Shock dynamic characteristics of hard disk drives

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Shock Dynamic Characteristics of Hard Disk Drives

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Abstract

The interest in the effects of shock on hard disk drives (HDDs) has come into currency due to the increasingly hostile environments encountered in the usage of the portable computer as well as the application of HDDs in consumer devices. Their mechanical robustness under shock and other mechanical disturbances is of great concern. Shock dynamic characteristics of HDDs are investigated in this dissertation using numerical, analytical, and experimental techniques.

Firstly, the influence of the pulse width/duration and the pulse amplitude of single half-sine acceleration pulse on shock responses of the relative displacement of a head actuator assembly (HAA) was investigated by using FEM simulation and a single-degree-of-freedom (SDOF) model. For both the FEM and SDOF models, the peak relative displacement occurs at a critical frequency ratio (i.e. $\beta = \omega / \omega_n \approx 0.6$). In other words, a *pseudo resonance* phenomenon occurs at the critical frequency ratio.

Secondly, to investigate the pulse shape effects of acceleration pulse on shock responses, the HAA subjected to three pulse shapes of acceleration shock was analyzed by using finite element simulation. An abnormal phenomenon was observed in the shock responses where a higher energy pulse results in lower responses. This has been explained in terms of a power spectrum analysis. A cross-over point (or

small interval) was observed in the acceleration power spectrum curve. A simple *theorem* was developed to prove the existence and location of such a cross-over point. A *corollary* was derived based on this theorem. The prediction of the corollary was confirmed by numerical results.

Thirdly, to study the effect of the pivot bearing stiffness on the shock responses of the HAA, a simplified beam model with a torsional spring and a translational spring to simulate the pivot stiffness was developed. This effect was also analyzed by modal analyses and drop test simulation with the finite element model of the HAA.

Fourthly, dynamic characteristics of a micro-drive were investigated by both experimental and numerical techniques. Finite element models of the main components/assemblies of the drive were created for modal analysis, harmonic analysis, and operational shock analysis, including the analysis of temperature effect on the flying attitude of the drive.

Finally, the influence of the radial position and the skew angle of the slider, the rotating speed of the disk, and the shock simulation, of the air bearing slider were analyzed by CML code. Two simplified nonlinear models of the air bearing slider were developed. The air bearing slider has obvious nonlinear behavior for ultra-low flying heights.

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Nomenclature

A_1, A_2	Amplitude of function
$A_0, A, A_{\max}, A_\varepsilon$	Amplitude/Magnitude of acceleration pulse
b	Width
B1, B2, B3, B4	Boundary condition
c, c_s	Viscous damping coefficient
C_n	Coefficient of discrete Fourier transform
E	Young's Modulus [GPa]
f, f_0, f_1, f_2, f_n	Frequency
$F(t_j)$	Discrete time function
F_s	Force
h, FH, FH_{PT}, Z	Flying height
$\mathbf{F}_{N_i}, \mathbf{H}_{N_i}$	Matrix
g	Gravitational acceleration
$h(t), g(t), q(t), u(t)$	Waveform function
i	Imaginary number unit
I, I_θ, I_ϕ	Moment of inertia
k, k_1, k_2, k_3	Stiffness coefficient of spring

Nomenclature

k_s	Vertical stiffness of suspension
$k(Z)$	Nonlinear stiffness of air bearing
$k(u)$	Stiffness
k_r, k_1	Stiffness of spring
k_r^*, k_1^*	Dimensionless stiffness of spring
K_z	Translational stiffness
K_p	Pitch stiffness
K_r	Roll stiffness
L, L_0	Length
N, n, l	Integer number
m	Mass
$M()$	Magnitude of a quantity under consideration
$M_{s\theta}, M_{s\phi}$	Moment
P, p_a	Pressure
$p(x, t), p(t)$	External loading
$P_n, G(f)$	Power spectral value
Q	Slip correction factor
R, R_1, R_2	Radius
$\sin, \cos$	trigonometric function
sh, ch	hyperbolic function
$S(f), S_n$	Power spectral density
t	Time

Nomenclature

t_b	Thickness
T, T_e, T_0	Pulse duration/width
$u(t)$	Unknown variable
$\dot{u}(t), v, \dot{z}_0, \dot{z}, U, V$	Velocity
x, y	Coordinate
x_1, x_2	Displacement
x_g, y_g	Coordinates of the slider's center of gravity
$Y(t)$	Time dependent amplitude function
$\bar{y}_{\max}, \bar{z}_{\max}$	Relative displacement in dimensionless
$y_{\max}, z_{\max}, \hat{z}_{\max}, z_0, z$	Relative displacement
$\ddot{x}_1, \ddot{x}_2, \ddot{u}(t), \ddot{z}$	Acceleration
α	Pulse width conversion factor
β	Frequency ratio
ν	Poisson Ratio
ρ	Density [kg/m ³]
$\omega, \omega_n, \omega_d$	Angular frequency
ϖ	Parameter($=\sqrt[4]{\frac{\omega^2 m}{EI}}$)
ξ	Damping ratio
$\varepsilon, \gamma, e, \pi$	Constant
θ	Pitch angle
ϕ	Roll angle

Nomenclature

$\phi(x)$	Modal shape function
λ	Frequency parameter
φ	Shape function parameter
μ	Viscosity of air
Δt	Time interval

Chapter 1 Introduction

1.1 Background

Hard disk drive (HDD) is one of the most important and also one of the most interesting components within a computer. Current technological improvements require more robust shock testing of HDD. This type of testing is most relevant to applications in portable computers and other mobile products with an HDD. Desktop computers also need to withstand shock due to shipping and mishandling. Dropping, striking or bouncing a drive against a hard surface can damage it internally with no external evidence of damage. A drive that is subjected to this type of shock may fail on initial use. Or, the damage could simply cause the reliability of the drive to degrade over time. When an HDD is subjected to a high level of acceleration shock, the head suspension system lifts off the disk and lands on it in a very short time, and the impact during this slap often leads to failure of the magnetic head and disk.

Several investigations have been performed in the past concerning non-operational shock. In a previous study by Allen and Bogy (1996), the effect of shock on the head-disk interface was studied and experimental shock data for the head-disk interface were compared with numerical results from a finite element model. The model considered only the effects of the disk and suspension arm on the magnetic head for

the case of a linear shock where the impact surface is parallel to the disk.

A finite element analysis of an HDD was performed by Edwards (1999). In this study a complete model of a non-operational HDD was developed for a linear shock with a constant drop height for three different contact surfaces. Results were obtained for different shock pulse widths and peak accelerations experienced by the HDD. The model was verified using laser Doppler vibrometry (LDV) for modal analysis and shock response data.

Aristegui and Geers (2000) implemented a shock simulation using LS-DYNA as the finite element solver. They modeled a shock event by postulating that the impact velocity decreases exponentially with time. In their simulation only the actuator arm, the disk, the suspension and the head gimbal assembly (HGA) were considered and the disk was modeled as a rigid wall. The numerical calculations were found to be very useful in describing the behavior of the suspension and the HGA during a shock event.

Continuing the work by Aristegui and Geers, Jayson, *et al* (2001) implemented an improved model with a linear shock impulse. In their work the entire HDD was modeled including the base plate, the disk, the actuator arm, the suspension, and the HGA. The entire HDD was given an initial downward velocity and was allowed to impact a constrained surface. The simulation was performed for a linear shock impulse. The HDD model was restricted to only allow linear motion.

The above papers studied the effects of shock on a non-operational HDD. However, it is also important to understand the response of the HDD when in use, since shock and vibrations can occur during operation of the HDD. The operational model is structurally the same as the non-operational model but the operational model must account for the hydrodynamic lubrication or air bearing between the magnetic head and the disk. During flying, a stiff air bearing is present between the slider and the disk, and it is the presence of this air bearing that determines the shock response of the head/disk interface.

Zeng and Bogy (2002) studied the response of the head-disk interface in the operating state to a shock impulse. A finite element model of the disk, suspension, slider and air bearing was used to find the structural response of the HDD. A finite difference based solution of the time dependent Reynolds equation was implemented to find the response of the air bearing to the structural excitation and the dynamic flying attitudes of the slider. The shock response was determined for linear shocks with various amplitudes.

1.2 Objectives

The objective of the present research is to investigate the shock dynamic characteristics of HDDs using numerical, analytical, and experimental techniques.

This study aims to:

1) investigate the dynamic characteristics of an HAA subjected to a single half-sine acceleration pulse

The finite element model of an HAA will be created by using a commercially available finite element software. To investigate the dynamic characteristics of the HAA subjected to a single half-sine acceleration pulse, the influence of the pulse width/duration and the pulse amplitude of a single half-sine acceleration pulse on the shock response of the HAA during a drop test will be analyzed by using finite element simulation. A simplified model will be developed to perform analytical analysis.

2) investigate the dynamic characteristics of the HAA subjected to different acceleration pulse shapes

In addition to the pulse amplitude and pulse width, the acceleration pulse shape is an important parameter to define the acceleration excitation. Three types of acceleration shocks different in pulse shapes will be selected as input loadings to study the pulse shape effect on the shock responses of the HAA during a drop test. The shock responses will be explained in terms of power spectrum analysis based on Fourier transform theory.

3) develop a pivot bearing model and study its stiffness effect

A simplified beam model with a torsional spring and a translational spring to simulate the pivot stiffness will be developed. The pivot stiffness effect will be investigated by

using analytical and numerical techniques based on the simplified model and the finite element model; respectively.

4) study the dynamic characteristics of a micro-drive in operating state

The dynamic characteristics of a micro-drive will be investigated using both experimental and numerical techniques. Finite element models of the main components/assemblies, such as the disk, the HAA, the HSA, and the HDA, of the micro-drive will be created. These models will be verified and modified by modal analysis, harmonic analysis, transient analysis, as well as some modal tests and/or compared with some simulation results by others independently. Operational shock simulation will be investigated by considering the air bearing effect. Some design guidelines will be suggested for improving shock resistance.

5) develop two air bearing models and perform air bearing simulation

Two simplified air bearing slider models will be developed to investigate their nonlinear behavior for ultra-low flying heights. A suitable numerical method will be developed to solve the nonlinear equations. Moreover, investigation on the air bearing simulation will be carried out by using CML code. The influence of the radial position and the skew angle of the slider, the rotating speed of the disk, and the shock simulation, of the air bearing slider will be analyzed.

1.3 Outline

This thesis is organized into eight chapters. Chapter 1 presents a brief introduction to the background and the objectives of this project. Chapter 2 contains a literature review on the history and the structure of HDDs, the mechanics, the developments and the challenges of shock resistance of HDDs in the current state of the art.

Chapter 3 presents investigations on the shock dynamic characteristics of an HAA subjected to a single half-sine acceleration pulse by using an FEM model and an SDOF model; respectively.

Chapter 4 contains investigations on the shock dynamic characteristics of the HAA subjected to different acceleration pulse shapes together with power spectrum analyses on shock responses and acceleration pulse. A simple theorem was developed to explain a phenomenon observed in the shock responses and the power spectrum analysis. A corollary was derived based on the above theorem.

In Chapter 5, a simplified beam model with a torsional spring and a translational spring was developed for theoretical analysis of the pivot stiffness. The pivot stiffness effect was also investigated by modal analysis and drop test simulation using finite element analysis.

In Chapter 6, dynamic characteristics of a 1-inch micro-drive were investigated by both experimental and numerical techniques. Finite element models of the disk, the

HAA, the HSA, and the HDA, of the drive were created. Four types of dynamic analyses were carried out, including modal analysis, harmonic analysis, operational shock analysis, as well as power spectrum analysis. Some design guidelines were proposed for improving the shock resistance. Moreover, the temperature effect on the HDD slider flying attitude will also be investigated.

Chapter 7 demonstrates the air bearing simulation and modeling by using CML code and two nonlinear simplified models of the air bearing slider for ultra-low flying heights.

Finally, Chapter 8 summarizes the research work and main conclusions of this thesis, as well as recommendations for future work.

Chapter 2 Literature Review

Hard disk drive (HDD) is the technology representative of modern magnetic data storage devices and systems. Handling of damage of HDDs used in desktop computer is a common and persistent problem both during initial assembly and testing of HDDs in the factory as well as at customer's site during process handling installation into the computer chassis. Especially the interest in the effects of shock on HDDs has come into currency due to the increasingly hostile environments encountered in the usage of portable computer. A review is given here for the structure and details of the HDD, its developments and the challenges of the shock resistance in current time. Some important works are briefly reviewed, including those about the shock resistance of HDDs, the dynamics of HDDs, the pivot stiffness study and the air bearing stiffness study, as well as the theoretical analyses, experiments and numerical simulations have been used by the researchers in literature.

2.1 History of Hard Disk Drives (HDDs)

HDDs are one of the most important components within personal computers (PCs). They have a long and interesting history dating back to the early 1950s. Perhaps one reason that one can find them so fascinating is how well engineers over the last few

decades have done at improving them in every respect: reliability, capacity, speed, power usage, etc.

In 1950, Engineering Research Associates of Minneapolis built the first commercial magnetic drum storage unit for the U.S. Navy, the ERA 110. It could store one million bits of data and retrieve a word in 5 thousandths of a second. In 1956 IBM invented the first computer disk storage system, the 305 RAMAC (Random Access Method of Accounting and Control). This system could store five Mbytes. It had fifty, 24-inch diameter disks. By 1961 IBM had invented the first disk drive with air bearing heads and in 1963 they introduced the removable disk pack drive.

In 1970, an eight-inch floppy disk drive was introduced by IBM. In 1973, IBM shipped the model 3340 Winchester sealed hard disk drive, the predecessor of all current hard disk drives. The 3340 had two spindles each with a capacity of 30 MBytes, and the term "30/30 Winchester" was thus coined.

In 1980, Seagate Technology introduced the first hard disk drive for microcomputers, the ST506. It was a full height (twice as high as most current 5 1/4" drives) 5 1/4" drive, with a stepper motor, and held 5 Mbytes. In the early 80's, the first 5 1/4" hard disks with voice coil actuators (more on this later) started shipping in volume, but stepper motor drives continued in production into the early 1990's. In 1981, Sony shipped the first 3 1/2" floppy drives.

In 1986, the first 3 1/2" hard disks with voice coil actuators were introduced by

Conner in volume, but half (1.6") and full height 5 1/4" drives persisted for several years. In 1988, Conner introduced the first one inch high 3 1/2" hard disk drives. In the same year, PrairieTek shipped the first 2 1/2" hard disks. In 1991, Integral Peripherals' 1820 is the first hard disk with 1.8" platters, later used for PC-Card disk drives. In 1992, Hewlett Packard's C3013A is the first 1.3" drive.

In 1997, Seagate introduced the first 7,200 RPM, Ultra ATA hard disk drive for desktop. In 2000, IBM triples the capacity of the world's smallest hard disk drive. This drive holds one gigabyte on a disk, which is the size of an American quarter. In 2002, Seagate announces Cheetah 15K.3 - its third generation 15,000 RPM drive one year ahead of the competition, and delivers industry's first Serial ATA disc drive -- the Barracuda ATA V.

In 2004, Seagate announces Savvio, the industry's first 2.5-inch enterprise disc drive, and launches 12 new products -- an unprecedented array of HDDs to target a wide range of applications like MP3 players, consumer electronics, notebook computers, PCs, servers and corporate data centers.

In terms of their capacity, storage, reliability and other characteristics, HDDs have probably improved more than any other PC component. The areal density of hard disk platters continues to increase at an amazing rate even exceeding some of the optimistic predictions of a few years ago. The latest achievement in areal density is 100 Gb/in² (Giba-bits per inch square) in production and 160 Gb/in² at lab level. Great efforts are being made to push technology towards 1000 Gb/in² or 1 Tb/in²

(Tera-bits per inch square). Researchers are pushing technology innovation and development in all aspects as so to keep an annual areal density increase rate at 35 – 45%.

It is very clear that the trend of the HDD development is more and more high areal density and the speed, the size of the HDD and its components become smaller and smaller. The improved performance of the mechanical parts has been of great significance in this progress. Among these, the shock resistance of the more and more elaborate parts is one of the most important guideline in the design of HDDs.

2.2 Structure of an HDD

The hard disk drive is the “data center” of a computer. To many people, a hard disk is a "black box" of sorts--it is thought of as just a small device that "somehow" stores data. It is hard to really understand the factors that affect performance, reliability and interfacing without knowing how the drive works internally. Fortunately, most hard disks are basically the same on the inside. While the technology evolves, many of the basics are unchanged from the first PC hard disks in the early 1980s.

Fig. 2.1 shows a photograph of a modern SCSI hard disk, with major components annotated. The platters are mounted by cutting a hole in the center and stacking them onto a *spindle*. The platters rotate at high speed, driven by a special *spindle motor* connected to the spindle. Special electromagnetic read/write devices called *heads* are

mounted onto *sliders* and used to either record information onto the disk or read information from it. The sliders are mounted onto *suspension* and *arms*, all of which are mechanically connected into a single assembly and positioned over the surface of the disk by a device called *actuator*. A *logic board* controls the activity of the other components and communicates with the rest of the PC.

Fig. 2.1 Photograph of a modern SCSI hard disk [www.pcguide.com]

The hard disk platters are accessed for read and write operations using the read/write heads mounted on the top and bottom surfaces of each platter. Obviously, the read/write heads don't just float in space; they must be held in an exact position relative to the surfaces they are reading, and furthermore, they must be moved from

track to track to allow access to the entire surface of the disk. The heads are mounted onto a structure that facilitates this process. Often called the *head actuator assembly*, it is comprised of several different parts, as shown in Fig. 2.2.

Fig. 2.2 Structure of a typical PC head actuator assembly [www.pcguide.com]

▲ Head Sliders

Hard disk read/write heads are too small to be used without attaching them to a larger unit. This is especially true of modern hard disk heads. Each hard disk head is therefore mounted to a special device called a *head slider* or just *slider* for short. The function of the slider is to physically support the head and hold it in the correct position relative to the platter as the head floats over its surface.

Sliders are given a special shape to allow them to ride precisely over the platter. Usually they are shaped somewhat like a sled; there are two rails or runners on the outside that support the slider at the correct *flying height* over the surface of the disk, and in the middle the read/write head itself is mounted, possibly on another rail. As hard disk read/write heads have been shrinking in size, so have the sliders that carry them. The main advantage of using small sliders is that it reduces the weight that must be yanked around the surface of the platters, improving both positioning speed and accuracy. Smaller sliders also have less surface area to potentially contact the surface of the disk in the impact or drop leads the failure of the hard disk.

▲ Suspension and Arm

The suspensions and arms are thin pieces of metal, usually triangular in shape onto which the head sliders (carrying the read/write heads) are mounted. There is one suspension per read/write head, and all of them are lined up and mounted to the head actuator to form a single unit. This means that when the actuator moves, all of the heads move together in a synchronized fashion. Heads cannot be individually sent to different track numbers.

The suspensions and arms are made of a lightweight, thin material, to allow them to be moved rapidly from the inner to outer parts of the drive. Newer designs have replaced solid arms with structural shapes in order to reduce weight and improve performance. Newer drives achieve faster seek times in part by using faster and smarter actuators and lighter, more rigid head arms, allowing the time to switch

between tracks to be reduced. Therefore, the shock resistance of the suspensions is reduced and should be treated carefully.

▲ Head Actuator

The actuator is the device used to position the head to different tracks on the surface of the platter. The actuator is a very important part of the hard disk, because changing from track to track is the only operation on the hard disk that requires active movement: changing heads is an electronic function, and changing sectors involves waiting for the right sector number to spin around and come under the head (passive movement). Changing tracks means the heads must be shifted, and so making sure this movement can be done quickly and accurately is of paramount importance. This is especially so because physical motion is so slow compared to anything electronic--typically a factor of 1,000 times slower or more.

As shown in Fig. 2.3, the actuator in a modern hard disk uses a device called a *voice coil motor* (VCM) to move the head arms in and out over the surface of the platters, and a closed-loop feedback system called a *servo system* to dynamically position the heads directly over the data tracks. The VCM works using electromagnetic attraction and repulsion. A coil is wrapped around a metal protrusion on the end of the set of head arms. This is mounted within an assembly containing a strong permanent magnet. When current is fed to the coil, an electromagnetic field is generated that causes the heads to move in one direction or the other based on attraction or repulsion relative to the permanent magnet. By controlling the current, the heads can be told to

move in or out much more precisely than using a stepper motor. All PC hard disk voice coil actuators are *rotary*, meaning that the actuator changes position by rotating on an axis.

Fig. 2.3 Structure of the voice coil motor [www.pcguide.com]

2.3 Shock Robustness of HDDs

As shown in Fig. 2.4, there are many mechanical issues in an HDD design. We mainly focus the literature review and discussion on following four aspects: 1) shock robustness of HDDs, 2) dynamics of HDDs, 3) pivot stiffness study, and 4) air bearing stiffness study.

Fig. 2.4 Mechanics in HDD design

Shock performance of drives under operation and non-operation status is becoming an increasingly important issue across all form factors, and disk drive manufacturers have been steadily increasing the threshold of shock level that the drivers can withstand. When the HDD is subjected to a high level of acceleration shock, the head suspension system lifts off the disk and lands on it in a very short time, and the impact during this slap often leads to failure of the head mode. The resulting collision can produce cracks in the slider, gouges in the disk, and floating debris that cause read and write failures. Today's high-performance drives, with their lower flying

heights, more sensitive medias and read/write heads, as well as more delicate suspensions, must be able to withstand the rigors of manufacturing. Hence the imposition of relatively high shock requirements destined for the HDD design.

Due to the inherent difficulty of the dynamic theoretical analysis, experiment and the complex structure of the HDD, there is relatively little published work related to the shock resistance of the HDD [Ishimaru (1996), Lee *et al* (2001), Edwards (1999)]. As there are many sources of high amplitude, short duration shocks with spectral content that overlaps the primary resonance of drive components. Efforts to develop a data of consistent and accurate field measurements of environmental and pulse durations of PC integrators are summarized in five test specification audits by Henderson (1998).

Several authors have examined the problem of shock on the head-disk interface both experimentally and numerically. Kumar *et al* (1994) studied the mechanics at the head disk interface caused by an input shock. Experimental results and a dynamic impact model were used to analyze the problem. A six DOF model, which included disk bending as well as the bending mode of the suspension was developed. Their paper addressed the head disk interface (HDI) damage in a drive designed for contact start stop (CSS), when exposed to shock under non-operational conditions. The HDI failure occurs when the slider overcomes the preload and separates from the disk surface upon impact. Although this type of failure can be largely avoided through the use of a load-unload design [Sagar, 1993], it may still be desirable to retain a CSS design to other considerations such as space and reliability. Kumar, *et al*'s results indicated that disk vibration becomes significant for narrow pulse width shocks. In

such cases, the energy transfer from the disk to the slider can be large enough to lift the slider even when it is on the top surface of the disk. Their analysis underscored the importance of all individual components toward a single failure mechanism, and highlighted the need to consider the entire drive as a single unit. They measured the slider motion using the capacitance between the slider and the disk. However, the relationship between the slider and the slider motion was not addressed.

Marek, *et al* (1995) reduced the problem to a single degree of freedom (DOF) model, neglecting disk motion while focusing on the cantilever mode of the suspension. Some of the analytical tools and test methods that have been developed by Hutchinson Technology to assess shock performance of head suspensions were described in their article. Most of those methods are in the evolutionary stage. Harrison and Mundt (2000) studied the flying height response to mechanical shock during operation of an HDD. The suspension and slider were simplified as a lumped multiple degrees of freedom spring-mass system. They modeled the air bearing by the usual Reynolds equation, and solved it numerically. They measured the disk's response at the point under the slider and the arm's response at the end of the arm. This measured data were used as input into the air bearing simulation code, and they obtained the dynamic flying height. It is inconvenient to apply their method because it requires the measurement data for each case of interest.

During the shock (or impact) analysis of structure, the input loading can be approximated by a half-sine pulse, with narrower shocks corresponding to impact on hard surfaces such as drop [Harries, (1988)]. The extent of damage on the disk was

highly dependent, not only on the shock pulse amplitude, but also on the pulse width. Narrow pulse width shocks caused much more damage than shock with wide pulse widths. It has been shown that loading intensity is an important parameter to determine material or structural failure mechanism in addition to total pressure impulse, e.g., pressure-impulse diagram has been used frequently to assess structural damages and human injuries when subjected to blast pulses [Smith and Hetherington (1994), Mays and Smith (1995), Krauthammer (1998), Li and Meng (2002a,b)]. Several methods have been suggested to consider loading shape effects on structural response, e.g., Li and Meng (2002b).

A finite-element analysis of a Seagate Bali II disk drive actuator assembly was reported by Aristegui and Geers (2000). In this paper, two finite-element computer programs, MARC and LS-DYNA, were employed to perform static, modal and transient-response calculations. The preloaded state of the suspension is determined, and modeling verifications is secured through the comparison of calculated and measured natural frequencies and mode shapes. Shock-response calculations are performed for two dropping heights and two impacting surfaces. Their results showed that arm-assembly response is sensitive to drop height and surface stiffness. It was found that a detailed model of the head gimbal is required, as the slider often impacts the disk at its edges. Finally, they found that the suspension itself is quite shock resistant.

The previously recognized industry shock specifications (half-sine pulse acceleration profiles of 100 to 300g amplitudes and 3 to 11 millisecond in duration) have proven

to be insufficient for the identification of the mechanisms that cause damage to the components of small portable drives. Currently new standards are being developed to adequately test and evaluate the roughness of portable drives and their components. These new standards include higher shock amplitudes of 500-1000 g and shorter durations of 0.5 to 2.5 ms. Allen and Bogy (1996) designed and constructed an uniaxial shock testing apparatus to produce high amplitude, short duration shock levels. The effects of shock on the HDD substrates, suspensions, arms and sliders were studied by these apparatus. In addition to the experimental apparatus, a finite element model of the experiment was created to emulate the apparatus closely and provide numerical correlation for the observations made during the experiment. Because the amplitude and frequency of the vibration of the disk are governed by the material properties, they found that the potential for disk damage is affected by these properties since the relative velocity of the slider and disk, which govern the force and severity of the impact, are directly related to the disk and suspension vibrations being either in or out of phase at the time of contact. More energy was imparted to the slider when the disk velocity was increasing at the time of contact. Higher likelihood of damage to the disk is the result of higher impact velocity and energy. In their finite element model, they only considered the effects of the disk and the suspension arm on the magnetic head for the case of a linear shock where the impact surface is parallel to the disk.

Ishimaru (1996) also developed an experimental set-up to study an HDI subjected to half-sine shock acceleration during non-operating using a dummy drive. He measured the degradation of recorded signals to investigate the permanent damage to the

disk surface due to the collision between head and disk. He found that the slap movement of the HAA is the dominant phenomenon governing the shock-proof performance of an HDI. This movement does not occur until the inertia force acting on the HAA exceeds the loading force. It is more violent as the magnitude of an acceleration increase. Higher load force, smaller equivalent mass of the HAA, low rigidity, and higher hardness of the disk contribute to better shock-proof performance. The duration of shock acceleration is not related to resonance frequency of the disk but to the HAA. Therefore, it is possible to estimate the slap movement quantitatively from the loading force, the equivalent mass of the HAA, and the duration of acceleration. Similar works also have been conducted by Sohn *et al* (2000) and found that there are two important factors for the actuator to endure high shock level. One is shock transmissibility and the other is the beating between the arm and the suspension, and the first bending natural frequency of arm was found to be the most important factor for the low shock transmissibility.

The dynamic performance of three different suspensions during shock was studied by Jen *et al* (1997) through both finite element simulations and experiments to determine the frequency responses function for in-plane and out-of-plane vibrations of different suspensions during shock. Lee *et al* (2001) built a theoretical model for prediction the shock behavior of the suspension in order to get a shock improved design. Their model consisted of a single DOF mass-spring system and a continuous system model with a single disk and spindle. The calculated results obtained from their model were compared with the results of finite element simulations and experimental data and found that the head lift-off entirely depended on dynamic behavior of the

suspension at non-operation, the suspension geometry design was found to be very important to improve the shock performance of the HDI, especially the length of suspension. On the other hand, the dynamic response of the head was fully controlled by shock response of the disk when the head was put on the middle or outer diameter of the disk. 8 parameters of the suspension shape were also investigated by Takahashi *et al* (2002) through a series of experiments to optimize the design of the suspension for high shock resistance. The thickness of the suspension was found to be the most sensitive factor. The effect of the actuator pivot unbalance on the track-following response of modern disk drives to external linear vibration and shock was investigated by Radwan *et al* (1996).

Most of the above publications studied the shock at the component level. However, many researchers have tended to reduce the investigation to the component level while also noting that the drive dynamics are complex and that the problem can not be fully understood without evaluating the entire system [Allen and Bogoy (1996); Kumar *et al* (1994); Lee, *et al* (2001)]. Edwards (1999) attempted to evaluate the HDD system as a whole, his finite element model in ANSYS includes models of the HDD enclosure base and cover, the HAA, the disk pack/spindle motor assembly and the voice coil motor assemble (VCM). He studied the shock response of a hard disk drive dropped from a height onto a surface with a specific contact stiffness. In each simulation the HDD model was dropped from a 25.4 mm height, but the contact stiffness of the impact surface was varied for each simulation. His simulations demonstrate the effect the surface stiffness characteristics have on both the magnitude and the pulse width of the impact shock received by the HDD and the dramatic

changes in the response of the internal components of the HDD to these different shocks. His study also showed that the acceleration amplification factors experienced by key components can be many times the values expected of single DOF systems due to the combined effect of multiple modes of vibration and that in particular the head stack assembly will experience the highest acceleration amplifications when subjected to shock with pulse widths in the 0.5 to 1.0 ms range due to the frequencies of the modes of vibration excited in this assembly. This sensitivity of the HAA to short duration shocks is a key factor in the resultant head-disk interface response leading to head-disk separation and subsequent head slaps. His experimental results showed good agreement to the analytical results that the frequencies of the modes of vibration excited in the disk pack/spindle motor and the HAA as well as with respect to their acceleration amplification factors.

A finite element model of a hard disk drive was developed by Jayson, *et al* (2003a) to investigate the response of the HDD to a shock impulse. Two types of shock were of interest in their model, a linear shock and a rotary shock. The linear model corresponds to an HDD being dropped flat onto an impact surface. The rotary model was constrained to rotate about an axis and simulated an HDD standing on one edge that was allowed to drop and impacted the opposite edge. The transient solution was performed using a finite element solver (LS-DYNA). Comparison of the simulation results for the two models was used to develop a correlation between the linear and rotary shock tests.

Shu *et al.* (2003, 2006, 2007) performed a drop test simulation of an HAA. A *pseudo-*

resonance phenomenon was observed and investigated by FE simulation and a simplified SDOF model. They found that, for both FE simulation and theoretical analysis by the SDOF model, when subjected to half-sine acceleration shock, the peak relative displacement occurred at a critical frequency ratio of the characteristic frequency of the acceleration shock and the natural frequency of the dynamic system. Shi *et al.* (2004-2007) investigated the pulse shape effect on the shock responses of an actuator arm subjected to three types of different acceleration shocks. Their numerical results showed that for the three acceleration pulses, the peak displacements have opposite behaviors for 0.1- and 1-ms pulse widths. The pseudo-resonance phenomena occurred for the maximum relative displacement, but at different pulse widths for these different acceleration shocks. They found that the relative magnitude of the peak displacement of the actuator arm is mainly determined by the power magnitude of the acceleration pulse at the resonant frequency.

Some new techniques were used by the researchers to identify and improve the shock characteristics when the hard disk drive is subject to external shock vibration. A 4-D shock-sensing and servo over-sampling scheme were presented and discussed by Chew (1999), this method to maintain data integrity when a drive is subjected to external shocks through re-design of the preamp, read channel and control software without additional cost of the drive.

Some researcher and engineers gave out some modifications to improve the shock durability through different ways. Kouhei *et al* (1996) viewed the motion of the complete slider-suspension assembly using a still camera and a stroboscope under

shock acceleration, and confirmed that the slider tilted when it collided with the disk. They proposed fitting a “Jump stop” over the slider to decrease tilt angle and collision velocity, they confirmed that this stopper enables shock durability up to 6 times larger than without it. However, add such a large component will cause many other problems. Through the high speed video camera, observation of a slider behavior after an external shock revealed that the slider jumps first and the re-contacts with the disks and its leading edge. Tokuyama *et al* (1999) developed suspension system that has a roof in the load beam that reduces the re-contact angle and the contact stress. They found that a magnetic disk device incorporating the system sustained no read errors for the external shocks up to 700 G. Lee *et al* (2000) added a dynamic absorber in the actuator arm to assist in controlling the amplitude and acceleration of the actuator arm in disk drive so that it can improve the shock handling capability in both operation and non-operation condition. Maxtor company developed a shock protection system (SPS) to minimize the problem of operation and non-operation shock that can occur in rack-mount computer environments. Maxtor’s SPS includes (i) SPS Head Slap Prevention, design the mechanical system to minimize the lifting of the head, and (ii) SPS Write Data Protection, when an operating shock is detected that might be severe enough to affect data integrity, disk drives will immediately suspend the write operation and wait until the shock transient has passed before completing the operation.

A modification of the HDD based on the numerical analyses and experiments were introduced by Lee *et al* (2002) to reduce the shock acceleration. The idea of this modification is changing the shock transfer path from shock input point to

the points of interest. Although this modification has significant in mechanical modeling theoretically, it was hard to overcome the constraint of the given system and doesn't affect the shock behavior of the HDD. Therefore, they tried to change the impact environment in a viewpoint of contacting part in next turn. With the proposed isolator attached system, contact force was decreased and it gave small amplitude of shock acceleration and longer time duration. They concluded that the most effective way to increase the shock resistance of the HDD is modifying the contacting part with attaching soft material to the base of the HDD.

Murthy, *et al.* (2006) investigated the shock performance of two form factor HDDs, 3.5 and 2.5 inch, respectively. The displacement of the actuator arm, the suspension, and the disk due to linear shock loads was studied experimentally for both non-operating and operating states of the HDDs. A finite element model was developed to simulate the shock response. Their numerical results compared well with their experimental results. However, they did not investigate the pulse width effect and pulse shape effect on the shock responses.

As nontraditional applications of hard disk drives emerge, their mechanical robustness during the operating state is of greater concern. In recent years, there has been an increasing application of small form factor (1 in. and smaller) HDDs in portable consumer appliances. Bhargava and Bogy (2007) proposed a procedure for simulating the operational shock response of a disk-suspension-slider air bearing system. The modeling of the structural components was done in ANSYS. The air bearing modeling was done using the CML dynamic air bearing simulator. The two

modules are coupled and each is iterated to convergence at every time step. They simulated shocks using a half-sine acceleration pulse with a pulse width of 0.5 ms and varying amplitude from -800 to 600 G. However, this method was inefficient and computationally expensive due to the exchange of data between the two modules at each time step. Moreover, they did not investigate the pulse width effect and the pulse shape effect on the shock responses.

Due to the relatively complicated components in the HDD, a full finite element analysis is time consuming; not only in building the model, but also in performing the system-level analysis. Gao *et al.* (2005) adopted a flexible multi-body dynamics formulation for non-operational analysis. They pointed out that this method is significantly faster compared to a full FE approach. Harmoko *et al.* (2007) proposed a more efficient method for predicting the shock tolerance of the HDD using state-space formulation to model the structural components of the HDD and quasi-static concept to model the non-linearity of the air bearing. They also proposed a procedure to conduct parametric study with this method and investigated the effect of overmold and voice coil stiffness on the shock tolerance.

2.4 Dynamics of HDDs

Early studies were mainly on the dynamics of head/suspension components, in which the suspension mode such as sway mode was the primary source of radial (in lateral direction) slider motions and contributed significantly to the TMR [Jeans, (1992);

Chiou and Miu, (1992)]. With rapid progress of the HDD technology, the lateral resonant frequencies of the suspension have become very high and can be neglected in understanding the arm resonance. It is the vibration modes of entire head actuator assembly rather than individual head/suspension that affect the dynamic behavior. It is therefore more important to analyze the entire head actuator assembly under actual operating condition [Lin, *et al* (2000)].

Since the rapid advance in hard disk drive technology for high capacity, rapid access, and reliability, there is a great challenge for mechanical design of the HAA in an HDD. The HAA has many higher frequency resonance, whose oscillation can persist even after the seeking process is completed and the read or write operation is started, thus leading to tract mis-registration (TMR) [Hospodor, (1995)]. And as be presented in the last section, the HAA's dynamic properties are also very important to control the shock resistance of the HDD. Therefore, improving the dynamic properties of the HAA has a profound effect on the performance of disk drive, and has received extensive research interests in both experimental and numerical studies [Radwan, (1993); Zeng and Body, (1998)]. The dynamic behavior of the head actuator assembly must be fully characterized before solution can be found.

Shaker is a common type of exciter used for the HDDs [Yoneoka, *et al* (1989); Henze, *et al* (1990)]. However, the shaker doesn't produce a point force on specimen, and it is often difficult to separate the dynamics of the test stand from those of the components. Frees and Miu (1990) introduced an "air-hammer" for free vibration analysis and a piezoelectric transducer for forced vibration analysis. Mori, *et al* (1993)

developed a piezoelectric tripod shaker to evaluate the dynamic characteristics of the suspension along various directions. Fu and Body (1994) applied a small stroke motion to a slider-suspension system to generate the desired magnitude of acceleration. Wilson and Bogy (1994a) designed an electromagnetic exciter to analyze the suspension and disk integrated into a system. The first three methods have not been often used because of their accuracy limitation. Wilson and Bogy (1994a) refined the design of the electromagnetic exciter and successfully applied it in modal experiments of components and systems of the HDDs. However, the effects of the mass of the ferromagnetic target on the tested structures could change their frequencies.

An alternative approach is being studied to use the voice coil of the HAA as the excitation device to study the entire HAA dynamics under the operating condition. The force portion of the transfer function is approximated by the sine sweeping current used as a stimulus to the voice coil [Jiang and Miles, (1999)]. Such experimental system is suitable for the HAA under actual operating condition.

The dynamic characteristics of the HAA are closely related to the pivot, particularly related to the ball bearings, and hence attract a fair amount of research interest [Ono, *et al* (1996); Decyiengyang, *et al* (2000)]. The lateral quasi-rigid body mode, caused by the flexibility of the pivot, and the mass and structure of the head actuator assembly, significantly contributes to the off-track error. The mode is usually located in the frequency range of 3-6 kHz dependent on particular drive. Aruga, *et al* (1996) first presented that the HAA has a quasi-rigid (QR) translation motion on

arm/suspension at the lateral QR mode vibration. Stricklin and Blanks (1996) described the vibrations of HAA at about 2 kHz was the in-plane bending of the arm. Jiang and Miles (1999) called such mode as system mode that the entire HAA vibrated and coupled with the motion of arm and suspension sway mode. This mode was also referred as butterfly mode by Radwan, *et al* (1999) and involved in-phase and in-plane lateral motion of all arms and the coil. It is evident that the vibration characteristics of the lateral quasi-rigid body mode are not fully understood in the operating condition with or without cover. Furthermore, most of study was done by the numerical modeling analysis, and there is little comparison between the analysis and corresponding measurement of the HAA. Therefore analysis and measurement of the lateral quasi-rigid body mode for entire head actuator assembly and their comparison were conducted to study the QR modes in the HAA and improve the dynamic properties of the HAA [Xu, *et al.* (2000); Zeng, *et al* (2001a); Xu, *et al* (2002)].

Due to its complexity in structure, numerical modeling such as the finite element (FE) method has been used in analysis of the HAA dynamics. Early finite element analyses in hard disk drives are mainly focused on the dynamic behaviour of the individual components such as suspension or arm [Frees and Miu, (1990); Miu and Karam, (1991)]. For example, Jeans (1992) developed a FE model of the Type 4 suspension, and calculated the sensitivity of modal frequencies to design parameters of the load beam. Yumura, *et al* (1992) investigated the effects of the head load on the gain of the resonance, and proposed a new design method for in-line suspensions. The dynamic characteristics of various suspension designs have been studied by

modal analysis and finite element modelling (FEM) [Ruiz and Body (1990a,b); Wilson and Bogy, (1994b); Zeng and Bogy, (1998); Jeong, *et al* (1998)].

However, it is more important to analyse the entire HAA dynamic characteristics. Radwan and Whaley (1993) and Aruga, *et al* (1996) established two finite element models of actuator without suspension and slider; respectively. Jiang and Miles (1999) built an HAA FE model in which the VCM was not included. Radwan, *et al* (1999) presented an HAA FE model including VCM, suspension and slider. These finite element models have linearized spring constant of ball bearings.

Some simplified models were also be used together with the FEM to study the dynamic characteristics of the HAA. Zeng *et al* (2001b) used a simplified beam model of the HAA together with the numerical simulations and experiments to propose a method for minimizing track seeking residual vibrations of the HDD. They found that the residual vibration of the HAA depends strongly on the waveform and the time duration of the driving impulse force.

2.5 Pivot Stiffness Study

Pivot is the rotating subsystem exists in a hard disk drive for supporting and rotating a head actuator arm. The pivot consists of a shaft, two ball bearings and a sleeve, and the pivot is attached to the head actuator arm by a screw or clip in most of current designs. The stiffness of the pivot assembly is a very important factor for the dynamic

behavior of the head actuator assembly [Gao *et al* (2000), Zeng *et al* (2001a,b)]. The important components in pivot are ball bearings, and many mechanical problems may occur due to the rotating mechanism and flexibility of the bearing.

Figure 2.5 illustrates a schematic diagram of the pivot that is used in most HDDs; the two ball bearings are biased against each other and with balls that are only about 1 mm in diameter. Although the balls are made of hardened steel, they are significant springy due to the very tiny areas of contact.

Fig. 2.5 The pivot in the head actuator assembly

The ball bearings are preloaded to reduce axial and radial play. The preload force level together with the bearing, inner and outer raceway curvatures, ball geometry,

and material determines the resultant stiffness of the pivot, so one of the most important the contribution to determine the stiffness of pivot is the flexibility of the ball bearings. The pivot shaft and sleeve are much more rigid than the ball bearing and hence their contributions to the overall stiffness can be neglected. Another most important contribution and never be studied before is the connect layer between the shaft, the ball bearing, the housing, the arm and the VCM. This layer always be made up of the soft glue material such as epoxy and the deformation of that in the transient response under the high load can cause large contribution to the overall dynamic response of the HAA.

The mechanical problems of the ball bearing or pivot in the HAA have attracted a fair amount of research interest. Ono *et al* (1996) and Deeyiengyang *et al* (2000) have analyzed numerically the non-repeatable radial vibration of a rotating shaft supported by a ball bearing with sinusoidal waviness errors on the outer and inner races and one of the rolling elements (balls). Some methods have been proposed to study bearing characteristics in disk drives. For example, a matrix method has been used to determine how the bearing design parameters affect both the disk drive and actuator performance [Prater, (1996)]. The restoring-force surface method has been applied to characterize the nonlinear track-following dynamics of the bearing within the rotating actuator [Hurst *et al* (1996)]. A fuzzy logic based pivot non-linearity model has also been proposed to cope with the non-linearity of the ball bearing [Huang *et al* (1998)]. Wang (1995) used a mathematical model of the ball bearing in the disk spindle motor to study the dynamics of rocking motion of the platters, but still no literature about the contributions of the pivot stiffness to such response of the HAA. Xu *et al*

(2001) studied the stiffness-related issues among ball bearing, pivot and HAA theoretically and experimentally. Their results showed that the pivot stiffness effective radial stiffness depends not only on the stiffness of the ball bearing, but also on the connection method used to attach the pivot to the head actuator arm. It is also shown from the measurement and analysis that the resonance frequency of the QR mode, determined by the pivot effective radial stiffness and inherent properties of the head actuator arm, can be moved to a higher frequency by using a more rigid pivot and a more stiff joint. Heo and Shen (2005) studied rocking vibration of hard disk drive spindle motors with fluid-dynamic bearings and rotating-shaft design. Their parametric study showed that the transverse mass moment of inertia of the rotating part is the most critical parameter affecting the rocking amplitude. The in-line and cross stiffness coefficients of the upper bearing are the most critical factors affecting the frequency and amplitude of the half-speed whirl.

Almost all of those studies in the literature, the stiffness of the pivot were simplified to the stiffness alone the axial direction and the radial direction. It is suitable to analysis the dynamic response of the HAA under the read/write operating to avoid the TMR. When the HDD is studied under the shock/impact, as the dynamic response at the slider end of the HAA is the key index to evaluate the shock resistance of the HDD, the complex stiffness such as the torsional stiffness of the pivot is very important and its contribution must be considered. In most of the studies about the HAA dynamic response, the ball bearing are simplified as linear springs, however, it is a challenge to obtain the actual stiffness value of the pivot or bearings. Furthermore, there is a lack of clear understanding on many stiffness-related issues among ball

bearing, pivot and the HAA, which is neglected in practice.

2.6 Air Bearing Stiffness Study

For a modern drive, the heads float over the surface of the disk and do all of their work without ever physically touching the platters they are magnetizing. As shown in Fig. 2.6, when the disk spins up to operating speed, the high speed causes air to flow under the sliders and lift them off the surface of the disk. The amount of space between the heads and the platters is called the *flying height*. It is also sometimes called the *head gap*, and some hard disk manufacturers refer to the heads as riding on an "air bearing".

Fig. 2.6 Flying principle of air bearing slider

As the preload pressure acting at the slider end on the suspension assemble to stabilize the sliding air bearing and prevent the slider lift off and slap the disk surface to lead the failure of the head mode during operating and keep the slider is effectively parked during non-operating, it indicates that there exist a vertical reaction between the head and the disk to support the HAA. The amplitude of the reaction is determined by the stiffness of the air bearing during the head did not touch the disk and the contact stiffness between the head and the disk while the head contact the disk due to the HDD subject to a external excite such as impact or vibration. Therefore, the stiffness between the head and the disk is a very important factor to determine the dynamic characteristics of the HAA and should be studied carefully to obtain the accuracy results of the HAA impact problem. It is a pity that little literature studies the contributions of it to the HAA dynamic response under impact. However some researchers investigate the effects of the head-disk contact in load/unload process and damage of the disk induced by the head-disk contact stresses [Suk and Gillis (1998); Fu and Bogy (2000), Hua, *et al* (2000)].

The action of the air bearing throughout contact is simulated to a set of four identical translational springs placed on the center lines of the two slider rails and one quarter of the untapered slider length away from the untapered ends by Leo and Sinclari (1991) to estimate how hard does a head hit a disk. Liu *et al* (1997) and Sheng, *et al* (1997) proposed a theoretical model of one-degree-of-freedom-system for the head-disk contact vibration study, and gave out a bilinear model of the system restoring

force.

In order to understand the fundamental destabilizing mechanism of the contact slider, a simple two-degree-of-freedom slider model for a single contact pad slider was first used for analysis by Ono and Suzuki (1997). In their model, it was assumed that the slider is allowed for both vertical and angular (pitch) vibrations to the running disk but it unable to move in the sliding direction, the slider was assumed to be in contact with the disk surface at a single point. The contact stiffness was represented by vertical stiffness (translational spring) and angular stiffness (rotational spring), and the air-bearing stiffness also be simulated to a series spring to the vertical contact spring. And Peng (1999) added another spring to represent the suspension flexure stiffness between the suspension and the slider.

Sheng *et al* (2000a) developed a theoretical model to simulate the slider suspension during the unloading process. For simplification, the roll modes of the slider and suspension are ignored in their model. The load beam is modeled as a lumped parameter system that is characterized by an effective load beam mass and effective load beam stiffness. The effective stiffness is calculated by using the effective mass and the bending frequency of the load beam. Similar simplified were done to the slider.

The shock responses of a disk-suspension-slider air bearing system were studied by Zeng and Boggy (2002), they found that the air bearing has different responses of upwards and downward shocks and the slider-asperity contacts occur when a strong

shock is applied. Jayson, *et al* (2002, 2003b) developed a finite element model of an HDD to investigate the transient response of an operational status subject to shock and vibration. In their model, the air bearing stiffness of the head disk interface was determined from a finite element solution of the Reynolds equation and approximated with linear springs. The structural response was analyzed for several types of sliders with a wide range of air bearing stiffness. However, they were not able to capture the true behavior of the air bearing because of the approximate modeling of the air bearing stiffness using linear springs.

Knigge *et al.* (2001) used joint-time frequency analysis to study the time-frequency evolution of a slider's response to a lump impact. Sheng *et al.* (2000b) introduced a nonlinear model for the air-bearing film and stated that as the slider's response increases in amplitude, the resonance frequency decrease. However, nonlinear dynamics is generally more complicated than what was described by their analysis. Thornton and Bogy (2003) proposed 1-DOF and 2-DOF analytical models and method of analysis for understanding the dynamical behavior of ultra-low flying height air-bearing sliders in proximity based on nonlinear dynamics. They found that for sub-5-nm flying height air-bearing sliders, the nonlinear effects couldn't be neglected. However, they did not consider the effect of suspension in their models. In their 2-DOF model, the two springs were located at fixed positions which may not represent the real cases.

Hua and Liu (2006) studied the multiple flying states of a negative pressure slider by simulation. They pointed out that the slider might have one to three balanced flying

states under certain conditions. The three states are connected with each other, and may interchange with each other under certain circumstances. Ono *et al.* (2005) studied the bouncing vibrations of a flying head slider in near-contact and contact regimes by experimental and analytical techniques. Their analytical results qualitatively agreed well with the experimental ones. They found that the bouncing vibration of a slider in near-contact and contact regimes is a self-excited vibration caused by the combination effect of an adhesion force and a friction force. Li *et al.* (2005) analyzed patterned sliders with ultra-low spacing flying above a rotating disk with smooth surface in a hard disk drive for high areal density recording. They discussed three types of pattern (slender, square, and broad) with the same bump area and found that the flying height of the broad patterned slider is the greater than either that of the square patterned slider or the slender patterned slider.

2.7 Study on Temperature/Thermal Effect

Modern HDDs are required to run at high revolutions (10000~15000rpm). As higher disk rotational speeds are required for evolution of HDDs, the internal running temperature increased. The motors required to drive the disks of these drives are required to generate a large amount of mechanical energy and thus dissipate a vast amount of heat. Shen *et al.* (2000) performed experiments to show that HDD disk media shift their natural frequencies at elevated temperatures. The thermal frequency shifts could result either from residual stresses (e.g., for aluminum disk media) or thermal membrane stresses caused by mismatch of coefficient of thermal expansions

(CTE) between the disk media and the clamp-spacer-hub assembly of the spindle motor. Tseng *et al.* (2002) show experimentally and theoretically that ball bearings in HDD spindle motors can lose their preload and reduce their radial bearing stiffness at elevated temperatures, if a CTE mismatch is present between the ball bearings and the hub of the spindle motors. Tseng *et al.* (2002) also develop a mathematical model to predict the reduction of the radial stiffness at elevated temperatures. Later, Jang *et al.* (2005) extended the work by Tseng *et al.* to include the effect of axial bearing stiffness. Unfortunately, the cited references above [Tseng *et al.* (2002), Jang *et al.* (2005)], only apply to spindle motors with ball bearings. HDD spindle motors currently on the market place, however, all employ fluid-dynamic bearings (FDBs).

With the increase of areal density in hard disk drives the physical spacing (or flying height (FH)) between the read/write element and the surface of the disk has been continuously decreased. A spacing of ~ 2.5 nm is said to be required for a density of 1 Tbit/ in². At such a low FH, static losses of the FH due to manufacturing tolerance, ambient pressure changes, and temperature variations can cause head-disk contact and result in data loss. Furthermore, slider disk contacts must be avoided during load/unload processes and operational shocks. The dynamic instability caused by FH modulations (FHMs) and nanoscale adhesion forces, such as electrostatic and intermolecular forces, should be minimized. Those challenges make a conventional air-bearing surface (ABS) slider an unlikely choice for 1 Tbit/ in². One potential solution is a FH adjustment or controlled slider that is capable of adjusting its gap FH. Because of their quick response and low power consumption, piezoelectric materials have been proposed as active elements for adjusting the FH [Suzuki, *et al.*,

(2003), Juang and Bogy, (2005)]. However, the requirements of the piezoelectric materials and the necessary modification of the slider design pose challenges in integration of the fabrication process and increase the manufacturing cost.

The read/write elements of a magnetic head slider consist of thin layers of different materials, including write poles, sensor, shields, undercoat, overcoat, coil, insulation layer, and the substrate. Because of the mismatch of the coefficients of thermal expansion of the various materials, the pole-tip of the write pole protrudes below the ABS plane when the ambient temperature varies and/or when an internal heat source is generated by Joule heating of the write current. These temperature-induced pole-tip protrusions (T-PTP) and write-induced pole-tip protrusions (W-PTP) adversely reduce the FH by several nanometers and increase the risk of head-disk contact. Kurita *et al.* (2006) developed an active head slider with a nanothermal actuator. They used a finite element method to calculate the temperature distribution and thermal protrusion of their slider. They found that the additional air pressure increase caused by the protrusion lifted the slider upward and the amount of FH reduction was 30% less than the protrusion. In their study, the distribution of the heat transfer coefficient on the ABS was assumed to be constant. However, the effect of heat conducted from the slider to the disk through the ABS is a strong function of both FH and air pressure distributions, and hence, these two factors have to be considered in the model.

Juang *et al.* (2006) studied the actuation performance of an ABS slider with consideration of the effect of FH and pressure distributions. They found that even

though the protruded area was relatively small, there was still considerable air bearing coupling with the resulting actuation efficiency (defined as the ratio of FH reduction to protrusion) of only 63%, which suggested that ABS played a key role in the actuation performance. Juang and Bogy (2007) studied the effects of the ABS on thermal actuation by numerical simulation. A three-dimensional (3D) thermal-structural coupled field finite element model was created with detailed structures of read/write and heating elements. The cooling effect of the air bearing was included in the model as thermal boundary conditions. They found that a properly designed ABS could significantly improve the actuation efficiency and power consumption of a FH control slider with thermal actuation.

Recently, small form factor (SFF) HDDs have received increasing attention in the marketplace because of their wide applications in commercial electronics, such as cell phones and digital video recorders. Due to their relatively short history, the thermal effect of SFF HDDs are not as well studied as those of 3.5-in or 2.5-in HDDs that have been in the marketplace for a long time.

2.8 Challenges in the Shock Simulation of the Drop Test of HDDs

Although HDDs have been studied by some researchers under the shock/impact conditions and have been developed for new applications, there are still a number of challenging issues that prevent the enhancement of the shock resistance of HDDs, such as the influence of pulse width/duration and pulse amplitude, the pulse shape

effect, the pivot modeling and its stiffness effect, the operational shock simulation, and the air bearing slider modeling and simulation for ultra-low flying heights.

Firstly, in the actual industrial vertical drop tests, the half-sine acceleration pulse is accepted as the standard input loading to study the dynamic response and assess the damage of an HAA. The extent of damage on the disk was highly dependent on the shock pulse width/duration and the amplitude. Moreover, although the half-sine acceleration pulse is accepted as the industry standard, the recorded acceleration pulse is usually not in exactly half-sine shape. Therefore, it's important to study the influence of the pulse shape changes on the dynamic responses of an HDD. To our knowledge, these effects have not been fully studied and reported in open literature.

Secondly, the modeling and simulation of the pivot in an HDD is another important issue. In most of the studies about the pivot, each ball of the pivot bearings was simplified into an axial spring and a radial spring. However, it is a challenge to obtain the actual stiffness values of the pivot bearings. Almost all the studies about the pivot show that the radial and axial stiffnesses of the pivot can affect the dynamic response of the HAA. However, little literature is available about how strong this effect is.

Thirdly, as HDDs continue to be made smaller in size, higher in capacity, and lower in cost, more and stronger disturbance during the operating state are encountered. Their mechanical robustness during the operational state is of great concern. Meanwhile, as the areal recording density increases in HDDs, the flying physical spacing between the head/slider and the disk media decreases and the likelihood of

head/disk contact during full speed rotation increases. Therefore, the simulation and modeling of the stiffness effect of the air bearing slider with ultra-low flying height becomes an important issue for the operational shock simulation. Moreover, as HDDs are scaled down in size, the volume-to-surface-area ratio decreases resulting in increased thermal resistance. Therefore, performance of SFF HDDs under various temperature environments is an important issue to study.

The above issues related to the shock dynamic characteristics of HDDs will be investigated in this project.

Chapter 3 Shock Dynamic Characteristics of an HAA Subjected to a Single Half-Sine Acceleration Pulse

3.1 Introduction

In practice, different impacts produce different pulse histories on a structure. For example, low contact stiffness impact produces pressure pulses a smaller amplitude and longer duration than those by high contact stiffness [Mays, 1995; Edwards, 1999]. Studies on this topic have been conducted for various structures, such as beams, plates and shells, within the category of dynamic plastic response [Jones, 1997; Zhu *et al*, 1986; Li *et al*, 1995, 2002a, 2002b]. However, this issue has not attracted enough attention for researchers and designers of HDD impact resistance engineering. Edwards (1999) studied the shock response of an HDD dropped from a height onto a surface with different contact stiffness, i.e., with different amplitudes and pulse widths of the input acceleration pulse loading. His analytical and numerical results showed that a disk drive was particularly sensitive to shock inputs with narrow pulse durations on the order of 0.5 to 1 milliseconds. Actually, the sensitivity range of the duration of the shock pulses depends on the dynamic properties of a head actuator assembly (HAA), which is the most important mechanical component of an HDD. A

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different amplitude of the shock pulse can lead to a different amplitude of the response of an HAA, and can affect the shock resistance capability of the HAA. To our knowledge, these effects have not been fully studied and reported in open literature.

The extent of damage on the disk was highly dependent on the shock pulse amplitude, and also on the pulse duration. Narrow pulse duration shocks caused much more damage than shock with wide pulse duration. Several authors have examined the problem of shock on the head-disk interface both experimentally and numerically [Kumar et. al. (1994), Marek, et. al. (1995), Allen and Bogy (1996)]. A finite element analysis of a Seagate Bali II disk drive actuator assembly was reported by Aristegui and Geers (2000). In their work, shock response calculations were performed for two-drop heights and two impacting surfaces. Their results showed that arm-assembly response was sensitive to drop height and surface stiffness.

In the actual industrial vertical drop tests, the half-sine acceleration pulse is accepted as the standard input loading to study the dynamic response and assess the damage of an HAA. For an HAA design, the maximum relative deflection between the tip of the actuator arm and the pivot of the HAA can be used as the key index to assess the possible failure of the HAA.

One of the objectives of the present work is to study the influence of the pulse width/duration and the pulse amplitude of single half-sine pulse shocks on the response of the maximum relative displacement of an HAA during a drop test, and

the conditions under which the peak value will occur. In section 3.2, the effects of the pulse width and the pulse amplitude of single half-sine pulse loadings are studied by finite element method with LSDYNA3D software. The maximum relative displacement is discussed. Then in section 3.3, the modeling of the actuator arm is simplified as a single-degree-of-freedom (SDOF) system and the theoretical solution of the SDOF system is illustrated. The maximum relative displacement of by the simplified SDOF system is also discussed.

3.2 Finite Element Modelling and a Pseudo-Resonance Phenomenon

3.2.1 Finite Element Modeling of C2 Drive

The finite element method (FEM) has been successfully used to study the dynamic responses of assemblies in HDDs. In this section, the effect of the pulse width and the pulse amplitude of single half-sine acceleration pulse loadings will be investigated using finite element analysis.

FE Model in Details. Based on the data from Seagate's ANSYS model, the finite element model of the HAA (without suspension) of Seagate hard disk type C2, which is shown in Fig. 3.1, was created and gradually improved using a commercial finite element (FE) software package (HYPERMESH), and analyzed with LS-DYNA3D software for vertical drop test simulation. Several investigators have contributions in the construction and improvements of this FE model [Shu *et al*, 2003-2007, Shi *et al*,

2004-2007]. This model consists of an arm, a pivot and a voice coil motor (VCM). The overall dimensions of the HAA are about 65 x 35 x 1.4 mm in length, width and thickness; respectively. The material of the arm is aluminum, and stainless steel is used for the pivot. The VCM consists of epoxy, copper and bobbin materials. The copper (C2 CU-CLAD+10%) is an anisotropic material and was simplified as an isotropic material. The pivot bearings were simulated by using spring elements with stiffnesses being defined in the radial and axial directions. For a given distance between the two rows of balls, the torsional stiffness is directly proportional to the radial stiffness.

Fig. 3.1 Finite element model of an actuator arm

The 8-node hexahedrons solid elements (SOLID164) are used for the elements all over the FE model except that the one-dimensional spring elements are used for the ball bearing; 22 springs represent the radial stiffness (along the radials of the shaft) and 22 springs for the axial stiffness (along the axis of the shaft). The FE

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model includes 15041 nodes and 10002 elements. This model is divided into 15 different parts and created one by one for ease to define the material and connection between the different parts, and then combined them to the whole model. The nodes in the interfaces between different parts were used together by the different parts. All these parts are connected directly except the part tail to the plate; they are defined as a pair of the contact with the CONTACT_AUTOMATIC_SURFACE_TO_SURFACE function of the LS-DYNA. It means that the tail can move freely and contact the plate during the drop test. Thus, it is a nonlinear model with contact defined and small deformation involved. A linear bulk viscosity coefficient of 0.06 is applied in the FE model through the hourglass control in the explicit algorithm of the LS-DYNA software. The properties of the materials used in this model are listed in table 3.1.

Table 3.1 Properties of materials

Material	Young's Modulus (E) [GPa]	Poisson Ratio (ν)	Density (ρ) [kg/m ³]
Aliminum	73.1	0.33	2.7E+3
Stainless_steel	200	0.28	7.6E+3
Copper	88	0.29	8.1E+3
Epoxy	2.5	0.29	1.6E+3
Bobbin Material	3.5	0.29	1.65E+3

Assumptions and Limitations. This model is consisted of several parts with

different materials. All the components were assumed to be connected directly with the same element nodes on their common interfaces between them. The anisotropic material was simplified as an isotropic material in order to avoid too complex simulation of an anisotropic material. The temperature effect is not considered in this FE model.

Convergence and Mesh Selection. In a previous work of this project [Shu *et al*, 2003], in order to verify the FE model of the HAA in drop test using LS-DYNA package, a typical cantilever beam model (similar as the HAA model) was used to verify the convergence and accuracy of the FE model. The beam model was meshed with three different layers of solid elements in vertical direction, i.e., 2-layer, 3-layer and 4-layer of solid elements. Theoretical results and numerical results were compared. From the comparison with the theoretical analyses and the FEM results with different layers of the model, it can be found that the 3 and 4 layers model can obtained the results with enough accuracy and good convergence. As for the convergence in time domain, the minimum time step size for explicit time integration depends on the minimum element length and the sonic speed.

Initial Conditions and Boundary Conditions. While the hard disk drive vertical freely drops to the ground, the whole system impacts the ground with the initial velocity $v = \sqrt{2gh}$ at the time $t=0$, where g is the gravitational acceleration and h is the drop height; the base of the hard disk drive contacts the ground and the velocity suddenly decreases to zero. The boundary conditions (BCs) of the HAA FE model are

as follows. As the HAA connects to the base through the shaft and after it contacts the ground, the shaft is fixed and cannot move, the inner surface of the pivot is restrained by the shaft and cannot move along any direction. The whole model has an initial velocity $v=5.0\text{mm/ms}$ at time $t=0$ to simulate the hard disk drive dropping about 1250mm to the ground.

Element Formulation and Time Step. The type of element formulation employed in the numerical simulation is a displacement based formulation. Shell elements are selected to mesh those of the HDD components with very thin thickness, instead of solid elements. In the transient analysis, explicit time integration method is employed. A central difference time integration method is used for the explicit transient analysis. The minimum time step size, $\Delta t_{\min}$, for explicit time integration depends on the minimum element length, $l_{\min}$, and the sonic speed, c , i.e., $\Delta t_{\min} = l_{\min} / c$; and

$$c = \sqrt{\frac{E}{(1-\nu^2)\rho}}, \text{ where, } \nu \text{ is Poisson's ratio, } \rho \text{ is specific mass density, } E \text{ is Young's}$$

Modulus.

3.2.2 Drop Test Simulation

In order to study the effects of the pulse width and amplitude of a single half-sine acceleration pulse loading applied on the inner surface of the pivot (B in Fig. 3.1) on the relative displacement between the tip of the actuator arm (A in Fig. 3.1) and the pivot shaft of the actuator arm (connected to the base), several half-sine acceleration pulse loadings with the same amplitude of 600 g (g being the gravitational

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acceleration) and different pulse widths/durations from 0.04 ms to 4 ms were inputted into the numerical model. The acceleration level of 600 g is close to the design shock limit of HDDs. Figure 3.2 shows three cases of the input acceleration pulse loadings in which the abscissa represents the pulse widths, and the y-coordinate represents the amplitudes of these loadings.

Figure 3.3 shows the time histories of the relative displacement for the three different pulse widths of a single half-sine acceleration pulse loading with 600 g amplitude, corresponding to Fig. 3.2. In Fig. 3.3, the abscissa is the time of response and the y-coordinate is the shock response of the relative displacement.

Fig. 3.2 Half-sine pulses with 600 g and different pulse widths/durations

Fig. 3.3 Relative displacement with 600 g and different widths/durations

From Fig. 3.3, it can be found that, the maximum relative displacements occur at different times for different pulse widths with the same amplitude. For different pulse widths, the maximum relative displacement occurs at the first oscillation of vibration for some cases, and/or occurs at the second or third oscillation for some other cases depending on the pulse width.

To investigate the pulse amplitude effect on the shock responses, the amplitude of single half-sine pulse loadings were increased from 600 g to 900 g. Similar numerical simulations were performed to the actuator arm subjected to another group of half-sine acceleration pulse loadings with 900 g amplitude and different pulse widths from 0.1 ms to 4 ms, three cases of them are shown in Fig. 3.4. The coordinates are the

same as those shown in Fig. 3.2.

Figure 3.5 shows the relative displacement historical data for different pulse durations with 900 g amplitude corresponding to Fig. 3.4. The behavior of relative displacement responses, as shown in Fig. 3.5, of the actuator arm subjected to this group of shock loadings is similar to that in Fig. 3.3.

By using the finite element analysis for an actuator arm, the relative displacements were obtained for different pulse widths and different pulse amplitudes of single half-sine acceleration pulse loadings. It is found that, for the same amplitude of the loadings, the maximum relative displacements may be quite different for different pulse widths. As expected, for different amplitudes, the similar behavior but different amplitude of relative displacement responses occurs.

Fig. 3.4 Half-sine pulses with 900 g and different widths/durations

Fig. 3.5 Relative displacement with 900 g and different pulse widths/durations

3.2.3 A Pseudo-Resonance Phenomenon

The variation of the maximum relative displacements ($y_{\max}$) of the actuator arm subjected to half-sine acceleration pulses with amplitude of 600 g and different pulse widths from 0.1 ms to 4 ms is shown in Fig. 3.6.

From Fig. 3.6, it can be found that, the maximum relative displacement increases sharply for pulse widths less than 0.5 ms, reaches the peak value at the pulse width of about 0.6 ms, decreases quickly to 0.2 mm at the pulse width of 1.0 ms, then decreases slowly and approaches a constant value after about 2 ms pulse width.

Fig. 3.6 Variation of the maximum relative displacement ($y_{\max}$) versus pulse width/duration of 600 g amplitude

The dependence of the maximum relative displacements ($y_{\max}$) to the pulse amplitudes of 600 g and 900 g for different pulse widths from 0.1 ms to 4 ms is compared in Fig. 3.7. It is noted that, from Fig. 3.7, for the different amplitudes of a single half-sine acceleration shock, the maximum relative displacements have similar trends versus pulse width. For the both amplitudes of 600 g and 900 g, the maximum relative displacements reach their peak values of $y_{\max}$ at the same pulse width of about 0.6 ms except that their numerical values increases with the amplitude.

What would be the results if certain transformation were performed to Fig. 3.6 and Fig. 3.7, namely transforming the pulse width/duration and/or magnitude of the shock

responses?

Fig. 3.7 Variation of the maximum relative displacement ($y_{\max}$) versus pulse width/duration of 600 g and 900 g amplitudes

Figure 3.8 shows the maximum relative displacement subjected to single half-sine acceleration shocks of an amplitude of 600 g, which is a transformation of Fig. 3.6, in which the abscissa is transformed from pulse duration/width into frequency ratio ($\beta = \omega / \omega_n$), which is the frequency ratio of the *characteristic frequency* of a half-sine acceleration pulse, defined as $\omega = \pi / T$, where T is the pulse duration/width, of the pulse loading to the first natural frequency of the system ω_n .

From Fig. 3.8, it can be found that, for the frequency ratio β of less than a critical value of about 0.6 ($\beta = \omega / \omega_n \approx 0.6$), the maximum relative displacement increases

sharply as the frequency ratio increases, and reaches the peak value at the critical frequency ratio, then decreases gradually after the critical point.

Fig. 3.8 The maximum relative displacement ($y_{\max}$) versus the frequency ratio β of 600 g amplitude

Figure 3.9 shows the dimensionless maximum relative displacements $\bar{y}_{\max} = y_{\max} / (A_0 / \omega_n^2)$ subjected to the same input loadings, which is a transformation of Fig. 3.6 or 3.8, in which the abscissa is transformed from the pulse widths into the frequency ratio; and the y-coordinate is transformed from a physical dimension into a dimensionless one, showing the relationship of the dimensionless maximum relative displacement versus the frequency ratio.

Fig. 3.9 The dimensionless maximum relative displacement ($\bar{y}_{\max}$) versus the frequency ratio β of 600 g amplitude

Fig. 3.10 shows the maximum relative displacement of the actuator arm subjected to a series of single half-sine acceleration pulses with two amplitudes of 600 g and 900 g, which is a transformation of Fig. 3.7, in which the abscissa is transformed from the pulse durations into the frequency ratio.

Figure 3.11 shows the dimensionless maximum relative displacements subjected to the same input loadings, which is a transformation of Fig. 3.10, in which the y-coordinate is transformed from physical dimension into dimensionless one. It can be found that they are almost coincident each other for the dimensionless maximum relative displacements subjected to the two sets of acceleration shocks with the two amplitudes of 600 g and 900 g.

Fig. 3.10 The max relative displacement ($y_{\max}$) versus the frequency ratio β of two amplitudes 600 g and 900 g

Fig. 3.11 The dimensionless maximum relative displacement ($\bar{y}_{\max}$) versus the frequency ratio β of two amplitudes 600 g and 900 g

From Fig. 3.6 to Fig. 3.11, it can be found that, for the both acceleration pulse amplitudes of 600 g and 900 g, the maximum relative displacements of the actuator arm reach the peak value at the same frequency ratio of about 0.6. This resembles a

resonance phenomenon in vibration, and is termed *pseudo-resonance* for convenience in this thesis.

It can be concluded that the dimensionless peak relative displacement of the actuator arm only depends on the frequency ratio β and not the amplitude of the acceleration pulse. This may be useful in some cases. For example, if we obtain the responses of a single half-sine acceleration pulse with a given pulse amplitude and pulse width, we can then predict the responses of another acceleration pulse with the same pulse width, but at different amplitude. This could be useful in those cases where measurements are not possible. This suggests that a scaling law exists between the maximum relative displacement and the amplitude of the acceleration pulse loading.

3.3 Study with an SDOF Model

Studying the vibration of an SDOF system is fundamental to the understanding of more advanced topics in vibrations. Generally speaking, whenever the natural frequency of vibration of a machine or structure coincides with the frequency of external excitation, a phenomenon, known as resonance, will occur, which may lead to excessive deflections and failures. The literature is replete with accounts of system failures brought about by resonance and excessive vibration of components and systems.

3.3.1 A Simplified SDOF Model

To further investigate above pseudo-resonance phenomenon observed in the finite element simulation of the actuator arm, in this section, the actuator arm is simplified as an SDOF system as shown in Fig. 3.12.

The theoretical solution of the SDOF system subjected to a single half-sine pulse loading is then derived by Laplace transform and inverse Laplace transform [Ayre, 2002].

where,

m is the mass,

k is the stiffness,

c is the viscous damping coefficient,

x_1 is the base input displacement,

x_2 is the absolute displacement of the mass.

Fig. 3.12 A simplified model of the actuator arm as an SDOF system

Suppose the SDOF system, as shown in Fig. 3.12, undergoes a harmonic motion. The equation of motion of the system is

$$m\ddot{x}_2 + c(\dot{x}_2 - \dot{x}_1) + k(x_2 - x_1) = 0, \quad (3.1)$$

where, x_1 is usually called as the base excitation.

Consider a single half-sine acceleration pulse as given by equation (3.2).

$$\ddot{x}_1(t) = \begin{cases} A_0 \sin\left(\frac{\pi t}{T}\right), & 0 \leq t \leq T \\ 0, & t > T \end{cases}, \quad (3.2)$$

where, T is the pulse width of the half-sine acceleration pulse, A_0 is the amplitude of the acceleration pulse.

The *characteristic frequency* of the single half-sine acceleration pulse is defined as follows,

$$\omega = \frac{\pi}{T}. \quad (3.3)$$

By convention, let $(c/m) = 2\xi\omega_n$, $(k/m) = \omega_n^2$, where, ω_n is the natural frequency in radians/sec, and ξ is the damping ratio.

And let variable z represents the relative displacement,

$$z = x_2 - x_1. \quad (3.4)$$

The equation of the motion then becomes

$$\ddot{z} + 2\omega_n\xi\dot{z} + \omega_n^2z = -A_0 \sin(\omega t), \quad 0 \leq t \leq T. \quad (3.5)$$

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By taking Laplace transform and inverse Laplace transform, one can obtain the close form solution of the total relative displacement of equation (3.5), which is shown in equation (3.6).

$$\begin{aligned}
 z(t) = & e^{-\xi\omega_n t} \left\{ z_0 \cos(\omega_d t) + \left\{ \frac{\dot{z}_0 + \xi\omega_n z_0}{\omega_d} \right\} \sin(\omega_d t) \right\} \\
 & + \frac{A_0}{\left[(\omega^2 - \omega_n^2)^2 + (2\xi\omega\omega_n)^2 \right]} \left[2\xi\omega\omega_n \cos(\omega t) + (\omega^2 - \omega_n^2) \sin(\omega t) \right] \\
 & - \frac{A_0\omega e^{-\xi\omega_n t} / \omega_d}{\left[(\omega^2 - \omega_n^2)^2 + (2\xi\omega\omega_n)^2 \right]} \left[2\xi\omega_d\omega_n \cos(\omega_d t) + (\omega^2 - \omega_n^2) (1 - 2\xi^2) \sin(\omega_d t) \right]
 \end{aligned} \quad (3.6)$$

where z_0 and $\dot{z}_0$ are the initial relative displacement and initial velocity; respectively.

$\omega_d = \omega_n \sqrt{1 - \xi^2}$ is the damped natural frequency.

The total absolute acceleration for time $0 \leq t \leq T$ is

$$\begin{aligned}
 \ddot{x}_2 = & \ddot{z} + \ddot{x}_1 \\
 = & -\omega_n e^{-\xi\omega_n t} \left\{ (\omega_n z_0 + 2\xi\dot{z}_0) \cos(\omega_d t) + \frac{\omega_n}{\omega_d} [\xi\omega_n z_0 + (1 - 2\xi^2)\dot{z}_0] \sin(\omega_d t) \right\} \\
 & + \frac{A_0\omega^2}{\left[(\omega^2 - \omega_n^2)^2 + (2\xi\omega\omega_n)^2 \right]} \left[-2\xi\omega\omega_n \cos(\omega t) - (\omega^2 - \omega_n^2) \sin(\omega t) \right] \\
 & - \frac{A_0\omega\omega_n e^{-\xi\omega_n t}}{\left[(\omega^2 - \omega_n^2)^2 + (2\xi\omega\omega_n)^2 \right]} \left[2\xi\omega^2 \cos(\omega_d t) + \frac{\omega_n}{\omega_d} (-\omega_n^2 + \omega^2 (1 - 2\xi^2)) \sin(\omega_d t) \right] \\
 & + A_0 \sin(\omega t)
 \end{aligned} \quad (3.7)$$

The relative displacement for time $t > T$ is

$$z(t) = e^{-\xi\omega_n t} \left(z(T) \cos \omega_d (t-T) + \frac{\dot{z}(T) + (\xi\omega_n) z(T)}{\omega_d} \sin \omega_d (t-T) \right), \quad (3.8)$$

where $z(T)$, $\dot{z}(T)$ represent the relative displacement and relative velocity of the system at the end of the duration of the input excitation; respectively. Note that the absolute displacement is equal to the relative displacement for $t > T$.

The relative acceleration for time $t > T$ is

$$\ddot{z}(t) = -\omega_n e^{-\xi\omega_n (t-T)} \left[\omega_n z(T) + 2\xi\dot{z}(T) \right] \cos(\omega_d (t-T)) - \omega_n e^{-\xi\omega_n (t-T)} \left\{ \frac{\omega_n}{\omega_d} \left[-\xi\omega_n z(T) + (1-2\xi^2)\dot{z}(T) \right] \sin(\omega_d (t-T)) \right\}. \quad (3.9)$$

Note that the absolute acceleration is equal to the relative acceleration for $t > T$.

3.3.2 The Pseudo-Resonance Phenomenon

For simplicity, consider an SDOF system with small damping, and assume the initial relative displacement and the initial relative velocity are all equal to zero, i.e., $z_0 = 0$, $\dot{z}_0 = 0$ and ξ is small, in which case, $\omega_d = \omega_n \sqrt{1-\xi^2} \approx \omega_n$. The relative displacement of the responses then reduces to,

$$z(t) = \frac{A_0}{(\omega^2 - \omega_n^2)} \left(\sin(\omega t) - \frac{\omega}{\omega_n} \sin(\omega_n t) \right), \quad (0 \leq t \leq T), \quad (3.10)$$

$$z(t) = z(T) \cos \omega_d (t-T) + \frac{\dot{z}(T)}{\omega_d} \sin \omega_d (t-T), \quad (t > T). \quad (3.11)$$

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Differentiating equation (3.10) with respect to time t , one obtains

$$\dot{z}(t) = \frac{A_0 \omega}{(\omega^2 - \omega_n^2)} (\cos(\omega t) - \cos(\omega_n t)) \quad , \quad (0 \leq t \leq T). \quad (3.12)$$

Let $\dot{z}(t)=0$, one arrives at the following condition in which case the maximum relative displacement will be obtained,

$$\cos(\omega t) - \cos(\omega_n t) = 0. \quad (3.13)$$

From equation (3.13), one can obtain two cases for the relation between ωt and $\omega_n t$.

$$\text{Case 1: } \omega_n t = 2n\pi - \omega t \quad \text{and/or} \quad \omega t = 2n\pi - \omega_n t, \quad (3.14)$$

$$\text{Case 2: } \omega_n t = 2n\pi + \omega t \quad \text{and/or} \quad \omega t = 2n\pi + \omega_n t, \quad (3.15)$$

where n can be any integer.

For Case 1, one has the following relation:

$$\sin(\omega_n t) = -\sin(\omega t). \quad (3.16)$$

Substituting equation (3.16) into equation (3.10), one obtains the following equation

$$z_{\max} = \frac{A_0}{(\omega - \omega_n) \omega_n} \sin(\omega t) \quad , \quad (0 \leq t \leq T). \quad (3.17)$$

For Case 2, one has the following relation:

$$\sin(\omega_n t) = \sin(\omega t) \quad (3.18)$$

Substituting equation (3.18) into equation (3.10), one obtains the following equation:

$$\hat{z}_{\max} = \frac{A_0}{(\omega_n + \omega) \omega_n} \sin(\omega t) \quad , \quad (0 \leq t \leq T). \quad (3.19)$$

Comparing equations (3.17) and (3.19), one can find that the $\hat{z}_{\max}$ from the Case 2

will be always less than the $z_{\max}$ from the Case 1 for the same time t . So we just need to consider the Case 1 in our later discussion.

From equation (3.14), one can get

$$t = \frac{2n\pi}{\omega_n + \omega} \leq T, \quad (3.20)$$

where the coefficient n can only be greater than zero; otherwise it is impossible and meaningless in practice.

Substituting equation (3.20) into equation (3.17), one gets

$$z_{\max} = \frac{A_0}{(\omega - \omega_n)\omega_n} \sin\left(\frac{2n\pi\omega}{\omega + \omega_n}\right), \quad (0 \leq t \leq T). \quad (3.21)$$

Let

$$\beta = \omega / \omega_n, \quad (3.22)$$

which is called the frequency ratio between the angular frequency (as shown in equation 3.3, of the half-sine acceleration pulse) and the natural angular frequency of the system, then one obtains the following equation:

$$z_{\max} = \frac{A_0}{\omega_n^2(\beta - 1)} \sin\left(\frac{2n\pi\beta}{1 + \beta}\right), \quad (0 \leq t \leq T), \quad (3.23)$$

where n can be any integer, $\beta = \omega / \omega_n$.

Referring to equation (3.20), equation (3.23) can only exist under the following condition,

$$\beta \leq \frac{1}{2n-1}. \quad (3.24)$$

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In other words, the equation (3.23) can only be meaningful for $\beta \leq 1$ when n is equal to 1 and for $\beta \leq 1/3$ when n is equal to 2 and so on.

By differentiating equation (3.23) with respect to β , one obtains equation (3.25) under which the peak value of $z_{\max}$ can be derived.

$$\tan\left(\frac{2n\pi\beta}{1+\beta}\right) = \frac{2n\pi(\beta-1)}{(1+\beta)^2} . \quad (3.25)$$

By examination of equations (3.23) and (3.25), one can find that the peak value of $z_{\max}$ will be obtained at $\beta \approx 0.6$ for any natural frequency ω_n and initial amplitude A_0 when n is equal to 1 for $0 \leq t \leq T$. If n is greater than 1, the peak value of $z_{\max}$ within $0 \leq t \leq T$ will be always less than that in the case of $n=1$. Thus, it should only be concerned for the case of $n=1$, which will be discussed later.

The maximum relative displacement should be calculated from equation (3.26) for the time $t > T$, i.e., $\beta > 1$, according to equation (3.11).

$$z_{\max} = \sqrt{\left[z(T)\right]^2 + \left[\frac{\dot{z}(T)}{\omega_n}\right]^2}, \quad (t > T) \quad (3.26)$$

Figure 3.13 shows the relationship between the maximum relative displacement $z_{\max}$ for an SDOF system subjected to single half-sine acceleration pulse loadings, and the frequency ratio β according to equation (3.23).

Figure 3.14 shows the relationship between the dimensionless maximum relative

displacement $\bar{z}_{\max}$ ($= z_{\max} / (A_0 / \omega_n^2)$) and the frequency ratio β for the same case as those in Fig. 3.13.

Fig. 3.13 The maximum relative displacement ($z_{\max}$) with respect to the frequency ratio (β) by SDOF model

Fig. 3.14 The dimensionless maximum relative displacement ($\bar{z}_{\max}$) versus the frequency ratio (β) by SDOF model

Figure 3.15 shows the relationship between the dimensionless maximum relative

displacement $\bar{z}_{\max}$ ($= z_{\max} / (A_0 / \omega_n^2)$) and the frequency ratio β in which a dash curve is included, which is a modification to equation (3.23) for the case of $\beta > 1$.

From Fig. 3.13 to Fig. 3.15, it can be found that, for an SDOF system with small damping ratio subjected to single half-sine acceleration pulse loadings, the maximum relative displacements reach the peak value at a definite frequency ratio of $\beta \approx 0.6$, which coincides with the observation by FE simulation results, as expected.

Fig. 3.15 Dimensionless maximum relative displacement ($\bar{z}_{\max}$) versus the frequency ratio (β) with a dashed curve for a modification

3.3.3 Parametric Study on Damping

As a parametric study, the influence of damping on the maximum relative displacement of an SDOF system with different damping coefficients from 0 to

0.10 subjected to half-sine acceleration pulse was investigated. Figure 3.16 shows four curves of the dimensionless maximum relative displacement ($\bar{z}_{\max}$) versus frequency ratio (β) of an SDOF system with different damping coefficients of 0, 0.01, 0.05, and 0.10; respectively.

Fig. 3.16 Four cases of the dimensionless maximum relative displacement ($\bar{z}_{\max}$) versus frequency ratio (β) of a SDOF system with different damping coefficients from 0 to 0.1.

From Fig. 3.16, it can be found that, for an SDOF system with different damping coefficients subjected to single half-sine acceleration pulse loadings, the maximum relative displacements reach the peak value at the same frequency ratio of $\beta \approx 0.6$. Since both the dimensionless maximum relative displacement and the frequency ratio are all in dimensionless, actual mass and stiffness of the SDOF system are not specified. We conclude that, for the short duration transient problem considered here,

damping does not have enough time to have strong influence to the maximum relative displacement.

The peak relative displacements for an SDOF system and those for an actuator arm are discussed by two models (FEM and SDOF). It is found that the pseudo resonance phenomenon occurs at about the same critical frequency ratio (0.6), by these two models.

3.4 Summary and Concluding Remarks

To investigate the shock dynamic characteristics of HDDs, the relative displacement of an HAA subjected to a single half-sine acceleration pulse during drop tests was simulated by using finite element (FE) model and a simplified single-degree-of-freedom (SDOF) model. The influence of the acceleration pulse width and the pulse amplitude on the shock response of the relative displacement was investigated by using these two models. It is concluded that,

(1) For both the FE model and the SDOF model, the dimensionless peak relative displacement, i.e. $\bar{y}_{\max}$ or $\bar{z}_{\max}$, occurs at a critical frequency ratio (i.e. $\beta = \omega / \omega_n \approx 0.6$), which is the frequency ratio of the *characteristic frequency* of the half-sine acceleration pulse to the first natural frequency of the system. In other words, a *pseudo resonance* phenomenon occurs at the critical frequency ratio.

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(2) Investigation of damping effect on the maximum relative displacement with the SDOF model shows that, for short duration transient problem, damping does not have enough time to have strong influence on the maximum relative displacement.

Chapter 4 Shock Dynamic Characteristics of the HAA Subjected to Different Acceleration Pulse Shapes

4.1 Introduction

In a shock analysis or experiment, in addition to the pulse amplitude and pulse width, the pulse shape is an important parameter to define the acceleration excitation. To emphasize on the fundamental aspects of the pulse shape effects, the dynamic system investigated is still a single actuator arm without inclusion of the head disk interface or suspension. Firstly, in section 4.2, the basic theory of discrete Fourier transform (DFT) is introduced and the power spectrum analysis is interpreted and verified by some trigonometric functions. Secondly, in section 4.3, the natural frequency of the actuator arm was analyzed subjected to half-sine acceleration shocks with an amplitude of 600 g and with two pulse widths of 0.5 ms and 1 ms. Thirdly, in section 4.4, three types of acceleration shocks different in pulse shapes (half-sine, triangular and dual-quadratic waveforms) were selected as input loadings. Dynamic analyses of the actuator arm subjected to these acceleration shocks and power spectrum analysis were performed. Fourthly, in section 4.5, a simple theorem was developed to explain a phenomenon observed in the power spectrum analysis and it was proved

mathematically. A corollary was derived based on the theorem. Lastly, in section 4.6, pulse width correlation between simulation and experiments were studied based on different pulse width definitions.

4.2 Fourier Transform Theory

4.2.1 Discrete Fourier Transform

The power spectrum analysis is based on the theory of discrete Fourier transform (DFT) described as follows. Consider a discrete time function $F(t_j)$, $j=0, 1, 2, \dots, N-1$, the definition of a DFT is given by [Mario and William (2004)]:

$$C_n = \frac{1}{N} \sum_{j=0}^{N-1} F(t_j) e^{-2\pi i (nj/N)}, \quad n = 0, 1, 2, \dots, (N-1), \quad (4.1)$$

where i is the imaginary number unit, defined as $i^2 = -1$. And the inverse DFT is

$$F(t_n) = \sum_{j=0}^{N-1} C_n e^{2\pi i (nj/N)}, \quad n = 0, 1, 2, \dots, (N-1). \quad (4.2)$$

It is important to realize that, in the calculation of the summation indicated in Eq. (4.2), the frequencies increase with increasing index n up to $n=N/2$. It is noted that, for $n>N/2$, the frequencies correspond to the negative of frequencies of order $N-n$. This fact restricts the harmonic components that may be represented in the series to a maximum of $N/2$.

The discrete power spectrum of the function $F(t_j)$ is represented by an array of power spectral values, P_n ($n = 0, 1, 2, \dots, N-1$), defined as

$$P_n = C_n C_n^* = |C_n|^2, \quad n = 0, 1, 2, \dots, (N-1), \quad (4.3)$$

where C_n^* is the complex conjugate of C_n .

The power spectrum is always symmetric about the middle value of n , i.e., $n=N/2$. Thus, $P_n = P_{N-n}$ for $n = 1, 2, \dots, N/2$. Each power spectral value corresponds to a partial frequency, f_n . For the first half of the power spectrum, P_n ($n = 0, 1, 2, \dots, N/2$), the partial frequency is related to n by

$$f_n = n f_0, \quad (4.4)$$

where, f_0 is the base frequency, defined as $f_0 = 1/L_0$, L_0 being the length of sample time. The second half of the power spectrum, P_n ($n = N/2+1, \dots, N-1$), corresponds to negative frequency values can be given by

$$f_n = -(N-n)f_0. \quad (4.5)$$

With the corresponding frequency values, the power spectrum is, then, a function of frequency in the domain from $-(N/2-1)f_0$ to $(N/2)f_0$.

There are many algorithms for performing DFT. For instance, Fast Fourier transform (FFT) is an efficient algorithm for doing it.

An alternative means of representing the power spectrum is the power spectral density, $S(f)$, given in a discrete form by

$$S_n = \frac{P_n}{f_0} = P_n L_0, \quad (4.6)$$

This function represents the power per unit frequency interval. Thus, the power in any given frequency interval can be obtained by integrating it with respect to the frequency.

4.2.2 Interpretation of the Power Spectrum Function

To obtain the power spectrum, one code to perform FFT was programmed by using Matlab software. Some trigonometric functions were tested to verify the correctness of the program.

[Example 4-1] Test Function I — A Simple Sinusoidal Function

Firstly, let us consider a simple sinusoidal function represented by

$$z_1(t) = A_1 \sin(2\pi f_1 t), \quad (4.7)$$

where, $A_1 = 0.2$ is the amplitude of the sinusoidal function, $f_1 = 3$ is the frequency of it.

We obtain the history response of the test function I as shown in Fig. 4.1. For a sample length of ten-unity, we calculate 1024 values using Eq. (4.7) and obtain the

power spectral values using Eq. (4.3). The first half of the power spectrum function of the sinusoidal function is shown in Fig. 4.2, where the one and only frequency component is represented by the only peak at frequency value of 3.

Fig. 4.1 History response of the first test function

Fig. 4.2 Power spectrum versus frequency of the first test function

[Example 4-2] Test Function II—— A Complex Sinusoidal Function

Secondly, let us consider a slightly complex sinusoidal function represented by

$$z_2(t) = A_1 \sin(2\pi \cdot f_1 t) + A_2 \sin(2\pi \cdot f_2 t), \quad (4.8)$$

where, $A_1 = 0.2$ is the first amplitude of the sinusoidal function,

$A_2 = 0.1$ is the second amplitude of the sinusoidal function,

$f_1 = 3$ is the first frequency of it.

$f_2 = 7$ is the second frequency of it.

Similarly, we obtain the history response of the test function as shown in Fig. 4.3.

And the first half of the power spectrum function of the sinusoidal function is shown in Fig. 4.4, where the two frequency components are represented by the two peaks at frequency values of 3 and 7; respectively.

Fig. 4.3 History response of the second test function

According to theorem of power spectrum analysis, the height of the a power peak is equal to $1/4$ times the square of the amplitude of the corresponding frequency component provided that the sample length is an integral multiple of the wavelength of that component. For example, the components with frequency values of 3 and 7 of the sinusoidal function yields power peak of heights equal to 0.01 and 0.0025; respectively, as shown in Fig. 4.4. When the sample length is not an integral multiple of the wavelength, the height of a power peak is slightly lower than $1/4$ times the square of the amplitude, and the peak power is spread to neighboring frequencies. This phenomenon is known as aliasing, and the aliasing error can be reduced if a sufficient number of periods of a frequency component reside in the sample length.

Fig. 4.4 Power spectrum versus frequency of the second test function

4.3 Natural Frequency Analysis from Shock Responses

By using vertical drop tests based on the half-sine acceleration pulse as the input loading, shock dynamic responses and damage may be studied and assessed. To perform a natural frequency analysis to the HAA, the half-sine acceleration pulses with a maximum acceleration of 600 g and with 0.5 ms and 1 ms pulse widths as shown in Fig. 4.5, was applied to the inner surface of the pivot (*B* in Fig. 3.1). The shock response was divided into two time periods: the response of full process, which corresponds to a forced vibration and that after shock was applied, which corresponds to a free vibration.

Fig. 4.5 Two half-sine acceleration pulses with pulse widths of 0.5 ms and 1 ms applied on the inner surface of the pivot

The response history data of the relative displacement between the tip of the actuator arm (A in Fig. 3.1) and the pivot of the HAA (base) for 0.5 ms and 1 ms shock durations are shown in Figs. 4.6, which corresponds to a response of a full process. The power spectra of the full process shock response for 1 ms and 0.5 ms pulse widths are shown in Figs. 4.7.

The response history data of the relative displacement for 0.5 ms and 1 ms shock durations after shocks applied, which correspond to the free vibration are shown in Fig. 4.8. The comparison of power spectrum for 1 ms shock between the forced vibration and the free vibration are shown in Fig. 4.9. The comparison of power spectrum for 0.5 ms shock between the forced vibration and the free vibration are

shown in Fig. 4.10. The excitation power spectra of the two half-sine acceleration shocks for 0.5 ms and 1 ms pulse widths are shown in Fig. 4.11.

Fig. 4.6 Response of the relative displacement of the actuator arm (Full Process)

Fig. 4.7 Power spectrum of the actuator arm (Full Process)

Fig. 4.8 Response of the relative displacement of the actuator arm (after shock)

Fig. 4.9 Comparison of response power spectrum for half-sine acceleration shock with 600g amplitude and 1 ms pulse width

Fig. 4.10 Comparison of response power spectrum for half-sine acceleration shock with 600 g amplitude and 0.5 ms pulse width

Fig. 4.11 Acceleration power of the half-sine shocks for 0.5 ms and 1 ms pulse widths

Comparing the acceleration power spectra of the shock for 1 ms and 0.5 ms pulse widths, it can be found that the acceleration power for the 0.5 ms case has a substantial magnitude at 1.3 kHz while the acceleration power for the 1 ms case is very small at 1.3 kHz. This explains why the response for 0.5 ms pulse width has relative higher magnitude than that for 1 ms pulse width.

By doing FFT analysis, we obtain two resonant frequencies (1.0 kHz, 1.3 kHz) in the forced vibration. They are close to the two natural frequencies (1.0 kHz, 1.33 kHz); respectively, indicating two vibration modes. The frequency of the 1st (lower) mode remains constant while that of the second mode has a shift to a higher value after the shock. The relative magnitude of displacement response for half-sine acceleration pulses is mainly determined by the power magnitude of the acceleration pulse at the resonant frequency.

4.4 Pulse Shape Effect Explained by Power Spectrum Analysis

Although the half-sine acceleration pulse is accepted as the industry standard, the recorded acceleration pulse usually does not have the same shape as that of half-sine one. Figures 4.12 to 4.14 are three typical acceleration pulse recorded by experiments. Obviously, they have some difference with a half-sine acceleration pulse. Therefore, it's essential to study the influence of the pulse shape changes on the dynamic

response of the drive.

Fig. 4.12 One of the experimental acceleration pulse

Fig. 4.13 One of the experimental acceleration pulse

Fig. 4.14 One of the experimental acceleration pulse

4.4.1 Pulse Shape Effect

The FE model of an actuator arm, as shown in Fig. 3.1, was analyzed to study the effects of the pulse shape during the dropping process. Here, three types of simple acceleration shocks different in shape (namely, the half-sine, triangular and dual-quadratic waveforms) with an amplitude of 600 g and with different pulse widths from 0.1 ms to 1 ms (1 ms case as shown in Fig. 4.15), were applied onto the FE model. Note that the dual-quadratic waveform consists of a rising portion in the form of βt^2 , where t is time and β a constant, and a falling portion as a mirror image of the foregoing.

Fig. 4.15 Three acceleration shocks with 1 ms pulse width applied on the inner surface of the pivot

The response history data of the relative displacement between the tip of the actuator arm (*A* in Fig. 3.1) and the pivot of the HAA (base) for 0.1 ms and 1 ms pulse widths are shown in Figs. 4.16 and 4.17; respectively.

From Fig. 4.16, it is observed that, for the 0.1 ms pulse width, the half-sine acceleration shock produces the largest peak displacement, the triangular shock gives the second largest, and the dual-quadratic shock gives the smallest peak displacement.

Fig. 4.16 Shock responses of input acceleration shocks with 0.1 ms pulse width applied on the inner surface of the pivot

Fig. 4.17 Shock responses of input acceleration shocks with 1 ms pulse width applied on the inner surface of the pivot

Letting $M()$ represents the magnitude of a quantity under consideration (peak relative

displacement, in this case) for a specific pulse shape, the results from the above comparison for the 0.1 ms shock duration can be written as

$$M(\text{half-sine}) > M(\text{triangular}) > M(\text{dual-quadratic}). \quad (4.9)$$

However, an abnormal phenomenon is observed for the 1 ms shock duration from Fig. 4.17, i.e., the relationship/sequence of the peak displacements for 1 ms duration is as shown as

$$M(\text{dual-quadratic}) > M(\text{triangular}) > M(\text{half-sine}). \quad (4.10)$$

Fig. 4.18 Peak response of relative displacement vs. pulse widths

It can also be noticed that, from Fig. 4.15, in the time domain, let $M()$ represents the root-mean-square (RMS) for the acceleration pulses of 1 ms and/or 0.1 ms pulse width, the sequence of the RMS for the three different acceleration pulses from large

to small has the same relationship as shown in relation (4.9) for the both pulse widths.

Meanwhile, from Fig. 4.15, let $M()$ represents the area under the pulse curve for the acceleration pulses of 1 ms and/or 0.1 ms pulse width, the sequence of the areas for the three different acceleration pulses from large to small also has the same relationship as shown in relation (4.9) for the both pulse widths.

Fig. 4.19 Maximum relative displacement ($y_{\max}$) versus pulse width

In short, the above opposite phenomena in response for the three different acceleration pulse with different pulse widths are summarized in Fig. 4.18, in which the relation (4.9) and (4.10) are shown as those corresponding to the pulse widths of 0.1 ms and 1.0 ms; respectively. The abnormal phenomena observed above, as given

by relations (4.9) and (4.10), will be explained with a power spectrum analysis that follows.

Figure 4.19 shows the variation of the maximum relative displacements ($y_{\max}$) of the actuator arm subjected to these three types of acceleration shocks with different pulse widths of 0.1 ms, 0.2 ms, 0.4 ms, 0.6 ms, 0.8 ms and 1.0 ms. It can be noted that, the maximum relative displacement experiences a peak value with an increase of pulse width, i.e., a pseudo-resonance phenomenon occurs for each case of the three pulse shapes. For the acceleration shock in a triangular shape, the pseudo-resonance takes place approximately at the same pulse width (~ 0.6 ms) as that for the acceleration shock in a half-sine shape. However, for the acceleration shock in a dual-quadratic shape, it occurs at a different pulse width of about 0.8 ms. There is a critical pulse width for each type of acceleration shock at which the pseudo-resonance phenomenon appears.

4.4.2 Explanation by Power Spectrum Analysis

In the power spectrum analysis, first, the response power spectra (Figs. 4.20 and 4.21) were obtained by performing FFT on the relative displacement responses (Figs. 4.16 and 4.17) for 0.1 ms and 1 ms shock durations; respectively. In Figs. 4.20 and 4.21, the abscissa represents the frequency in kHz while the y-coordinate is the response power in mm^2 .

Fig. 4.20 Shock response power spectra for input acceleration shocks with 0.1 ms pulse width applied on the inner surface of the pivot

From Figs. 4.20 and 4.21, it is found that, at the natural frequency 1.3 kHz, the sequence of the response power for the three different acceleration pulses of 0.1 ms pulse width from large to small has the same relationship as shown in relation (4.9), if the symbol $M()$ is used to represent the magnitude of the response power. However, the case of 1 ms pulse width is the reverse, exhibiting an opposite sequence as shown in relation (4.10). These observations of the power spectra are consistent with the relationship of the relative displacement responses in the time domain as shown in Figs. 4.16 and 4.17.

Fig. 4.21 Shock response power spectra for input acceleration shocks with 1 ms pulse width applied on the inner surface of the pivot

Fig. 4.22 Acceleration power spectra for input acceleration shocks with 0.1 ms pulse width applied on the inner surface of the pivot

Fig. 4.23 Acceleration power spectra for input acceleration shocks with 1 ms pulse width applied on the inner surface of the pivot

Next, the power spectra of the acceleration pulses with three different shapes for the 0.1 ms and 1 ms pulse widths are shown in Figs. 4.22 and 4.23; respectively, in which the abscissa represents the frequency in kHz while the y-coordinate is the acceleration power in mm^2/ms^4 .

From Fig. 4.22, it is found that, for the 0.1 ms shock duration, at the natural frequency of 1.3 kHz, the power magnitude of the half-sine acceleration waveform is the largest, followed those of triangular and dual-quadratic waveforms, i.e., the power magnitudes have the same relationship as that shown in relation (4.9) in this case. Consequently, the corresponding peak displacement for the half-sine waveform is the largest, followed by the triangular and dual-quadratic waveforms, as shown in Fig.

4.16, i.e., the relationship of the peak displacements is consistent with that shown in relation (4.9).

From Fig. 4.23, for the 1 ms shock duration, still at the natural frequency of 1.3 kHz, the reverse phenomenon occurs. The power magnitude of the dual-quadratic waveform is the largest, followed by those of the triangular and half-sine waveforms, i.e., the power magnitudes have the same relationship as that shown in relation (4.10). Therefore, the corresponding peak displacement for the dual-quadratic waveform is the largest, followed by the triangular and half-sine waveforms, as shown in Fig. 4.17, i.e. the relationship of the peak displacements is consistent with that shown in relation (4.10).

In Figs. 4.22 and 4.23, it is also noticed that, there is a cross-over point or small interval for the power spectrum curves of the three pulse shapes. When the natural frequency falls in the region to the left of the cross-over point, the peak displacement of the half-sine acceleration waveform is the largest, followed by those of the triangular and dual-quadratic waveforms. However, when the natural frequency falls in the region to the right of this cross-over point (but to the left of any possible second crossing point), the peak displacement of the dual-quadratic is the largest, followed by those of the triangular and half-sine waveforms. Near the cross-over point, the three pulse shapes give similar magnitudes of the peak displacement responses and the power spectra, which are shown as those in Figs. 4.18 and 4.24 corresponding to the pulse width of 0.8-ms. It can also be found that the largest displacement response and response power occurs when the pulse width is equal to 0.8 ms for the same pulse

shape which is close to the resonant frequency of the system.

In short, the above conclusions can be summarized in two figures as shown in Figs. 4.18 and 4.24, in which the relation (4.9) and (4.10) are shown as those corresponding to the pulse widths of 0.1 ms and 1.0 ms; respectively.

Figure 4.25 shows the variations of acceleration power at the natural frequency of 1.3 kHz for these three types of acceleration shocks with pulse widths of 0.1 ms, 0.2 ms, 0.4 ms, 0.6 ms, 0.8 ms and 1 ms. Comparing Figs. 4.19 and 4.25, it can be found that, the relative relation of the acceleration powers is consistent with that of the corresponding maximum relative displacements as shown in Fig. 4.19.

Fig. 4.24 Power of acceleration at resonant frequency versus pulse width

Fig. 4.25 Power of acceleration at resonant frequency vs. pulse widths

It can also be noted from Fig. 4.25 that, for both the half-sine and the triangular acceleration shocks, the maximum acceleration powers, for a resonant frequency of 1.3 kHz, are reached at the approximately same pulse width of about 0.6 ms. However, for the dual-quadratic acceleration shock, it is reached at about 0.8 ms pulse width. This observation of the acceleration powers gives a reasonable explanation to the pseudo-resonance phenomena observed in Fig. 4.19.

In summary, we conclude that the relative magnitude of the peak displacement of the actuator arm is mainly determined by the power magnitude of the acceleration pulse at the resonant frequency.

4.5 A Simple Theorem and a Corollary

4.5.1 A Theorem for the Coincident Power Magnitude

From Figs. 4.22 and 4.23, it has been noted that, a cross-over point (or small interval) for the three different power spectrum curves occurs at a frequency value (location) of $1/T$, where T is the pulse width (pulse duration). When $T = 0.1$ and 1 ms, the cross-over points occur at about 10 and 1 kHz; respectively. Based on a note by Wang (2004), a simple mathematical theory was developed to explain this phenomenon.

Theorem: *If a given waveform, $h(t)$, can be expressed as a sum of an arbitrary reference waveform, $q(t)$, with a pulse width of T , and two identical and adjacent waveforms, $g(t)$ and $g(t - T/2)$, with the pulse width of each equal to $T/2$, then, the power spectrum curve of the given waveform will coincide with the power spectrum curve of the reference waveform at the frequency values given by $f_n = (2n-1)/T$, $n = 1, 2, 3...$*

Considering the triangular pulse as the reference waveform, for $n = 1$, the frequency of the first coincident point is $f_1 = 1/T$, just as that of the observed cross-over point. It can be shown that the dual-quadratic waveform satisfies the conditions of the theorem exactly, with many coincident points being observed at the predicted frequency values. The conditions of the theorem are an approximation to the situation of the half-sine waveform. Nevertheless, the numerical results shown in Figs. 4.22

and 4.23 demonstrate that the theorem predicts the frequency of its first cross-over (coincident) point with reasonable accuracy.

To prove this theorem, considering the difference between the given excitation, $h(t)$, and the reference waveform, $q(t)$, a function, $u(t)$, is defined as

$$u(t) = h(t) - q(t) = g(t) + g\left(t - \frac{T}{2}\right). \quad (4.11)$$

The coincident points of the power spectra of $h(t)$ and $q(t)$ correspond to the zero power magnitude for $u(t)$. Thus, the frequency for such a zero magnitude can be found as follows.

The Fourier transform of $u(t)$ is given by

$$F[u(t)] = F[g(t)] + F\left[g\left(t - \frac{T}{2}\right)\right]. \quad (4.12)$$

Using the translation theorem for the term involving $t - T/2$ and denoting $F[g(t)]$ by $G(f)$, where f is the frequency, Eq. (4.12) becomes

$$F[u(t)] = G(f)(1 + e^{-i\pi T f}). \quad (4.13)$$

For a zero power, let

$$F[u(t)] = 0. \quad (4.14)$$

If $G(f) = 0$, this is already a zero-power point.

If $G(f) \neq 0$ for any frequency under consideration, the following equation must hold,

$$1 + e^{-i\pi T f} = 0. \quad (4.15)$$

Using the Euler formula for a complex number, Eq. (4.15) can be rewritten as

$$1 + \cos(\pi T f) - i \sin(\pi T f) = 0. \quad (4.16)$$

The real part of Eq. (4.16) gives

$$1 + \cos(\pi T f) = 0, \quad (4.17)$$

and the imaginary part of Eq. (4.16) gives

$$\sin(\pi T f) = 0. \quad (4.18)$$

It can be found that the solutions to Eq. (4.17) and (4.18) are given by $1/T, 3/T, \dots$, or, in summary, $(2n - 1)/T, n = 1, 2, 3, \dots$

4.5.2 A Corollary Based on the Theorem

The coincident points stated in the above theorem are observed as cross-over points/intervals. For $n = 1$, the first cross-over point occurs at $f_1 = 1/T$, which can be defined as the *characteristic frequency* of the acceleration shock. According to this theorem, a corollary can be deduced as follows.

Corollary: *When the characteristic frequencies of a group of acceleration shocks*

with different pulse shapes are very close to the resonant frequency of the dynamic system, these acceleration shocks will have nearly equal acceleration powers at the resonant frequency; consequently, they will produce nearly equal shock responses.

In Fig. 4.25 shown above, it can also be found that there is a narrow cross-over point/interval for the three acceleration power curves around a pulse width of 0.8 ms, and a similar narrow cross-over point/interval also exists in Fig. 4.19 around the same pulse width for the maximum relative displacement. This is because when the pulse width is approximately equal to 0.8 ms, the characteristic frequencies ($1/0.8 = 1.25$ kHz) of these acceleration shocks are very close to the resonant frequency (1.3 kHz) of the actuator arm. These observations confirm the prediction of the corollary stated above.

4.6 Pulse Width Correlation

In numerical simulation, the pulse width is usually defined as a time interval that begins and ends at a zero value of the acceleration (refers to Fig. 4.26). In a shock experiment, however, with a maximum value of the acceleration pulse, $A_{\max}$, the pulse width (duration) is usually defined as a time interval that starts at the instant when the acceleration reaches a certain threshold, A_{ε} ($= \varepsilon A_{\max}$, for example, $\varepsilon = 0.1$ means 10% of the maximum acceleration) and ends at the instant when the acceleration decreases to the same threshold value after passing its maximum. For meaningful comparison of experimental and numerical data, a pulse width conversion is

necessary.

Fig. 4.26 Pulse widths correlation: Experiment versus Simulation

Let T_ε represent the pulse width defined on the basis of the threshold ε ($0 < \varepsilon < 1$) as used in experiments (also termed PW_γ , where $\gamma = 100\varepsilon$, e.g., PW_{50} for $\varepsilon = 0.5$). Thus, T_0 represents the pulse width defined for $\varepsilon = 0$, as in simulation (PW_0). For any given pulse, the correlation between the T_ε and T_0 , as shown in Fig. 4.26, can be expressed as

$$T_\varepsilon = \alpha \cdot T_0, \quad (4.19)$$

where α is a pulse width conversion factor.

Table 4.1 shows the formulae developed for this factor and its values in the case of $\varepsilon = 0.1$. The dual-quadratic pulse shape possesses the smallest value of α , indicating

the largest difference between a pulse shape usually defined in an experiment with a 10% threshold and that used in simulation.

Table 4.1 Pulse width conversion factor for various pulse shapes

Pulse shape	Formula of α	α for $\varepsilon=0.1$
Half-sine	$1 - \sin^{-1}(\varepsilon)/90$	0.936
Triangular	$1 - \varepsilon$	0.900
Dual-quadratic	$1 - \sqrt{\varepsilon}$	0.684
Raised-cosine	$1 - \cos^{-1}(1 - 2\varepsilon)/180$	0.795

Notes: The four acceleration pulses in table 4.1 are defined as follows:

Half-sine waveform: $A = A_{\max} \sin(\pi t/T_0)$

Triangular waveform: $A = 2A_{\max} (t/T_0)$

Dual-quadratic waveform: $A = 4A_{\max} (t/T_0)^2$

Raised-cosine waveform: $A = 0.5A_{\max} [1 - \cos(2\pi t/T_0)]$

4.7 Summary and Concluding Remarks

To study the shock dynamic characteristics of HDDs, this chapter contains the investigation on the pulse shape effect of acceleration pulses on shock responses of the HAA and the corresponding explanation by power spectrum analysis based on the theory of discrete Fourier transform (DFT). Based on the results obtained in this

study, the following conclusions are made.

(1) In the shock responses of the HAA, an abnormal phenomenon was observed where a stronger single half-sine acceleration pulse results in a lower relative displacement compared with those of other two pulse shapes. This has been explained in terms of a power spectrum analysis. We conclude that the relative magnitude of the maximum relative displacement is mainly determined by the power magnitude of the acceleration pulse at the resonant frequency.

(2) A cross-over point (or small interval) was observed in the acceleration power spectrum curves. A simple theorem was developed and proved to illustrate the existence and location of such a cross-over point. When the resonant frequency resides in the region to the left of the cross-over point, the half-sine pulse gives the largest peak displacement, followed by the triangular and dual-quadratic pulses; the reverse is true when the resonant frequency resides in the right of the cross-over point but to the left of any higher-frequency cross-over points. A corollary was derived based on this theorem. The prediction of the corollary was confirmed by numerical results. The theorem and the corollary read as follows.

Theorem: *If a given waveform, $h(t)$, can be expressed as a sum of an arbitrary reference waveform, $q(t)$, with a pulse width of T , and two identical and adjacent waveforms, $g(t)$ and $g(t - T/2)$, with the pulse width of each equal to $T/2$, then, the power spectrum curve of the given waveform will coincide with the power spectrum curve of the reference waveform at the frequency values given by $f_n = (2n-1)/T$, $n =$*

1, 2, 3...

Corollary: *When the characteristic frequencies of a group of acceleration shocks with different pulse shapes are very close to the resonant frequency of the dynamic system, these acceleration shocks will have nearly equal acceleration powers at the resonant frequency; consequently, they will produce nearly equal shock responses.*

(3) Due to noise in experiment, a threshold value is usually used to determine the pulse width. This results in a narrower pulse width than the actual one. Simple formulae were developed for a pulse width conversion between experiment and simulation for equivalency in representing the physical phenomenon.

Chapter 5 Pivot Bearing Modeling and its Stiffness Effect

5.1 Introduction

The pivot assembly is an important rotating subsystem and is used to support the actuator arm and allows its rotation in the horizontal plane. Figure 5.1 shows a schematic diagram of an HDD without a base, with major components annotated. The slider is mounted onto a suspension and an actuator arm, which is in turn fixed to a base plate via a pivot, as shown in Fig. 2.5. The pivot is connected to the base plate through its shaft. The stiffness of the pivot assembly has been regarded as a very important factor for the dynamic behavior of an HAA. The most important components in the pivot assembly are the ball bearings, and many mechanical problems may occur due to the rotating mechanism and the flexibility of the bearings.

In most of the studies about the dynamic response of the HAA, each ball of the pivot bearings was simplified into two linear springs: the first has the stiffness defined in the axial direction and the second in the radial direction. However, it is a challenge to obtain the actual stiffness values of the pivot bearings. Furthermore, there is a lack of clear understanding of many stiffness-related issues relevant to the ball bearing, pivot

and HAA.

Fig. 5.1 Schematic diagram of an HDD without a base

Almost all the studies about the pivot have a consensus that the radial and axial stiffnesses of the pivot can affect the dynamic response of the HAA. However, little literature is available about how strong this effect is. In the present chapter, based on the HAA of a Seagate hard disk drive, a simplified beam model with a torsional spring and a translational spring at the pivot end of the actuator arm was developed. The finite element model of the HAA stated in Chapter 3 was also analyzed for modal analysis with the ANSYS software, and drop test simulation with the LS-DYNA3D software to study the effects of the pivot bearing stiffness on the dynamic characteristics of the HAA.

5.2 A Simplified Beam Model

It is generally accepted that the HAA can be simplified as a beam in theoretical analysis and numerical simulation. However, some differences in boundary conditions and simplified methods exist among various models. The HAA is connected to the base of the drive through the pivot shaft, as shown in Fig. 2.5. Thus, the pivot shaft can be regarded as being fully constrained to the base in the vertical drop analysis. The contribution of the pivot bearing to the vertical dynamic response of the HAA comes from a linear stiffness (in the axial direction of the shaft) and a torsional stiffness, and the later is caused by a combination of the radial stiffnesses (in the radial direction of the shaft) of the two sets of balls in the pivot assembly.

Figure 5.2 shows a simplified beam model of the HAA (without a suspension), in which the beam simulates the actuator arm, the torsional spring represents the torsional stiffness and the translational spring represents the vertical stiffness of the pivot bearing. In Fig. 5.2, k_r is the stiffness of the torsional spring, k_l is the stiffness of the translational spring, E is Young's modulus of the beam material, I is the moment of inertia of a representative cross section and m represents the mass per unit length of the beam, L is the length of the beam. As the failure of the head is mainly caused by the vertical dynamic response of the HAA, this simplified model was used to analyze the drop test response of the HAA.

Fig. 5.2 A simplified beam model of an HAA (without a suspension)

The governing equations of motion of this simplified beam model can be written as follows [Rao, 2004]:

$$EI \frac{\partial^4 y}{\partial x^4} + m \frac{\partial^2 y}{\partial t^2} = p(x, t), \quad (5.1)$$

where $p(x, t)$ on the right hand side is the external loading. When $p(x, t) = 0$, Eq. (5.1) becomes a homogeneous equation and can be written as

$$EI \frac{\partial^4 y}{\partial x^4} + m \frac{\partial^2 y}{\partial t^2} = 0. \quad (5.2)$$

The solution of Eq. (5.2) can be obtained by separating the variables and assuming that the solution of Eq. (5.2) has the form,

$$y(x, t) = \phi(x)Y(t). \quad (5.3)$$

In other words, it is assumed that the free-vibration consists of a constant shape

function $\phi(x)$ and a time dependent amplitude function $Y(t)$.

By separating the variables, following equations can be derived from Eq. (5.2)

$$\frac{\partial^4 \phi}{\partial x^4} - \bar{\omega}^4 \phi = 0. \quad (5.4)$$

where parameter $\bar{\omega} = \sqrt[4]{\frac{\omega^2 m}{EI}}$ and ω is the natural circular frequency of the model.

Assuming the general solution of Eq. (5.4) as follows,

$$\phi(x) = b_1 \cos(\bar{\omega}x) + b_2 \sin(\bar{\omega}x) + b_3 \cosh(\bar{\omega}x) + b_4 \sinh(\bar{\omega}x) \quad (5.5)$$

The boundary conditions of the simplified beam model at the two end points A and B, as shown in Fig. 5.2, are,

$$\frac{\partial^4 y}{\partial x^4}(0, t) = 0, \quad \frac{\partial^3 y}{\partial x^3}(0, t) = 0; \quad (5.6)$$

$$EI \frac{\partial^2 y}{\partial x^2}(l, t) = k_r \frac{\partial y}{\partial x}(l, t), \quad -EI \frac{\partial^3 y}{\partial x^3}(l, t) = k_1 y(l, t); \quad (5.7)$$

respectively.

The dimensionless stiffnesses of the torsional spring, k_r^* , and the translational spring, k_1^* , are defined as

$$k_r^* = \frac{k_r L}{EI}, \quad k_1^* = \frac{k_1 L^3}{EI}, \quad (5.8)$$

and the frequency parameter is defined as

$$\lambda = \varpi L; \quad (5.9)$$

respectively.

By considering the boundary conditions shown in Eqs. (5.6) and (5.7), the frequency parameter can be obtained by solving,

$$\begin{aligned} & k_1^{*2} + k_1^* \frac{\lambda \frac{k_1^*}{k_r} [\text{sh}(\lambda) \cos(\lambda) - \text{ch}(\lambda) \sin(\lambda)] - \lambda^3 [\sin(\lambda) \text{ch}(\lambda) + \cos(\lambda) \text{sh}(\lambda)]}{1 + \text{ch}(\lambda) \cos(\lambda)} \\ & + \lambda^4 \frac{k_1^* [1 - \text{ch}(\lambda) \cos(\lambda)]}{k_r [1 + \text{ch}(\lambda) \cos(\lambda)]} = 0 \end{aligned} \quad (5.10)$$

and the modal shape is given by

$$\phi(x) = \sin(\varpi x) + \text{sh}(\varpi x) + \phi [\cos(\varpi x) + \text{ch}(\varpi x)], \quad (5.11)$$

where,

$$\phi = \frac{\lambda^3 [\cos(\lambda) - \text{ch}(\lambda)] + k_1^* [\sin(\lambda) + \text{sh}(\lambda)]}{\lambda^3 [\sin(\lambda) + \text{sh}(\lambda)] - k_1^* [\cos(\lambda) + \text{ch}(\lambda)]} \quad (5.12)$$

is the shape function parameter.

The solution to the transcendental equation, Eq. (5.10), provides the values of λ which represents the natural frequencies of the model. An analytical solution cannot be obtained for the transcendental equation. However, the frequency parameter λ as a function of k_1^* and/or k_r^* can be derived for some special cases.

Figure 5.3 shows the first frequency parameter λ_1 as a function of k_1^* for the special case $k_r^* = \infty$, and Fig. 5.4 presents the first frequency parameter λ_1 as a function of k_r^* for the special case $k_1^* = \infty$. From these two figures, it is clear that the natural frequency parameter of the model increases together with the increase of the stiffness k_1^* and/or k_r^* , it increases very sharply while the value of the horizontal coordinate is small, and approaches to the asymptote $\lambda_1 = 1.8751$, which represents the special case for $k_r^* = \infty, k_1^* = \infty$ (cantilever beam), with the increase of the value of the horizontal coordinate.

While the stiffness of the two springs $k_r^* \neq \infty$ and $k_1^* \neq \infty$, Fig. 5.5 shows the variation of the first natural frequency parameter λ_1 with k_1^* and k_r^* . From this figure, it can be seen that when the two dimensionless stiffnesses k_1^* and k_r^* are not very high, the first natural frequency is very sensitive to them. However, the effects of the stiffnesses of the pivot bearing on the first natural frequency of the beam become insignificant when k_1^* and k_r^* are both larger than 80. For this case, the first natural frequency of the model approaches that of a cantilever beam. The effects of k_1^* and k_r^* on the other natural frequencies are the same as that on the first natural frequency of the model.

From the analysis results in this section above, while the dimensionless stiffnesses of the torsional and the translational springs of the simplified HAA model are not very large, it needs to be treated carefully in the dynamic analysis of the arm as the effects

of the ball bearing and pivot stiffnesses are very serious in the small range, this effect decreases together with the increase of the stiffnesses.

Fig. 5.3 Parameter λ_1 as a function of k_1^* for the special case $k_r^* = \infty$

Fig. 5.4 Parameter λ_1 as a function of k_r^* for the special case $k_1^* = \infty$.

Fig. 5.5 Parameter λ_1 as a function of k_r^* and k_1^*

5.3 Effect of Pivot Bearing Stiffness by FEM

5.3.1 Range of Pivot Bearing Stiffness

The common range of the stiffness along the axial and radial direction of per ball bearing under different preload can be found from the theoretical and experimental results in Xu *et. al.* (2001). The stiffnesses of the ball bearing are $7 \times 10^6 \sim 20 \times 10^7$ (N/m) along the radial direction and $1 \times 10^6 \sim 5 \times 10^6$ (N/m) along the axial direction.

A real HAA (without suspension) of the Seagate hard disk type C2 is shown in Fig.

5.6. It consists of an arm, a pivot and a voice coil motor (VCM). The material of the arm is aluminum, and stainless steel is used for the pivot. The VCM consists of epoxy, copper and bobbin materials. Its material properties are listed in table 3.1. The length of the arm is about $L=0.04m$, the thickness of the arm is about $t_b=0.0014m$ and the width is about $b=0.005\sim0.022m$. As the pivot of this type has two ball bearings and the axial distance between these two bearings is $0.004666m$, for the different preload of the ball bearing, the common range of the axial stiffness $k_1=(1\times10^6\sim5\times10^6)\times2=2\times10^6\sim1\times10^7 N/m$ and the common range of the torsional stiffness of the pivot is $k_r=(7\times10^6\sim2\times10^7)\times0.004666^2/2=76.2\sim217.7Nm$.

Substitute these data into Eq. (5.4) and use the mean value of the arm width 13.5mm as the uniform width of the arm, the dimensionless stiffnesses of the torsional and the translational springs in the simplified beam model can be approximately assessed as:

$$k_r^* = \frac{k_r L}{EI} \approx 13.5 \sim 38.6, \text{ and } k_1^* = \frac{k_1 L^3}{EI} \approx 567 \sim 2836$$

From the theoretical analyses in section 5.2, it can be obtained that in the common range of the stiffnesses of the pivot ball bearing, the effects of the ball bearing and pivot stiffnesses to the vertical dynamic properties of the arm is sensitive to the stiffness of the pivot ball bearing.

Fig. 5.6 The HAA (without a suspension) of the Seagate hard disk type C2

Table 5.1 Values of torsional stiffness k_r and axial stiffness k_1

Case	Torsional stiffness k_r (N-m)	Axial stiffness k_1 (N/m)
1	76.2	6×10^6
2	146.95	6×10^6
3	217.7	6×10^6
4	146.95	2×10^6
5	146.95	1×10^7

In order to study the effects of the pivot bearing stiffnesses on the vertical dynamic response of the arm of the HAA in the normal shock range for the hard disk drive industry, five cases of the pivot stiffnesses were investigated in the present work. The

values of the torsional stiffness k_r and axial stiffness k_1 are listed in table 5.1.

5.3.2 Modal Analysis

The FE model for modal analysis is still the same as that shown in Fig. 3.1. Modal analysis was conducted with this FE model with different pivot stiffnesses. The resonant analysis results of the vertical bending modes are listed in table 5.2 for the five cases of pivot stiffness as listed in table 5.1. The three mode shapes are shown in Fig. 5.7.

From the data in table 5.2, it can be seen that, for Cases 1, 2 and 3, with the axial stiffness k_1 being kept constant (refer to in table 5.1), the resonant frequencies of the arm bending modes increase as the torsional stiffness k_r increases. For Cases 4, 2 and 5, with the torsional stiffness k_r being kept constant (table 5.1), the resonant frequencies of the arm bending increase as the axial stiffness k_1 increases.

Table 5.2 The predicted resonant frequencies (kHz)

Case	First bending mode	Second bending mode	Third bending mode
1	1.251	4.879	8.760
2	1.320	5.107	8.759
3	1.361	5.238	8.897
4	1.243	5.023	8.538
5	1.363	5.528	8.905

Fig. 5.7 Three arm bending mode shapes

Figure 5.8 shows that the resonant frequency of the first arm bending mode increases with increasing stiffness k_1 or k_r , accompanied by a reduced slope. This resonant frequency increases only by 8-9% from the minimum value to the maximum value of the normal range of the pivot bearing stiffnesses for the hard disk drive industry (about 200% increase in k_r and about 400% increase in k_1). The resonant frequencies of the second and third arm bending modes have the same trend as that of the first mode does.

The numerical modal analysis shows that the resonant frequencies of the arm bending

modes of the HAA increase as the stiffnesses increase. In the normal range of the pivot bearing stiffnesses for the hard disk drive industry, the sensitivity of the resonant frequency of the arm bending modes to the pivot bearing stiffnesses is not very serious.

(a)

(b)

Fig. 5.8 The resonant frequency of the first arm bending mode as functions of (a) the torsional stiffness (b) the axial stiffness of the pivot bearing

5.3.3 Vertical Drop Test Simulation

The FE model shown in Fig. 3.1 was also used to investigate the dynamic response of the actuator arm in vertical drop test simulation for different pivot bearing stiffnesses with the LS-DYNA software. Since the HAA is connected to the base through the inner surface of the pivot, i.e., the pivot shaft, the boundary condition of the model is that the degrees of freedom of all the nodes on the pivot shaft are fully constrained. The initial velocity was specified as 5 m/s for the whole model at the time $t = 0$ to simulate the process in which the hard disk drive drops from a height of about 1.25 m.

The 5-ms history data of the relative displacement at the tip of the arm (Point *A* in Fig. 3.1) for Cases 1, 2 and 3 are shown in Fig. 5.9. In these cases, the axial stiffness k_1 is kept constant, and the torsional stiffness k_r is 76.2, 146.95 and 217.7 N-m; respectively. The relative displacement history data during the vertical drop tests for these three cases almost overlap one another and cannot be distinguished. It implies that the effect of the torsional stiffness of the pivot bearing on the responses of the arm during the vertical drop tests is very small. Therefore, when the torsional stiffness of the pivot bearing varies within the normal range for the hard disk drive industry, its effect can be neglected.

Figure 5.10 shows the relative displacement data for Cases 4, 2 and 5. For these three cases, the torsional stiffness k_r is kept constant, and the axial stiffness k_1 is set equal to 2×10^6 , 6×10^6 and 1×10^7 N/m; respectively. It can be seen in this figure that the maximum relative displacement at Point *A* of the arm decreases as k_1 increases.

The relative deviation of the maximum displacement among these three cases is about 20%. This phenomenon is partly because the large part of the HAA is concentrated around the pivot. Thus, when the axial stiffness decreases, the inertial contribution of the large mass block of the VCM and the arm around the pivot causes the larger displacement of the whole HAA.

Fig. 5.9 The displacement data for Cases 1, 2 and 3

Five cases of vertical drop tests from the same height and different axial and torsional stiffnesses of the pivot bearing of the HAA were simulated in this section using the explicit finite element software LS-DYNA. For the normal pivot stiffness range in the hard disk drive industry, numerical results show that the vertical displacement of the actuator arm during the vertical drop test is insensitive to the torsional stiffness of the pivot bearing. In contrast, the increase of the axial stiffness of the pivot bearing can

reduce the maximum displacement of the actuator arm during the vertical drop.

Fig. 5.10 The displacement data for Cases 4, 2 and 5

5.4 Summary and Concluding Remarks

To study the effects of the pivot bearing stiffness of the HAA, a simplified beam model was developed. The pivot bearing stiffness effects were analyzed by conducting modal analysis and drop test simulation with the FE model of the HAA. Based on the results obtained from this study, the following conclusions are made.

- (1) The frequency parameter λ of the simplified beam model was derived as a function of dimensionless stiffnesses, k_1^* and/or k_r^* , of a torsional spring and a

translational spring. It was noted that when the two dimensionless stiffnesses are not very high, the vertical bending modes of the beam model are sensitive to them and the stiffness effects need to be considered carefully in the dynamic analysis.

(2) FE numerical results of modal analysis and drop test simulation of the HAA show that the resonant frequencies of the arm bending modes increase with the stiffness. In the normal pivot stiffness range for the hard disk drive industry, the sensitivity of the arm bending modes to the stiffness of the pivot bearing is not very significant. The maximum relative displacement of the arm during the vertical drop test is insensitive to the torsional stiffness, but is sensitive to the axial stiffness, of the pivot bearing.

Chapter 6 Dynamic Characteristics of a Micro-Drive in Operating State

6.1 Introduction

The interest in the effects of shock on hard disk drives (HDDs) has come into currency due to the increasingly hostile environments encountered in the usage of the portable computer as well as the application of HDDs in consumer devices such as portable MP3 audio players, personal digital assistants (PDAs) and high-end digital cameras. The benefits of HDD storage in these applications are massive amounts of data storage at a low cost per giga-byte, rapid data transfer between the storage medium and the device, and close-to-instant access to any desired photo, musical selection or video program. As non-traditional application of HDDs emerges, their mechanical robustness under shock and other mechanical disturbances during different states are of great concern. Normal drives can not malfunction during the operating state, damage during initial assembly, testing, installation and all unfavorable situations caused by final users.

As HDDs continue to be made smaller in size, higher in capacity, and lower in cost, more and stronger disturbance during the operating state are encountered, especially

with the emergence of micro-drives for consumer electronics. The mechanical components of micro-drives become very delicate, and a better understanding of their dynamic property is essential. The dynamic characteristics of a micro-drive are investigated using both experimental and numerical techniques in this chapter.

Figure 6.1 shows a photograph of a 1-inch micro-drive, S1 from Seagate Technology, with major components/assemblies annotated and its top cover removed. The drive consists of a 1-inch disk rotated by a fixed-shaft spindle motor, two actuator arms that are driven by a voice coil motor, one or two suspension assemblies, a base and a cover. When power is applied, the disk rotates in steady state at a speed of 3,600 RPM. The slider(s) fly over the surface of the disk on the air bearing generated by the relative motion between the rotating disk and the stationary slider(s).

The main mechanical components/assemblies of the drive include the disk, the head arm assembly (HAA), the head stack assembly (HSA) and the head disk assembly (HDA) etc. Figure 6.2 shows a photograph of the HAA with major components annotated. Figure 6.3 shows a photograph of the HSA. The physical assembly relation of the HSA is shown in Fig. 6.4. There are two kinds of standard configurations, as shown in Fig. 6.5, of the HSA with one or two heads; respectively.

Fig. 6.1 Photograph of an 1-inch micro-drive with its top cover removed, S1 from Seagate Technology, with major components/assemblies annotated.

Fig. 6.2 Photograph of an HAA with major components annotated

Fig. 6.3 Photograph of an HSA

Fig. 6.4 Physical assembly relation of HSA

Fig. 6.5 Two kinds of standard configurations of HSA

This chapter is organized as follows. Two schematic diagrams of experimental set-up for modal test and damping measurement are introduced in section 6.2. Finite element models of the disk, the HAA, the HSA, and the HDA, of the S1 drive, are described in section 6.3. In section 6.4, several results by experimental and numerical techniques are presented, including the results by modal analysis, harmonic analysis, operational shock analysis, as well as power spectrum analysis and temperature effect analysis. Discussions about these results and comparison between them are presented. Some design considerations are suggested for reducing the shock responses. A summary and concluding remarks are given in the last section, section 6.5.

6.2 Experimental Setups with Laser Doppler Vibrometer

6.2.1 Experimental Setup for Modal Test

A schematic diagram for the experimental set-up is shown in Fig. 6.6. Figure 6.7 shows a photograph of the experimental setup with major experimental facilities annotated. Figure 6.8 shows the physical assembly relation for fixing the HAA specimen.

Fig. 6.6 Schematic diagram of experimental set-up for modal test

Fig. 6.7 Photograph of the experimental setup with major experimental facilities annotated

Fig. 6.8 Physical assembly for fixing the HAA specimen

As the size of its disk is only 1-inch in diameter, the mechanical components of the HAA of the micro-drive are very delicate. A special fixture, as shown in Fig. 6.9, was designed for fixing the HAA specimen. The Polytec LDV was used to measure the absolute velocity responses of the slider, the suspension and the arm by scanning at one point, or along a line or on an area. The Polytec Scanning Vibrometer system (analyzer) was adopted for data acquisition and analysis to obtain the frequency response functions (FRFs) between the output signals and the input signals. Chirp signal provided by the analyzer was selected for the input excitation. An accelerometer was used to measure the input acceleration from the shaker.

Fig. 6.9 Fixture designed for fixing the HAA specimen

Several cases for modal test of the HAA were performed based on the experimental set-up. First, it is necessary to do some preparations such as the calibration of the accelerometer, the parameter setup for the analyzer software before the formal experiment begins. Then, we perform a group of tests without using the accelerometer. In this group of experiments, three scanning points, points A, B, and C,

as shown in Fig. 6.10, were selected on the slider, the lift-tab of the suspension and the arm; respectively. Another point D is scanned on the fixture, as shown in Fig. 6.9, to obtain the base frequencies. Thirdly, we perform another group of tests by adding the accelerometer to measure the input signal. Transmissibility of the output signals over those of the input can be obtained. Finally, we defined and scanned some lines/areas to obtain the mode shapes for each mode.

(a) Slider

(b) Lift-tab

Fig. 6.10 Three scanning points: (A) point A on the slider; (B) point B on the lift-tab;
(C) point C on the arm tip

(c) Arm

Fig. 6.10 (continued) Three scanning points: (A) point A on the slider; (B) point B on the lift-tab; (C) point C on the arm tip

Experimental results will be shown in section 6.4 together with those by FEM simulation.

6.2.2 Experimental Setup for Damping Measurement

As shown in Fig. 6.11, the schematic diagram of experimental setup for damping measurement is very similar to that for modal test except that the use of the waveform generator, the power amplifier, the charge amplifier, the accelerometer, and the shaker is omitted. The shaker is just used as a table for mounting the fixture, which is, in turn, used for fixing the HAA specimen. The point A on the slider, as shown in Fig. 6.10, was selected as the scanning point for measuring the time-response (velocity) of the HAA. The acquisition settings for the analyzer software are set as shown in Fig. 6.12.

Fig. 6.11 Schematic diagram of experimental set-up for damping measurement

Fig. 6.12 Acquisition settings for damping measurement

Figure 6.13 shows the photograph of the main experimental equipments for the

damping measurement. When the suspension is given an initial perturbation on the lift-tab and released suddenly, a free vibration of the suspension will be excited and the vibration will be damped out gradually. The Polytec LDV was used to measure the free vibration time-response of the HAA. Then the most popular time-response method, the logarithmic decrement method will be used to measure the damping ratio of the HAA. The experimental results will be shown in section 6.4.3.

Fig. 6.13 Photograph of the major experimental facilities for the damping measurement

6.3 Finite Element Modeling of a Micro-Drive (S1 Drive)

We created several FE models of the S1 drive, including those of the disk, the head

arm assembly (HAA), the head stack assembly (HSA), and the head disk assembly (HDA). They are new and novel FE models. The details of these FE models are presented in the following sub-sections, separately.

6.3.1 Disk

It is essential to know the dynamic property of a disk for understanding its shock effects. The disk is connected to the base with a spindle bearing system. FE models of the disk meshed with shell or solid elements (SHELL181/SHELL163 or SOLID185/SOLID164) were created with different boundary conditions. Figure 6.14 shows the geometry of the disk with major dimensions annotated. Four boundary conditions (Fig. 6.14) were studied. The material property of the disk is listed in table 6.1. The results are shown in section 6.4. There are several assumptions and limitations of the current FE model of the disk. Firstly, the disk is simplified as an isotropic plate instead of the real structure, which consists of multi-layer different materials. Secondly, the effect of the spindle bearing is not considered. Thirdly, the temperature effect is also not considered here.

Table 6.1 Material Properties of the Disk

Young's Modulus (GPa)	Poisson Ratio	Density (kg/mm ³)	Thickness (mm)
83.5	0.33	2.554E-6	0.381

6.3.2 Head Arm Assembly (HAA)

There are two standard configurations for the one-inch micro-drive with one or two reading/writing heads and corresponding one or two head arm assemblies (HAAs). As was shown in Fig. 6.2, the HAA includes an arm, a hinge, a suspension/load beam, a gimbal, and a head/slider. The slider is made of SiC. All the other components of the HAA are made of stainless steel. The arm, the hinge, the suspension and the gimbal are welded together. The slider is glued to the gimbal. When a preload is

applied, the load beam presses down the gimbal through a dimple. When the load beam lifts off, it can separate from the gimbal at the dimple.

FE Model in Details. The FE model of the HAA is shown in Fig. 6.15. The arm is meshed with two layers of solid elements (SOLID185/SOLID164). The slider is also meshed with the solid elements. The hinge, the load beam and the gimbal are meshed with shell elements (SHELL181/SHELL163). The yellow part at the left end of the arm denotes the clamped area. The hinge is formed with an initial angle of about 10 degree at its middle line, as shown in the lower part in Fig. 6.15. Consequently, a preloading of about 1.2 gf will produce between the slider and the disk.

Fig. 6.15 FE model of HAA

Physically, the slider is glued with the gimbal; and the arm, hinge, suspension, gimbal are connected together through welding and the rotational degrees of freedom at the

welding point should be constrained. These welding connections were simulated with coupled degrees of freedom (CDOFs) between solid/shell elements and shell/shell elements. Even the solid elements only have translation degrees of freedom, the rotation degrees of freedom of the nodes at the welding point can be constrained through the constrained translation degrees of freedom at more than one node in a plane. This FE model of the HAA has 6357 nodes and 4111 elements, including 2590 solid elements and 1521 shell elements. The material properties of the HAA are listed in table 6.2.

Table 6.2 Material Properties of HAA

Materials	Young's Modulus (GPa)	Poisson Ratio	Density (kg/mm ³)
Slider (SiC)	190.3	0.49	4.4160E-6
Others (Stainless Steel)	190.3	0.32	8.0793E-6

Boundary Conditions and Preloadings. As shown in Fig. 6.15, the nodes on the clamped area are fully constrained. To apply the preloading on the FE model, the z-displacement loadings were applied to the four corners of the slider to pull the slider from the free state to the 'z-height' position. The preloading can be conducted by doing a nonlinear static analysis with the FE model subjected to the z-displacement loadings. After the static analysis, the locations of the nodes in the FE model should be updated for the following steps of dynamic analysis, such as modal analysis, harmonic analysis, and transient analysis.

Assumptions and Limitations. The real structure of the slider is very complicated. It consists of many thin layers of different materials (Gupta, 2001) The slider is simplified as a cubic solid structure with single material. The welding points between the arm and the hinge and those between the hinge and the suspension are simulated using CDOFs between them. The gluings between the slider and the gimbal are also simulated with CDOFs between them. CDOFs also defined between the load beam and the gimbal, and the T-limiter etc. We did not consider the temperature effect in the FE model for the dynamic analyses of the HAA.

6.3.3 Head Stack Assembly (HSA)

In the last two sections, the FE models of the HAA and the disk have been introduced. In this section, the FE model of the HSA with one head will be presented. Besides the various components of the HAA, the over mold with inlaid coil, the pivot bearing, a shim and a screw were incorporated in the model. The pivot bearing, the over mold, the HAA are screwed together. The pivot bearing consists of two columns of balls, a cartridge, a shaft and a sleeve. The cartridge, the shaft, the sleeve, the balls, the shim, and the screw are made of stainless steel. The over mold is made of epoxy. The coil is composed of layered-winding copper threads and inlaid into the over mold.

FE Model in Details. The FE model of the HSA with one head is shown in Fig. 6.16. The over mold, the shaft, the cartridge, and the shim are meshed with solid elements (SOLID185/SOLID164). Those components of the HAA in the HSA have a similar mesh as stated in section 6.3.2. The pivot bearing has two columns of nine balls each.

Each ball was simulated as being an inclined spring element (COMBIN14/COMBIN165), as shown in Fig. 6.17. The spring elements connect the shaft and the cartridge. The stiffness of the pivot bearing and the spring constant are listed in table 6.3. This FE model of the HSA has 6732 nodes and 4782 elements, including 3243 solid elements, 1521 shell elements and 18 spring elements. The material properties of the HSA are listed in table 6.4.

Boundary Conditions and Preloadings. A fully constrained boundary condition was applied on the shaft of the HSA. The preloading can also be applied to the HSA as stated in section 6.3.2 for further dynamic analysis.

Assumptions and Limitations. The assumptions and limitations related to the HAA are also applicable here. Moreover, the coil of the HSA has anisotropic property, and it was simplified as an isotropic material in order to avoid the too complex simulation of an anisotropic material. In the FE model, all the components of the HSA were assumed to have the same nodes on their common boundaries/surfaces.

Fig. 6.16 FE model of HSA with one head

Fig. 6.17 There are two columns of balls in the pivot bearing and each ball was simulated as being an inclined spring element.

Table 6.3 Stiffness of Pivot Bearing and Springs' Parameters

Items	Values	Units
Column Number of Ball Bearings	2	-
Number of Springs/Balls Per Column	9	-
Total Number of Springs/Balls	18	-
Overall Axial Stiffness	1119	N/mm
Overall Radial Stiffness	8124	N/mm
Springs' Constant	1027	N/mm
Springs' Inclined Angle	20.36	degree

Table 6.4 Material Properties of HSA

Materials	Young's Modulus (GPa)	Poisson Ratio	Density (kg/mm ³)
Slider (SiC)	190.3	0.49	4.4160E-6
Over Mold (Epoxy)	15.0	0.39	1.6500E-6
Coil (Copper)	87.9	0.29	8.07930E-6
Spring	10.2	0.29	8.07930E-6
Others (Stainless Steel)	190.3	0.32	8.07930E-6

6.3.4 Head Disk Assembly (HDA)

FE Model in Details. In the last three sections, the FE models of the HAA and the HSA, as well as the disk have been introduced. In this section, the FE model of the HDA will be presented. The HDA is composed of the disk, the HAA and the air bearing between the slider and the disk. The compositions of the disk and the HAA are the same as those stated in section 6.3.1 and 6.3.2. The overall FE model is shown in Fig. 6.18. In the model, the meshing of the disk and the HAA is similar to those stated in the previous sections. The air bearing between the slider and the disk was simulated as being five spring elements (COMBIN14/COMBIN165), as shown in Fig. 6.19. The three spring constants were calculated according to the air bearing stiffnesses, including its translational stiffness, its pitch stiffness and its roll stiffness, obtained by CML code. Table 6.5 shows the air bearing stiffness matrix obtained by CML code when the slider is located at the outer diameter of the disk. The

diagonal items represent the translational stiffness (K_z), the pitch stiffness (K_p) and the roll stiffness (K_r); respectively. The formulae for calculating the spring constants from the air bearing stiffnesses are shown in table 6.6. This FE model of the HDA has 4861 nodes and 3642 elements, including 1601 solid elements, 2036 shell elements and 5 spring elements. The material properties of the model are listed in table 6.7.

Boundary Conditions and Preloadings. Before performing the shock analysis, the HAA was preformed with an initial bending angle at the middle line of the hinge to produce a prescribed preloading force. CDOFs are defined between the slider and the disk for applying the preloadings. Due to the CDOFs between the slider and the disk, the disk will rotate the same angle simultaneously.

Fig. 6.18 FE model of HDA

Assumptions and Limitations. The assumptions and limitations of the disk, the HAA stated in previous sections are also applicable here for the HDA. We did not

consider the temperature effect in most of the dynamic analyses except the numerical results presented in the section 6.4.5.5, in which the temperature effect is considered.

Fig. 6.19 Air bearing simulation by five spring elements

Table 6.5 Air bearing stiffness matrix

Items	Height	Pitch	Roll
Load (g)	0.135598	0.613109E-1	0.586374E-3
*Pitch-torque ($\mu\text{N-M}$)	0.482297	0.328664	-0.484511E-2
*Roll-torque ($\mu\text{N-M}$)	-0.625791E-2	-0.116873E-1	0.268667E-1

*Notes: Refer Fig. 6.19, *pitch* is defined as the relative displacement between the points P_1 and P_2 on the slider divided by the length (L) of the slider. Similarly, *roll* is defined as the relative displacement between the points P_3 and P_4 on the slider divided by the width (b) of the slider.

Table 6.6 Spring Constants

Spring Constant	Formula
k_1	$2K_p / L^2$
k_2	$2K_r / b^2$
k_3	$K_z - 2k_1 - 2k_2$
<p>where, K_p is the pitch stiffness of the air bearing;</p> <p>K_r is the roll stiffness of the air bearing;</p> <p>K_z is the translational stiffness of the air bearing;</p> <p>L is the length of the slider;</p> <p>b is the width of the slider.</p>	

Table 6.7 Material Properties of HDA

Materials	Young's Modulus (GPa)	Poisson Ratio	Density (kg/mm ³)
Slider (SiC)	190.3	0.49	4.4160E-6
Disk	83.5	0.33	2.5540E-6
Others (Stainless Steel)	190.3	0.32	8.0793E-6

6.4 Results and Discussions

6.4.1 Modal Analysis of the Disk

A rotating disk has three types of modes in vibration. The first type is symmetric modes with l nodal circles and no nodal diameters, i.e., $(l, 0)$. These modes are also referred to as the ‘umbrella’ modes of vibration (Fig. 6.20a). The second type is asymmetric modes with zero nodal circles and n nodal diameters i.e., $(0, n)$ (Fig. 6.20b). The third type is coupled asymmetric modes i.e., (l, n) ($l > 0, n > 0$) (Fig. 6.20c), which are higher frequency modes.

Table 6.8 shows the modal analysis results of the disk modeled by shell and/or solid elements with different boundary conditions, as stated in Fig. 6.14. As a comparison, the corresponding experimental results are also listed in this table. In table 6.8, the $(0, 1)$ mode is also referred to as ‘tilt mode’ and the $(0, 2)$ mode is referred to as ‘flutter mode’. It can be found, from table 6.8, that in the simulation results annotated by one asterisk (*) with boundary conditions B1 to B3, the first umbrella mode $(0, 0)$ is not the first/lowest frequency mode and has large difference with the experimental results. Those annotated by double asterisk (**) with boundary condition B4 are closer to the experimental results, especially for the first/lowest frequency mode $(0, 0)$, i.e. the first umbrella mode. All the modes with B4 boundary condition are in the same order as those obtained by the experiment. This indicates that the simulated boundary condition B4 is close to the experiment.

a) Umbrella mode

b) Asymmetric mode

c) Coupled mode

Fig. 6.20 Vibration modes of the disk: a) umbrella mode, b) asymmetric mode, and c) coupled mode

Table 6.8 Numerical and experimental modal analysis results of the disk

No	B1-Shell*	B2-Shell*	B3-Solid*	B4-Shell**	Experiment#
1	1790(0, 1)	3240(0,1)	2470(0, 1)	2135(0, 0)	2150(0, 0)
2	1790(0, 1)	3240(0,1)	2470(0, 1)	2376(0, 1)	2425(0, 1)
3	1910(0, 0)	3360(0, 0)	2730(0, 0)	2378(0, 1)	2744(0, 1)
4	3330(0, 2)	4070(0, 2)	3650(0, 2)	3815(0, 2)	3750(0, 2)
5	3330(0, 2)	4070(0, 2)	3650(0, 2)	3815(0, 2)	3906(0, 2)
Notes: (0,0) —— umbrella mode ; (0,1) —— Tilt mode; (0,2) —— Flutter mode					
# H Harmoko performed the experimental results were Seagate.					

6.4.2 Modal Test/Analysis and Harmonic Analysis of the HAA

In the modal test, natural frequencies of the HAA were obtained by scanning on points A, B, C and D, as shown in Figs. 6.10 and 6.9. Figures 6.21 (a), (b), (c) and (d) show the measured frequency response function curves between the range of 0 to 5000 Hz on the four scanning points. It can be found, from Fig. 6.21, that there are about 8 obvious frequency peaks between the range of 0 to 5000 Hz, including the HAA's frequencies and the base frequencies. Four frequencies among them are the base frequencies of the fixture, as shown in Fig. 6.21(d), and the other four

frequencies of them represent the natural frequencies of the HAA. The mode shapes were obtained by scanning along curve(s) and/or on area(s), in the modal test. Figure 6.22(a) and (d) show the suspension rigid bending mode and the first bending mode; respectively. Figure 6.22(b) and (c) show the slider-gimbal bending mode and its torsion mode; respectively.

By using finite element simulation, the natural frequencies and the corresponding mode shapes were obtained by doing modal analysis with the FE model of the HAA. Some modifications were made to the FE model according to the modal test results. Figure 6.23 shows the mode shapes of the HAA by finite element simulation corresponding to the experimental results as shown in Fig. 6.22. Table 6.9 shows a few modal frequencies among the range of 0 to 4 kHz by two approaches, i.e. FE simulation and experiment. It is noted that most of the FE simulation results agree well with those by experiment except the gimbal and slider bending mode.

Harmonic analysis was also performed with the FE model of the HAA to obtain the dominant modes subjected to a sinusoidal load applied on the clamped area, as shown in Fig. 6.15. Table 6.10 shows the natural frequencies of the HAA by performing FE modal analysis and the corresponding results by harmonic analysis. In table 6.10, the symbolic mark (✓) in the column of the harmonic analysis implies that the corresponding natural frequency of the HAA is a dominant mode (DM). Figure 6.24 shows the harmonic response of displacements of the HAA on the slider.

Chapter 6: Operational Shock Simulation and Modal Test

(a) Slider

(b) Lift-tab

(c) Arm

(d) Base

Fig. 6.21 FRF curves obtained at four locations: (a) A on the slider, (b) B on the lift-tab, (c) C on the arm, and (d) D on the base

(a) Suspension rigid bending mode

(b) Slider-gimbal bending mode

(c) Slider-gimbal torsion mode

(d) Suspension first bending mode

Fig. 6.22 Four mode shapes of the HAA by experiment

(a) Suspension rigid bending mode

(b) Slider-gimbal bending mode

(c) Slider-gimbal torsion mode

(d) Suspension first bending mode

Fig. 6.23 Four mode shapes of the HAA by FE simulation

Table 6.9 Modal Analysis of the HAA in its Free State by FE simulation and Experiment [Unit: Hz]

No	FE Simulation	Experiment	Error (%)	Mode Statement
1	265	265	0	Suspension Rigid Bending
2	1588	775	?	Gimbal & Slider Bending
3	2014	1994	1.0	Gimbal & Slider Torsion
4	3883	3806	2.0	Suspension 1 st Bending

Fig. 6.24 Harmonic response of the HAA

Table 6.10 Modal Analysis of the HAA in its Free State [Unit: Hz]

No	Modal Frequency	Harmonic Analysis (DM)	Mode Statement
1	265	√	Suspension Rigid Bending
2	1588	√	Gimbal & Slider Bending
3	2014		Gimbal & Slider Torsion
4	3883	√	Suspension 1 st Bending
5	4788		Gimbal Bending
6	5816	√	Suspension 1 st Torsion
7	8404	√	Arm+Suspension Bending
8	8681		Gimbal Torsion
9	12140	√	Arm B+Suspension T
10	12768	√	Suspension Torsion
11	15075		Gimbal Bending
12	16502		Sway
13	16643	√	Suspension Torsion
14	20308		Gimbal Torsion
15	20958		Gimbal Bending

It can be found, from table 6.10 and Fig. 6.24, that there are about 13 natural frequencies in the range of 0 to 20 kHz for the HAA and there are mainly 8 dominant modes among these modes. These dominant modes mainly include the suspension rigid bending mode, the gimbal and slider bending mode, the first suspension bending

mode, the first suspension torsion mode, the arm and suspension bending mode and their torsion mode etc. If we reduce the frequency range of 0 to 6 kHz, there are only 6 natural frequencies in this range and there are only 4 dominant modes among them.

Table 6.11 shows the FE modal analysis results of the HAA after the preloading was applied. The experimental and simulation modal analysis results from Seagate are also shown in the table. From table 6.11, it is noted that the modal analysis simulation results of the HAA in its preloading state coincide well with those of experimental and simulation results from Seagate. This implies that the geometry, meshing and boundary conditions of the current FE model of the HAA in its preloading state are reasonable. Moreover, comparing the results in table 6.10 and table 6.11, it can be found that the first three modes of the HAA in its free state do not appear in its preloading state. In the preloading state, the lowest natural frequency of the HAA is the first suspension bending mode at about 3.7 kHz.

Table 6.11 Modal Analysis of the HAA in its Preloading State [Unit: Hz]

No	Current FE Simulation	Seagate		Modes
		Simulation	Experiment	
1	3703	3664	3625	1 st Bending
2	6303	6753	6438	1 st Torsion
3	11751	11597	11000	2 nd Bending
4	13492	14536	13690	2 nd Torsion
5	14997	16920	16380	Sway
6	28057	27384	-	Arm Bending

6.4.3 The Measured Damping of the HAA

To measure the damping of a system, the logarithmic decrement method is perhaps the most popular time-response method. When a single-DOF oscillatory system with viscous damping (see Equation 3.5) is excited by an impulse input (or an initial condition excitation), its response takes the form of a time decay (see Fig. 6.25), given by

$$z(t) = z_0 \exp(-\zeta \omega_n t) \sin(\omega_d t), \quad (6.1)$$

in which the damped natural frequency is given by

$$\omega_d = \sqrt{1 - \zeta^2} \omega_n. \quad (6.2)$$

Fig. 6.25 Impulse response of a simple oscillator

If the response at $t = t_i$ is denoted by z_i ; and the response at $t = t_i + 2\pi r / \omega_d$ is

denoted by z_{i+r} ; then, from Equation 6.1, we have

$$\frac{z_{i+r}}{z_i} = \exp\left(-\zeta \frac{\omega_n}{\omega_d} 2\pi r\right), \quad i = 1, 2, \dots, n \quad (6.3)$$

In particular, suppose that z_i corresponds to a peak point in the time decay function, having magnitude A_i ; and that z_{i+r} corresponds to the peak point r cycles later in the time history, and its magnitude is denoted by A_{i+r} (see Fig. 6.25). Then,

$$\frac{A_{i+r}}{A_i} = \exp\left(-\zeta \frac{\omega_n}{\omega_d} 2\pi r\right) = \exp\left[-\frac{\zeta}{\sqrt{1-\zeta^2}} 2\pi r\right], \quad (6.4)$$

where Equation 6.2 has been used. Then, the logarithmic decrement δ is given by

$$\delta = \frac{1}{r} \ln\left(\frac{A_{i+r}}{A_i}\right) = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}, \quad (6.5)$$

or the damping ratio may be expressed as

$$\zeta = \frac{\delta}{\sqrt{(2\pi)^2 + \delta^2}}. \quad (6.6)$$

Figure 6.26 shows time responses of the HAA subjected to an initial perturbation in two cases (a) and (b). In case (a), we can have the first peak point in the time decay response, having magnitude $A_1 = 0.29158$ mm; and that corresponds to the peak point 20th cycles later in the time history, having magnitude $A_{20} = 0.0462$ mm.

Thus, we can obtain the logarithmic decrement of case (a) according to equation (6.5),

(a) with initial perturbation of 0.4 mm

(b) with initial perturbation of 0.73 mm

Fig. 6.26 Time response of the HAA subjected to an initial perturbation: (a) with initial perturbation of 0.4 mm, (b) with initial perturbation of 0.73 mm.

$$\delta = \frac{1}{20-1} \ln \left(\frac{A_{20}}{A_1} \right) = \frac{1}{20-1} \ln \left(\frac{0.17323}{0.57548} \right) = 1.199019.$$

Then we can obtain the damping ratio of case (a) according to equation (6.6),

$$\zeta = \frac{1.199019}{\sqrt{(2\pi)^2 + 1.199019^2}} = 0.009866.$$

Similarly, we can obtain the logarithmic decrement and the damping ratio of case (b), as well as those of other cases. Table 6.12 gives a summary of the measured damping ratio with different initial perturbation of six cases. The average value of the measured damping ratio is 0.011381. It is noted that the damping ratio of the HAA is very small (only about one percent).

Table 6.12 The Measured Damping Ratio of the HAA

Case No	The 1 st Peak Displacement (mm)	The 20 th Peak Displacement (mm)	The Measured Damping Ratio
1	0.57548	0.17323	0.009866
2	0.883691	0.215447	0.011535
3	0.41765	0.13376	0.009385
4	0.878864	0.166309	0.01348
5	0.806945	0.170627	0.012635
6	1.50874	0.374889	0.011387
Average Value of the Damping Ratio			0.011381

6.4.4 Modal Analysis and Harmonic Analysis of the HSA

In the previous sections/paragraphs, the FE models of the disk and the HAA have been built up and verified by modal analysis and harmonic analysis. The numerical simulation results are compared with those by experimental techniques. In the following paragraphs, natural frequencies and the dominant modes of the HSA with one head/slider by performing modal analysis and harmonic analysis will be presented.

Table 6.13 shows the natural frequencies of the HSA by modal analysis and those of the dominant frequencies by harmonic analysis. The corresponding natural frequencies of the HAA are also listed in this table for comparison, in which the number in parenthesis in the column of HAA refers to the sequential number of the natural frequencies of the HAA. In table 6.13, the symbolic mark ($\checkmark$) in the column of DM implies that the corresponding natural frequency of the HSA is a dominant mode.

Figure 6.27 shows the harmonic response of displacements of the HSA on the slider. It can be found, from table 6.13 and Fig. 6.27, that there are about 19 natural frequencies in the range of 0 to 20 kHz for the HSA and there are mainly 9 dominant modes (DMs) among them. Compared with those of the HAA, the HSA has some new modes, such as the rocking mode at about 5.2 kHz, while some modes corresponding to those of the HAA have some changes in their frequency. For example, the frequency of the gimbal and slider bending mode increase from about 2 kHz to about 6 kHz. There are three DMs between the range of 0 and 6 kHz,

including the suspension rigid bending mode, the first suspension bending mode, and the rocking mode.

Table 6.13 Modal Analysis of the HSA in its Free State [Unit: Hz]

No	HSA	HAA	DM	Mode Statement
1	267	265(1)	√	Suspension Rigid Bending
2	3763	3883(4)	√	Suspension 1 st Bending
3	4772	4788(5)		Gimbal Bending
4	5197	-	√	Rocking (HSA)
5	5279	5816(6) 2014(3)		Suspension 1 st Torsion Gimbal + Slider Torsion
6	6047	1588(2)	√	Gimbal + Slider Bending
7	7450	8681(8)	√	Gimbal Torsion
8	8311	8404(7)	√	Arm + Suspension Bending
9	8877	-		VCM T + Suspension Bending (HSA)
10	9615	-		Arm + Suspension Bending
11	10060	-		Gimbal + Slider Torsion
12	13018	12140(9)		Arm Bending+Suspension Torsion
13	13711	12768(10)	√	Suspension Torsion/Sway
14	13841	-		VCM+Arm Bending (HSA)
15	15100	15075(11)		Gimbal Bending
16	16480	-	√	Gimbal + Suspension Bending
17	16753	16643(13)		VCM Torsion + Suspension Torsion (HSA)
18	17306	-	√	Arm+ Suspension+Gimbal Bending
19	17517	16502(12)		Sway
20	20921	20308(14)		Gimbal Torsion

Fig. 6.27 Harmonic response of the HSA

6.4.5 The Modal Analysis and Op-Shock Simulation of the HDA

6.4.5.1 Modal Analysis

The modal analysis results of the HDA are shown in table 6.14. The corresponding natural frequencies of the disk and the HAA are also listed in the table. Figure 6.28 shows six mode shapes of the HDA by FE simulation. From table 6.14, it can be found that some modes of the HDA are dominated by the corresponding modes of the disk such as the umbrella mode and the tilt mode. Meanwhile, some modes are dominated by the corresponding modes of the HAA such as the suspension bending mode and torsion mode.

Table 6.14 Modal Analysis of the HDA

No	HDA	Disk	HAA	Mode
1	/	/	265	Rigid bending mode
2	2120(a)*	2135	/	(0,0) Umbrella mode
3	2349	2376	/	(0,1) Tilt mode
4	2376(b)*	2378	/	(0,1) Tilt mode
5	3622(c)*	/	3793	Suspension bending mode
6	3809(e)*	3815	/	(0,2) Flutter mode
7	3815	3815	/	(0,2) Flutter mode
8	6812(d)*	/	6852	Suspension torsion mode
9	7085(f)*	7177	/	(0,3) Flutter mode
10	7184	7184	/	(0,3) Flutter mode
* The corresponding mode shapes are shown in Fig. 6.28 (a)-(f); respectively.

a) Disk umbrella mode (0,0)

b) Disk tilt mode (0,1)

c) Suspension bending mode

Fig. 6.28 Six mode shapes of the HDA by FE simulation: a) Disk umbrella mode (0, 0), b) Disk tilt mode (0, 1), c) Suspension bending mode, d) Suspension torsion mode, e) Disk flutter mode (0, 2), f) Disk flutter mode (0, 3).

d) Suspension torsion mode

e) Disk flutter mode (0,2)

f) Disk flutter mode (0,3)

Fig. 6.28 (continued) Six mode shapes of the HDA by FE simulation

6.4.5.2 Operational Shock Simulation

With the FE model of the HDA introduced in section 6.3.4, operational shock simulations of the HDA were carried out subjected to different magnitudes of half-sine acceleration shock. Figure 6.29 shows the shock responses of the HDA subjected to two positive and two negative half-sine acceleration pulses with magnitudes of 200g and 400g; respectively. All the load cases have the same pulse width of 1 ms. A positive shock is defined as an acceleration pulse in the positive z direction. Similarly, a negative shock is defined as one in the negative z direction. The first plot (Fig. 6.29a) shows the pitch responses of the slider under the four load cases. It is noted that the magnitude of the pitch response is proportional to that of the input shocks. For a positive shock and a negative shock with the same magnitude, for example, +200g and -200g, their shock responses are almost symmetric. The second plot (Fig. 6.29b) shows the roll responses of the slider under the four load cases. The roll responses have similar behavior as those of pitch responses. The third plot (Fig. 6.29c) shows the displacement at the center of the slider relative to the pivot. Again, the relative displacements have similar behavior as those of the pitch and roll responses. The forth plot (Fig. 6.29d) shows the spring elongation/contraction at the center of the slider. It is noted that some nonlinear behavior was observed in the spring relative displacements which is different with those shown in the above three plots.

a) Pitch of slider

b) Roll of slider

c) Displacement at the center of the slider relative to the pivot

d) Spring elongation/contraction at the center of the slider

Fig. 6.29 Shock responses of the HDA subjected to two positive and/or two negative half-sine acceleration pulses: a) Pitch of slider, b) Roll of slider, c) Relative displacement at the center of the slider, d) Spring's relative displacement at the center of the slider

A power spectrum analysis was performed on the shock responses of the HDA. Figure 6.30 shows the response spectra of the shock responses of the HDA subjected to half-sine acceleration pulse of 200g in magnitude and 1 ms in pulse width. One obvious peak is observed in the response spectrum curves at about 2.1 kHz. Recall the modal analysis results shown in table 6.14, it is noted that the umbrella mode is just at the frequency of about 2.1 kHz. And consequently, we conclude that the umbrella mode is the dominant mode in the shock responses of the HDA.

Fig. 6.30 Spectrum curves of the shock responses of the HDA as shown in Fig. 6.29

6.4.5.3 Pulse Width Effect

Chapter 3 shows that a half-sine acceleration pulse will produce the maximum shock responses when the frequency ratio β ($= \omega / \omega_n$) is about 0.6. Moreover, the study in Chapter 4 shows that the maximum relative displacement of shock responses is mainly determined by the acceleration power at the resonant frequency. These two conclusions will be verified for the HDA as follows.

Figure 6.31 shows seven half-sine acceleration pulses with the same magnitude of 200g but with different pulse widths of 0.1, 0.2, 0.4, 0.5, 0.6, 0.8 and 1.0 ms; respectively. Figure 6.32 shows the pitch responses of the slider of the HDA subjected to the seven load cases. It is noted, from Fig. 6.32, that the shock responses have different behaviors for the different pulse widths. The maximum values and

minimum values of the shock responses were picked up from each load cases. Figure 6.33 shows the maximum values (represented by symbol A) and the absolute values of the minimum values (represented by symbol B) and the sum of the two values (A+B) corresponding to the different pulse widths. It can be found, from Fig. 6.33, that peak values were obtained from all the three curves when the pulse width is about 0.4 ms in which case the critical frequency ratio $\beta = \omega / \omega_n \approx 0.6$. This verifies the conclusion obtained in Chapter 3.

Fig. 6.31 Seven half-sine acceleration pulses

Fig. 6.32 Pitch responses of the slider of the HDA subjected to the seven half-sine acceleration pulses

Fig. 6.33 The maximum/minimum value versus pulse width (PW).

To further explain the phenomenon observed above, a power spectrum analysis was performed on the input half-sine acceleration pulses, as shown in Fig. 6.31. Figure 6.34 shows the acceleration power spectrum curves with an additional vertical dashed line at 2.1 kHz, which is the dominant natural frequency of the HDA. It is noted, from Fig. 6.34, that the acceleration pulse with pulse width of 0.4 ms gives the maximum acceleration power at the natural frequency of 2.1 kHz. This observation gives us an explanation why the half-sine acceleration pulse with the pulse width of 0.4 ms produces the maximum shock response, which also verifies the conclusion obtained in Chapter 4.

Fig. 6.34 Acceleration power spectra of the seven half-sine acceleration pulses

6.4.5.4 Design Considerations

To explore possible approaches to reduce the maximum shock response, three ways were investigated, i.e., 1) change the Young's modulus, 2) change the thickness of the hinge, and 3) change the boundary condition of the disk. Firstly, as an academic investigation, we just doubled the Young's modulus of the HAA. Secondly, we reduce the thickness of the hinge to 80% and 50% of the original case. Thirdly, a clamped boundary condition was applied to the inner circle of the disk to simulate a more rigid connection between the disk and the spindle.

Figure 6.35 shows the comparison of the maximum/minimum shock responses for the double Young's modulus and the original case. There are six curves in this figure. The three thick curves represent the maximum value A, minimum value B, and the sum of A and B, of the shock response for the original case, which are the same as those shown in Fig. 6.33. The other three thin curves represent the corresponding values for the case with double Young's modulus. It is noted that, for the pulse width of 0.4 ms case, the maximum A, minimum B, and the sum of them of the shock response decreased by about 40% to 50%.

Figure 6.36 shows the comparison of the maximum/minimum shock responses for the three thicknesses of the hinge. It is noted that the maximum value of A decreases with thickness. It will decrease about 5% if the thickness of the hinge decreases 50%; while the minimum value of B increases with thickness. It will increase about 4% if the thickness of the hinge decreases 50%. The sum of them (A+B) decreases with

thickness. It will decrease about 1% if the thickness of the hinge decreases 50%.

Fig. 6.35 Comparison of the Maximum/Minimum for two cases of original & double Young's modulus of the HAA

Fig. 6.36 Changes of the Max/Min with the ratio of the hinge thickness

Figure 6.37 shows the comparison of the maximum/minimum shock responses with two boundary conditions, i.e., a simply-supported boundary condition (the original case) and a clamped boundary condition, of the disk. Again, there are six curves in this figure. The three thick curves are the same as those shown in Fig. 6.33 for the original case. The other three thin curves represent the corresponding values for the case with a clamped boundary condition of the disk. It is noted that the maximum A, minimum B, and the sum A+B of them of the shock responses decreased greatly for the clamped boundary condition case and the peak values of the three values, i.e., A, B and A+B, appear at a pulse width of 0.2 ms, which has a shift compared with the original case.

Fig. 6.37 Comparison of the Max/Min for original (simply-supported) & clamp boundary condition of the disk

Three possible ways are suggested for reducing the shock responses and improving the design of the drive. Among them, the third way with a clamped boundary condition for the disk produces the most significant effect on reducing the shock responses.

6.4.5.5 Temperature/Thermal Effect

FE Model and Temperature Loading. The same FE model of the HDA, as shown in Fig. 6.15, is used for the present study. In addition to the acceleration shock loading applied in the FE model, a temperature loading will also be applied for thermal analyses. The finite element application program ANSYS/LS-DYNA is used for analysis. The ANSYS/LS-DYNA program offers several methods of temperature loading: 1) Time-varying temperature applied to a nodal component. 2) Constant temperature applied to all nodes in the model. 3) Temperature results from an ANSYS thermal analysis applied as non-uniform temperature loads in a subsequent explicit dynamic analysis.

The first method utilizes the general loading procedure to apply a time-varying temperature to a specific nodal component. Two array parameters should be defined to represent the load; the first parameter contains the time values and the second parameter contains the temperature values. The second method allows you to apply a uniform constant temperature to all nodes in the model. This method may be used to model a structure subjected to steady-state thermal loading. The third method allows you to apply the temperatures calculated in an ANSYS thermal analysis as loads in an

explicit dynamic analysis. This method is useful for modeling temperature-dependent phenomena. For all three methods of temperature loading, a reference temperature can be input. The thermal loading is defined as the difference between the applied temperature and the reference temperature. If the reference temperature is not specifically defined, it defaults to zero.

In the present study, to investigate the temperature effect on the HDD slider flying attitude, we select the second method to apply the temperature loading to model a structure subjected to steady-state thermal loading. The uniform change of temperature loading condition is studied and three temperature cases of 25 °C, 50 °C and 75 °C are selected here to simulate the thermal change experienced by the HDD. Table 6.15 shows the mechanical properties of materials used in the present study.

Table 6.15 Mechanical Properties of Materials

Material	Young's Modulus (GPa)	Poisson's Ratio	Density (kg/mm ³)	Coefficient of Thermal Expansion (x 10 ⁻⁶ °C ⁻¹)
Slider (SiC)	190.3	0.49	4.4160E-6	4.3
Disk	83.5	0.33	2.5540E-6	0.9
Others (Stainless Steel)	190.3	0.32	8.0793E-6	10.2

Temperature/Thermal Effect on Flying Attitude. We focus on the flying attitude

of the pitch angle and the roll angle of the slider of the HDA in the present study. Figure 6.38 shows the pitch response of the slider of the HDA subjected to a half-sine acceleration pulse of 200g 1ms with three different temperatures of 25°C, 50°C, and 75°C. Figure 6.39 shows the roll response of the slider of the HDA subjected to the same three load cases as those in Fig. 6.38. Table 6.16 lists the maximum pitch and roll response of the HDA subjected to the same three load cases as those in Fig. 6.38. Table 6.17 lists the slider flying attitude (pitch & roll) sensitivity to temperature.

Fig. 6.38 Pitch response of the slider of the HDA subjected to a half-sine acceleration pulse of 200g 1ms with three different temperatures of 25°C, 50°C, and 75°C

Fig. 6.39 Roll response of the slider of the HDA subjected to a half-sine acceleration pulse of 200g 1ms with three different temperatures of 25°C, 50°C, and 75°C

Table 6.16 Maximum Pitch & Roll Response of the Slider of the HDA Subjected to a Half-Sine Acceleration Pulse of 200g 1ms with Three Different Temperatures

Temperature (°C)	Pitch (μRad)	Roll (μRad)
25	4.76E-04	2.07E-03
50	4.67E-04	2.04E-03
75	4.45E-04	1.99E-03

Table 6.17 Slider Flying Attitude (Pitch & Roll) Sensitivity to Temperature

Range of Temperature (°C)	Pitch Sensitivity ($\mu\text{Rad}/^{\circ}\text{C}$)	Roll Sensitivity ($\mu\text{Rad}/^{\circ}\text{C}$)
25 to 50	3.55E-07	1.2E-06
50 to 75	8.69E-07	2.0E-06
Average Sensitivity	6.12E-07	1.6E-06

From Figs. 6.38, 6.39 and Tables 6.16, 6.17, it can be found that maximum pitch and roll responses of the slider of the HDA subjected to a half-sine acceleration pulse of 200g 1ms decrease slightly with the increasing of temperatures from 25°C to 75°C. Meanwhile, the response frequencies of the pitch angle and the roll angle slightly increase simultaneously. The slider flying attitude (pitch & roll) sensitivity to temperature from 50°C to 75 °C is much larger than that from 25 °C to 50 °C. The average pitch sensitivity to temperature is 6. 12E-07 $\mu\text{Rad}/^{\circ}\text{C}$, and the average roll sensitivity to temperature is 1. 6E-06 $\mu\text{Rad}/^{\circ}\text{C}$. These numerical results indicate that the higher the temperature, the more sensitivity of the slider flying attitude will be.

6.6 Summary and Concluding Remarks

Dynamic characteristics of a 1-inch micro-drive, S1 from Seagate, were investigated by both experimental and numerical techniques. Finite element models of the disk,

the head arm assembly (HAA), the head stack assembly (HSA), and the head disk assembly (HDA), of the drive were created using ANSYS/LS-DYNA software. Four types of analysis were carried out, which include modal analysis, harmonic analysis, operational shock analysis, as well as power spectrum analysis. Based on these studies, the following conclusions are made.

- 1) Numerical results by modal analysis and harmonic analysis show that they agree well with those by experiment. Compared with the modes of the HAA, the HSA has some new modes, such as the rocking mode; meanwhile some modes have certain frequency shifted. There are mainly three dominant modes for the HSA between the frequency range of 0 and 6 kHz, including the suspension rigid bending mode, the suspension first bending mode, and the rocking mode. In addition, the damping ratio of the HAA is measured to be about 1.1 percent.
- 2) Numerical results of the HDA and the disk show that the umbrella mode of the disk is the dominant mode for operational shock according to their mode shapes and power spectrum analysis.
- 3) Investigation of the pulse width effect on the maximum shock responses shows that it will produce the maximum shock response when the characteristic frequency of the acceleration pulse is about 0.6 of the resonant frequency of the dynamic system, which correlates with the conclusion in Chapter 3. The result of a power spectrum analysis on the half-sine acceleration pulses also correlates with the conclusion in

Chapter 4.

4) Three possible ways were suggested for reducing the maximum shock response and improving the design of the drive. Among them, the third way with a more rigid clamped boundary condition for the disk produces the most significant effect on reducing the shock responses.

5) Investigation of the temperature/thermal effect on the slider flying attitude indicates that the higher the temperature, the more sensitivity of the slider flying attitude will be.

Chapter 7 Air Bearing Slider Simulation and Modeling

As the areal recording density increases in HDDs, the flying physical spacing between the head/slider and the disk media decreases and the likelihood of head/disk contact during full speed rotation increases. Therefore, the simulation and modeling of the stiffness effect of the air bearing slider with ultra-low flying heights becomes an important issue for the operational shock simulation.

7.1 Reynolds Equation and CML code

The air bearing stiffness of a particular slider design is determined by the geometry of the air bearing surface (ABS). The pressure of the air bearing is described using the compressible Reynolds equation,

$$\frac{\partial}{\partial x} \left(ph^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(ph^3 \frac{\partial p}{\partial y} \right) = 6\mu \left[U \frac{\partial(ph)}{\partial x} + V \frac{\partial(ph)}{\partial y} \right] + 12\mu \frac{\partial(ph)}{\partial t}, \quad (7.1)$$

where p is the pressure, h is the flying height distribution, μ is the viscosity of air, U and V are the local disk velocity in the x and y direction on the disk surface plane; respectively.

Due to the extremely small spacing of the head disk interface, the standard Reynolds equation for pivoting slider bearings must be adjusted to account for rarefaction effects or slip at the boundaries. The Reynolds equation including a slip correction factor Q is shown in Eq. 7.2.

$$\frac{\partial}{\partial x} \left(ph^3 Q \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(ph^3 Q \frac{\partial p}{\partial y} \right) = 6\mu \left[U \frac{\partial(ph)}{\partial x} + V \frac{\partial(ph)}{\partial y} \right] + 12\mu \frac{\partial(ph)}{\partial t} . \quad (7.2)$$

The slip correction factor Q is a function of the Knudsen number and must be considered when the Knudsen number is greater than 0.1. For current air bearing simulation, Q was developed from the Boltzmann equation for kinetic gas theory. The last term on the right hand side of Eq. 7.2 is the time dependent term or “squeeze” term, and is not included for the steady state Reynolds equation.

The dynamic model of a slider is described by the following equations with 3 degrees of freedom:

$$\left. \begin{aligned} m\ddot{z} &= F_s + \int_A (p - p_a) dA \\ I_\theta \ddot{\theta} &= M_{s\theta} + \int_A (p - p_a) (x_g - x) dA \\ I_\phi \ddot{\phi} &= M_{s\phi} + \int_A (p - p_a) (y_g - y) dA \end{aligned} \right\}, \quad (7.3)$$

where z , θ , and ϕ are vertical displacement, pitch angle, and roll angle; respectively; F_s , $M_{s\theta}$, $M_{s\phi}$ are respectively the force and moments applied by suspension in the z , θ , and ϕ directions; p_a is the ambient pressure, and x_g and y_g are the positions of the slider's center of gravity; I_θ , I_ϕ are the slider's moments of inertia and m is the

slider's mass.

Satisfying these equations simultaneously results in the slider's dynamic response in the vertical, pitch, and roll DOFs to various inputs. These equations are based on an assumption of no contact between the slider and disk, and they are highly coupled and nonlinear. Dynamic analysis of a slider flying over a rotating disk requires simultaneous solution of the time-dependent generalized Reynolds equation and the equation of motion of the slider and its suspension. The slider's motion is completely determined by the balance of the air bearing pressure, the suspension force and its inertia.

The simulations conducted in the following two sections, sections 7.2 and 7.3, are based on an air bearing design program, called CML code, developed by the computer mechanics laboratory in University of California, Berkeley [Lu, 1997]. A partial contact model was implemented in the CML code. The model is statistical, and does not model actual impacts of the slider with asperities. Instead, it adds a contact force to the air bearing force based on the statistical amount of contact. The contact force is calculated based on the disk surface parameters and the flying characteristics of the slider. While actual impact effects cannot be predicted, the model provides a useful qualitative analysis of partial contact.

A typical slider (pico slider), as shown in Fig. 7.1, whose main dimensions are 1.23 mm x 1.01 mm with suspension force of 1.25 g was examined by the code. An 1-in disk was selected with its roughness characterized by the asperity number density,

radius of curvature of the asperity summits, and standard deviation of the asperity heights. The reduced Young's modulus for the contact was chosen to be $1.2\text{E}+11$ Pa. The characteristic parameters of the asperities are similar to those given in Bhushan *et al.* [Bhushan, 1995]. The asperity number density is $1\text{E}+11$ per square meter. The standard deviation of asperity heights is 8 nm. The radius of curvature for asperity is $10\text{ }\mu\text{m}$. The glide height of the disk is 2.5 nm. All results were obtained using a grid size of 162×162 .

Fig. 7.1 A typical slider geometry

Figure 7.2(a)-(c) shows the air pressure distribution, the mass flow, and the contact pressure, of the slider at the outer diameter with the preloading force of 1.25 g; respectively. The maximum air bearing pressure and the maximum contact pressure

near the trailing edge are 13.63 and 1071 times the ambient pressure; respectively.

a) Air pressure distribution

b) Mass flow

Fig. 7.2 (a) air pressure distribution, (b) the mass flow, and (c) the contact pressure

c) Contact pressure

Fig. 7.2 (continued) (a) air pressure distribution, (b) the mass flow, and (c) the contact pressure

7.2 Effects of Radial Position and Rotating Speed

7.2.1 Effect of Radial Position and Skew Angle

At the same rotating speed (RPM), the slider flies over the disk with a different skew angle and disk linear velocity, depending on its radial position, as shown in Fig. 7.3. Four positions were considered here at a disk rotating speed of 3600 rpm: (1) at 11.682 mm radial position with a 9.861 degree skew angle; (2) at 10.93 mm radial position with a 7.588 degree skew angle; (3) at 7.696 mm radial position with a -4.44 degree skew angle; and (4) at 6.218 mm radial position with a -12.004 degree skew

angle.

Fig. 7.3 Schematic diagram for radial position and skew angle

The simulation results by CML code are shown in Fig. 7.4. Figure 7.4(a) shows the maximum contact pressure. It is known that the contact pressure is a function of the number of contact asperities and velocity. The flying height first increases and reduces the number of contact asperities, as a result, the contact pressure decreases. As the slider moves from the inner diameter (ID) to the outer diameter (OD), the skew angle changes from negative value to positive one as shown in Fig. 7.4(b), and the disk velocity increases, which resulted in the flying height decreases, as shown in Fig. 7.4(c), and the contact pressure increases. As shown in Fig. 7.4(d) and 7.4(e), the pitch increases while the roll decreases. The maximum air bearing pressure increases

with the radius, as shown in Fig. 7.4(f).

(a) Maximum contact pressure

(b) Skew angle

(c) Flying height

(d) Pitch

(e) Roll

(f) Maximum air bearing pressure

Fig. 7.4 Effect of radial position: (a) maximum contact pressure, (b) skew angle, (c) flying height, (d) pitch, (e) roll, (f) maximum air bearing pressure.

7.2.2 Effect of Rotating Speed

The range of the rotating speed of the disk was selected from 3000 to 4800 rpm. The effect of the rotating speed (RPM) was investigated when the slider was located at different radii of the disk with different skew angles as stated in the previous section. Figure 7.5 shows (a) the flying height, (b) the pitch angle, and (c) the roll angle as a function of the disk radius under different rotating speeds of the disk.

It is noted, from Fig. 7.5(a), that the flying height increases with the RPM when the slider locates at ID of the disk. While the opposite phenomena occurs when the slider locates at OD of the disk. Generally speaking, for a given rotating speed, the slider flies lower over OD than ID; and for different rotating speeds, a lower RPM will produce a more stable flying height between ID and OD. From Fig. 7.5(b), it is noted that the pitch angle increases with the RPM whether the slider locates at ID or OD. From Fig. 7.5(c), it can be found that a smaller roll angle occurs over OD than ID, and a higher RPM will result in a larger change of the roll angle between ID and OD.

(a) Flying height

(b) Pitch angle

(c) Roll angle

Fig. 7.5 Effect of rotating speed (RPM)

7.3 Shock Simulation of Air Bearing Slider

In the CML code, shock is simulated by applying a normal force (N), a pitch moment (Nm) and a roll moment (Nm) to the slider at the load point. The responses of the disk and other components are neglected. A simple half sinusoidal shock of 200 g and 1 ms was specified for the simulation of the dynamic property of the air bearing slider. Figure 7.6 shows some simulation results during the shock, including (a) the air bearing force, (b) the normal contact force, (c) the pitch contact moment, (d) the roll contact moment, (e) the minimum spacing, (f) the displacement at nominal trailing edge center (TEC), (g) the pitch, and (h) the roll. It is noted, from Fig. 7.6(a), (e) and (f), that there are obvious oscillations in the air bearing force, the minimum spacing and the displacement at the TEC when contact occurs.

(a) Air bearing force

(b) Normal contact force

(c) Pitch contact moment

(d) Roll contact moment

(e) Minimum spacing

(f) Displacement at nominal TEC

(g) Pitch

(h) Roll

Fig. 7.6 Some simulation results during the shock, (a) the air bearing force, (b) the normal contact force, (c) the pitch contact moment, (d) the roll contact moment, (e) the minimum spacing, (f) the displacement at nominal TEC, (g) the pitch, (h) the roll.

7.4 A Nonlinear SDOF Model and Modified Newmark β Method

7.4.1 A SDOF Model of Air Bearing Slider

A SDOF model of the air bearing slider may be inaccurate for modeling most air bearing slider dynamics. However, it can be useful in providing a basic understanding of the nonlinear effects. A schematic diagram of the SDOF model is shown in Fig. 7.7.

Fig. 7.7 A SDOF model schematic diagram of air bearing slider

The equation of motion for the system can be written as,

$$m\ddot{Z} + (c + c_s)\dot{Z} + [k(Z) + k_s]Z = 0, \quad (7.4)$$

where c and c_s are the damping coefficients of the air bearing and the suspension, Z is the flying height with a zero mean ($Z = FH - \text{steady state FH}$), k_s is the vertical stiffness of suspension. $k(Z)$ is the nonlinear stiffness in units MN per meter as a function of flying height at the pole-tip (PT) location in units of nanometers, and is represented as [Thornton and Bogy, 2003]

$$k(FH_{PT}) = \varepsilon(FH_{PT})^\gamma . \quad (7.5)$$

For the given slider, the coefficients γ and ε were determined to be -0.480 and 5.1 ; respectively. Figure 7.8 is a plot of the stiffness as a function of flying height at the PT location. It is noted, from Fig. 7.8, that 1) as FH decreases, the stiffness increases exponentially and 2) the lower FH, the greater nonlinear effects become due to small perturbations. The nonlinear equation (7.4) can be solved numerically. We solve it by a modified Newmark β method (see section 7.4.2 in details).

Fig. 7.8 Nonlinear vertical air bearing stiffness as a function of FH at the pole-tip

7.4.2 Modified Newmark β Method

The objective of this section is to obtain a numerical solution for a nonlinear equation,

$$m\ddot{u}(t) + c\dot{u}(t) + k(u)u(t) = p(t) . \quad (7.6)$$

Usually, the closed-form solution is not possible under arbitrary excitations, especially for a nonlinear equation as shown in Eq. (7.6). Thus it is needed to resort to numerical methods. Among various numerical methods, we focus on Newmark β Method because of its versatility in accuracy, convergence and stability-time step etc. For solving $u(t)$, $\dot{u}(t)$, and $\ddot{u}(t)$ from Eq. (7.6), we also have:

$$\left. \begin{aligned} \dot{u}(t) &= \dot{u}(t_0) + \int_{t_0}^t \ddot{u}(s) ds \\ u(t) &= u(t_0) + \int_{t_0}^t \dot{u}(s) ds \end{aligned} \right\} . \quad (7.7)$$

The discretized forms of Eqs. (7.6) and (7.7) can be written as follows:

$$\left. \begin{aligned} m\ddot{u}_i + c\dot{u}_i + k_i u_i &= p_i \\ m\ddot{u}_{i+1} + c\dot{u}_{i+1} + k_{i+1} u_{i+1} &= p_{i+1} \end{aligned} \right\} , \quad (7.8)$$

$$\left. \begin{aligned} \dot{u}_{i+1} &= \dot{u}_i + \int_{t_i}^{t_{i+1}} \ddot{u}(s) ds \\ u_{i+1} &= u_i + \int_{t_i}^{t_{i+1}} \dot{u}(s) ds \end{aligned} \right\} . \quad (7.9)$$

The idea is to calculate u_{i+1} , $\dot{u}_{i+1}$, and $\ddot{u}_{i+1}$, and from u_i , $\dot{u}_i$, and $\ddot{u}_i$, while satisfying the above equations. The key step for Newmark β Method is to remove the

integration sign from Eq. (7.9). The modified Newmark β Method for solving nonlinear equation can be written in unified form as follows,

$$\left. \begin{aligned} \dot{u}_{i+1} &= \dot{u}_i + (1-\gamma)\ddot{u}_i\Delta t + \gamma\ddot{u}_{i+1}\Delta t \\ u_{i+1} &= u_i + \dot{u}_i\Delta t + (0.5-\beta)\ddot{u}_i(\Delta t)^2 + \beta\ddot{u}_{i+1}(\Delta t)^2 \\ \ddot{u}_{i+1} + \frac{c}{m}\dot{u}_{i+1} + \frac{k_i}{m}u_{i+1} &= \frac{p_{i+1}}{m} \end{aligned} \right\}, \quad (7.10)$$

where, γ and β are Newmark parameters (constants) for different acceleration approximate methods. $\gamma = 1/2$ or 0 , $1/6 \leq \beta \leq 1/4$.

Solving for $\ddot{u}_{i+1}$ from Eq. (7.10), one can obtain,

$$\ddot{u}_{i+1} = \frac{1}{mG_i} \left[-k_i u_i - (c + k_i \Delta t) \dot{u}_i - Q_i \ddot{u}_i + p_{i+1} \right], \quad (7.11)$$

where

$$\begin{aligned} G_i &= 1 + \frac{c}{m}\gamma\Delta t + \frac{k_i}{m}\beta(\Delta t)^2 \\ Q_i &= \frac{c}{m}(1-\gamma)\Delta t + \frac{k_i}{m}(0.5-\beta)(\Delta t)^2 \end{aligned} \quad (7.12)$$

In summary, the formula of the modified Newmark β Method can be written as follows:

$$\begin{Bmatrix} u_{i+1} \\ \dot{u}_{i+1} \\ \ddot{u}_{i+1} \end{Bmatrix} = \mathbf{F}_{N_i} \begin{Bmatrix} u_i \\ \dot{u}_i \\ \ddot{u}_i \end{Bmatrix} + \mathbf{H}_{N_i} p_{i+1}, \quad (7.13)$$

where,

$$\mathbf{F}_{N_i} = \begin{bmatrix} 1 - \frac{k_i}{mG_i} \beta (\Delta t)^2 & \Delta t \left(1 - \frac{c + k_i \Delta t}{mG_i} \beta \Delta t \right) & \frac{(\Delta t)^2}{2} - \frac{G_i + Q_i}{G_i} \beta (\Delta t)^2 \\ -\frac{k_i}{mG_i} \gamma \Delta t & 1 - \frac{c + k_i \Delta t}{mG_i} \gamma \Delta t & \Delta t - \frac{G_i + Q_i}{G_i} \gamma \Delta t \\ -\frac{k_i}{mG_i} & -\frac{c + k_i \Delta t}{mG_i} & -\frac{Q_i}{G_i} \end{bmatrix}, \quad (7.14)$$

$$\mathbf{H}_{N_i} = \frac{1}{mG_i} \begin{Bmatrix} \beta (\Delta t)^2 \\ \gamma \Delta t \\ 1 \end{Bmatrix}. \quad (7.15)$$

7.4.3 Dynamic Responses and Nonlinear Behavior

The nonlinear Eq. (7.4) can be solved by the modified Newmark β Method stated in section 7.4.2. Dynamic responses of the air bearing slider system can be obtained for a given initial perturbation. Figures 7.9 and 7.10 show the dynamic responses, displacement and velocity, of the system in time domain for two given initial perturbations of 0.02 nm and 2.0 nm; respectively.

It is noted, from Figs. 7.9 and 7.10, that there are little differences for the dynamic responses under different initial perturbations except their amplitudes in the time

domain. But there are obvious differences between them if they are converted into frequency domain. Power spectrum analyses were conducted for the dynamic responses shown in Figs. 7.9 and 7.10.

(a) Displacement

(b) Velocity

Fig. 7.9 Dynamic responses of the air bearing slider system in time domain for initial perturbations, $z_0 = 0.02$ nm.

(a) Displacement

(b) Velocity

Fig. 7.10 Dynamic responses of the air bearing slider system in time domain for initial perturbation, $z_0 = 2.0$ nm.

Figure 7.11 shows the power spectra of the system for three initial perturbations, i.e., (a) $z_0 = 0.02$ nm, (b) $z_0 = 2.0$ nm, and (c) $z_0 = 3.0$ nm. It is noted, that for a small initial perturbation, say, $z_0 = 0.02$ nm, the system behaves linearly, showing only one resonance peak in its power spectrum curve. As the initial amplitude is increased, more than one resonance peaks appear in the power spectrum curves. The system

shows obvious nonlinear behavior. Further efforts should be given to investigate this nonlinear phenomenon.

Fig. 7.11 Power spectra of dynamic responses of the air bearing slider system for three initial perturbations, $z_0 = 0.02$ nm, 2.0 nm, and 3.0 nm.

As a parametric study, we also investigated the effect of the stiffness of suspension. We found that the stiffness of suspension has little influence on the dynamic property of the air bearing slider. This is because the air bearing stiffness is much higher than that of the suspension.

7.5 A Nonlinear 2-DOF Model and Uncoupling the Equations of Motion

7.5.1 A 2-DOF Model of Air Bearing Slider

The SDOF model over simplifies the dynamics of the air bearing slider system at the pole-tip (PT) location. Furthermore, we assume that the ABS designs are symmetric and the test condition is at 0° skew angle, and the roll mode will not contribute to the slider's response at the PT location. This can be modeled by using a 2-DOF analytical model as shown in Fig. 7.12.

Fig. 7.12 A 2-DOF model schematic diagram of air bearing slider

The equations of motion of this model system for unforced free vibration can be written as,

$$\begin{bmatrix} m & 0 \\ 0 & I \end{bmatrix} \begin{Bmatrix} \ddot{z} \\ \ddot{\theta} \end{Bmatrix} + \begin{bmatrix} c_{11} & c_{12} \\ c_{21} & c_{22} \end{bmatrix} \begin{Bmatrix} \dot{z} \\ \dot{\theta} \end{Bmatrix} + \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{Bmatrix} z \\ \theta \end{Bmatrix} = 0, \quad (7.16)$$

where, m is the mass of the slider, I is the slider's moment of inertia, k_{11} , k_{12} , k_{21} , and

k_{22} , are given by,

$$\left. \begin{aligned} k_{11} &= k_1 + k_2 + k_s \\ k_{12} &= -(k_1 d_1 - k_2 d_2) = k_{21} \\ k_{22} &= k_1 d_1^2 + k_2 d_2^2 + k_\theta \end{aligned} \right\}, \quad (7.17)$$

where, $k_1(z, \theta)$ and k_2 are the stiffness at the trailing edge (TE) and leading edge (LE) of the slider; respectively; d_1 and d_2 are the pressure center at the TE and the LE; respectively; k_s and k_θ are the vertical stiffness and torsional stiffness of the suspension.

The damping coefficient c_{ij} is written in the same form as the stiffness k_{ij} . The stiffness at the TE, $k_1(z, \theta)$, is a nonlinear function of both z and θ , and is represented as $k_1(z, \theta) = \varepsilon(z - d_1\theta + FH)^\gamma$.

7.5.2 Uncoupling and Solving the Equations of Motion

The two equations in (7.16) are nonlinear and coupled. In order to facilitate solving for transient responses governed by the coupled nonlinear differential equations, it is useful to transform the coupled differential equations to uncoupled ones by transforming them from the physical coordinate system to a principal coordinate system. For present dynamic system, we ignore all the non-diagonal items in the damping matrix, which may not cause much error for the systems with small damping

for shock analysis.

The uncoupled equations in the principal coordinate system can then be solved for the responses in the principal coordinate system using the modified Newmark β method for the single degree of freedom system suggested in the section 7.4.2. The responses in the principal coordinate system can then be transformed back to the physical coordinate system to provide the actual responses in the physical coordinate system.

7.5.3 Dynamic Responses for Given Initial Perturbations

The initial perturbations refer to the initial disturbance inputs of the flying height and/or pitch angle here. Figures 7.13 and 7.14 show the dynamic responses, a) flying height, and b) velocity in decibel (DB) representation, of the system for given initial perturbations of flying heights of 0.02 nm and 2.0 nm, respectively. It is noted that, for a small initial perturbation, the 2-DOF system shows two resonant peaks, which represents the two dominant modes (two pitch modes); while for a large perturbation, many resonant peaks appear in the power spectrum curves, representing its nonlinear behavior.

In short, the 2-DOF system mainly shows two resonance peaks, which represent the first pitch mode and the second pitch mode, respectively, for a small initial perturbation; and some higher harmonic modes resulting from the nonlinear stiffness for a large initial perturbation.

(a) Flying height

(b) Velocity in DB representation

Fig. 7.13 Dynamic responses simulated by a 2-DOF model for an initial perturbation of 0.02 nm.

(a) Flying height

(b) Velocity in DB representation

Fig. 7.14 Dynamic responses simulated by a 2-DOF model for an initial perturbation of 2.0 nm.

7.5.4 Parametric Study

To further investigate the dynamic property of the 2-DOF system, we select three

cases for parametric studies: Firstly, we investigated the stiffness effect of the spring k_2 at the LE of the slider, supposing $k_2 = nE5$ N/m, where n is the stiffness coefficient and varies from 1 to 30. The spring k_2 represents the stiffness effect at the LE of the slider and its stiffness is much smaller than that of the spring k_1 at the TE of the slider, which is in the units of MN/m. Secondly, we investigate the flying height (FH) effect on the two resonant pitch modes for a given stiffness of k_2 . Thirdly, we investigate the location effect of the two springs at the TE and the LE of the slider on the two resonant pitch modes to find the optimal positions of the two springs.

Figure 7.15 (a) and (b) show the frequencies f_1 and f_2 of the two pitch modes and their corresponding amplitudes in DB versus the coefficient n . It is noted that, from Fig. 7.15, for the coefficient n varies from 1 to 20, the first frequency, f_1 , increases up to 291%; while that of the second frequency, f_2 , increases only about 13%. Meanwhile, the magnitude of the first frequency increases with the increasing of the frequency and the coefficient n , while that of the second frequency simultaneously decreases greatly. When the coefficient n increases to about 25 and above, the second pitch mode disappears in the frequency domain.

To investigate the FH effect on the two pitch modes, the range of the FHs is selected between 1 to 30 nm. Figure 7.16 shows the FH effect on the two pitch modes for a given stiffness of k_2 . It is noticed that, from Fig. 7.16, for a small stiffness of k_2 , the frequency of the first pitch mode has no change with the increase of the FH. With the stiffness k_2 increases, the frequency of the first pitch mode decreases slowly with the increase of the FH. The frequency of the second pitch mode decreases exponentially

with the increase of the FH for a small FH. However, with the FH increasing, the frequency of the second pitch mode decreases slowly.

(a) Resonant frequencies

(b) Magnitudes of two resonant peaks in DB

Fig. 7.15 Two pitch modes and their magnitudes of the 2-DOF model vs. coefficient n of spring stiffness k_2 at the LE of the slider.

(a) The first frequency vs. flying height

(b) The second frequency vs. flying height

Fig. 7.16 Two pitch modes of the 2-DOF model for different coefficient n of spring stiffness (k_2).

Figure 7.17 shows the location effect of the two springs, k_1 and k_2 , at the TE and the

LE of the slider, respectively, on the two pitch modes for a given stiffness k_2 and a given flying height. In the Fig. 7.17, the abscissa represents the ratio of the spring's location d_1 or d_2 over the length of the slider, while the ordinate represents the frequency values of the two resonant modes.

Fig. 7.17 Two main frequencies vs. the locations, d_1 (d_2), of the two springs at the TE and the LE of the slider.

Following observations can be made from Fig. 7.17. Firstly, the location of the spring k_1 at the TE has obvious influence on the frequency of the second pitch mode, while the location of the spring k_2 at the LE has no influence on it. At the extreme case, the two curves have an intersection, point A, as shown in Fig. 7.17. Secondly, there is an optimal position for the spring k_1 to reach a maximum frequency value for the first pitch mode even, in general, the location of the spring k_1 does not have much

influence on the frequency of the first pitch mode, while the frequency of the first pitch mode increases linearly with the increasing of the ratio of d_2 over the slider length, i.e., with the moving of the location of the spring k_2 to the LE of the slider. Again, at the extreme case, the two curves also have an intersection, point B, as shown in Fig. 7.17.

Moreover, as a parametric study, we also investigated the effect of the stiffness of the suspension. We noticed that the stiffness of the suspension has little influence on the dynamic property of the air-bearing slider. This is because the air-bearing stiffness is much higher than that of the suspension.

7.6 Summary and Concluding Remarks

The static/dynamic properties, including the influence of the radial position and the skew angle of the slider, the rotating speed of the disk, and the shock simulation, of the air bearing slider were analyzed by CML code. Two nonlinear simplified air-bearing models are proposed to investigate the dynamic behavior of air-bearing sliders for HDDs with ultra-low flying heights. The main conclusions are:

1) Generally speaking, for a given rotating speed of the disk, as the slider moves from the inner diameter (ID) to the outer diameter (OD), the maximum contact pressure, the skew angle, the pitch, and the maximum air bearing pressure increase; while the flying height, and roll decrease. These trends are strengthened by a faster rotating

speed of the disk. There are obvious oscillations in the air bearing force, the minimum spacing, and the displacement at the TEC when contact occurs during a shock.

2) Based on the numerical results of the two simplified air bearing slider models, we can find the following observations: a) For a large initial perturbation applied on the air-bearing slider system, many resonance peaks appear in the power spectrum curves. Numerical results show obvious nonlinear behavior for air-bearing sliders with ultra-low flying heights. b) There are two main pitch modes for the 2-DOF air-bearing slider model. With the increase of the spring stiffness, the magnitude of the frequency of the first pitch mode increases, while that of the second pitch mode decreases greatly. d) The location of the spring at the trailing edge (TE) has obvious influence on the frequency of the second pitch mode, while the location of the spring at the leading edge (LE) has no influence on it. There is an optimal position for the spring at the TE to reach a maximum frequency for the first pitch mode.

3) A modified Newmark β method was developed for solving nonlinear equation of motion of the air bearing slider. The stiffness of suspension has little influence on the property of the air bearing slider because the air bearing stiffness is much higher than that of the suspension.

Chapter 8 Conclusion and Recommendation

8.1 Conclusions

The effect of shock on hard disk drives (HDDs) is of great interest due to the increasingly hostile environments encountered in the usage of the portable computer. Especially as non-traditional applications of HDDs, such as those in consumer devices, emerges, their mechanical robustness under shock and other mechanical disturbances during different states are of great concern. This dissertation has focused on studying the shock dynamic characteristics of HDDs. Analytical, numerical and experimental techniques have been employed in this project.

- (1) To study the shock dynamic characteristics of HDDs, the relative displacements of an actuator arm subjected to a single half-sine acceleration pulse during drop tests were simulated by using finite element (FE) model and a simplified single-degree-of-freedom (SDOF) model. The influence of the pulse width/duration and the pulse amplitude on the shock responses of the relative displacement was investigated by using these two models. It was found that, for both the FE model and the SDOF model, the dimensionless peak relative displacement occurs at a

critical frequency ratio (i.e. $\beta = \omega / \omega_n \approx 0.6$), which is the frequency ratio of the *characteristic frequency* of the half-sine acceleration pulse to the first natural frequency of the system. In other words, a *pseudo resonance* phenomenon occurs at the critical frequency ratio. The pseudo resonance phenomenon was first observed in the shock responses of the HAA by using the FE model and then explained in general by using the simplified SDOF model. Moreover, it is noted that for short duration transient problem, damping does not have enough time to have strong influence on the maximum relative displacement.

- (2) The shock dynamic characteristics of the actuator arm subjected to different acceleration pulse shapes, including those in half-sine, triangular and dual-quadratic waveforms, were investigated by using FE simulation. In the shock responses of the dynamic system, an abnormal phenomenon was observed where a stronger single half-sine acceleration pulse results in a lower relative displacement compared with those of other two pulse shapes. This phenomenon has been explained in terms of a power spectrum analysis. We conclude that the relative magnitude of the maximum relative displacement is mainly determined by the power magnitude of the acceleration pulse at the resonant frequency of the dynamic system.
- (3) Moreover, a cross-over point (or small interval) was observed in the acceleration power spectrum curves. A simple theorem was developed and proved to illustrate the existence and location of such a cross-over point. As another shock dynamic characteristic of the dynamic system subjected to different acceleration

pulse shapes, we conclude that when the resonant frequency of the dynamic system resides in the region to the left of the cross-over point, the half-sine pulse shape gives the largest peak displacement, followed by the triangular and dual-quadratic; the reverse is true when the resonant frequency resides in the right of the cross-over point but to the left of any higher-frequency cross-over points. A corollary was derived based on this theorem. The prediction of the corollary was verified by numerical results. The theorem and the corollary read as follows.

Theorem: *If a given waveform, $h(t)$, can be expressed as a sum of an arbitrary reference waveform, $q(t)$, with a pulse width of T , and two identical and adjacent waveforms, $g(t)$ and $g(t - T/2)$, with the pulse width of each equal to $T/2$, then, the power spectrum curve of the given waveform will coincide with the power spectrum curve of the reference waveform at the frequency values given by $f_n = (2n-1)/T$, $n = 1, 2, 3...$*

Corollary: *When the characteristic frequencies of a group of acceleration shocks with different pulse shapes are very close to the resonant frequency of the dynamic system, these acceleration shocks will have nearly equal acceleration powers at the resonant frequency; consequently, they will produce nearly equal shock responses.*

- (4) To study the effect of the pivot bearing stiffness on shock responses of HDDs, a simplified beam model with a torsional spring and a translational spring to simulate the pivot stiffness of HDDs was developed. Dynamic analyses of the

simplified beam model show that, when the two dimensionless stiffnesses of the torsional and translational springs are not very high, the vertical bending modes are sensitive to them and the stiffness effects need to be considered carefully in the dynamic analysis. The shock responses are insensitive to the torsional stiffness, but are sensitive to the axial stiffness, of the pivot bearing.

(5) Dynamic characteristics of a micro-drive were investigated by using both experimental and numerical techniques. Finite element models of the disk, the head arm assembly (HAA), the head stack assembly (HSA), and the head disk assembly (HDA), of the drive were created. Numerical results by modal analysis and harmonic analysis show that they agree well with those by experiment. It was found that the umbrella mode of the disk is the dominant mode for operational shock according to their mode shapes and power spectrum analysis. Investigation of the pulse width effect on the maximum shock responses presents a further verification to the first and the second conclusions stated above. Three possible approaches were suggested for design consideration to reduce the maximum shock response and to improve the design of the drive. Moreover, investigation of the temperature/thermal effect on the flying attitude indicates that the higher the temperature, the more sensitivity will be of the slider flying attitude of HDDs subjected to a shock loading.

(6) In air bearing slider simulation, the influences of the radial position and the skew angle of the slider, the rotating speed of the disk, and the shock simulation, of the air bearing slider, were analyzed by using CML code. Generally speaking, for a

given rotating speed of the disk, as the slider moves from the inner diameter (ID) to the outer diameter (OD), the maximum contact pressure, the skew angle, the pitch angle, and the maximum air bearing pressure increase; while the flying height, and roll angle decrease. These trends are strengthened by a faster rotating speed of the disk. There are obvious oscillations in the air bearing force, the minimum spacing, and the displacement at the TEC when contact occurs during a shock.

- (7) Two nonlinear simplified models of the air bearing slider were developed in providing a basic understanding of the nonlinear effects of the air bearing for ultra-low flying heights. It is found that the air bearing slider has obvious nonlinear behavior for ultra-low flying heights. For a large initial perturbation applied on the air-bearing slider system, many resonance peaks appear in the power spectrum curves. There are two main pitch modes for the 2-DOF air-bearing slider model. With the increasing of the spring stiffness, the magnitude of the frequency of the first pitch mode increases, while that of the second pitch mode decreases greatly. There is an optimal position for the spring at the TE to reach a maximum frequency for the first pitch mode. Moreover, a modified Newmark β method was developed for solving a nonlinear equation.

8.2 Recommendations

Based on the research work finished in this project, the following aspects are

recommended for further investigation.

- (1) The stiffness effect of the spindle bearing on the shock responses.
- (2) More complicated air bearing slider model to consider the contact force, friction force, electrostatic and intermolecular forces between the slider and the disk.
- (3) To couple the air bearing slider (ABS) model together with the pivot bearing model and to compare air bearing stiffness properties derived from analysis of dynamic response to static deflections from steady state solution.
- (4) To take into account non-linear air bearing stiffness and model effect of flow on off-track, on-track vibrations.
- (5) To predict the actual head-disk performance by performing probabilistic analysis for HDDs with a flying height of a few nanometers.

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