Temperature	ZnO thickness (μm)	d_{31} (pC/N)
Feng <i>et al</i> [103]	RF Magnetron sputtering	Not mentioned	2	-2.7
DeVoe <i>et al</i> [101]	RF Magnetron sputtering	Post annealing at 500°C	0.5-1.5	-2.3
Gardeniers <i>et al</i> [104]	RF magnetron reactive sputtering with Zn target	425°C	0.68-2.07	-0.02~-5.27
Shibata <i>et al</i> [105]	RF Magnetron sputtering	350°C	1	-3.5
Wang <i>et al</i> (in this work)	RF Magnetron sputtering with ZnO target	Room temperature	0.81	-3.21

7.4 Characterization of ZnO Sensor for Vibration and Impact Detection

The sample of the ZnO piezoelectric cantilever is shown in Fig. 7.19. A Si cantilever beam (20 mm×5 mm×3.0 μm) with a small area thin film ZnO sensor was

investigated. ZnO sensor ($500\text{ }\mu\text{m}\times 500\text{ }\mu\text{m}\times 1.3\text{ }\mu\text{m}$) was located at the fixed side of the cantilever, It can collect most of the vibration information. The size of ZnO sensor was much smaller than the size of the Si cantilever, therefore, ZnO sensor can be treated as a point sensor and the strain is uniformly distributed over the area that is covered by the sensor. The experimental setup used for sensor characterization is shown in Fig. 7.20. The cantilever beam was bonded on the housing which was bonded on the shaker. An accelerometer was located on the housing of cantilever. A LDV was used to verify the sensor signals. The LDV measured the vertical displacement at the tip of the cantilever.

Fig. 7.19 The sample of ZnO piezoelectric cantilever.

In the first experiment, a sweep voltage signal was applied on the shaker. The velocity of cantilever tip versus input vibration frequency is shown in Fig. 7.21. It can be seen that the natural frequency of the cantilever with ZnO sensor is 1004.7 Hz.

Fig. 7.20 The experimental setup used for sensor characterization.

Fig. 7.21 The velocity of cantilever tip versus input vibration frequency.

The second experiment was performed to verify whether or not the sensor detects the vibration signal that was measured by LDV. A single harmonic vibration was applied on the housing which simultaneously excited several vibration modes of the cantilever beam, shown in Fig. 7.22. The experimental results show that the ZnO

sensor can generate similar output waveform with the input acceleration and the response of LDV.

(a)

(b)

(c)

Fig. 7.22 LDV and sensor measurements of the cantilever when a harmonic vibration is applied at the housing: (a) the acceleration of input vibration versus time ($f=850$ Hz, $a_{max}=51$ m/s²), (b) LDV output, (c) sensor output.

Fig. 7.23 shows the relationship of open-circuit voltage of ZnO sensor with different input accelerations under different frequencies. The natural frequency of the cantilever was 1004.7Hz which was derived previously. It can be found that the output voltage of ZnO sensor increases with input acceleration linearly under fixed frequency. Besides, under the same input acceleration, the output voltage of ZnO sensor increases with the frequency of input vibration. In the mean time, the increasing rate of output voltage of ZnO sensor is larger when under the higher frequency.

Fig. 7.23 The open-circuit voltage of ZnO sensor versus different input accelerations under different frequencies.

Fig. 7.24 shows the experimental setup for impact testing of cantilever beam. A steel ball of diameter of 1 mm drops from a given height to the housing.

Fig. 7.24 The experimental setup for impact testing.

Fig. 7.25 LDV and sensor measurements of the cantilever when an impact is applied at the housing: (a) the acceleration of input vibration versus time; (b) LDV output; (c) sensor output.

An impact was applied on the housing which simultaneously excited several vibration modes of the cantilever beam, shown in Fig. 7.25. The experimental results show that the response from the sensor and the LDV measurements are very similar to each other. So the ZnO piezoelectric cantilever has a good sensitivity to the mechanical impact.

7.5 Summary

Piezoelectric micro cantilever with high quality ZnO films were fabricated and characterized in this work.

In the first experiment, the ZnO thin film was deposited in the room temperature by RF magnetron sputtering. The optimized deposition parameters were investigated.

When the gas ration of $O_2/(O_2+A_r)$ is 25% and the working pressure is 0.8 Pa, the ZnO film has a highly preferred c-axis orientation.

In the later experiment, a piezoelectric micro ZnO cantilever was fabricated using micromachining techniques. The dynamic performance of the piezoelectric cantilever was characterized by impedance analyzer and LDV. The phase spectra of the ZnO cantilever measured by impedance analyzer shows that the resonant frequency of the micro cantilever is 24988 Hz. The velocity of the cantilever tip in frequency domain measured by LDV indicated that the resonant frequency of the cantilever is 25031 Hz which is very close to that value from the impedance analysis. Moreover, the LDV result shows that the tip deflection of the cantilever is linear with the drive voltage. The transverse piezoelectric constant d_{31} of the ZnO film calculated from the LDV data is -3.21p C/N, which is essential for our incoming design and simulation of slider with thin film ZnO sensor and actuator used in smart sliders in HDD.

Moreover, a silicon cantilever with a small ZnO sensor which was deposited on the fixed side of the cantilever was fabricated. The dynamic performance of this cantilever excited by a mechanical impact was investigated. The experimental results show that the response from the sensor and the LDV measurements are very similar to each other, which indicated that as a sensor, the thin film ZnO have a good sensitivity.

CHAPTER 8

CONCLUSION

8.1 Conclusion

This thesis presents an investigation on the tribology of head/disk interface and active slider for magnetic recording slider flying on a disk at spacing below 10 nm.

First, the finite element method based on commercial software ANSYS was used to simulate an elastic-plastic contact analysis for single asperity. In order to verify its validity, the results were compared with Hertz solution and CEB solution. A contact analysis of the slider-patterned media interface was subsequently performed using a surface topography description and mechanical properties resembling those of magnetic recording heads and thin-film rigid disks. The patterned media surface was approximated by a sinusoidal curve. Typical patterned media cases were modeled and investigated. Finite element results for the contact pressure at elastic-plastic asperity micro contacts and subsurface stress distribution were obtained.

To reduce flying height and prevent possible contact between head and disk, an active slider based on piezoelectric shear-mode actuation was proposed and investigated for flying height control. The electromechanical simulation results showed that the flying height and pitch angle of the PZT slider were significantly reduced with the increase in the drive voltage on the PZT sheet. However, beyond the drive voltage of 80 V for the proposed PZT slider, the reduction in the flying height of the PZT slider was found to be limited owing to the high air-bearing stiffness at low flying height region. Furthermore, the PZT slider can be rotated and balanced at a negative pitch angle. The parametric simulation results showed that a short distance of the PZT sheet to the trailing edge of the slider can help to achieve a low flying height. A small center-

trailing pad of the slider can also help to achieve a low flying height, but cannot prevent the reduction in pitch angle. The depth of the center-trailing pad did not change the reduction ratio of the pitch angle when increasing the drive voltage. A big pitch angle value was needed to avoid the pitch angle falling below zero at a high drive voltage.

A novel structure design was proposed for an active slider with a piezoelectric actuator at the top of the slider body and a soft layer between the substrate and head elements. The PZT actuator was bonded on the top of the slider. A parametric simulation was performed to obtain an optimized design for the PZT actuator. Three types of soft materials were compared and it was found that PTFE was most effective to achieve large deformation towards disk. By using the optimized dimension of PZT actuator and the most effective soft material PTFE, three different fillings of soft material were studied and compared.

In addition, a ZnO piezoelectric contact sensor integrated with thermal flying height control (TFC) slider for contact detection was proposed and investigated. A finite element model was developed to investigate the effects of contact force and thermal-stress on voltage output of the built-in contact sensor in TFC slider. The simulation results showed that the shear-mode ZnO sensor was more sensitive to contact force than the transverse-mode ZnO sensor. The sensitivity of contact sensor can be increased by reducing the cross-sectional area of sensor, increasing the thickness of sensor, choosing a short distance of sensor to air bearing surface. In addition, the thermal-stress effect from TFC heater on contact sensor was significantly larger and the amplitude of thermal-stress inducing output voltage was orders of magnitude larger than that induced by contact force. However, by optimizing the distance of contact sensor to air bearing surface, it was possible to eliminate the

thermal-stress effect.

Finally, piezoelectric micro cantilevers with high quality ZnO films were fabricated and characterized to demonstrate the feasibility of an integrated piezoelectric contact sensor and possible active slider. The dynamic performance of the piezoelectric cantilever was characterized by impedance analyzer and LDV. The LDV result showed that the tip deflection of the cantilever was linear with the drive voltage. The transverse piezoelectric constant d_{31} of the ZnO film calculated from the LDV data is -3.21p C/N, which is essential for our incoming design and simulation of slider with thin film ZnO sensor and actuator used for smart sliders in HDD. Subsequently, a silicon cantilever with a small ZnO sensor was fabricated. The dynamic performance of this cantilever excited by a mechanical impact was investigated. The experimental results showed that the response from the sensor and the LDV measurements were very similar to each other, which indicated that the thin film ZnO sensor had good sensitivity.

8.2 Future work

Piezoelectric micro cantilevers with high quality ZnO films have been fabricated and characterized in this work. Sensing and actuating mechanism of transverse mode ZnO films are investigated. Possible future work includes the study on sensing and actuating mechanism of shear mode ZnO films, integration of ZnO thin film into the smart slider by micro-electro-mechanical systems process as sensor and actuator to detect head-disk contact and control the flying height between slider and disk.

1. Sensing and actuating mechanism of shear mode ZnO films will be investigated. Shear mode ZnO thin film will be deposited using RF magnetron sputtering. Performance and characterization of shear mode ZnO film will also be investigated. Sensitivity of shear mode ZnO thin film as sensor and actuator

will be obtained from experiment.

2. ZnO thin film will be integrated into the smart slider based on the optimized structure design. Wafer and slider level fabrication process will be developed to achieve the integrated slider/sensor/actuator. The capability of the built-in ZnO thin film sensor and actuator will be tested and demonstrated.

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