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Xu, Jianfeng

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Head/Disk Interface Tribology in the Nanometer Regime

A dissertation submitted in partial satisfaction of
the requirements for the degree of Doctor of Philosophy

in

Engineering Sciences (Mechanical Engineering)

by

Jianfeng Xu

Committee in charge:

Professor Frank E. Talke, Chair
Professor David J. Benson
Professor Michael J. Holst
Professor Sutanu Sarkar
Professor Jack K. Wolf

2008

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University of California, San Diego

2008

TABLE OF CONTENTS

Signature Page.....	iii
Table of Contents.....	iv
List of Figures.....	viii
List of Tables.....	xii
Acknowledgements.....	xiii
Vita.....	xiii
Publications.....	xvi
Abstract.....	xvii
 Chapter 1 Introduction.....	 1
1.1 Magnetic Recording Hard Drives and Their Evolutions.....	1
1.2 Structure of a Hard Disk Drive and Magnetic Recording	5
1.3 The Read/Write Process.....	6
1.4 Scaling Laws for Magnetic Recording.....	11
1.5 The Head/Disk Interface.....	13
1.6 Content of Thesis.....	15
1.7 References.....	16
 Chapter 2 Introduction to Tribology and Its Application to the Head/Disk Interface.....	 17
2.1 Introduction.....	17
2.2 Surface Roughness.....	19
2.3 Contact Theory.....	22
2.3.1 Deformation of a Single Asperity.....	23
2.3.2 Contact of Multiple Asperities.....	26
2.4 Adhesion and Friction Theory.....	30
2.4.1 Surface Forces.....	30
2.4.2 Adhesion Theory.....	33
2.4.3 Friction Models.....	40
2.5 Lubrication.....	41
2.5.1 Boundary Lubrication.....	44
2.5.2 Hydrodynamic Lubrication.....	50
2.5.3 Rarefaction effects.....	52
2.6 Wear	54
2.6.1 Adhesive Wear.....	55
2.6.2 Abrasive Wear.....	57
2.6.3 Tribochemical Wear.....	59
2.7 Summary.....	60
2.8 References.....	61
 Chapter 3 Mechanical Scaling Laws for Hard Disk Drives.....	 67
3.1 Introduction.....	68
3.2 A Simplified Tapered Beam Model for Suspensions.....	69
3.3 Scaling Laws for disks.....	73

3.4 Summary and Conclusion.....	75
3.5 References.....	76
Chapter 4 Failure of Brittle and Ductile Hard Disks Due to High Shock Levels.....	77
4.1 Introduction.....	77
4.2 Definition of the Problem.....	80
4.3 Theoretical Background.....	81
4.3.1 Brittle vs. Ductile Material.....	82
4.3.2 Failure Criteria and Strain Hardening.....	82
4.4 Model of disk drive and numerical implementation.....	83
4.5 Crack Formations on Glass Disks and Plastic Deformation on Aluminum Disks.....	88
4.6 Design Factors Affecting the Failure inception of Glass and Aluminum Disks.....	93
4.6.1 Disk Thickness.....	93
4.6.2 Clamping Conditions.....	94
4.6.3 Disk Form Factor.....	96
4.7 Discussion.....	98
4.8 Conclusions.....	101
4.9 Acknowledgement.....	102
4.10 References.....	102
Chapter 5 Experimental Setup for the Measurement of Slider Dynamics in Five Degrees-of-freedom.....	105
5.1 Introduction.....	105
5.2 Laser Doppler Vibrometer Used in Measurement of Slider Vibrations.....	106
5.3 Five Degrees-of-freedom Experimental Systems.....	109
5.3.1 Experimental Set-up for Five Degrees-of-freedom Measurement..	109
5.3.2 Investigation of Slider Vibration in Five Degrees-of-freedom.....	112
5.4 Summary.....	119
5.5 References.....	119
Chapter 6 Experimental Study of Slider Vibrations Using the 5-DOF Measurement System.....	121
6.1 Effect of Disk Topography on Slider Dynamics.....	121
6.1.1 Introduction.....	121
6.1.2 Experimental Conditions.....	124
6.1.3 Disk Topography and Slider Response in the Time Domain.....	126
6.1.4 Spectral Analysis of Disk Topography and Slider Response.....	129
6.1.5 Summary.....	133
6.2 Dynamics of Ultra Low Flying Sliders during Contact with a Lubricated Disk.....	133
6.2.1 Introduction.....	133
6.2.2 Experimental Conditions.....	134
6.2.3 Slider-lubricant Interactions in the Time Domain.....	135
6.2.4 Slider-lubricant Interactions in the Frequency Domain.....	136

6.2.5 Slider-lubricant Interactions Using Joint-time-frequency-analysis (JTFA)	142
6.2.6 Summary.....	143
6.3 Summary and Conclusion.....	145
6.4 Acknowledgement.....	145
6.5 References.....	146
 Chapter 7 Study of Slider/Disk Contact Due to Current-Induced Thermal Pole Tip Protrusion.....	148
7.1 Introduction.....	148
7.2 Experimental Conditions.....	152
7.3 Experimental Results and Discussion.....	153
7.3.1 Vibration characteristics of slider when write channel is engaged.....	153
7.3.2 “Rough” disk vs. “smooth” disk.....	156
7.3.3 Lubricated disk vs. de-lubricated disk.....	158
7.4 Modeling of Thermal Pole Tip Protrusion.....	160
7.4.1 Theoretical background.....	160
7.4.2 Simulation of Thermal Pole Tip Protrusion.....	162
7.5 Numerical Simulation of Slider Vibration in Five Degrees-of-freedom.....	166
7.5.1 Coordinate System.....	166
7.5.2 Equation of Motion for the Slider.....	167
7.5.3 Calculation of Air Bearing Shear Forces.....	168
7.5.4 Calculation of Contact, Adhesion and Friction Forces.....	170
7.5.5 Numerical Solution of the Coupled Slider Dynamics Equation and the Reynolds Equation.....	171
7.6 Simulation Results for Slider Vibrations.....	172
7.7 Summary and Conclusion.....	176
7.8 Acknowledgement.....	176
7.9 References.....	177
 Chapter 8 Magnetic Spacing Measurement Using the Read-back Signal.....	180
8.1 Introduction.....	180
8.2 Theoretical Background.....	181
8.3 Experimental Procedure.....	183
8.4 Experimental Results and Analysis.....	186
8.5 Summary and Conclusion.....	190
8.6 Acknowledgement.....	191
8.7 References.....	191
 Chapter 9 Conclusion.....	193
 Appendix A Calibration of the Position-Sensitive Detector (PSD)	196
Appendix B Measurement Issues and Solutions.....	205
a) Effect of Disk Flutter and Run-out.....	205
b) Effect of Signal Averaging.....	206
Appendix C A Rough Surface Contact/Adhesion/Friction Model [C1, C2]	208

Appendix D Flowchart for Solving Dynamics Equation.....	213
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LIST OF FIGURES

Figure 1.1 The first hard disk drive system (IBM 305 RAMAC) [1.1]	1
Figure 1.2 The first hard disk drive system with “air bearing” (IBM 1301) [1.2]	2
Figure 1.3 The disk drive unit (IBM 3340) with “Winchester Technology” [1.3]	3
Figure 1.4 A disk drive (ST506) made for micro-computers [1.4]	4
Figure 1.5 The structure of a state-of-the-art hard disk drive.....	5
Figure 1.6 A typical suspension and slider assembly.....	6
Figure 1.7 Digital magnetic writing and reading.....	7
Figure 1.8 Magnetic heads as an integration of a reading head and a writing head.....	9
Figure 1.9 Composite structure of a GMR sensor [1.10]	10
Figure 1.10 Scaling for magnetic recording.....	11
Figure 1.11 Schematic view of the head/disk interface.....	13
Figure 2.1 Schematic view of two interacting surfaces in relative motion.....	18
Figure 2.2 Roughness profile and definitions.....	19
Figure 2.3 Amplitude density function $f(z)$	21
Figure 2.4 Power spectral density of surface profile of three different disks.....	22
Figure 2.5 Schematic view of a spherical asperity in contact with a flat surface.....	23
Figure 2.6 Deformation of an asperity using a volume conservation model [2.8]	26
Figure 2.7 Model for contact between a rough surface and a smooth rigid plane.....	27
Figure 2.8 Lennard-Jones potential U vs. r	31
Figure 2.9 A schematic diagram of the meniscus formed at the contact interface of an idealized smooth sphere against a smooth surface [2.15]	32
Figure 2.10 Applicability of the Bradley’s model.....	34
Figure 2.11 Sketch of the JKR (Johnson, Kendall and Roberts) model.....	34
Figure 2.12 Sketch of the DMT (Derjagin, Muller, Toropov) model [2.19]	36
Figure 2.13 Sketch of the Maugis model [2.20]	37
Figure 2.14 Adhesion map for the elastic behavior of bodies [2.21]	38
Figure 2.15 Stribeck diagram: coefficient of friction as a function of viscosity-velocity-load parameter [2.35]	42
Figure 2.16 Schematic view of hydrodynamic lubrication [2.1]	43
Figure 2.17 Schematic view of boundary lubrication [2.1]	43
Figure 2.18 Schematic view of elasto-hydrodynamic lubrication (the dotted line indicates the original shape of the body) [2.36]	44
Figure 2.19 Overview of boundary lubrication for rough surfaces.....	45
Figure 2.20 Stiction as a function of lubricant film thickness and texture height [2.52] ..	50
Figure 2.21 Schematic view of four primary wear modes [2.72]	55
Figure 2.22 Schematic view of single asperity contact in sliding surfaces. a) Maximum contact area of radius a . b) Contact area after sliding at a distance y . c) After sliding at a distance $2a$, the contact area is reduced to zero [2.74]	56
Figure 2.23 Schematic view of a simplified abrasive wear model [2.75]	58
Figure 3.1 A simplified tapered beam model for suspension.....	70

Figure 3.2 A suspension design with its scaled shape simulated using finite element method.....	72
Figure 3.3 Results from finite element analysis of suspensions.....	73
Figure 3.4 Disks modeled at different dimensions.....	74
Figure 3.5 Results from finite element analysis of disks.....	75
Figure 4.1 The structure of a state-of-the-art hard disk drive.....	80
Figure 4.2 Typical stress-strain curves for brittle and ductile materials.....	81
Figure 4.3 Finite element model of a micro disk drive (disk diameter 27.4 mm)	84
Figure 4.4 Finite element model of the disk clamp.....	85
Figure 4.5 Point force applied on the slider.....	85
Figure 4.6 Acceleration profile.....	86
Figure 4.7 Crack formations on a glass disk (disk diameter is 27.4 mm)	89
Figure 4.8 Contours of plastic strain on disk surface (disk diameter is 27.4 mm)	92
Figure 4.9 Critical shock level vs. disk thickness (other design parameters are kept constant)	94
Figure 4.10 Critical shock level vs. clamping force (other design parameters are kept constant)	95
Figure 4.11 Critical shock level vs. clamp outer diameter (other design parameters are kept constant)	96
Figure 4.12 Finite element models for hard disk drives with different form factors.....	97
Figure 4.13 Critical shock level vs. disk form factor.....	98
Figure 4.14 Bending of a flat circular plate (outer edge free, inner edge clamped) [4.30]	99
Figure 5.1 Six components of slider vibration.....	106
Figure 5.2 Diagram denoting theory of laser Doppler vibrometry.....	108
Figure 5.3 System set-up for measurement of slider vibrations for five degrees-of-freedom.....	110
Figure 5.4 Illustration of measurement of slider motion using three independent laser Doppler vibrometers and one position-sensitive detector to detect the slider motion for five degrees of freedom: vertical, down-track, off-track, pitch, and roll.....	111
Figure 5.5 Air bearing slider used in the experiment.....	113
Figure 5.6 Waveforms of slider vibrations for five degrees-of-freedom when flying on a rotating disk. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll.....	114
Figure 5.7 Power spectral density of slider motion for five degrees-of-freedom obtained from experimental measurements. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll.....	115
Figure 5.8 Waveforms of slider vibrations for five degrees-of-freedom when contacting with a rotating disk. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll.....	117
Figure 5.9 Power spectral density of slider motion for five degrees-of-freedom obtained from experimental measurements. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll.....	118

Figure 6.1 Schematic view of the effect of circumferential and radial component of disk topography on the slider vibrations.....	123
Figure 6.2 The slider air bearing design for the experiment.....	124
Figure 6.3 Wavelength vs. frequency at radius of 25.4 mm for disk rotating at 7200 rpm.....	126
Figure 6.4 Waveforms of slider modulation and disk topography.....	127
Figure 6.5 Power spectral density of three components of topography of three different disks: (a) vertical component, (b) pitch component, and (c) roll component.....	129
Figure 6.6 Slider gain for (a) vertical, (b) pitch, and (c) roll.....	131
Figure 6.7 The slider air bearing design for the experiment.....	135
Figure 6.8 Vertical displacement of slider during slider-lubricant interactions at time intervals of $t = 2$ minutes.....	137
Figure 6.9 Images of disk after completion of experiment, obtained with optical surface analyzer (image on top is view of P-specula, image on bottom is view of Q-phase)	138
Figure 6.10 Image of the disk “wear-track” after completion of the experiment (left) and the slider trailing edge pad after completion of the experiment (right)	139
Figure 6.11 Power spectral density of slider displacement in (a) vertical, (b) down-track, (c) off-track, (d) pitch, and (e) roll direction.....	140
Figure 6.12 Time-frequency plot of slider displacement in vertical direction at time (a) $t = 0$ min, (b) $t = 8$ min, (c) $t = 16$ min.....	143
Figure 7.1 Schematic view of slider flying on a rotating disk at conditions a) without thermal pole tip protrusion, and b) with thermal pole tip protrusion.....	149
Figure 7.2 Air bearing slider used in the experiment.....	152
Figure 7.3 Slider vibrations for five degrees-of-freedom.....	154
Figure 7.4 Joint time frequency analysis of slider vibrations in five degrees-of-freedom.....	155
Figure 7.5 Disk surface topography measured using atomic force microscopy.....	157
Figure 7.6 Effect of surface roughness on slider vibrations.....	158
Figure 7.7 Effect of lubricant mobility layer on slider vibrations.....	159
Figure 7.8 Schematic view of the write head.....	163
Figure 7.9 Simulation model and simulated pole tip protrusion.....	164
Figure 7.10 Pole tip protrusion versus write current.....	165
Figure 7.11 Pressure profile of the slider with pole tip protrusion.....	166
Figure 7.12 Definition of coordinate system for slider vibrations.....	167
Figure 7.13 Simulation result for slider vibrations due to thermal pole tip protrusion...	173
Figure 7.14 Slider vibration amplitudes versus surface roughness.....	174
Figure 7.15 Slider vibration amplitudes versus lubricant thickness.....	175
Figure 8.1 Schematic of experimental setup for the magnetic spacing measurement....	184
Figure 8.2 Slider design and predicted pole tip flying height change as a function of disk rotational speed.....	185
Figure 8.3 V_{THR} as a function of disk position and RPM. The output reproduces variations in the magnetic spacing for a range of rotational speeds.....	187
Figure 8.4 Typical plot of scaled V_{THR} vs RPM, demonstrating the linearity in the scaled V_{THR}	188

Figure 8.5 Change of PTFH (pole tip flying height) as a function of disk RPM.....	189
Figure 8.6 Magnetic spacing (HMS) and independently-measured track average amplitude (TAA) as a function of time.....	190
Figure A1 Position-sensitive detector.....	197
Figure A2 Determination the radius (R) of the laser spot.....	198
Figure A3 Measured pitch voltage vs. predicted pitch voltage when laser is on a) disk surface, b) slider.....	199
Figure A4 Sketch of the angular change of the slider to the position change of the laser spot.....	200
Figure A5 Laser Doppler vibrometer measurement of slider angular vibration.....	202
Figure A6 Measured pitch angle vs. predicted pitch angle when laser is on a) disk surface, b) slider.....	203
Figure B1 Vertical displacement of a rotating disk, a) signal captured before analog filtering, b) signal captured after analog high-pass filtering at 10 kHz.....	206
Figure B2 Vertical displacement of a slider, a) single shot signal, b) signal averaged 25 times.....	207
Figure C1 Illustration of continuum mechanics based contact model.....	208
Figure C2 Force calculated using the model for a slider and a smooth disk.....	211
Figure D1 Flowchart on solving the dynamics equation.....	214
Figure D2 Flowchart for implementation of Newmark method ($\beta=0.25$, $\gamma=0.5$) [D1]...	215

LIST OF TABLES

Table 2.1 Typical value of wear coefficient.....	57
Table 4.1 Key material properties.....	82
Table C1 Parameters for material properties used in contact force calculation.....	211

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Chapter 6, in part, is a reprint of the following two publications: J. Xu, J. D. Kiely, Y.-T. Hsia, and F. E. Talke. “Effect of Disk Topography on Slider Dynamics,” IEEE Transactions on Magnetics, 42, 10, 2519-2521, (2006). And J. Xu, J. D. Keily, Y.-T. Hsia and F. E. Talke, “Dynamics of Ultra Low Flying Sliders During Contact with A Lubricated Disk,” Microsystem Technologies, 13, 10, 1371-1375, (2007). The dissertation author was the primary investigator and author of the publication.

Chapter 7, in part, has been submitted for publication: J. Xu, J. D. Keily, Y.-T. Hsia and F. E. Talke, “Slider/Disk Contact Due to Current-Induced Thermal Pole Tip Protrusion,” Microsystem Technologies, (2008). The dissertation author was the primary investigator and author of the publication.

Chapter 8, in part, is a reprint of the following publication: J. Xu, J. D. Keily, Y.-T. Hsia and F. E. Talke “Head Medium Spacing Measurement Using the Read-back Signal,” IEEE Transactions on Magnetics, 42, 10, 2486-2488, (2006). The dissertation author was the primary investigator and author of the publication.

VITA

2001	B.Eng., Huazhong University of Science and Technology, Wuhan, Hubei, China
2002	M.S., National University of Singapore (Singapore-MIT Alliance)
2002-2003	Research Engineer, Data Storage Institute, Singapore
2005/2006 /2007	Summer Internship, Seagate Research, Pittsburgh, Pa
2005/2006	Teaching Assistant, Department of Mechanical and Aerospace Engineering, University of California, San Diego
2003-2007	Research Assistant, Center for Magnetic Recording Research, University of California, San Diego
2008	Ph.D., University of California, San Diego

PUBLICATIONS

J. Xu and B. Liu, “Flying Height Modulation and Femto Slider Design”, *IEEE Transactions on Magnetics*, 39, 5, 2438-2440, (2003).

J. Xu, J. D. Kiely, Y.-T.Hsia, and F. E. Talke. “Effect of Disk Topography on Slider Dynamics,” *IEEE Transactions on Magnetics*, 42, 10, 2519-2521, (2006).

J. Xu, J. D. Keily, Y.-T. Hsia and F. E. Talke “Head Medium Spacing Measurement Using the Read-back Signal,” *IEEE Transactions on Magnetics*, 42, 10, 2486-2488, (2006).

J. Xu, J. D. Keily, Y.-T. Hsia and F. E. Talke, “Dynamics of Ultra Low Flying Sliders During Contact with A Lubricated Disk,” *Microsystem Technologies*, 13, 10, 1371-1375, (2007).

J. Xu, I. Etsion, F. E. Talke, “Failure Investigation of Brittle and Ductile Hard Disks Due to High Shock Levels,” submitted for publication in *ASME Journal of Mechanical Design*, (2007).

J. Xu, J. D. Keily, Y.-T. Hsia and F. E. Talke, “Slider/Disk Contact Due to Current-Induced Thermal Pole Tip Protrusion,” submitted for publication in *Microsystem Technologies*, (2008).

Z. Wang, W. Cao, X. Shan, J. Xu, S. Lim, W. Noell, N. de Rooij, “Development of 1x4 MEMS-based Optical Switch,” *Sensor and Actuators A-Physical*, 114, 1, 80-87, (2004).

A. N. Murthy, M. Pfabe, J. Xu, and F. E. Talke, “Dynamic Response of 1-inch Form Factor Disk Drives to External Shock and Vibration Loads,” published online in *Microsystems Technologies* (2006)

ABSTRACT OF THE DISSERTATION

Head/Disk Interface Tribology in the Nanometer Regime

by

Jianfeng Xu

Doctor of Philosophy in Engineering Sciences (Mechanical Engineering)

University of California, San Diego, 2008

Professor Frank E. Talke, Chair

This thesis presents experimental and theoretical studies of the characteristics of the head/disk interface at very low flying height. The study starts with a discussion of the tribological background of the head/disk interface and presents a review of the literature related to studies of the head/disk interface. Then, mechanical scaling laws for hard disk drives are discussed. Numerical results for failure inception of brittle and ductile hard disks due to high shock levels are presented. An experimental setup for measuring slider dynamics in five degrees-of-freedom (DOF) is presented. This is followed by experimental studies of slider vibrations due to slider/disk contacts. Thereafter, a study of slider vibrations due to write-head induced “thermal” pole-tip-protrusion is presented. Numerical simulations of slider vibrations are compared with experimental results. A method for measuring the magnetic spacing based on the read-back signal is presented. Finally, the results of this thesis are summarized and directions for future research are given.

Chapter 1 Introduction

1.1 Magnetic Recording Hard Drives and Their Evolutions

Since the first introduction of electronic computer in the early 1940's, computer technology has affected almost every aspect of life from business to home. The hardware of the computer itself has also evolved and improved over the course of time. Among the most important components of computers are magnetic hard disk drives, which have played a significant role in computer technology.



Figure 1.1 The first hard disk drive system (IBM 305 RAMAC) [1.1]

In 1956, International Business Machines (IBM) introduced the 305 RAMAC (Random Access Method of Accounting and Control) as the first magnetic hard disk for data storage [1.1] (Fig. 1.1). Though the overall capacity of the 305 RAMAC was only 5

megabytes (MB), it had 50 spinning disks of 600 mm (“24-inch”) diameter. The areal density of this magnetic hard disk drive was only 2,000 bits per square-inch (3.1 bits per square-mm), and its head-to-disk spacing was approximately 20 μm .

In 1962, IBM introduced the 1301 series disk drive [1.2]. The key advance of this disk drive was the implementation of heads that were flying above the surface of the disk on an "air bearing", reducing the distance from the head to the surface of the disk to around 6 μm .

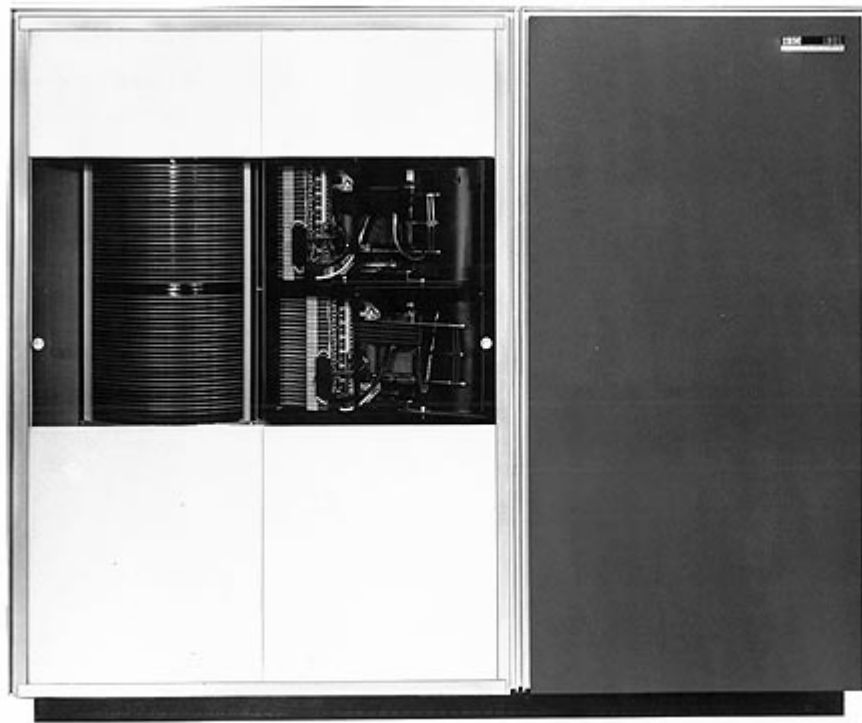


Figure 1.2 The first hard disk drive system with “air bearing” (IBM 1301) [1.2]

Another milestone for the hard drive industry occurred in 1973 with the introduction of the IBM 3340 series disk drive [1.3], which offered an advanced head and disk

technology known as "Winchester Technology". The 3340 disk drive featured a smaller read/write head that was flying closer to the disk surface, on an air film of about $0.5\text{ }\mu\text{m}$ in thickness. The 3340 disk drive could record data on the disks at a density of 1.7 million bits per square-inch (2635 bits per square-mm). A picture of the IBM 3340 drive unit is shown in Fig. 1.3.

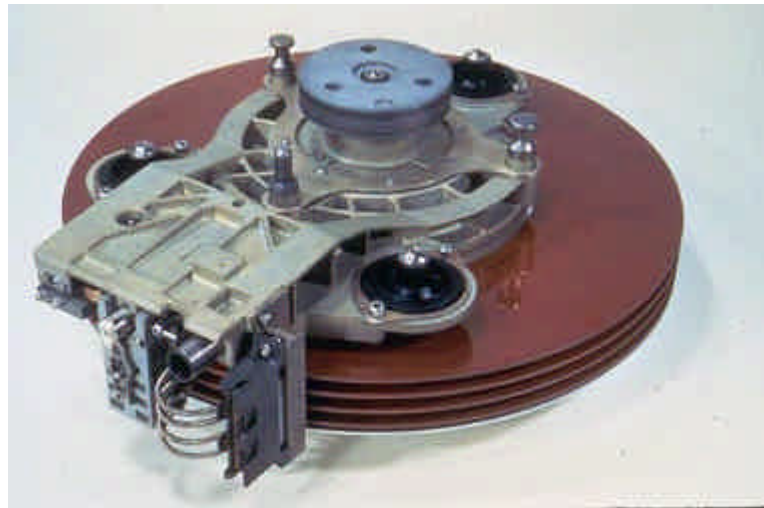


Figure 1.3 The disk drive unit (IBM 3340) with “Winchester Technology” [1.3]

In 1979, Seagate Technology introduced a disk drive (ST506) for micro-computers, as shown in Fig. 1.4. The disk drive (ST506) featured disks with 130 mm (“5.25-inch”) diameter and a recording density of 8 million bits per square-inch (12.4 kilo-bits per square-mm).

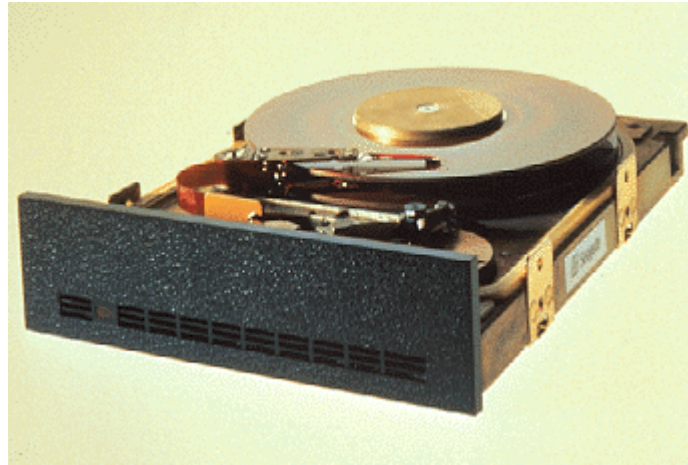


Figure 1.4 A disk drive (ST506) made for micro-computers [1.4]

Modern hard disk drives have a recording density of about 150 billion bits per square-inch (232.5 million bits per square-mm) and offer a number of different form factors such as 95 mm (“3.5 inch”), 65 mm (“2.5 inch”), 45 mm (“1.8 inch”) and 27.4 mm (“1.0 inch”). The demand for high quality graphics and media content requires greater storage capacity than ever before. The recording density of today’s hard disk drives is 75 million times larger than that of the first hard disk drive in 1956. At the same time, the head-to-disk spacing of hard disk drives has decreased by a factor of 2000. The growing demand for data storage is accelerating the development of small form factor hard disk drives, i.e., disk drives for which the disk diameter is less than 50 mm. Many mechanical design aspects of small form factor disk drives are influenced by the size of the individual components, and special attention must be given to the dynamic performance and shock response of small form factor drives.

1.2 Structure of a Hard Disk Drive and Magnetic Recording

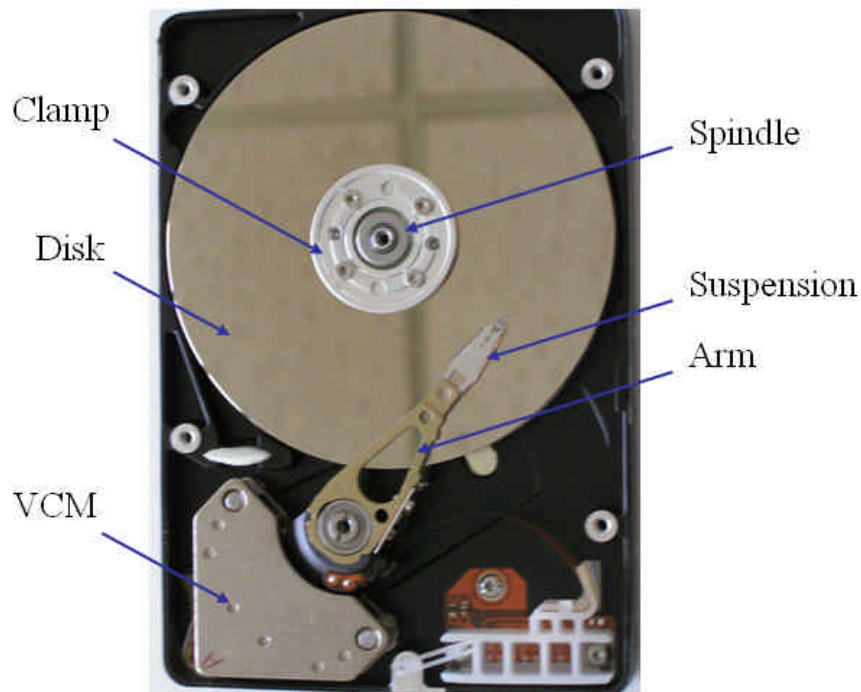


Figure 1.5 The structure of a state-of-the-art hard disk drive

The internal structure of a state-of-the-art hard disk drive is shown in Fig. 1.5. Each hard disk drive contains one or more magnetic disks, mounted on the shaft of a spindle motor. The suspension/arm is mounted on the side of the disk and driven by the voice coil motor. The sliders, each equipped with integrated thin film read/write heads, are mounted on the backside of the suspension (Fig. 1.6). When the hard disk drive is powered on, the disks start to rotate until they reach a constant velocity. A converging channel between the slider and the disk generates a so-called “air bearing”. As a result of the air bearing, the slider flies over the disk, enabling the read/write heads to record information on circumferential tracks of the disk, and to read information from the disk. The voice coil

motor moves the suspension and the slider from track to track to record or read data on the entire surface of the disk.

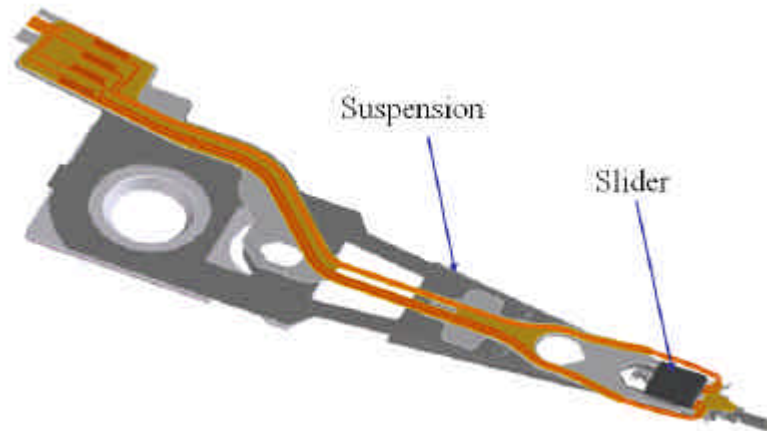


Figure 1.6 A typical suspension and slider assembly

During the operation of a disk drive, the slider flies over the disk at a fixed head-to-disk spacing. To increase the recording density, the head-to-disk spacing of the slider must be minimized. In today's disk drives, the head-to-disk spacing is approximately 8 nm.

1.3 The Read/Write Process

A schematic view of the recording process is shown in Fig. 1.7. Data are stored on a magnetic disk in the form of binary bits [1.5]. During the write process, a write current in the head coil generates a magnetic field between the poles of the head. This field magnetizes the disk medium. The write current is synchronized with a so-called “time clock”. To write a “1” onto the rotating magnetic disk at a given “time clock”, the direction

of the write current is changed, resulting in a magnetic transition being recorded in the disk medium. The absence of a transition at the clock time implies that a “0” is stored in the medium.

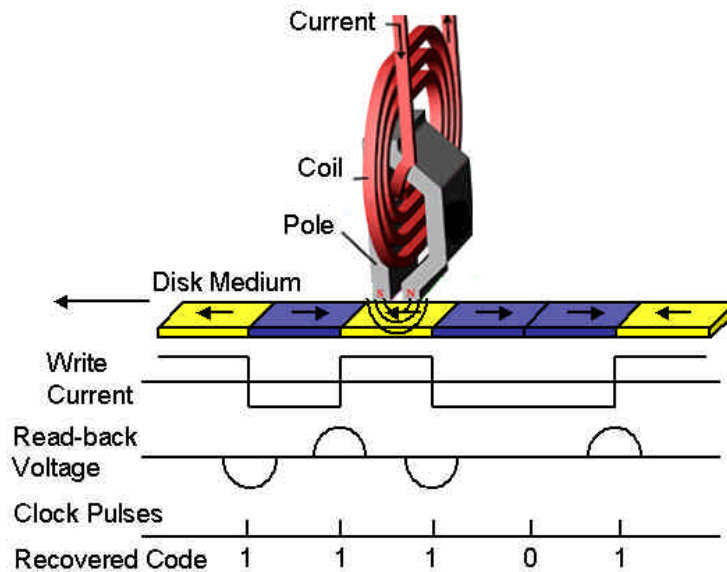


Figure 1.7 Digital magnetic writing and reading

For the heads used until the 1990's, the read process was performed with the same head structure that was used for the write process. During reading, a voltage pulse is induced in the head whenever a change in the magnetization direction in the disk medium occurs. If there is no voltage pulse at the given time clock, a “0” is read from the disk medium.

Based on the Wallace equation [1.6], the voltage signal E from a recording head for sine-wave recording is given by:

$$E(\bar{x}) = Ce^{-kd} \left[\frac{1-e^{-kd}}{kd} \right] \left[\frac{\sin \frac{kg}{2}}{\frac{kg}{2}} \right] \cos k\bar{x} \quad (1.1)$$

where $\bar{x}$ is the distance that the medium moves; C is a constant related to the material properties of the medium and the recording head; k is the wave-number of the recorded sine wave defined as $\frac{2\pi}{l}$; l is the wavelength; d is the magnetic spacing, defined as the distance between the bottom surface of the read/write element and the top surface of the magnetic layer on the disk; d is the thickness of the magnetic medium; and g is the gap length of the recording head. The term e^{-kd} is the so-called spacing loss term. The term $\left[\frac{1-e^{-kd}}{kd} \right]$ is the loss term related to the thickness of the magnetic medium, and the term $\left[\frac{\sin \frac{kg}{2}}{\frac{kg}{2}} \right]$ is the loss term related to the gap length of the recording head.

In modern hard disk drives, the read and write function of a magnetic head is separated into two individual entities, as shown in Fig. 1.8. The write head has the same structure as the head described previously. A separate read head is added next to the write head, consisting of two adjacent magnetic shields with a magneto-resistive (MR) sensor sandwiched in between them.

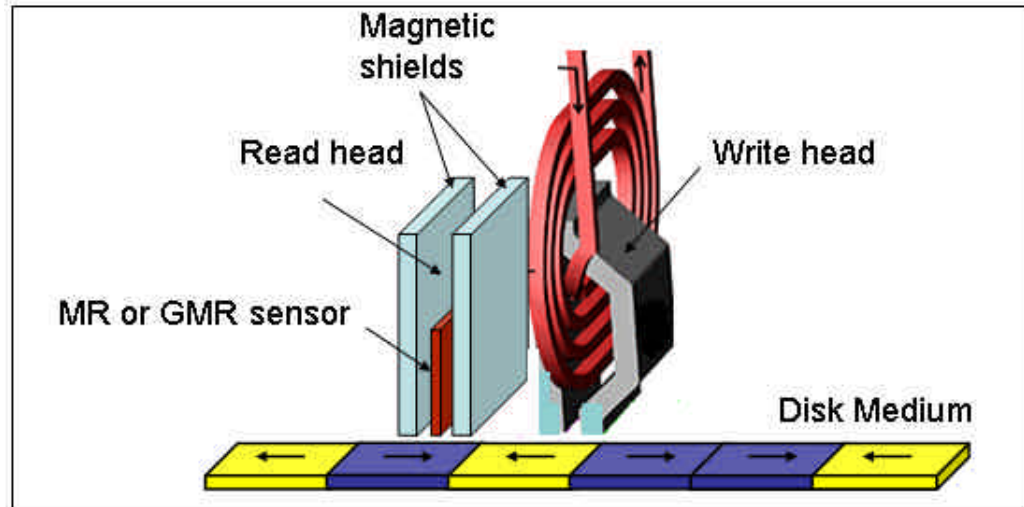


Figure 1.8 Magnetic heads as an integration of a reading head and a writing head.

The MR sensor is based on the characteristics of certain metals to change their resistance in the presence of a magnetic field [1.5]. In the first implementation of MR sensors, the film between the two shields was an alloy of Ni and Fe. When the head passes over a magnetic field of one polarity, say, a "0" on the disk, no transverse magnetic field is applied and the electrical resistance is high. When the head passes over a magnetic field of the opposite polarity ("1"), the magnetic field rotates the magnetic orientation of the sensor which lowers the electrical resistance. The resistance change can be measured and as a result, data are read back from the disk medium.

In modern magnetic heads, the read sensor uses the so-called giant-magneto-resistive (GMR) effect discovered by Gruenberg and Fert et al. [1.7, 1.8]. The GMR effect is related to the fact that conduction electrons with a spin direction parallel to the magnetic orientation in a material move freely, resulting in low resistance. On the other hand, if the spin direction of conduction electrons is opposite to the magnetic

orientation of the material, frequent collisions occur with atoms in the material, resulting in high resistance [1.9].

A typical GMR sensor is composed of four thin layers that are sandwiched into a single structure, as shown in Fig. 1.9. The four layers are the so-called free layer, the spacer layer, the pinned layer and the exchange layer. When the head passes over a magnetic field of one polarity, say, a "0" on the disk, the electrons of the free layer are aligned with those of the pinned layer; this creates a low resistance in the entire head structure. When the head passes over a magnetic field of the opposite polarity ("1"), the electrons of the free layer rotate so that they are not aligned with those of the pinned layer. This causes an increase in the resistance of the overall structure. Similar to the conventional MR effect, the change in resistance allows the detection of a transition in the magnetic medium, and thus, the reading of stored information. The GMR sensor is much more sensitive in detecting a change of the magnetic field than a MR sensor.

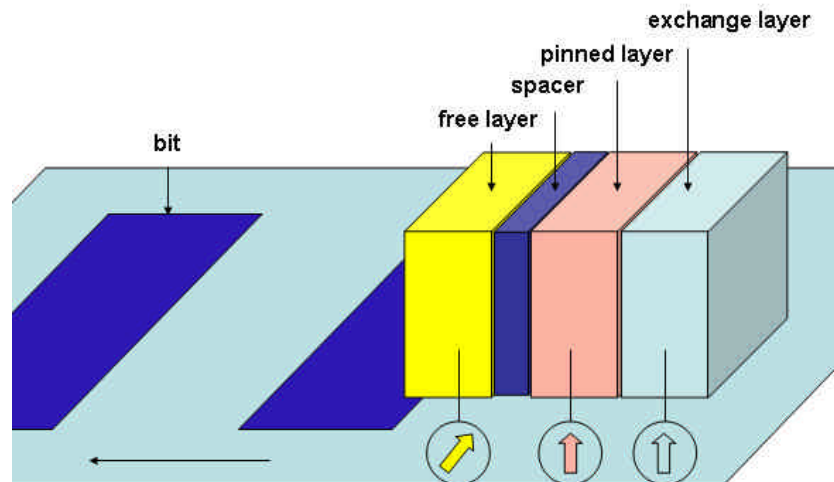


Figure 1.9 Composite structure of a GMR sensor [1.10]

1.4 Scaling Laws for Magnetic Recording

As discussed in Section 1.1, the size of typical hard disk drives has decreased substantially in the past fifty years. At the same time, the storage capacity of hard disk drives has increased by many orders of magnitude. Thus, the smallest disk drives today have a much higher storage capacity than the largest disk drive systems in the 1950s or 1960s.

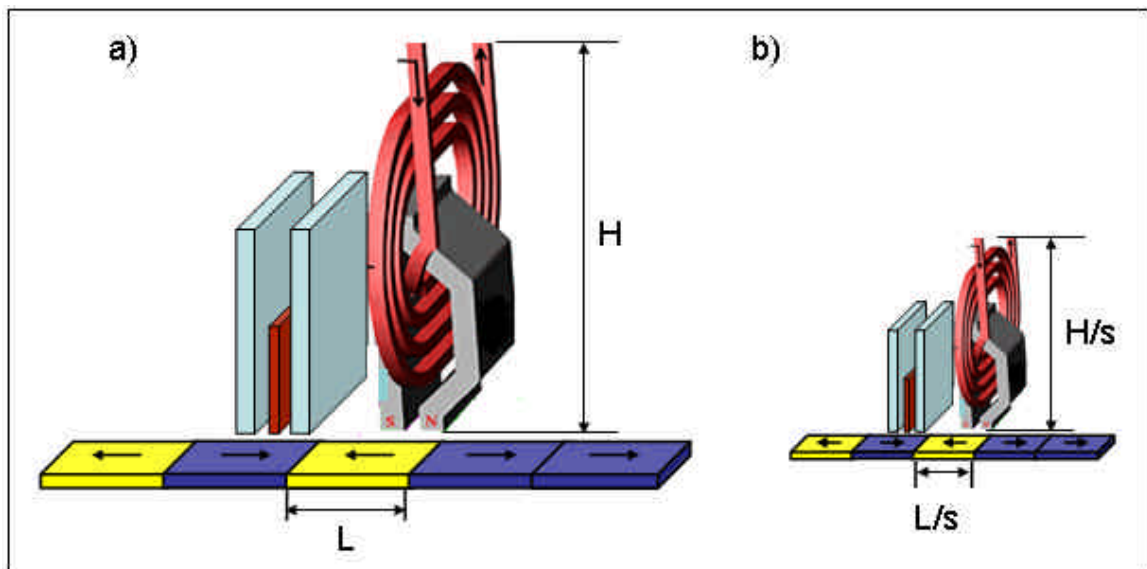


Figure 1.10 Scaling for magnetic recording

To predict the magnetic performance of a future disk drive with increased capacity, researchers [1.11] have used so-called “scaling laws”. A simple magnetic recording system, as shown in Fig. 1.10a, is composed of a magnetic head and a magnetic medium. The set-up in Fig. 1.10a describes the “present” design, while the set-up in Fig. 1.10b describes the

“new” or “future” design. To keep the magnetic performance of the “present” and “new” designs the same, we require that the signal to noise ratio (SNR) remain the same. Thus, if the dimensions of the magnetic recording system are decreased by a factor of s , a scaled magnetic recording system is obtained, as shown in Fig. 1.10b. In this “new” magnetic recording system, the width, length and height of the magnetic system are decreased by a factor of s . Since each recorded bit in the “new” design has decreased in width and length by the scale factor s , the “new” magnetic recording system has an increased areal density corresponding to s^2 times the areal density of the “present” system. As every dimension is scaled by the same factor s , the flying height has to be reduced at the same rate. Failure to do so would result in a greatly decreased readback signal, and a write field with insufficient gradient to write the smaller bits required. This is the reason why the ratio of bit length to flying height has not changed significantly in 30 years [1.12].

The signal to noise ratio (SNR) in magnetic recording systems is proportional to $\frac{wb}{D^2}$, where w and b denote the width and length of a recorded bit, respectively, and D denotes the diameter of a magnetic grain [1.13]. To maintain the same signal to noise ratio, the grain size (D) has to be reduced. If the grain size becomes too small, thermal energy may destabilize the magnetization and cause the recorded magnetization to decay. Innovations in magnetic recording technology have to be made in order to maintain the growth of areal density of hard disk drives.

1.5 The Head/Disk Interface

A schematic of the head/disk interface is shown in Fig. 1.11. The read/write element is embedded in a so-called “slider”, which flies over the disk. Both the slider surface and the magnetic layer of the disk are covered by a diamond-like-carbon (DLC) overcoat to protect the surfaces from wear. The DLC overcoat is covered by a thin lubricant layer that is applied to the disk surface by dip coating of the disk in a lubricant bath. In small form factor disk drives, the disk substrate is generally made of glass. For glass disks, the magnetic layer is directly deposited onto the glass substrate with a soft under-layer. For aluminum disks, a nickel-phosphorus layer (Ni-P layer) is first deposited on the aluminum substrate to create a hard surface. Then the magnetic layer and the carbon overcoat are deposited by sputtering or chemical vapor deposition.

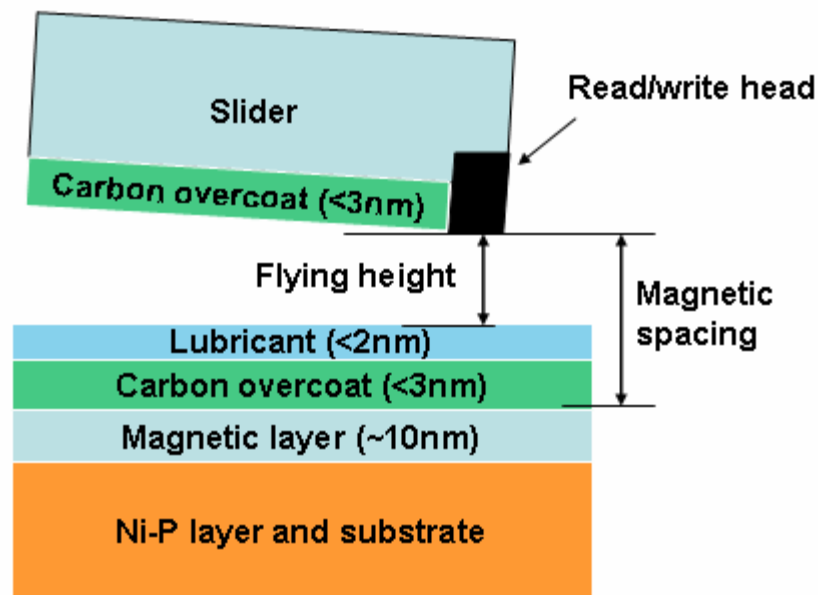


Figure 1.11 Schematic view of the head/disk interface

During operation of a disk drive, a nearly constant spacing between the slider and the disk is maintained. This spacing is the so-called “flying height”. In Fig. 1.11, the flying height between slider and disk is shown as well as the magnetic spacing. The magnetic spacing is defined as the distance between the bottom surface of the read/write element and the top surface of the magnetic layer on the disk. If the flying height is much larger than the sum of the lubricant thickness and the carbon overcoat, the effect of the carbon overcoat thickness and lubricant thickness can be neglected. In present day disk drives, for which the flying height is approximately 8 nm, the effect of the lubricant thickness and carbon overcoat are very important and have to be taken into account.

As shown in the Eq. 1.1, the readback voltage decreases exponentially with an increase in the magnetic spacing. An increase in linear bit density requires a decrease in the magnetic spacing between the head and the disk. This, in turn, requires that the head flying height must be reduced and that a thinner protective overcoat must be implemented.

In order to achieve areal densities on the order of 1 trillion bits per square-inch (1.55 billion bits per square-mm), the magnetic spacing must be reduced to 5 nm [1.14]. For a recording density of 1 trillion bits per square-inch (1.55 billion bits per square-mm), the length of each recorded bit (in the circumferential direction) should be approximately 12.5 nm and the width of each recorded bit (in the radial direction) should be approximately 50 nm.

As the magnetic spacing and, therefore, the flying height becomes very small, intermittent contacts may occur between slider and disk during normal disk drive operation.

Such contacts can cause instability of the head/disk interface [1.15]. It is therefore very important to study slider dynamics at very close spacing, i.e., at spacing of less than 5 nm.

1.6 Content of Thesis

This thesis presents experimental and theoretical studies of the characteristics of the head/disk interface at very low flying height on the order of several nanometers. The study starts with a discussion of the tribological background of the head/disk interface and presents a review of the literature related to studies of the head/disk interface. Then, mechanical scaling laws for hard disk drives are discussed. Numerical results for failure inception of brittle and ductile hard disks due to high shock levels are presented. An experimental setup for measuring slider dynamics in five degrees-of-freedom (DOF) is presented. This is followed by experimental studies of slider vibrations due to slider/disk contacts. Thereafter, a study of slider vibrations due to write-head induced “thermal” pole-tip-protrusion is presented. Numerical simulations of slider vibrations are compared with experimental results. A method for measuring the magnetic spacing based on the read-back signal is presented. Finally, the results of this thesis are summarized and directions for future research are given.

A brief synopsis of each chapter is given below.

Chapter 2 gives a brief review and analysis of the tribology of the head/disk interface.

Chapter 3 outlines the so-called “mechanical scaling laws” for hard disk drives in terms of mechanical parameters.

Chapter 4 discusses the breakage of glass disks and the plastic deformation of aluminum disks.

Chapter 5 describes a measurement system for the detection of slider vibrations in five degrees-of-freedom.

Chapter 6 investigates the relationship between disk surface topography and slider vibrations and studies slider/lubricant interactions for flying at very low flying heights.

Chapter 7 analyses slider/disk interactions using a thermally actuated slider and compares numerical results for slider vibrations with experimental measurements.

Chapter 8 presents and evaluates a technique for obtaining the flying height based on the read-back signal.

Chapter 9 provides a summary of the work contained in this thesis.

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Chapter 2 Introduction to Tribology and Its Application to the Head/Disk Interface

2.1 Introduction

Tribology is the study of friction, wear and lubrication of interacting surfaces in relative motion [2.1, 2.2]. A schematic view of two interacting surfaces in relative motion is shown in Fig. 2.1. Friction is the force that opposes relative tangential motion. Wear is the removal of material from either one of the two surfaces. Lubrication occurs when two opposing surfaces move relative to each other in the presence of a lubricant film between the surfaces.

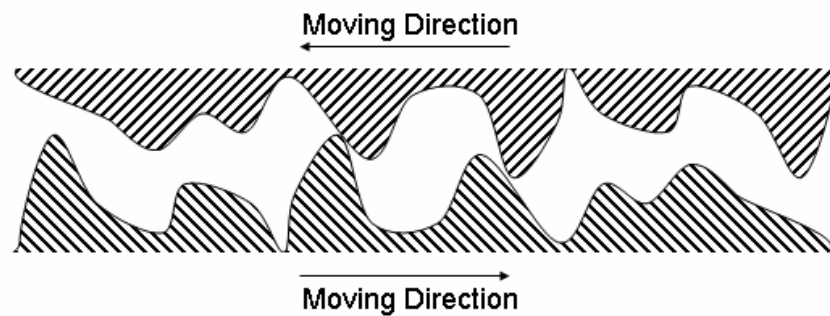


Figure 2.1 Schematic view of two interacting surfaces in relative motion

Tribology is an interdisciplinary field dealing with surfaces, contact mechanics, material science, lubricant chemistry, instrumentation and related areas. The head/disk interface is unique in terms of its tribological characteristics. A magnetic recording slider is positioned above the disk on a thin air film. During operation of the disk drive, the slider

flies above the disk. This situation is called hydrodynamic lubrication, and contact between the slider and the disk is absent. However, if the spacing between the slider and the disk is very small, contacts between the slider and the disk occur, leading to friction and wear of the interface. Contacts can also occur during start/stop of a disk drive or during the load/unload process. Thus, the tribology of the head/disk interface is concerned with all areas of tribology, i.e., hydrodynamic lubrication, boundary lubrication, friction and wear. In order to understand the tribology of the head/disk interface, an overview of the general principles of tribology will be presented first.

2.2 Surface Roughness

On a microscopic scale, all solid surfaces are rough. When surfaces are placed in contact, surface roughness causes contact at discrete points. This causes friction and wear (Fig. 2.1). To understand the behavior of surfaces in contact, it is important to characterize surface roughness. Two commonly used measures exist to describe surface roughness, i.e., center line average roughness (R_a) and root-mean-square roughness (R_q or RMS). A typical surface profile is shown in Fig. 2.2.

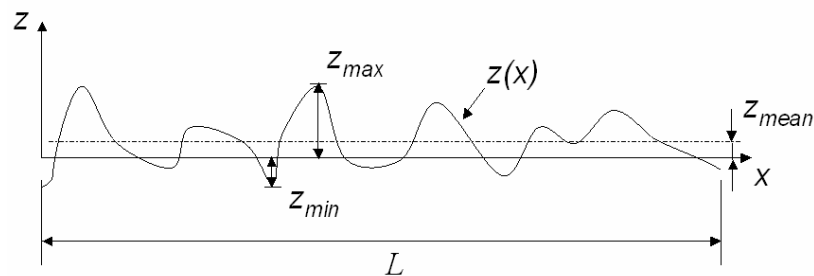


Figure 2.2 Roughness profile and definitions

With reference to Fig. 2.2, the center line average roughness R_a is defined as the arithmetic mean of the absolute value of the surface height from the mean z_{mean} through the profile [2.3]. For a profile of length L , the center line average roughness R_a is given by

$$R_a = \frac{1}{L} \int_0^L |z(x) - z_{mean}| dx \quad (2.1)$$

where $z(x)$ is the amplitude of the surface profile from a given coordinate system, and z_{mean} is defined as the mean value of $z(x)$ for a given profile of length L , i.e., $z_{mean} = \frac{1}{L} \int_0^L z(x) dx$.

The root-mean-square roughness R_q (or *RMS*) is defined as

$$R_q = \sqrt{\frac{1}{L} \int_0^L (z(x) - z_{mean})^2 dx} \quad (2.2)$$

In many cases, the values of R_a and R_q are close to each other. For example, if the amplitude variation $z(x)$ follows a Gaussian distribution between 0 and L , $R_q = 1.25 R_a$ [2.3].

Another parameter used often in characterizing surface roughness is the peak-to-valley (or peak to peak) distance z_{pv} , given by

$$z_{pv} = |z_{max} - z_{min}| \quad (2.3)$$

where z_{max} is the largest value of $z(x)$ and z_{min} is the smallest value of $z(x)$ for a given profile of length L (see Fig. 2.2).

The values of R_a and R_q are not characterizing a surface uniquely, i.e., they lack important information about surface topography, such as the shape and the sharpness of the

surface peaks. Often, a so-called amplitude density function $f(z)$ is used to describe surface topography (Fig. 2.3).

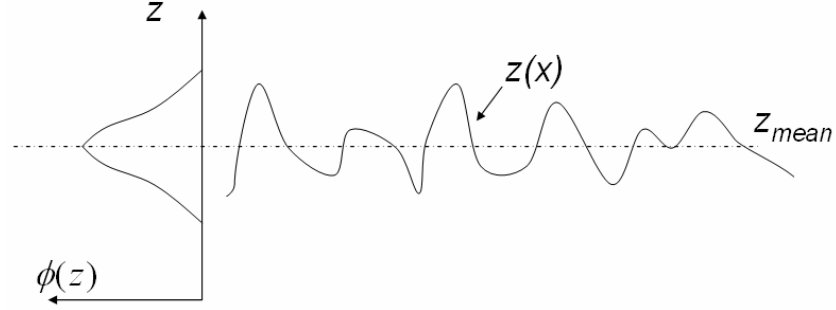


Figure 2.3 Amplitude density function $f(z)$

Other parameters used to characterize surfaces are skewness and kurtosis. The skewness (S_{sk}) of a surface profile is defined as

$$S_{sk} = \frac{1}{R_q^3} \int_{-\infty}^{+\infty} (z - z_{mean})^3 f(z) dz \quad (2.4)$$

The kurtosis (S_{ku}) of a surface profile provides information about the sharpness of the peak of the amplitude density curve. It is defined as

$$S_{ku} = \frac{1}{R_q^4} \int_{-\infty}^{+\infty} (z - z_{mean})^4 f(z) dz \quad (2.5)$$

In addition, the autocorrelation function and the power spectral density function [2.4] are frequently used parameters. The autocorrelation function provides a measure of the correlation between the heights of a surface at positions separated by a distance D along the surface. The autocorrelation function $C(D)$ is defined as

$$C(\Delta) = \frac{1}{L} \int_0^L z(x) z(x + \Delta) d\Delta \quad (2.6)$$

If the distance D is zero, the value of the autocorrelation function reaches its maximum and equals the square of the root-mean-square roughness (R_a).

The power spectral density [2.4] describes the spatial frequencies f that are present in a surface profile. It is defined as the Fourier transform of the autocorrelation function $C(D)$, i.e.,

$$P(f) = \frac{2}{p} \int_0^\infty C(\Delta) \cos(2p f \Delta) d\Delta \quad (2.7)$$

An example of the power spectral density of a disk surface measured using a laser Doppler vibrometer is shown in Fig. 2.4 for three different disks. The measurement was taken at a radius of 25.4 mm in the circumferential direction for a full rotation. This figure shows that disk 1 has the lowest spectral density in the frequency range between 50 kHz and 300 kHz while disk 3 has largest spectral density.

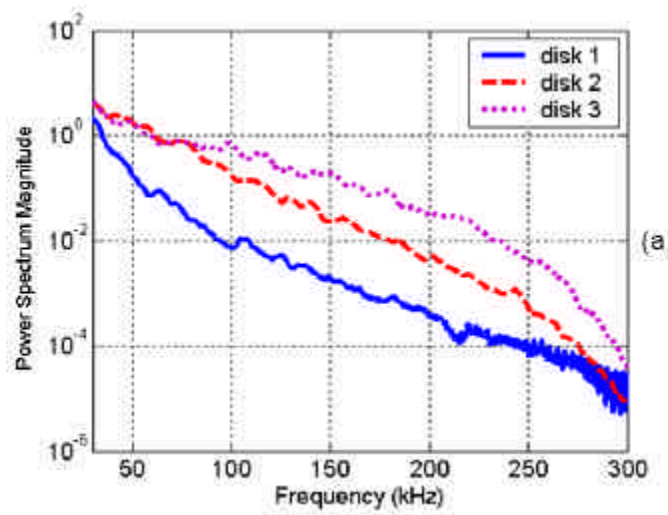


Figure 2.4 Power spectral density of surface profile of three different disks

2.3 Contact Theory

2.3.1 Deformation of a Single Asperity

If two surfaces are brought together, contact will occur as a result of the normal load. Due to the presence of surface roughness, contact occurs at the asperities (high areas) of the two surfaces. These asperities are responsible for supporting the normal load between the two surfaces. They are also the origin of friction and wear.

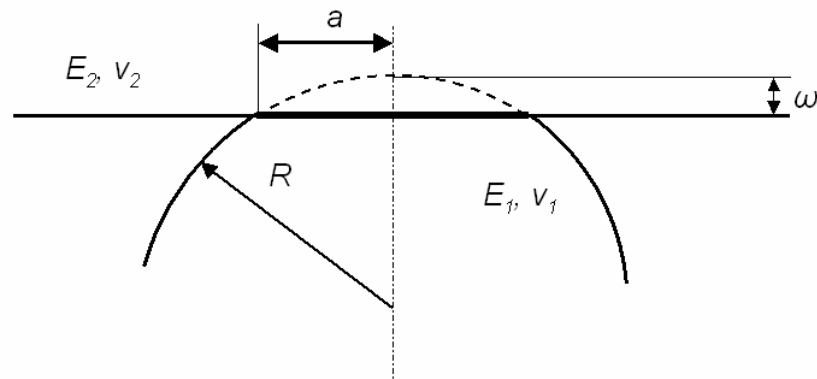


Figure 2.5 Schematic view of a spherical asperity in contact with a flat surface

The Hertz contact model [2.5] examines the elastic deformation of an individual spherical asperity in contact with a nominally flat surface (Fig. 2.5). The model assumes that the deformation of the asperity is purely elastic. In addition, it assumes that the material is homogeneous and isotropic. The radius of the spherical asperity is denoted by R . During contact, two points outside of the contact zone move closer by a distance w , the so-called interference height. The radius a and the area A of the contact zone can be calculated as [2.5]

$$w = \frac{a^2}{R} \quad (2.8)$$

and

$$A = p a^2 = p R w \quad (2.9)$$

Denoting the Young's modulus and Poisson's ratio as E_1, ν_1 and E_2, ν_2 for the spherical asperity and the flat surface, respectively, one can define a composite modulus E as

$$E = \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)^{-1} \quad (2.10)$$

The normal load W between a single asperity and a flat surface can then be obtained as [2.5]

$$W = \frac{4}{3} E R^{\frac{1}{2}} w^{\frac{3}{2}} \quad (2.11)$$

while the mean contact pressure P_{mean} is given by

$$P_{\text{mean}} = \frac{W}{p a^2} \quad (2.12)$$

Due to the spherical shape of the asperity, the contact pressure is not uniform over the contact area but reaches its highest value at the center of the contact area and falls to zero at the edge of the contact area. The maximum Hertzian contact pressure P_0 occurs at the center of the contact zone and is given in terms of the mean contact pressure as

$$P_0 = \frac{3}{2} P_{\text{mean}} \quad (2.13)$$

Combining Eqns. 2.8 to 2.13, one can obtain the displacement w due to contact as

$$w = \left(\frac{p}{2} \right)^2 \left(\frac{P_0}{E} \right)^2 R \quad (2.14)$$

Hertz [2.5] only considered the elastic deformation of a spherical asperity and a flat surface. As the normal load between a spherical asperity and a plane increases, plastic deformation can occur at the contact interface. Greenwood and Williamson [2.6] introduced a contact model which accounts for the plastic deformation of the contact area. They assumed that the maximum contact pressure P_0 is proportional to the surface hardness H . Defining K as the ratio of the maximum contact pressure and the surface hardness, the critical value w_c (critical interference) for plastic deformation to occur is given by

$$w_c = \left(\frac{p}{2}\right)^2 \left(\frac{KH}{E}\right)^2 R \quad (2.15)$$

The normal load needed for the critical value w_c to be reached is [2.6]

$$W = \frac{4}{3} E R^{\frac{1}{2}} w_c^{\frac{3}{2}} \quad (2.16)$$

Tabor [2.7] showed experimentally that the onset of plastic flow occurs when the maximum contact pressure P_0 between a sphere and a plane reaches a value of $0.6H$, where H is the hardness. Greenwood and Williamson [2.6] used the information from Tabor's experiment and obtained the critical value of displacement for which plastic deformation occurs at the contact between a spherical asperity and a flat surface as

$$w_c = \left(\frac{p}{2}\right)^2 \left(\frac{0.6H}{E}\right)^2 R \approx 0.89 \left(\frac{H}{E}\right)^2 R \quad (2.17)$$

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

Chang et al. [2.8, 2.9] reinterpreted the concept from Greenwood and Williamson's model by considering volume conservation for plastically deformed asperities (Fig. 2.6).

For $w > w_c$, the contact area of a plastically deformed asperity is

$$A = p R w \left(2 - \frac{w_c}{w} \right) \quad (2.18)$$

Thus, the contact load for such an asperity is

$$W = p R w \left(2 - \frac{w_c}{w} \right) K H \quad (2.19)$$

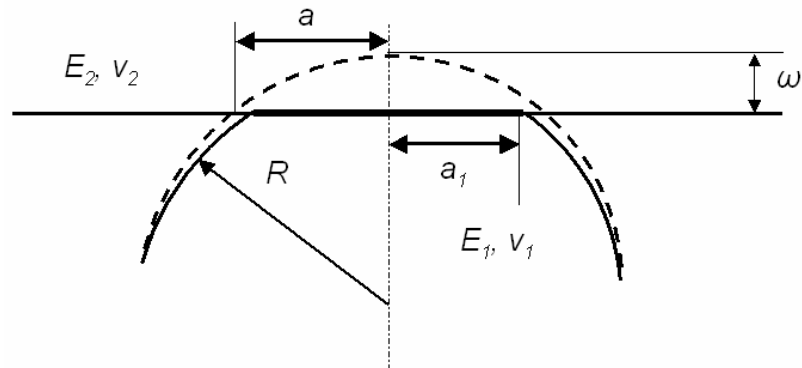


Figure 2.6 Deformation of an asperity using a volume conservation model [2.8]

Kogut and Etsion [2.10] further improved the elastic-plastic contact model of a spherical asperity contact with a flat surface. They separated the contact in four regions ($w/w_c < 1$; $1 \leq w/w_c < 6$; $6 \leq w/w_c < 110$; $110 \leq w/w_c$) and used curve fitting of the contact pressure, contact area and contact load as a function of the dimensionless interference w/w_c to predict elastic contact, elastic-plastic contact and plastic contact conditions.

2.3.2 Contact of Multiple Asperities

In most practical cases, contact between two surfaces occurs in multiple places. One of the first statistical theories for the contact of rough surfaces was introduced by

Greenwood and Williamson [2.6]. In their model it is assumed that all contacting asperities are spherical in shape and that they have the same radius R . In addition, Greenwood and Williamson [2.6] assumed that the height of the asperities follows a height distribution $f(z)$. Thus, the probability that a particular asperity has a height between z and $z+dz$ above some reference plane will be $\int_z^{z+dz} f(z) dz$. Fig. 2.7 illustrates the contact between a rough surface and a rigid plane surface assumed in the model.

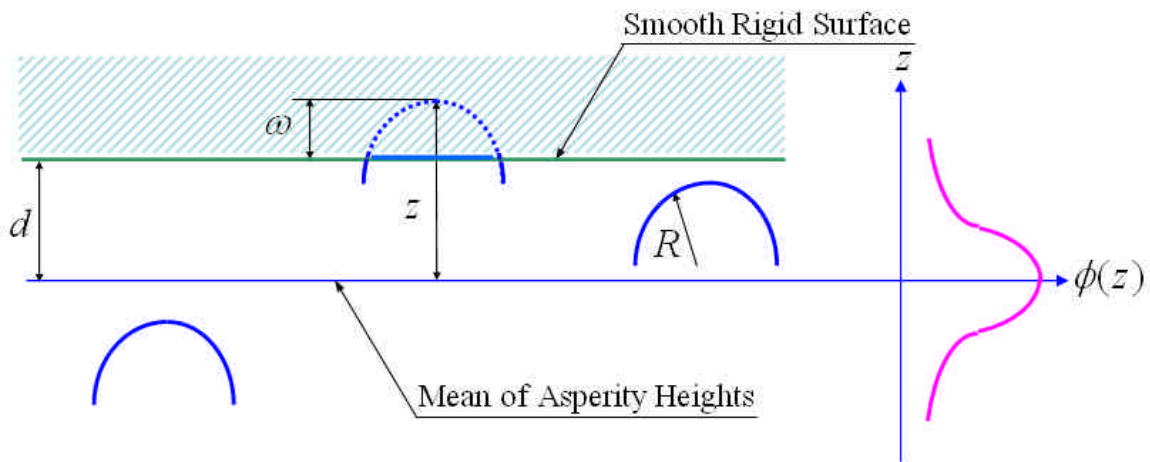


Figure 2.7 Model for contact between a rough surface and a smooth rigid plane

As indicated in Fig. 2.7, asperities of contact radius R are present on the bottom surface, following the probability distribution $f(z)$. The flat surface, shaded in Fig. 2.7, is brought in contact with the bottom surface. After contact, the two surfaces are separated by a distance d measured from the smooth rigid surface to the mean of the asperity heights of the rough surface. For all asperities with height z greater than d , the probability of making contact at that given asperity is

$$prob(z > d) = \int_d^{\infty} f(z) dz \quad (2.20)$$

If the asperity density is denoted by h and the nominal contact area by A_n , the contact area A_e and the contact load W_e for elastic contact can be calculated as

$$A_e(d) = h A_n \int_d^{\infty} A(z-d) f(z) dz \quad (2.21)$$

$$W_e(d) = h A_n \int_d^{\infty} W(z-d) f(z) dz \quad (2.22)$$

where $z-d$ is the interference height w , A is the contact area (Eqn. 2.9) and W is the normal load (Eqn. 2.11).

Assuming that plastic flow first occurs internally, Greenwood and Williamson [2.6] took the criterion for detectable plastic flow to be

$$w_c = \left(\frac{H}{E} \right)^2 R \quad (2.23)$$

For this situation, the probability of plastic contact is

$$prob(z > d + w_c) = \int_{d+w_c}^{\infty} f(z) dz \quad (2.24)$$

The expected area A_p of plastic contacts will be

$$A_p(d) = h A_n \int_{d+w_c}^{\infty} A(z-d) f(z) dz \quad (2.25)$$

The contact load due to plastic contact will be the area multiplied by the mean contact pressure which is a function of the hardness H .

In Chang et al. [2.8, 2.9]'s model, the total contact area A_t and contact load W_t for elastic-plastic contact using the volume conservation model is calculated as

$$A_t(d) = A_e(d) + A_p(d) \quad (2.26)$$

where

$$A_e(d) = h A_n p R \int_d^{w_c} (z-d) f(z) dz \quad (2.27)$$

$$\text{and } A_p(d) = h A_n p R \int_{d+w_c}^{\infty} [2(z-d) - w_c] f(z) dz \quad (2.28)$$

Furthermore, the total contact load W_t is given by

$$W_t(d) = W_e(d) + W_p(d) \quad (2.29)$$

where W_e is the contact load due to elastic contact and W_p is the contact load due to plastic contact. These loads are calculated as [2.8]

$$W_e(d) = h A_n E \frac{4}{3} R^{\frac{1}{2}} \int_d^{d+w_c} (z-d)^{\frac{3}{2}} f(z) dz \quad (2.30)$$

and

$$W_p(d) = h A_n E p R K \frac{H}{E} \int_{d+w_c}^{\infty} [2(z-d) - w_c] f(z) dz \quad (2.31)$$

In Kogut and Etsion [2.10]'s model, the contact area and load are separated into four regions, i.e., $w/w_c < 1$; $1 \leq w/w_c < 6$; $6 \leq w/w_c < 110$; $110 \leq w/w_c$. They have used the finite element method to solve the problem of a deformable sphere in contact with a rigid flat. The contact evolves in four distinct zones. At $w/w_c < 1$, the contact is fully elastic; at $1 \leq w/w_c < 6$, the plastic region develops below the surface of the sphere and the entire contact interface is still elastic; at $6 \leq w/w_c < 110$, the contact area becomes elastic-plastic and after that, the contact is entirely plastic.

2.4 Adhesion and Friction Theory

2.4.1 Surface Forces

Surface forces are the forces that are present when two bodies are brought together. Three types of forces will be discussed in this section: electrostatic forces, molecular forces, and surface tension forces.

The electrostatic forces or Coulomb force [2.11] is the force that occurs due to electric fields between charges and dipoles. The free energy E for the Coulomb interaction between two charges q_1 and q_2 is given by

$$E(r) = -\frac{q_1 q_2}{4\pi\epsilon_0 e r} \quad (2.32)$$

where r is the distance between the two charges, ϵ_0 is the permittivity constant (8.85×10^{-12} C²/Nm²), and e is the relative permittivity for the medium across which the force acts.

The electrostatic force F is the derivative of the free energy E between the charges given by

$$F = -\frac{dE(r)}{dr} = \frac{q_1 q_2}{4\pi e \epsilon_0 r^2} \quad (2.33)$$

As indicated in the above equation, the magnitude of the electrostatic force between two point charges depends linearly on the magnitude of each charge and quadratically on the inverse of the distance between the charges [2.11].

Electrodynamic forces result from oscillating electrons. Electrons generate a fluctuating dipole field. This field interacts with nearby atoms, inducing dipole moments among them. The electrodynamic forces are also known as van der Waals forces or London forces [2.12].

The van der Waals force is often approximated by the Lennard-Jones potential model. The Lennard-Jones potential U is a function of distance between two atoms [2.13]

$$U(r) = -\frac{B}{r^6} + \frac{A}{r^{12}} \quad (2.34)$$

where A and B are constants, and r is the atomic distance. The dependence of the Lennard-Jones potential U on the distance r is shown in Fig. 2.8, indicating that the potential is repulsive at small distances and attractive at large distances.

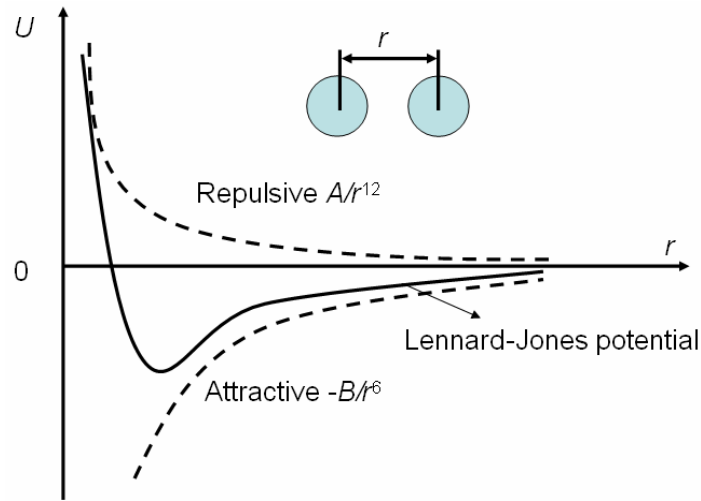


Figure 2.8 Lennard-Jones potential U vs. r

The potential y between an atom and an infinite plane at a distance d can be obtained by integration as

$$y(d) = -\frac{pAr}{6d^3} + \frac{pBr}{45d^9} \quad (2.35)$$

where r is the density of atoms contained in the plane and A and B are constants [2.14].

Bowden and Tabor [2.15] analyzed the effect of meniscus forces between two contacting bodies in the presence of a liquid film. For a smooth sphere against a smooth plate, an increase in adhesion force is generally observed when a thin film of liquid is introduced at the contact interface. They concluded that the increase in adhesion was due to meniscus formation in the contact area. A physical model of meniscus formation between a smooth sphere and a flat surface with a liquid bridge is shown in Fig. 2.9.

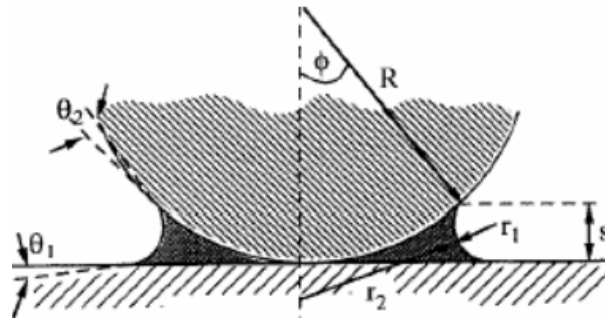


Figure 2.9 A schematic diagram of the meniscus formed at the contact interface of an idealized smooth sphere against a smooth surface [2.15]

The meniscus force F_m , for the case of contact between a smooth sphere and a smooth flat surface is given by

$$F_m = 2p Rg (\cos q_1 + \cos q_2) \quad (2.36)$$

where R is the radius of the sphere, g is the surface tension of the liquid, and q_1 and q_2 are the contact angles of the liquid on the flat surface and the spherical surface, respectively.

2.4.2 Adhesion Theory

The basic theoretical work on adhesion is concerned mainly with the contact between a sphere and a rigid flat surface. Early investigations on adhesion have been presented by Tabor and Winterton [2.16] who described experimental methods for the direct measurement of normal van der Waals forces for separations less than 10 nm and retarded van der Waals forces for separations greater than 20 nm. Here, retarded van der Waals forces are the atomic attraction forces when the atoms are at a “large” distance. For this situation, the interaction time for an instantaneous dipole and its neighbor may be comparable with the lifetime of the dipole. This leads to a smaller force of attraction. Typically, retarded van der Waals forces are an order of magnitude smaller than “normal” van der Waals forces. Bradley [2.17] presented a model for van der Waals forces for two surfaces. He assumed that both surfaces in contact are rigid, and that the intermolecular surface forces are confined to the area close to the contact point. In Bradley's model, elastic material deformations due to the effect of attractive interaction forces are neglected. A sketch defining the applicability of Bradley's model is shown in Fig. 2.10.

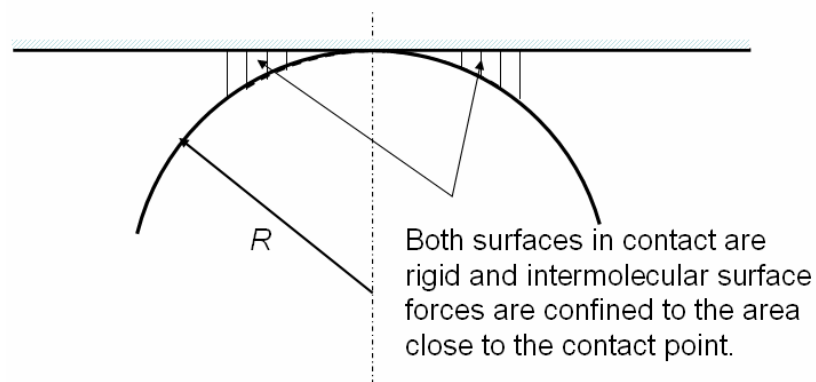


Figure 2.10 Sketch of the Bradley's model

A number of adhesion models for the contact between a sphere and a rigid flat surface have been proposed. The JKR (Johnson, Kendall and Roberts) model [2.18] accounts for the influence of intermolecular surface forces (van der Waals forces) within the contact zone. The JKR model assumes that the intermolecular surface forces are confined to the area of contact. The existence of intermolecular surface forces causes an increase in the deformation of the sphere. A schematic of the JKR model is shown in Fig. 2.11. The intermolecular surface forces cause increased deformation of the sphere, and thus, an increase in the contact area as compared to the Hertz contact model.

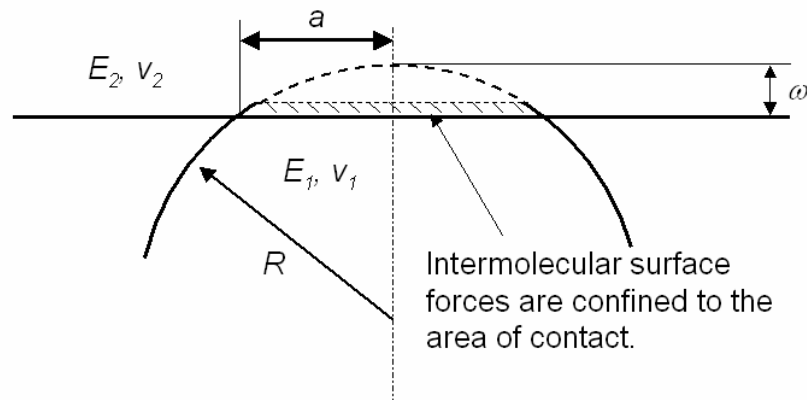


Figure 2.11 Sketch of the JKR (Johnson, Kendall and Roberts) model

In the JKR model, the relationship between the contact radius a and the penetration depth w is obtained as follows [2.18]

$$w = \frac{a^2}{R} - \sqrt{\frac{2p a \Delta g}{E}} \quad (2.37)$$

where a is the contact radius, R is the radius of the spherical asperity, E is the composite modulus and Δg is the work of adhesion, evaluated at the contact zone, given by

$$\Delta g = g_1 + g_2 - g_{12} \quad (2.38)$$

In Eqn. 2.38, g_1 and g_2 are the surface energies of the two contacting surfaces, and g_{12} is the energy of the contact interface after contact.

Comparing Eqn. 2.37 with Eqn. 2.8, we note that the first term in Eqn. 2.37 is identical to the Hertz solution and that the second term denotes the adhesion due to van der Waals forces.

The relation between the loading force W and the contact radius a for a fixed load is as follows

$$W = \frac{4}{3} \frac{Ea^3}{R} - \frac{3}{2} p \Delta g R \quad (2.39)$$

Another model, the so-called DMT (Derjagin, Muller, Toropov) model [2.19], assumes that the intermolecular surface forces (van der Waals forces) act along the contact area perimeter, as shown in Fig. 2.12. In this case, the intermolecular surface forces do not change the deformed profile of the sphere. The DMT model proposes a correction term to the relationship between the loading force and the penetration depth ($w = \frac{a^2}{R}$),

$$W = \frac{4}{3} \frac{Ea^3}{R} - 2p \Delta g R \quad (2.40)$$

where Dg is the work of adhesion evaluated at the contact zone.

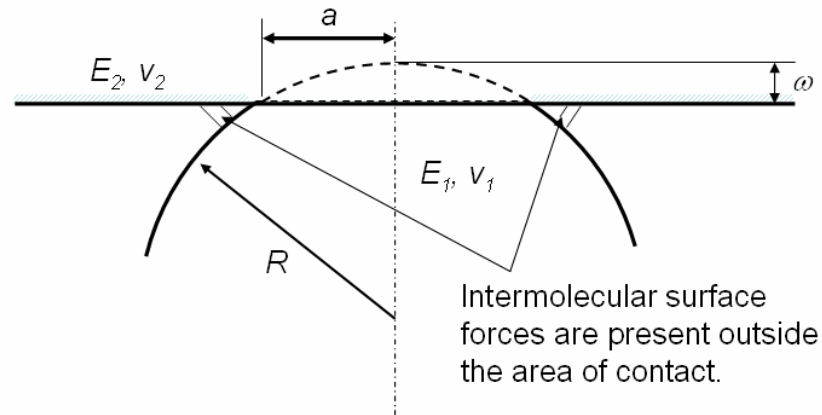


Figure 2.12 Sketch of the DMT (Derjagin, Muller, Toropov) model [2.19]

A still further model, the Maugis model [2.20], assumes that intermolecular surface forces act within a ring zone at the contact area border, as shown in Fig. 2.13.

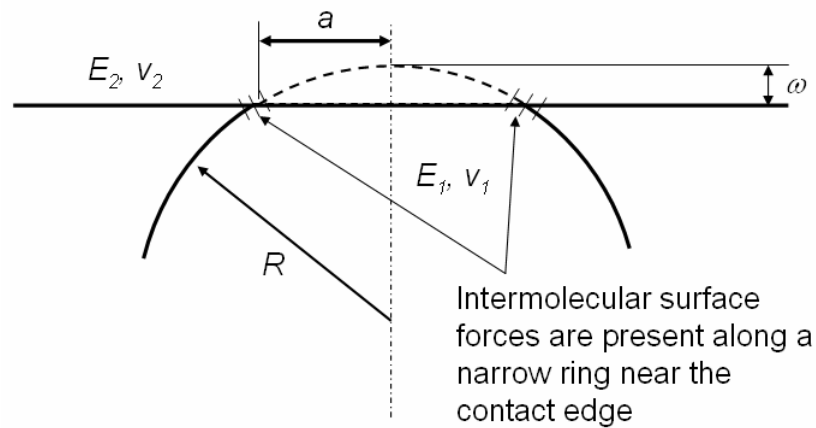


Figure 2.13 Sketch of the Maugis model [2.20]

Two non-dimensional parameters, l and A , are introduced.

$$l = \frac{s_0 (9R)^{\frac{1}{3}}}{(2p \Delta g E^2)^{\frac{1}{3}}} \text{ and } A = \frac{a \left(\frac{4}{3} E \right)^{\frac{1}{3}}}{(p \Delta g R^2)^{\frac{1}{3}}} \quad (2.41)$$

where s_0 denotes the tension acting over the contact circle.

In the Maugis model, the relationship between penetration depth w and contact radius a is given as follows

$$w = \left(\frac{9p^2 \Delta g^2 R}{16E^2} \right)^{\frac{1}{3}} \left(A^2 - \frac{4}{3} l A \sqrt{m^2 - 1} \right) \quad (2.42)$$

where m represents the ratio of the width of the annular region to the radius of the contact area. Again, the first term in Eqn. 2.42 denotes the Hertz solution.

The relation between the loading force W and the contact radius a is

$$W = p \Delta g R \left[A^3 - l A^2 \left(\sqrt{m^2 - 1} + m^2 \arctan \sqrt{m^2 - 1} \right) \right] \quad (2.43)$$

where Δg is the work of adhesion evaluated at the contact zone.

Johnson and Greenwood [2.21] introduced a so-called “adhesion map” to examine the validity of the various adhesion models. To examine the adhesion map, two parameters were introduced. The first parameter is the Tabor number m defined as

$$m = \left(\frac{R \Delta g^2}{E^2 e^3} \right)^{\frac{1}{3}} \quad (2.44)$$

where R is the composite radius, Δg is the work of adhesion, E is the composite Young's modulus and e is the effective range of surface forces. The second parameter is the transition parameter l which is related to m by $l = 1.16 m$ [2.21].

The adhesion map for the elastic behavior of bodies is shown in Fig. 2.14, reproduced from [2.21]

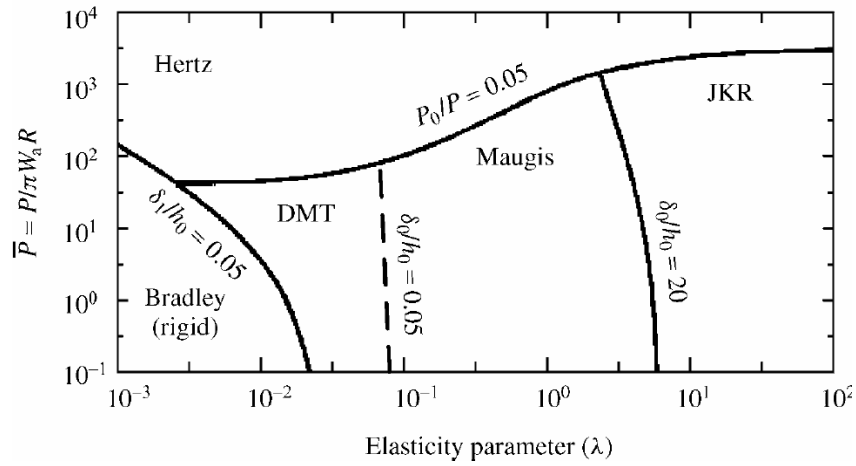


Figure 2.14 Adhesion map for the elastic behavior of bodies [2.21]

In Fig 14, $\bar{P}$ is the ratio between the total load and the adhesion contribution of the load. When adhesion is negligible, the deformation falls in the Hertz limit. d_1 is the elastic compression, and h_0 is the equilibrium distance. When $d_1 \ll h_0$, the bodies are rigid and follow the Bradley theory. d_0 is the deformation due to adhesion. When the adhesion is small, the DMT model is applicable. The JKR model, on the other hand, predicts the behavior of highly adhesive bodies. The Maugis model is valid in the intermediate region.

The above models all assume elastic deformation between contacting region. Several papers deal with plastic deformation. Pollock [2.22], and Pethica and Tabor [2.23] studied the effect of an oxide layer on adhesion. It is shown that surface forces are sufficient to cause local plastic deformation. Roy and Pollock [2.24] extended the work using the Greenwood and Williamson model [2.6] to multi-asperity rough surfaces of metallic

contacts. Chang, Etsion and Bogy [2.25] summarized the above studies and obtained the total adhesion force F_s for rough surfaces. They divided the integrations into two parts: one for contacting asperities and the other for non-contacting asperities. Based on the requirement of volume conservation, an “adjusted interference” w' was introduced

$$w' = w \left(2 - \frac{w_c}{w} \right) \quad (2.45)$$

The total adhesion force F_s was given by

$$F_s(d) = \frac{8p}{3} h A_n R \Delta g \left\{ \int_{-\infty}^d \left[\left(\frac{e}{e-w'} \right)^2 - 0.25 \left(\frac{e}{e-w'} \right)^8 \right] f(z) \right. \\ \left. + \int_d^{\infty} \int_0^{\infty} \frac{2}{Re} \left[\left(\frac{e}{Z(w',s)} \right)^3 - \left(\frac{e}{Z(w',s)} \right)^9 \right] f(z) s ds dz \right\} \quad (2.46)$$

where h is the areal density of asperities, A_n is the nominal contact area, e is the intermolecular distance, $s = \sqrt{r^2 - a^2}$ and r is the radial distance measured from the center of the contact area, and z is a function of w' and s .

2.4.3 Friction Models

The friction force between surfaces is a tangential force that resists relative motion between the two surfaces when both are pressed against each other with a normal force. Da Vinci [2.27] discovered that the friction force is proportional to the load, opposite to the direction of motion and independent of contact area. Amonton [2.28] believed that friction was predominately a result of the work done to lift one surface over the roughness of the other. In addition, he is credited for the following two laws of friction. The first law states

that friction is independent of the apparent area of contact. The second law is that the friction force is proportional to the normal load.

Coulomb [2.28] found that “strength due to friction is proportional to compressive force”, and “although for large bodies friction does not follow exactly this law”. Coulomb [2.28] published the work referring to Amontons [2.28]. The second law of friction is commonly recognized as the "Amontons-Coulomb Law". This classical approximation of the force of friction between two solid surfaces is described by the equation

$$F_f = \mu W \quad (2.47)$$

where μ is the coefficient of friction, W is the normal force between the contacting surfaces, and F_f is the frictional force between the two surfaces. Tabor [2.29] reviewed the understanding of friction in the early 80's. He summarized that three main factors are involved in friction: 1) the true contact area, 2) the strength of the interfacial bond, and 3) the deformation process involved when these interfacial bonds are broken during sliding. Clearly, contact and adhesion modeling are essential parts of any friction model.

Bowden and Tabor [2.30] assumed that friction is proportional to both the real contact area and a mean lateral force per unit area, the so-called shear strength

$$F_f = s A_r \quad (2.48)$$

where A_r is the real area of contact and s is the shear strength. They considered purely plastic deformation of the asperities until the contact area of each junction had grown large enough to carry its part of the normal load W . The contact area can be expressed as

$$A_r = \frac{W}{p_y} \quad (2.49)$$

where p_y is the yield pressure of the asperity, which is significantly smaller than the yield pressure of the bulk material due to the small size of asperities [2.30].

The assumption of totally plastic contacts is not realistic for most sliding situations, since the sliding surfaces would be completely damaged after a short time. Hence, elastic deformation must be considered. Archard [2.31] assumed a load dependent number of asperities. As the normal force increases, more asperities come into contact and the average area of each asperity contact grows. This in turn results in an increase in the friction force.

Following their work on multi-asperity contact between rough surfaces, Chang, Etsion and Bogy [2.32] developed a model for calculating the static friction force based on the surface roughness and the normal force. They assumed that when relative sliding of the surface begins, all contacting asperities must have failed. The tangential force corresponding to this limiting condition is the static friction force. They calculated the stress field at the point of failure inception for an asperity from the work of Hamilton [2.33]. The von Mises failure criterion was used as a measure when sliding will first occur. The static friction can then be obtained as [2.32]

$$F_f(d) = h A_n E \frac{4}{3} R^{\frac{1}{2}} \int_d^{d+w_c} (z-d)^{\frac{3}{2}} f\left(\frac{z-d}{w_c}, n\right) f(z) dz \quad (2.50)$$

where f is a function depending on the failure inception location.

2.5 Lubrication

Lubrication occurs when surfaces move relative to each other in the presence of a lubricant film. Two basic lubrication regions exist: the hydrodynamic lubrication region

and the boundary lubrication region [2.34]. These regions are characterized by the type of lubricant/surface interactions and the resulting friction. One way of separating these regimes is via the use of the Stribeck diagram [2.35], as shown in Fig. 2.15.

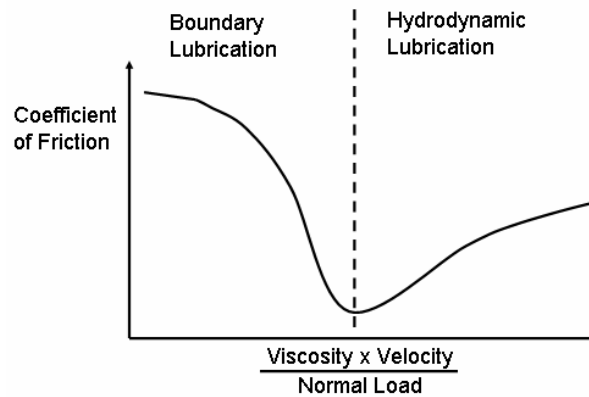


Figure 2.15 Stribeck diagram: coefficient of friction as a function of viscosity-velocity-load parameter [2.35]

As indicated in Fig. 2.15, the coefficient of friction is a function of the viscosity of the lubricating fluid, the sliding velocity, and the applied normal load. The two lubrication regimes are a function of the value of the non-dimensional parameter formed by viscosity, velocity and normal load. In Fig. 2.15, the region on the left is the boundary lubrication region, and the region on the right is the hydrodynamic lubrication region [2.35].

Under conditions of hydrodynamic lubrication, the sliding surfaces are separated by a continuous lubricant film, as shown in Fig. 2.16, and contacts between asperities are absent. The applied load is carried by pressure generated within the fluid. The frictional

resistance to motion arises entirely from the shearing of the viscous fluid. Hydrodynamic lubrication occurs generally at high velocity and low normal load.

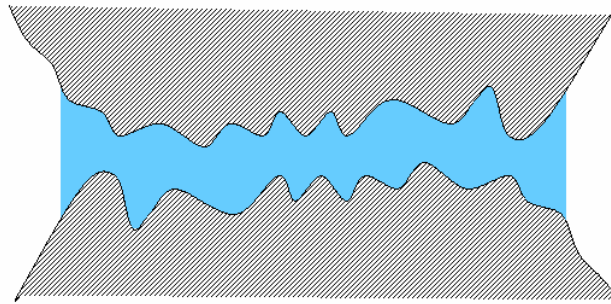


Figure 2.16 Schematic view of hydrodynamic lubrication [2.1]

Boundary lubrication occurs when asperities of the sliding surfaces come into contact with each other. This normally occurs at low velocity or high normal loads. A schematic view of boundary lubrication is shown in Fig. 2.17. As indicated in this figure, direct contact occurs between the sliding surface asperities. Boundary lubrication is often characterized by high friction and wear due to intimate contact between the two sliding surfaces.

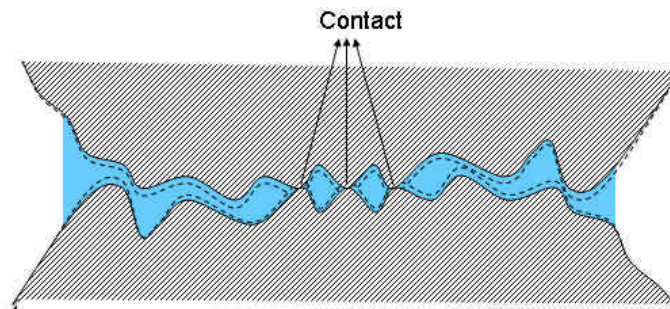


Figure 2.17 Schematic view of boundary lubrication [2.1]

Another state of lubrication, called “elasto-hydrodynamic lubrication” is often encountered. Elasto-hydrodynamic lubrication exhibits characteristics of both boundary and hydrodynamic lubrication. Elasto-hydrodynamic lubrication is characterized by high local pressures in the lubrication film, i.e., pressures that are significantly higher than encountered normally in hydrodynamic lubrication. These high pressures cause an elastic deformation of the sliding surfaces. Load is shared by contacting asperities as well as hydrodynamic pressure generated between the two surfaces. A schematic view of elasto-hydrodynamic lubrication is shown in Fig. 2.18. As indicated in this figure, the original body is deformed elastically due to the force transmitted through the lubricant.

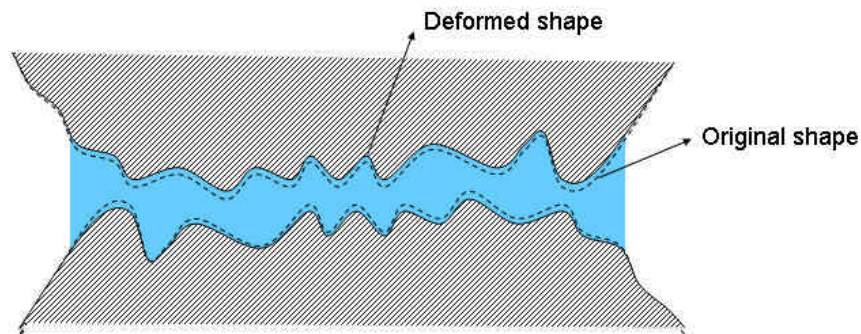


Figure 2.18 Schematic view of elasto-hydrodynamic lubrication (the dotted line indicates the original shape of the body) [2.36]

2.5.1 Boundary Lubrication

There are several situations when boundary lubrication occurs at the head/disk interface. For contact-start-stop hard disk drives, boundary lubrication occurs during start/stop when the slider is sliding on the disk surface. When the slider is flying over the

rotating disk surface, surface asperities of slider and disk surfaces often come into intermittent contact. This is another situation where boundary lubrication occurs at the head/disk interface.

Bowden and Tabor [2.37] suggested that the normal load is supported by the intimate contact of the surface asperities and to some extent, by the pressure generated by distorting the lubricant layer. In situations when boundary lubrication exist for rough surfaces, the loading force W between two surfaces must be separated into two portions: one portion is due to asperity contact forces and the other portion is due to hydrodynamic fluid pressure forces. The total load force can then be calculated from

$$W = \sum_{\text{asperities}} W_i + \int_{\text{total area} - \text{real contact area}} p_L dA \quad (2.51)$$

where W_i is the contact force at the i -th asperity and p_L is the hydrodynamic pressure. A schematic which explains boundary lubrication is shown in Fig. 2.19.

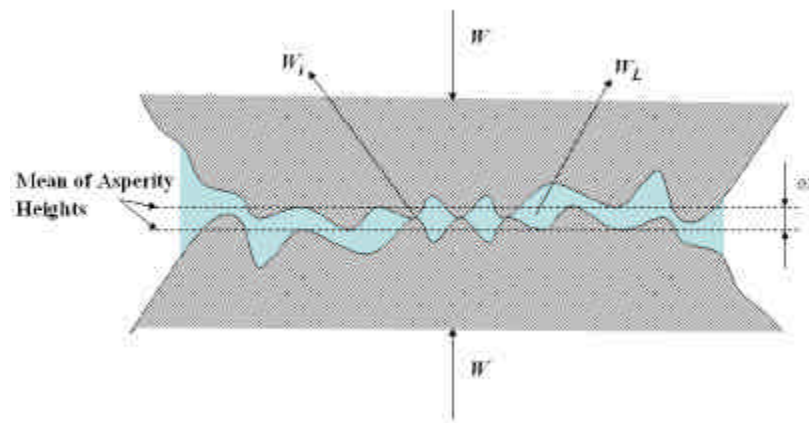
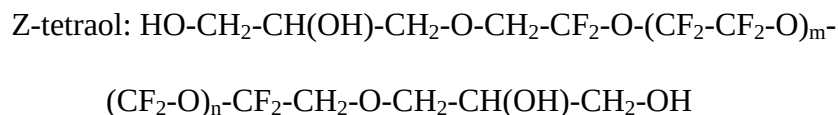


Figure 2.19 Overview of boundary lubrication for rough surfaces

Persson [2.38] investigated a boundary lubrication model for an elastic block sliding on a substrate with a layer of lubricat molecules. Ziberman et al. [2.39] studied the

squeeze-out dynamics in boundary lubrication. They investigated the dynamics of expulsion of the liquid-like mono-layers of molecules between two surfaces and solved the two-dimensional Navier-Stokes equation combined with kinetic Monte Carlo simulations. Thompson and Robbins [2.40, 2.41] used a Lennard-Jones potential to model the interaction between fluid molecules and discrete wall atoms.

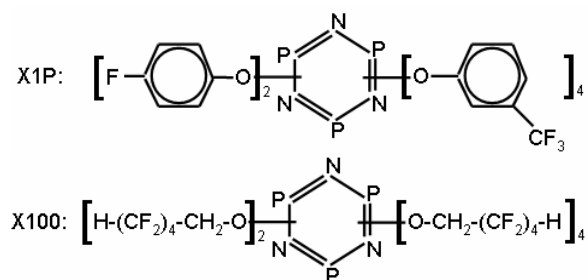
Magnetic recording disks are lubricated with a very thin lubricant film for wear and corrosion protection. Lubricants for the head/disk interface must have good load carrying capability, mobility, low surface energy and good thermal stability. Commonly used lubricants are perfluoropolyether (PFPE) type lubricants. In hard disk drives, perfluoropolyether (PFPE) such as Z-DOL and Z-tetraol, are often used [2.42]. The chemical structures of Z-DOL and Z-tetraol is given by



where the ratio of m/n is about $2/3$ for both Z-DOL and Z-tetraol.

As can be seen from the chemical structure of the above lubricant, Z-DOL has two polar endgroups (-OH) while Z-tetraol has 4 polar endgroups (-OH). Thus, Z-DOL has a lower bonding strength to carbon overcoat than Z-tetraol, as discussed by Zhao and Bhushan [2.43]. However, PFPE type lubricants are known to have high stiction and they degrade catalytically when exposed to alumina [2.44]. In order to improve the performance of the head/disk interface, additives have been investigated. Zhao et al. [2.45] studied a number of additives for Z-DOL and found that additives such as X-1P and X-100 improved

the stiction performance of the head/disk interface. The chemical structure of X-1P and X-100 is given by



Yang et al. [2.46] and Kang et al. [2.47] found that X-1P improved the degradation performance. Z-DOL and Z-tetraol have been widely used with X-1P additive in hard disk drives.

Most of the lubrication studies for the head/disk interface are concerned with lubricant motion and lubricant transfer from disk to slider. Mate [2.48] indicated that there were four types of factors that to drive the lubricant flow: 1) disjoining pressure gradient, 2) capillary pressure gradient, 3) centrifugal force due to rotation of the disk, 4) shear force due to the air flow over the disk surface.

The disjoining pressure is the pressure acting in a molecularly-thin fluid film that is in excess of the pressure acting in a bulk fluid of the same material [2.49]. The disjoining pressure arises from the interaction of a molecularly-thin fluid film with a solid surface. There are multiple components to the disjoining pressure due to the various interactions that can occur between the fluid molecules and the surface. For the PFPE type of lubricants, two dominant components exist: the polar and the dispersive component. The dispersive component is due to the van der Waals interaction forces operating at the interface, and the

polar (or structural) component is due to effects such as specific hydrogen-bonding or acid-base interactions. Mathematically, the disjoining pressure is defined as

$$\Pi(h) = -\frac{dG}{dh} \quad (2.52)$$

where G is the Gibbs free energy, which is related to the surface energy γ of the lubricant film, i.e.,

$$G = 2\gamma, \quad (2.53)$$

and h is the thickness of the lubricant film. As indicated in Eqn. 2.52, the magnitude of the disjoining pressure is dependent on film thickness. It is also worth noting that the disjoining pressure is not an isotropic pressure, but rather contributes to the normal component of the pressure tensor.

Deoras et al. [2.50] used Fick's second law of diffusivity [2.51] to calculate the diffusivity constant (diffusion coefficient in Fick's Law) of a lubricant as it spreads at the boundary of a step in lubricant thickness. They observed that diffusivity was a strong function of the functional end group and the molecular structure of the lubricant.

Karis et al. [2.52] studied spin-off behavior of lubricant due to the rotation of the disk. They found that the spin-off removal rate was proportional to the square of the lubricant film thickness, and inversely proportional to the viscosity of the lubricant. A similar result was observed by Tani and Matsumoto [2.53].

When the slider flies over a rotating disk, shear forces in the air bearing can result in displacement of lubricant as well. Corrugation-like variations of the lubricant thickness in the slider "wear" track have been reported by Ma et al. [2.54]. Lubricant moguls have been observed and studied by Dai et al. [2.55]. Mate and Marchon [2.56] studied the shear effect

of the air flow to lubricant motion and demonstrated that lubricant ripples originate from air shear effects. Scarpulla et al. [2.57] proposed a shear mobility parameter to describe the ability of molecularly-thin films of lubricants moving by an applied shear force. They discovered that the effective viscosity and thickness of the mobile layer of the lubricant determines the motion of molecularly-thin films of lubricants.

Lubricant transfer from disk to slider has also been studied by a number of researchers. Marchon et al. [2.58] proposed a model for lubricant flow from disk to slider at steady-state equilibrium. Their model takes into account inflow and outflow due to condensation and evaporation, and lubricant outflow due to air shear. They found that the accumulation rate of lubricant on a slider decreases exponentially as the molecular weight of the lubricant increases. Ma et al. [2.59] also proposed a model for lubricant transfer from disk to slider. They found that reducing the size of the slider trailing edge pad helps to reduce the amount of lubricant transfer from disk to slider.

Stiction is another important concern for the head/disk interface with thin boundary lubricants. For contact-start-stop technology, a laser textured landing zone was created on the inner perimeter of a disk to “park” the slider during power-off [2.60, 2.61]. Gui and Marchon [2.62] studied head/disk stiction for contact-start-stop technology. They plotted head/disk stiction as a function of lubricant film thickness and texture height (Fig. 2.20).

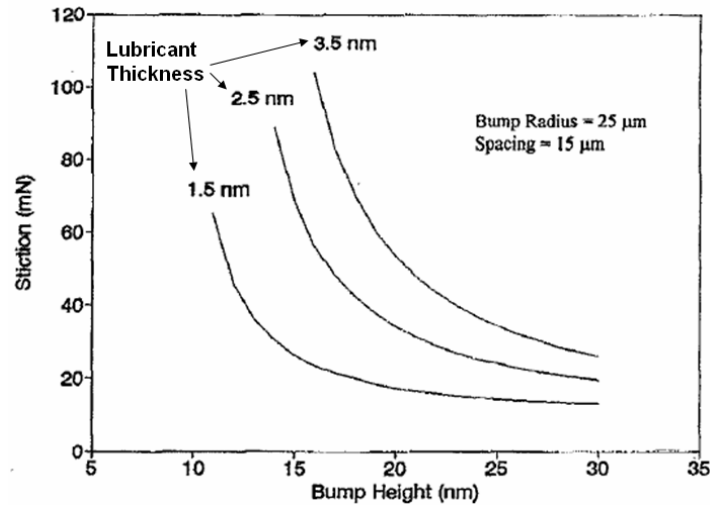


Figure 2.20 Stiction as a function of lubricant film thickness and texture height [2.52]

As shown in Fig. 2.20, stiction increases with increasing lubricant thickness and decreasing bump height of the surface texture [2.62].

To reduce stiction between the head/disk interface during flying, Zhou et al. [2.63] investigated texturing of the slider air bearing surface. Texturing has been proven to be an effective way to reduce the stiction force.

As the areal density increases, an improved understanding of slider dynamics and slider/lubricant interaction is needed. This topic will be addressed in detail in chapter 6 of this thesis.

2.5.2 Hydrodynamic Lubrication

During the operation of a magnetic hard disk drive, the slider is separated from the disk by a viscous air film, which lubricates the interface between the slider and the disk. The pressure distribution in the air bearing is governed by the Reynolds equation [2.64],

which is derived from the Navier-Stokes equation and the continuity equation with the following assumptions

- 1) The flow is laminar;
- 2) The fluid film has small film thickness compared to the other dimensions;
- 3) The no-slip boundary condition is valid at the wall;
- 4) The viscosity of the fluid is Newtonian;
- 5) The inertial and body force of the lubricant film are neglected.

The general form of the compressible Reynolds equation for application to air bearings is

$$\frac{\partial}{\partial x} [ph^3 \frac{\partial p}{\partial x}] + \frac{\partial}{\partial y} [ph^3 \frac{\partial p}{\partial y}] = 6m \frac{\partial (Uph)}{\partial x} + 6m \frac{\partial (Vph)}{\partial y} + 12m \frac{\partial (ph)}{\partial t} \quad (2.54)$$

where p is the pressure, h is the local slider-disk separation, m is the viscosity of the air, U and V are the slider velocities in the x and y directions, respectively, and t is the time.

By defining the non-dimensional parameters

$$X = \frac{x}{L}, Y = \frac{y}{L}, H = \frac{h}{h_0}, P = \frac{p}{p_a}, T = \omega t \quad (2.55)$$

where L is the length of the slider, h_0 is the minimum flying height, p_a is the ambient pressure and ω is an appropriate angular frequency, the non-dimensional form of the Reynolds equation can be written as

$$\frac{\partial}{\partial X} [PH^3 \frac{\partial P}{\partial X}] + \frac{\partial}{\partial Y} [PH^3 \frac{\partial P}{\partial Y}] = \frac{\partial}{\partial X} \Lambda_x PH + \frac{\partial}{\partial Y} \Lambda_y PH + s \frac{\partial}{\partial T} [PH] \quad (2.56)$$

where Λ_x and Λ_y are the bearing numbers in the x and y directions, respectively, defined by

$$\Lambda_x = \frac{6mUL}{p_a h_0^2}, \Lambda_y = \frac{6mVL}{p_a h_0^2} \quad (2.57)$$

and s is the squeeze number defined by

$$s = \frac{12m\omega L^2}{p_a h_0^2} \quad (2.58)$$

2.5.3 Rarefaction effects

In the derivation of the compressible Reynolds equation (Eqn. 2.56), no-slip boundary conditions are used. Since the minimum spacing between slider and disk is smaller than the molecular mean free path of air (approximately 67 nm for air [2.65]), the no-slip boundary condition at the wall cannot be satisfied, i.e., the flow cannot be treated as continuum flow and rarefaction effects must be considered.

The Knudsen number ($K_n = l / h_0$), defined as the function of the mean free path l of the molecules and the minimum fluid film thickness h_0 , is used to determine whether rarefaction effects are important in a given flow. The classification of flows based on the Knudsen number K_n is shown below

$$\left\{ \begin{array}{ll} K_n < 0.01 & \text{continuum flow} \\ 0.01 < K_n < 3 & \text{slip flow} \\ K_n > 3 \ \& \ \frac{K_n}{R_e^{0.5}} < 10 & \text{transition flow} \\ K_n > 3 \ \& \ \frac{K_n}{R_e^{0.5}} > 10 & \text{free molecular flow} \end{array} \right. \quad (2.59)$$

where R_e is the Reynolds number

In a typical head/disk interface, the minimum flying height is on the order of 10 nm, while the mean free path of air is approximately 67 nm. Thus, the Knudsen number is 6.7 and rarefaction effects have to be considered. Various models have been proposed to account for rarefaction effects. All models introduce a correction term Q in the non-dimensional Reynolds equation as follows

$$\frac{\partial}{\partial X}[QPH^3 \frac{\partial P}{\partial X}] + \frac{\partial}{\partial Y}[QPH^3 \frac{\partial P}{\partial Y}] = \frac{\partial}{\partial X}\Lambda_x PH + \frac{\partial}{\partial Y}\Lambda_y PH + s \frac{\partial}{\partial T}[PH] \quad (2.60)$$

Burgdorfer [2.66] proposed a first order slip correction ($Q = 1 + 6 \frac{K_n}{PH}$) based on the Taylor expansion of the mean velocity field. Hsia and Domoto [2.67] introduced a second order slip correction ($Q = 1 + 6 \frac{K_n}{PH} + 6[\frac{K_n}{PH}]^2$). Fukui and Kaneko [2.68] derived a molecular gas lubrication model ($Q = 1 + f\left(\frac{K_n}{PH}\right)$) based on the linearized Boltzmann equation. A recent particle-based simulation study (Alexander *et al.* [2.69]) has been found to be in good agreement with the Fukui and Kaneko model for slider-disk spacing as small as 10 nm.

In this dissertation, the rarefaction effect is taken into account using the Fukui and Kaneko model [2.68]. This approach has shown to give good agreement between experimental data and numerical calculations for slider flying heights in the sub-10 nm region [2.69].

2.6 Wear

Wear is the loss of material from one or both of the contacting surfaces when subjected to relative motion. Burwell [2.70] defined different wear mechanisms and pointed out four fundamental wear modes: adhesive wear, abrasive wear, fatigue wear and corrosive wear.

Both adhesive wear and abrasive wear involve plastic flow of the contacting surfaces. If the contacting surfaces have enough adhesive bonding strength to resist relative sliding, large plastic flow caused by dislocation occurs in the contact region under compression and shearing. As a result, cracks initiate and propagate in the contact interface. This type of wear is called adhesive wear. A schematic view of adhesive wear is shown in Fig. 2.21a.

Abrasive wear often involves plastic flow and brittle fracture. If interlocking of asperities occurs, the harder surface tends to penetrate into the softer surface. This can cause fracture and generate particles between the contacting surfaces. This type of wear is called abrasive wear. Another type of abrasive wear occurs if hard abrasive particles are present at an interface. A schematic view of adhesive wear is shown in Fig. 2.21b.

When the contacting surfaces in relative motion experience many repeated friction cycles, plastic flow and fracture tends to occur at the contacting surfaces. This type of wear is called fatigue wear (Fig. 2.21c). If chemical reactions occur at the interface of contacting surface in relative motion, removal of surface layers may occur. The resultant wear is called corrosive wear (Fig. 2.21d). At the head/disk interface, surface characteristics may change due to tribochemical reactions. This is commonly known as “tribochemical wear” [2.71].

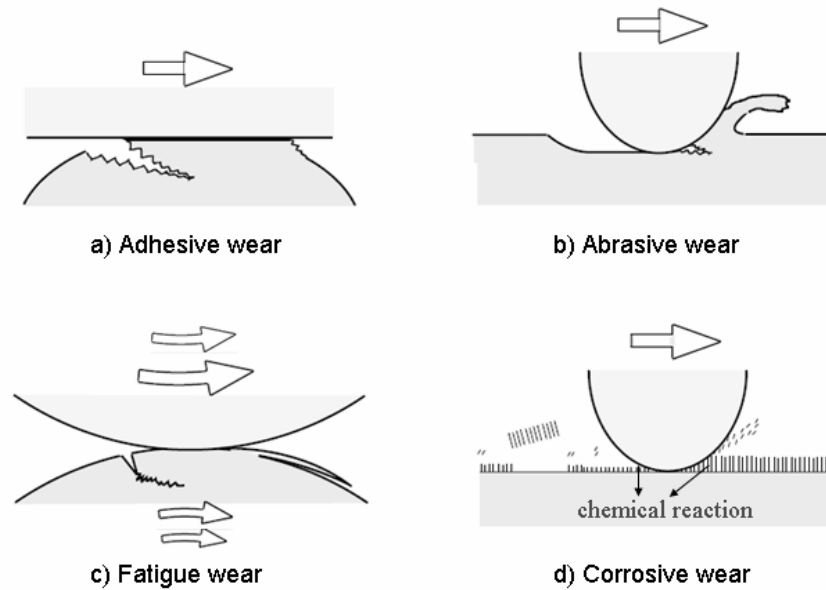


Figure 2.21 Schematic view of four primary wear modes [2.72]

Adhesive and abrasive wear are the most common failure modes of the head/disk interface. A quantitative measure of adhesive wear and abrasive wear will be discussed in the next paragraph.

2.6.1 Adhesive Wear

Holm [2.73] and Archard [2.74] have developed a model of adhesive wear based on the following assumptions

- 1) The surfaces in contact contain spherical asperities of equal radius of curvature.
- 2) The duration of contact of two asperities follows the following sequence. At zero time, the contact areas form a fully established circular contact of radius a , as shown in Fig. 2.22a. After sliding begins, the contact area is reduced as shown in Fig. 2.22b. After sliding at a distance of $2a$, the contact area is reduced to zero (Fig. 2.22c).

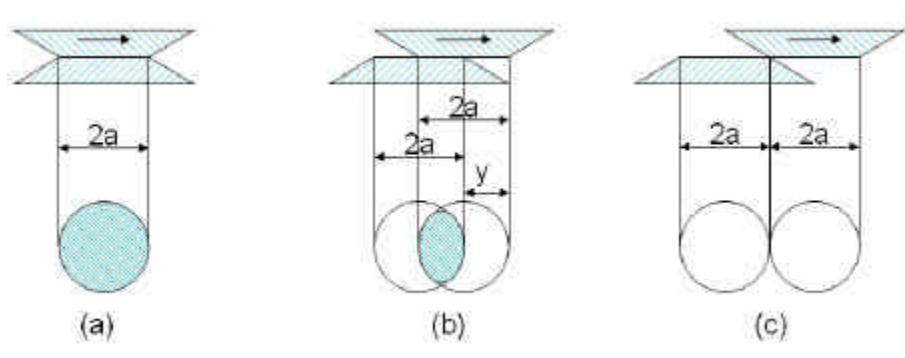


Figure 2.22 Schematic view of single asperity contact in sliding surfaces. a) Maximum contact area of radius a . b) Contact area after sliding at a distance y . c) After sliding at a distance $2a$, the contact area is reduced to zero [2.74]

3) Wear particles are assumed to be hemispherical and have the same radius as the contact area a , i.e.,

$$dV = \frac{2pa^3}{3} \quad (2.61)$$

4) Only a proportion of contact spheres will produce wear particles. The proportional factor is denoted by k .

Based on the above assumptions, the average volume of material dQ worn away per unit sliding distance for one pair of asperities through a distance $2a$ is given by

$$dQ = \frac{kdV}{2a} = \frac{kp a^2}{3} \quad (2.62)$$

Thus, the overall wear rate from all asperity contacts is the summation of the whole area of contact.

$$Q = \sum dQ = \frac{k}{3} \sum p a^2 \quad (2.63)$$

Then, the total normal load supported by the whole area of contact can be calculated as

$$W = P \sum p a^2 \quad (2.64)$$

where P is the yield pressure for the plastically deforming asperity.

By assuming $P = H$, where H is the indentation hardness, and $k = k/3$, and substituting Eqn. 64 into Eqn. 63, we obtain the Archard's wear equation [2.74].

$$Q = \frac{kW}{H} \quad (2.65)$$

As indicated in this equation, the wear volume per unit sliding distance, Q , is related to the normal load W , the indentation hardness H , and the dimensionless wear coefficient k .

Typical values of the wear coefficient k are shown below [2.1]

Table 2.1 Typical value of wear coefficient [2.1]

Material	Wear coefficient k
mild steel (against mild steel)	7×10^{-3}
α/β brass	6×10^{-4}
PTFE	2.5×10^{-5}
α brass	1.7×10^{-4}
PMMA	7×10^{-6}
copper–beryllium	3.7×10^{-5}
hard tool steel	1.3×10^{-4}
Stellite 1	5.5×10^{-5}
stainless steel (ferritic)	1.7×10^{-5}
polyethylene	1.3×10^{-7}

2.6.2 Abrasive Wear

When two surfaces in contact are sliding relative to each other, the harder surface tends to dig into the softer surface, plowing a series of grooves. This type of wear is called abrasive wear. Rabinowicz [2.75] derived a simplified quantitative expression for abrasive wear. In his model he assumed that the asperities on the harder surface (or the abrasive particle) are conical, as shown in Fig. 2.23.

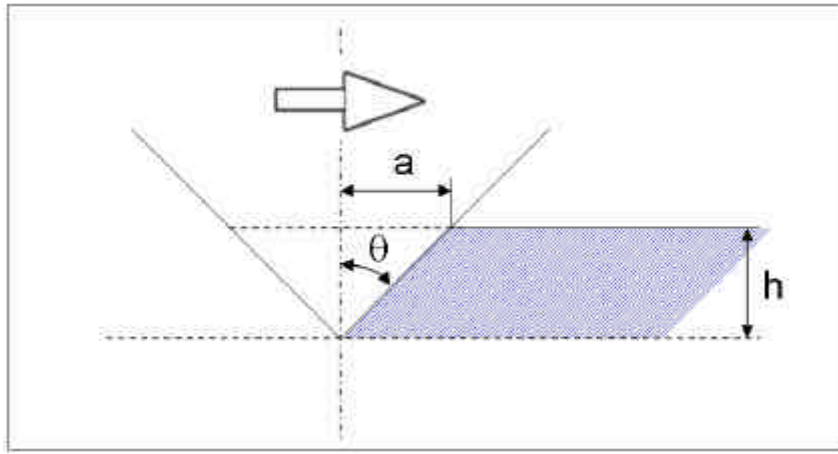


Figure 2.23 Schematic view of a simplified abrasive wear model [2.75]

When the asperity is moving, only contact of the front surface is considered. We assume that a pressure p acts over the area of contact, resulting in penetration of the conical asperity into the softer material (Fig. 2.23). The normal load dW supported by this asperity is

$$dW = p \frac{p a^2}{2} \quad (2.66)$$

If the angle q is defined as shown in Fig. 2.23, the geometrical relationship between the penetration depth h and the radius a is $h = a \tan q$. The volume of material displaced by the asperity per unit sliding distance is

$$dQ = h \, da = h a^2 \tan q \quad (2.67)$$

where h denotes a fraction of material removed as wear debris.

Combining Eqn. 66 and Eqn. 67, we obtain

$$dQ = \frac{2dW}{pP} h \tan q \quad (2.68)$$

If one accounts for the contributions of all the asperities, introduces a constant

$K = \frac{2h \tan q}{p}$ and assumes that $P = H$, where H is the indentation hardness of the softer

material, then the total volume removed per unit sliding distance is

$$Q = \frac{KW}{H} \quad (2.69)$$

As indicated by Eqn. 2.69, the wear volume Q per unit sliding distance is related to the normal load W , the indentation hardness H , and the dimensionless wear coefficient K .

Equation 2.69 derived by Rabinowicz [2.75] for abrasive wear is identical to Archard's equation [2.74] for adhesive wear.

2.6.3 Tribochemical Wear

At the head/disk interface, surface characteristics may change due to tribochemical reactions. This is commonly known as "tribochemical wear" [2.71]. Molecules inside the atmosphere react with the surface, resulting in a change of tribological properties of the

surface. Marchon et al. [2.76] observed that friction increases during a sliding test between an oxide-based ceramic slider and an amorphous carbon surface. They concluded that an increase of friction was primarily due to tribochemical wear of the surface where carbon atoms of the surface react with oxygen molecules in the atmosphere. Yang et al. [2.77] performed similar tests in an environment of dry nitrogen and observed a dramatic decrease in the coefficient of friction. Their observation confirms the discussion by Marchon et al. [2.76]. The tribochemical wear was recently reviewed by Talke [2.71] and he concluded that a thin layer of lubricant film can be considered as a high-energy barrier for oxygen molecules and thereby reducing the occurrence of tribochemical wear in favor of abrasive and adhesive wear.

Another type of tribochemical wear of the head/disk interface can occur during contact sliding tests. After contact sliding tests, “wear” tracks can occur on disk surfaces. Sinha et al. [2.78] observed that the lubricant on the “wear” track was bonded to the substrate and did not move back to a low-stress situation after sliding. They suggested chemical changes of the bonded PFPE molecules by the processes of thermal degradation or molecular scission. Bhatia et al. [2.79] analyzed the “wear” track with X-ray spectroscopy. They discovered that the formation of carboxylic bonds on the “wear” track due to contact sliding tests. Further study involving actual disk drive slider may be necessary to determine the effects of contact sliding on lubricant and its implications on the performance of the disk drive.

2.7 Summary

In this chapter, an overview of the principles of tribology has been presented with the view towards applications for the head/disk interface. Contact mechanics has been discussed along with boundary and hydrodynamic lubrication. The Reynolds equation was introduced for the analysis of slider air bearings. In addition, various wear mechanisms have been discussed and reviewed.

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Chapter 3 Mechanical Scaling Laws for Hard Disk Drives

Looking at past generations of disk drives, one sees that the size of disk drives has become smaller and smaller. The first disk drive, the IBM RAMAC, featured disks with 24 inch (“600 mm”) in diameter. The next generation of disk drives used disks of 14 inch (“355 mm”) in diameter. Thereafter, 8.5 inch (“215 mm”) and 5.5 inch (“140 mm”) drives were produced. Today, 3.5 inch (“95 mm”) disk drives are used for desktop computers and 2.5 inch (“65 mm”) disk drives are used for laptop computers. In addition, 1.8 inch (“45 mm”) disk drives are widely used in consumer electronics and disk drives as small as 0.85 inch (“20 mm”) and 1.0 inch (“27.4 mm”) are manufactured. In looking at disk drives of decreasing size, the following questions arise from a mechanical point of view: 1) What is the effect of decreasing size on the dynamics of the disk drive and the head/disk interface, 2) Are there any guide exists in designing a new disk drive with smaller diameter based on the characteristics of a larger disk drive already in production, and 3) Are there any scaling laws similar to the magnetic scaling laws to help in the design of smaller disk drives?

To answer these questions, we will investigate the influence of size reduction of individual disk drive components on the static and dynamic performance of a disk drive, and discuss mechanical “scaling laws” useful for the design of future miniaturized disk drives.

3.1 Introduction

The growing demand for data storage applications is accelerating the development of small form-factor hard disk drives, and, at the same time, increasing the need for higher areal density disk drives. The areal density of a recording disk is the product of linear density (number of recorded bits per unit length along the circumference of the disk) and track density (number of tracks per unit length in the radial direction). As reviewed in chapter 1, a typical hard disk drive consists of a number of mechanical components such as the disks, the heads, the suspensions, the arms and the actuators. The heads and suspensions are typically integrated in a hard disk drive in the so-called “head/gimbal assembly”. The mechanical stiffness and natural frequency of these components are important for the operation and performance of a hard disk drive.

In-plane and out-of-plane vibrations of suspensions pose a serious limitation for further increases in the track density of hard disk drives. In a recent study by Tarnopolsky [3.1], a simplified triangular shape of a suspension was used and the maximum vibration amplitude at the tip of the suspension was investigated. The results showed that a reduction in the dimensions of the suspension spring reduces vibrations of the actuator/suspension assembly. Liu et al. [3.2] have investigated suspension dynamics during track seeking and the role of vibrations in the failure of the head/disk interface. Kilian et al. [3.3] have optimized the mechanical design of suspensions with respect to minimizing the amplitude of in-plane and out-of-plane vibrations. Miu and Bhat [3.4] have studied minimization of vibrations of the head/gimbal assembly during dynamic accessing using control schemes.

Disk vibrations are likely to cause track mis-registration and thus limit the maximum achievable number of tracks per inch (TPI). McAllister [3.6] described measurement methods for the characterization of disk vibrations. Srikrishna and Kasetty [3.7] predicted track mis-registration due to vibration of disk substrate materials.

Although the above studies deal with the dynamics of head/gimbal assemblies and vibrations of disks, they did not specifically address the design region of current disk drives, i.e., form factors in the range from 1.0 inch (“27.4 mm”) to 3.5 inch (“95 mm”). Many mechanical design aspects of small form factor disk drives are influenced by the size of the individual components, and special attention must be given to the dynamic performance of small form factor drives.

In this study, we will investigate scaling laws for the stiffness and natural frequency of suspensions and disks. The results will be compared with analytical solutions and finite element calculations.

3.2 A Simplified Tapered Beam Model for Suspensions

The suspension of a hard disk drive can be modeled as a tapered beam clamped at one end, as shown in Fig. 3.1. The length and thickness of the beam is denoted by l and t , respectively. The width of the beam at the clamped end is denoted by w_0 and at the free end is denoted by w_1 .

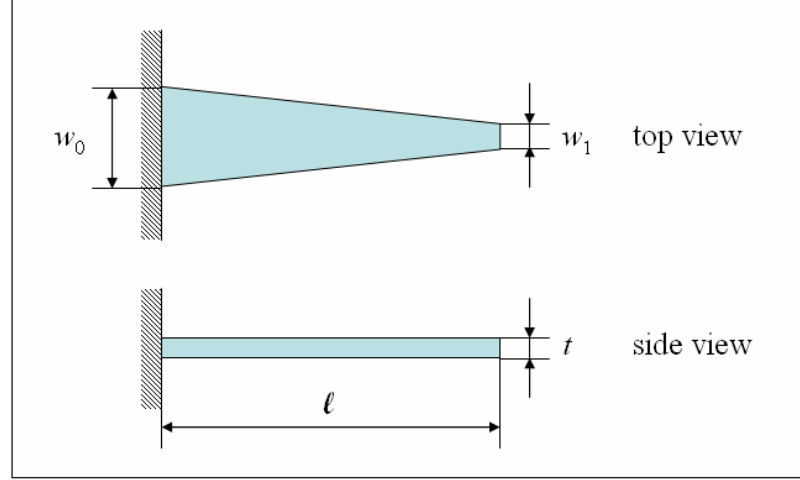


Figure 3.1 A simplified tapered beam model for suspension

For a linearly tapered beam, the taper ratio can be defined as

$$a = \frac{w_1}{w_0} \quad (3.1)$$

Timoshenko et al. [3.8] presented the stiffness of such a tapered beam as

$$k = \frac{c_1 EI_0}{l^3} \quad (3.2)$$

and the first resonance frequency as

$$f = \frac{c_2}{2pl^2} \sqrt{\frac{EI_0}{r A_0}} \quad (3.3)$$

where E is the Young's modulus of the beam, r is the density, c_1 and c_2 are constants related the taper ratio. In addition, A_0 and I_0 are the area and the area moment of inertia of the beam at the clamped end, respectively, given by

$$A_0 = w_0 t \text{ and } I_0 = \frac{w_0 t^3}{12} \quad (3.4)$$

Thus, Eqn. 3.2 and 3.3 can be rewritten as

$$k = \frac{c_1 E w_0 t^3}{12 l^3} \quad (3.5)$$

and

$$f = \frac{c_2}{2p} \sqrt{\frac{E}{12 r}} \frac{t}{l^2} \quad (3.6)$$

The values for c_1 and c_2 are listed in [3.8] for a change of the taper ratio a from 0 to 1.

Hoffmann et al. [3.9] approximated c_1 and c_2 as a function of the taper ratio a , i.e.,

$$c_1 = \left(\frac{1}{(1-a)^2} - \frac{1}{(1-a)^3} \right) [1 + a \ln(a) - a] + \frac{1}{2(1-a)} \quad (3.7)$$

and

$$c_2 = 1.14 - 1.95a + 3.35a^2 - 3.08a^3 + 1.10a^4 \quad (3.8)$$

To investigate the effect of reduced beam dimensions, it is useful to consider a standard and a miniaturized system where all dimensions are reduced by the same scale factor, say $s = 0.5$. The dimensions of the scaled suspension are $l_{\min} = 0.50l$, $w_{0\min} = 0.50w_0$, $w_{1\min} = 0.50w_1$, and $t_{\min} = 0.50t$. Using the scaled values in Eqns. 3.5 and 3.6, we observe that the stiffness of the miniaturized suspension is one half that of the full-scale suspension, while the natural frequency is twice that of the full scale suspension. Thus, down-scaling the dimension of a suspension increases the resonance frequency of a suspension and reduces its stiffness. The increased resonance frequency improves the dynamics of a suspension during track following and track-seeking and, thus, improves the recording performance of a hard disk drive.

Since the approximation of a suspension as a tapered beam is a very simplified approach, we have investigated the frequency characteristics of a suspension using finite element analysis [3.10]. By varying the length, width and thickness of the suspension, as shown in Fig. 3.2, we have determined the suspension characteristics as a function of design dimensions.

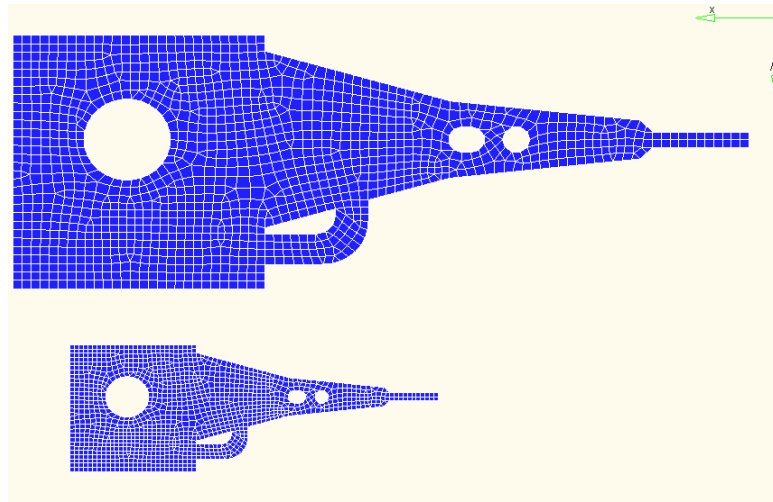


Figure 3.2 A suspension design with its scaled shape simulated using finite element method

In Fig. 3.3, the natural frequencies of the first four mode shapes of a typical suspension are shown as a function of scaling the appropriate mechanical design dimensions using finite element analysis. We observe from Fig. 3.3 that scaling of the suspension in length, width and thickness by 50 percent results in nearly doubling of the natural frequency at a given mode. This result is in good agreement with the predictions from Eqn. 3.6, i.e., all vibration modes increase with reduced dimensions.

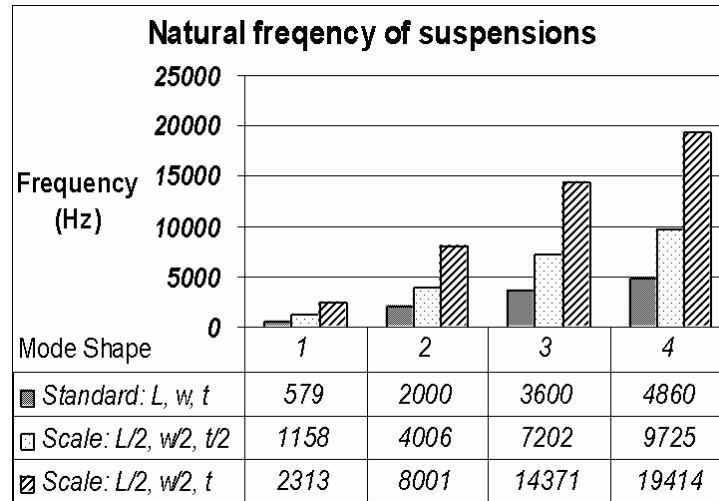


Figure 3.3 Results from finite element analysis of suspensions

3.3 Scaling Laws for disks

Magnetic disks are the most important mechanical components of hard disk drives. For a rotating disk, both bending and membrane stress are present. Benson and Bogy [3.11] have studied the deflection of a flexible spinning disk due to stationary loading. They defined a non-dimensional stiffness as

$$k = \frac{2Eh^2}{3r\Omega^2(3+\nu)(1-\nu^2)b^4} \quad (3.9)$$

where ν is the Poisson's ratio, h is the thickness, r is the areal density, Ω is the angular velocity, and b is the outer disk radius.

The resonance frequencies of a rotating disk can be calculated from

$$P = \sqrt{P_1^2 + P_2^2} \quad (3.10)$$

where P_1 are the frequencies of a rotating membrane disk without bending stiffness, and P_2 are the frequencies of a stationary disk with bending stiffness but no centrifugal forces. For the lowest frequency, P_1 and P_2 are given by

$$\frac{P_1^2}{w^2} = 1 \quad (3.11)$$

where w is the rotational frequency and

$$P_2 = \frac{1.928^2 h}{b^2} \sqrt{\frac{E}{3(1-\nu^2)r}} \quad (3.12)$$

Reducing all disk dimensions by 50 percent ($b_{\min} = 0.50b$, $h_{\min} = 0.50h$), we observe from Eqn. 3.9 and Eqn. 3.10 that both the stiffness and the resonance frequencies increase. To verify these predictions, we have used finite element analysis [3.10] to calculate the resonance frequencies of the three disks shown in Fig. 3.4.

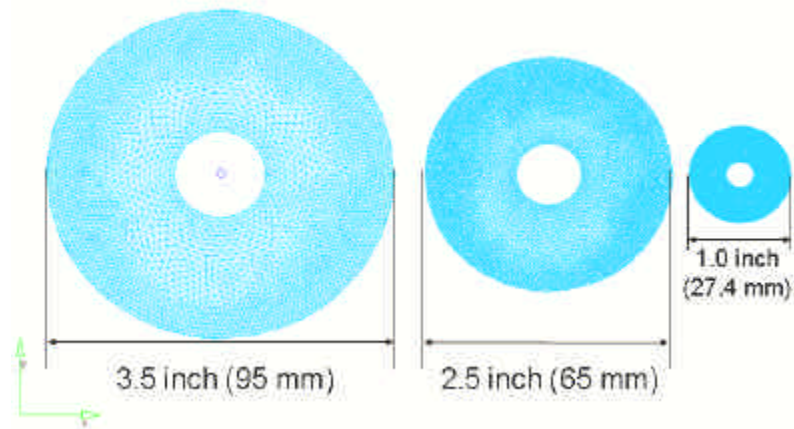


Figure 3.4 Disks of various diameters

The natural frequencies of the first four mode shapes of the disks from Fig. 3.4 with thickness and diameter scaled at the same rate are shown in Fig. 3.5. As can be observed from Fig. 3.5, the numerical values of the natural frequency at a given mode will increase with decreasing disk diameter. This result is in agreement with the theoretical trend predicted by Eqn. 3.10. As discussed by Srikrishna and Kasetty [3.7], disks with increased stiffness and frequency exhibit less track mis-registration caused by disk vibration.

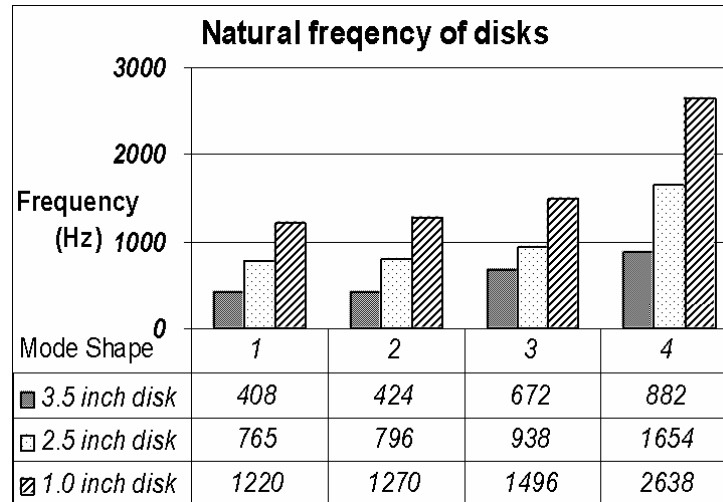


Figure 3.5 Results from finite element analysis of disks

3.4 Summary and Conclusion

The influence of size reduction of suspensions and disks on the static and dynamic performance of a disk drive was evaluated. We have observed that all vibration modes increase with decreasing dimensions for both suspensions and disks. Reducing the size of

disk drive components is therefore beneficial for both the static and dynamic characteristics of individual components.

3.5 References

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Chapter 4 Failure of Brittle and Ductile Hard Disks Due to High Shock Levels

The failure of magnetic recording disks made of brittle or ductile materials is of great interest in the design of small form factor hard disk drives. In this study, fracture of glass disks (brittle material) and plastic deformation of aluminum disks (ductile material) at very high shock levels is investigated using finite element analysis. Failure inception for both disk types occurs at the inside perimeter of the disk. For glass disks, cracks are found to propagate towards the outer perimeter of the disk. The permissible shock level increases with an increase in the clamp diameter, a reduction of the disk diameter, and an increase in the thickness of the disk.

4.1 Introduction

Disk drives with disk diameters as small as 27.4 mm (so-called “micro” drives) are presently used in many consumer applications such as hand held music players, personal digital assistants and mobile phones. Further application of disk drive technology in cars and other mobile devices are anticipated. In many of the above uses of hard disk drives, high shock levels, on the order of thousands of Gs, are expected. To guarantee the reliable performance of hard disk drives under those conditions, the characteristics and response of hard disks to high shock levels has become an important design consideration.

A number of studies have been undertaken in the past concerning the dynamic response of hard disk drives due to shock loads. Schreck [4.1] studied the problem of

shock on the head-disk interface experimentally and used the magnetic read back signal to evaluate the damage. Allen and Bogy [4.2] carried out both experimental work and finite element analysis to determine the effect of shock loads on hard disk substrates, suspensions and sliders. Edwards [4.3] performed a finite element analysis of a hard disk drive to study the transient response of the head disk interface as a function of non-operational shock loads. Lee et al. [4.4] studied the transient response of a suspension subject to a shock impulse. Lin [4.5] investigated the effects of drive enclosure stiffness, disk thickness, actuator stiffness, and bearing stiffness for a 3.5 inch hard disk drive under shock level below 100 G. Zeng and Bogy [4.6] studied the shock response of a hard disk drive in term of disk-suspension-slider air bearing systems. Jayson et al. [4.7] studied numerically the effect of shock on the head disk interface using a simplified air bearing model to simulate operational conditions. More recently, Murthy et al. [4.8] investigated the shock response for 3.5 and 2.5 inch form factor hard disk drives and compared numerical results with experimental results. Feliss et al. [4.9] studied operational and non-operational shock for microdrives. They used a scanning laser Doppler vibrometer to determine the amplitude and frequency of vibrations of the disk enclosure and investigated whether these vibrations contribute to failure of the head/disk interface. Bhargava and Bogy [4.10] studied operational-shock response for small form factor hard disk drives numerically. They used commercially available finite element software programs for the suspension and the disk, and a transient air bearing simulation code to study the response of the disk drive to external shocks.

Failure of materials due to shock or impact has also been investigated for situations other than hard disk drives. Bless et al. [4.11] observed shock waves propagating in a glass rod under dynamic compression. A series of plate impact experiments, bar impact experiments, and impact recovery experiments were performed on brittle materials under various impact conditions by Rosenberg et al. [4.12], Bourne et al. [4.13], and Millett et al. [4.14]. A comprehensive model for failure of brittle materials was discussed by Grady [4.15]. The model incorporates time-dependent fracture and rate-dependent plasticity. Govindjee et al. [4.16] introduced a constitutive model for the tensile failure of plain concrete and implemented this model using finite element analysis. Their model is also applicable to brittle materials.

Failure of ductile materials typically consists of the nucleation, growth and coalescence of voids [4.17 and 4.18]. Failure modes are affected by the state of stress, plastic strain, pressure, strain rate and temperature. A set of dynamic experiments for the failure of ductile materials have been discussed by Costin et al. [4.19]. They used a dynamically loaded circumferentially notched round bar to study material properties at extremely high loading rates. Many experimental models were developed to simulate the material response under dynamic loading [4.20, 4.21]. A thorough review has been provided by Rosakis et al. [4.22] and Voyiadjis et al. [4.23].

Although many studies have been performed on the shock response of hard disk drives and on the general topic of failure of materials, none of these studies considered material failure of disks for small form factor hard disk drives due to high shock load. To fill this gap, this paper investigates the fracture and plastic deformation of brittle or ductile

magnetic recording disks, respectively, due to shock load using finite element analysis. We are using the so-called “radial return plasticity approach” of Belytschko et al. [4.24] to numerically simulate the plastic hardening effect in ductile materials. Different failure criteria are proposed for crack formation in glass disks and the onset of plastic deformation of aluminum disks due to high shock levels. The results provide design guidelines for the maximum allowable shock levels for disk drives with glass or aluminum disks.

4.2 Definition of the Problem

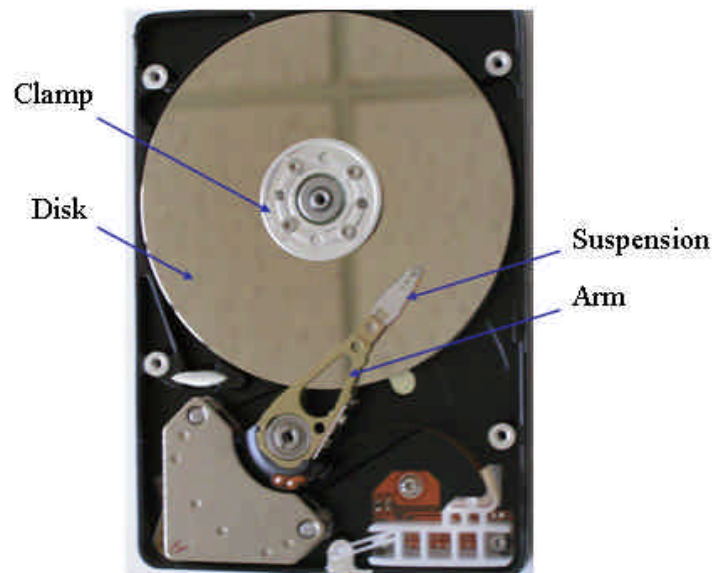


Figure 4.1 The structure of a state-of-the-art hard disk drive

The structure of a state-of-the-art hard disk drive is shown in Fig. 4.1. Each disk drive contains a number of magnetic disks, mounted on the shaft of a spindle motor. The suspension/arm is positioned on the side of the disk and is driven by a voice coil motor. The

slider with an integrated thin film read/write element is attached to the backside of the suspension.

To simulate a shock event, a physical model with all the major components of a hard disk drive must first be developed. Then, material properties and boundary conditions need to be defined. Finally, the response of the disk drive to a shock load is determined.

4.3 Theoretical Background

4.3.1 Brittle vs. Ductile Material

Brittle and ductile materials are typically classified by their behavior during mechanical testing. One of the most notable differences between brittle and ductile materials is their stress-strain relationship (Fig. 4.2).

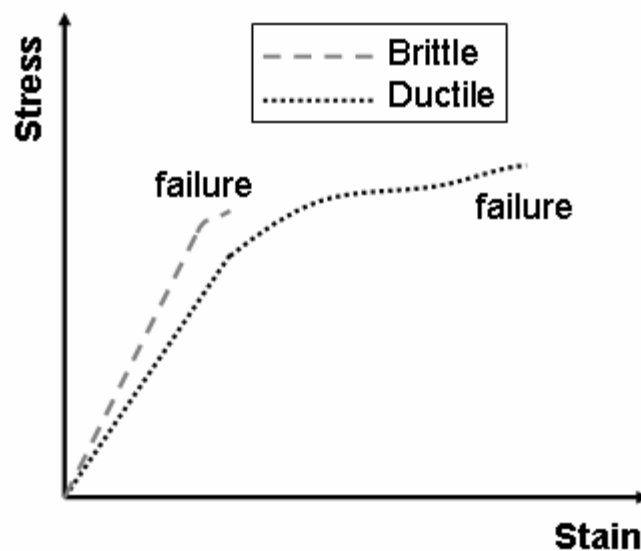


Figure 4.2 Typical stress-strain curves for brittle and ductile materials

As shown in Fig. 4.2, brittle materials can sustain only a very small amount of strain before failure. In general, failure occurs suddenly and without much warning. On the other hand, ductile materials can sustain a much larger amount of strain. Furthermore, they can experience strain hardening before failure.

Glass and aluminum are typical examples for brittle and ductile materials, respectively. The material properties of glass and aluminum used in hard disk drives are shown in Table I.

Table 4.1 Key material properties

Property	Glass	Aluminum
Young's modulus (GPa)	85	72
Density (kg/m ³)	2.5e3	2.7e3
Poisson's ratio	0.25	0.33
Yield strength (MPa)	150	200
Fracture toughness (MPa-m ^{1/2})	0.7	25
Hardness (GPa)	5.5	0.9
Damping ratio	0.009	0.01

As indicated in the table, disks made of glass have higher Young's modulus and higher hardness, but lower yield strength, fracture toughness, and Poisson's ratio as compared to disks made of aluminum. In this paper, we will use glass and aluminum for disks made of brittle and ductile material, respectively.

4.3.2 Failure Criteria and Strain Hardening

Brittle materials cannot withstand large strains before fracture occurs. In our simulation, a critical plastic strain of 5.0e-6 was selected as the failure criterion for failure

inception of glass disks. This value has been used previously in similar investigations reported in the literature [4.25]. From Table 1 we can see that the yield strain for glass is $s_y/E = 1.76e-3$ and hence, the critical plastic strain of $5.0e-6$ is a very small fraction, about 0.3%, of the yield strain. The maximum tensile strain failure criterion was used in our work for the glass disks. The effects of the magnetic overcoat and of residual stresses on the failure of the glass disks were neglected.

The von Mises failure criterion, typical for ductile materials, was used in our work for the aluminum disks. The hardening behavior of aluminum was simulated using values for the effective stress versus the effective plastic strain according to data provided in Ref. [4.25]. The data fit a linear hardening law with a tangent modulus that is about 1% of the Young's modulus for the aluminium given in Table 1.

4.4 Model of disk drive and numerical implementation

The numerical model developed for the disk drive consists of the circular disk, the head gimbal assembly, the disk enclosure and two clamps (see Fig. 4.3). The disk simulated in our model is stationary and thus no centrifugal force is considered.

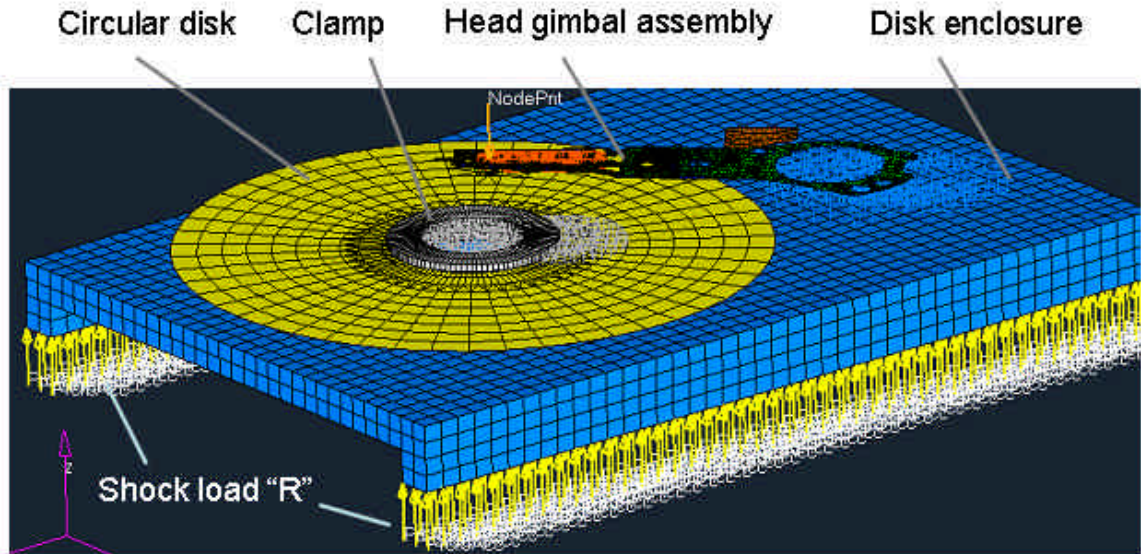


Figure 4.3 Finite element model of a micro disk drive (disk diameter 27.4 mm)

The model was first created in Pro/E [4.27] and then imported into “Hypermesh”, a commercially available pre- and post- processor for finite element analysis [4.28]. The model includes mesh, boundary conditions, contact surfaces and loads. The shock load, denoted by “R” in Fig. 4.3, is applied through the two rails of the disk enclosure. The disk is meshed with 602 shell elements while the disk enclosure is meshed using 2,486 solid elements. The disk and the disk enclosure are connected by clamps that are meshed using 634 solid elements. The surface of the clamps is parallel to the disk and the clamping force is distributed evenly along the clamp (see Fig. 4.4).

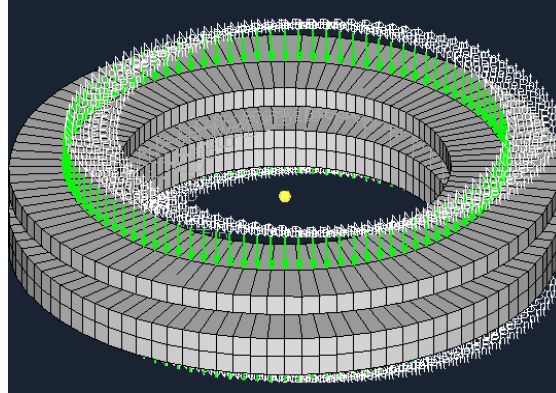


Figure 4.4 Finite element model of the disk clamp

The head/gimbal assembly (see Fig. 4.3) is modeled using 3219 shell elements and 96 solid elements. The assembly is connected at one end to the disk enclosure through rigid elements. The slider is positioned over the disk and contacts the disk. Contact surfaces are defined between the clamp and the disk and between the slider and the disk. The clamping force used is 20 N and the load applied to the slider by the head gimbal assembly is 25 mN (Fig. 4.5).

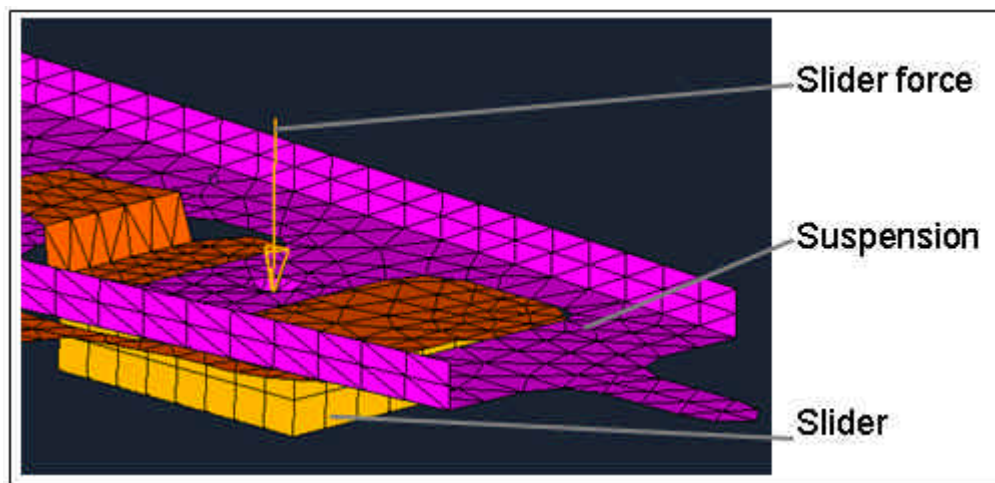


Figure 4.5 Point force applied on the slider

The applied shock load is shown in Fig. 4.6 for five different shock levels ranging from 1000 G to 5000 G. A shock load is modeled by an acceleration profile consisting of a half-sine wave with duration of one millisecond. Throughout this paper, the peak amplitude of the shock acceleration is used to identify the level of the shock load. Commercially available finite element software, LS-DYNA, was used as the solver [4.29].

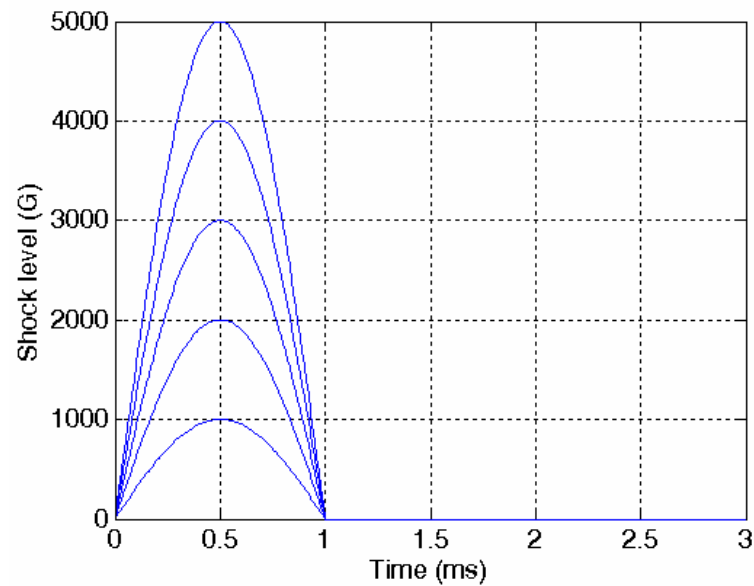


Figure 4.6 Acceleration profile

The “critical shock level” is defined as the shock level for which failure inception corresponds to the peak of the acceleration sine wave. During the process of determining the “critical shock level”, we start with an arbitrary shock level and calculate the tensile strains over the entire disk surface throughout the acceleration cycle for time steps of 0.01 ms. If no failure is detected we increase the shock level in steps of 100 G until first failure

can be observed within a time interval of ± 0.05 ms about the 0.5 ms time corresponding to the shock peak amplitude. In the case that this first failure occurs below the peak amplitude, outside the ± 0.05 ms time interval, the acceleration value at this instant is selected as the shock level for the next trial.

A number of cases were investigated numerically for shock levels higher than the critical one. This was done to study propagation of cracks in glass disks and plastic strain evolution in aluminum disks, beyond their failure inception.

For glass disks, the calculated plastic tensile strain is compared with the value of the critical plastic strain at each time step. If the plastic tensile strain of any element reaches the critical plastic strain, this element is deleted, and crack propagation is initiated.

For aluminum disks, plastic hardening based on the “radial return plasticity approach” by Belytschko et al. [4.24] is considered after the maximum von Mises stress reaches the yield strength. This model is able to deal with any type of hardening and is therefore suitable for general plastic behavior in finite element analyses. The numerical implementation for the radial return approach is described in detail in Ref. [4.26]. Basically, the model presents the relation between stress and strain in the plastic regime by a piece-wise linear approximation. The radial return plasticity procedure consists of the following steps:

a) At each time step the plastic strain increment Δe^P is calculated using

$$\Delta e^P = \frac{s^* - s_y^n}{3G + E^P} \quad (4.1)$$

where s^* is an effective trial stress, defined by $s^* = \left(\frac{3}{2} s_{ij}^{*n+1} s_{ij}^{*n+1} \right)^{\frac{1}{2}}$. Here s_{ij}^{*n+1} is the trial stress at the yield surface for time step $n+1$, s_y^n is the yield strength at time step n , G is the shear modulus, and E^P is the step wise linear tangent modulus in the plastic regime.

b) The plastic strain e_{n+1}^P is updated for time step $n+1$ using

$$e_{n+1}^P = e_n^P + \Delta e^P \quad (4.2)$$

c) The yield strength s_y^{n+1} is calculated for time step $n+1$ by

$$s_y^{n+1} = s_y^n + E^P \Delta e^P \quad (4.3)$$

d) The scale factor m using the yield strength at time step $n+1$ is computed as

$$m = \frac{s_y^{n+1}}{s^*} \quad (4.4)$$

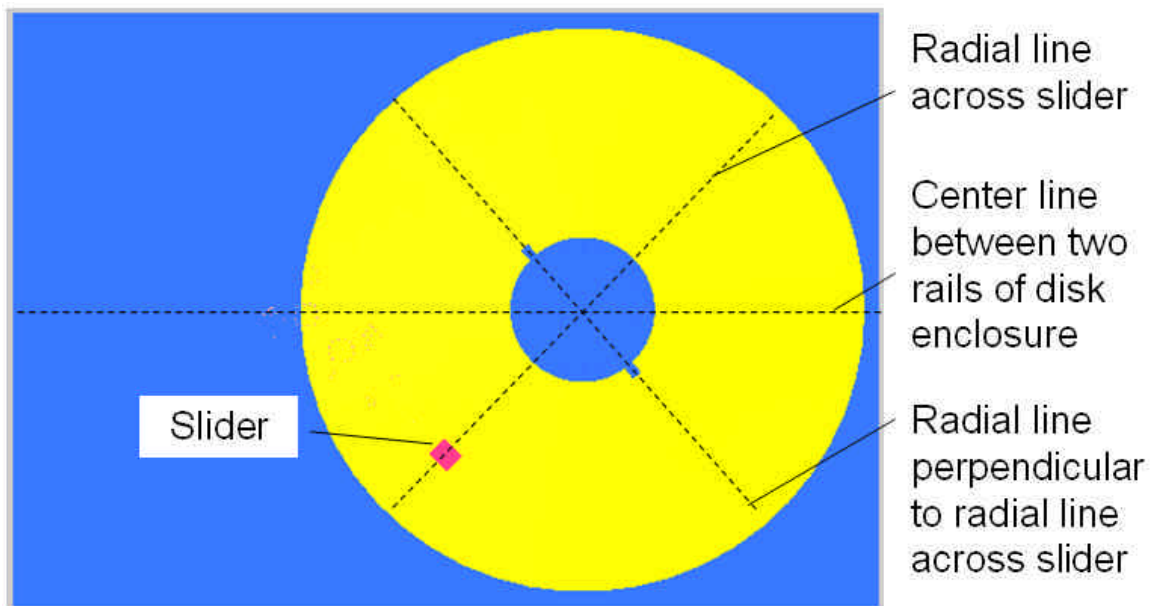
e) Finally, the deviatoric stress s_{ij}^{n+1} for time step $n+1$ is returned to the yield surface as

$$s_{ij}^{n+1} = m s_{ij}^{*n+1} \quad (4.5)$$

4.5 Crack Formation in Glass Disks and Plastic Deformation in Aluminum Disks

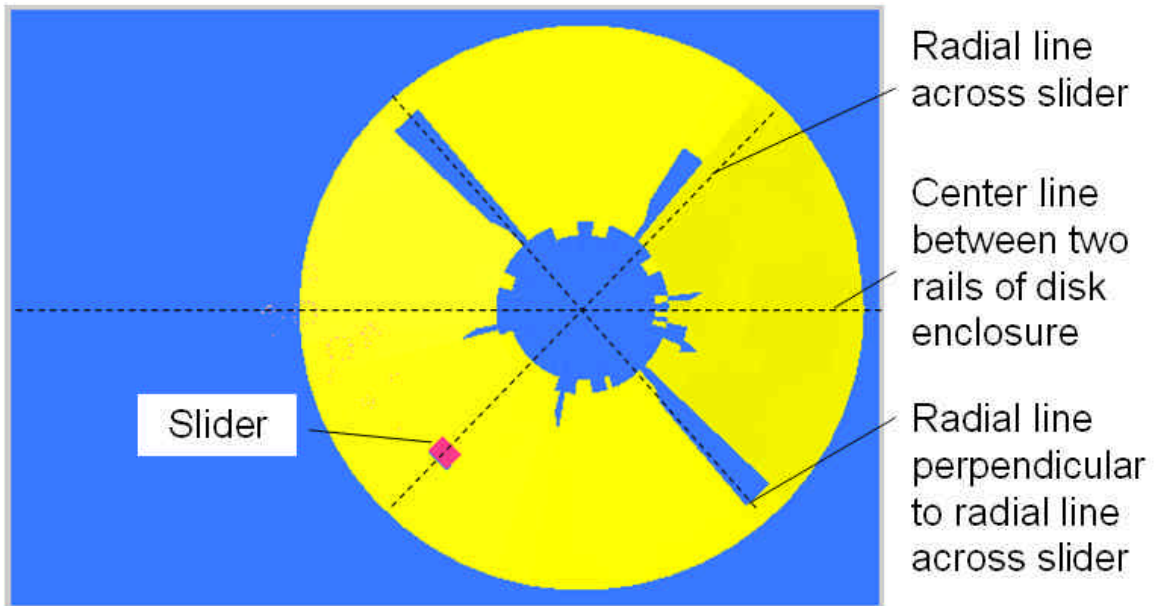
Crack formation on a glass disk in response to a shock level of 4,500 G (which is 500 G above the critical shock level) is shown in Fig. 4.7. Failure inception in the presence of a slider is presented in Fig. 4.7a. Contact between slider and disk during the simulation of a shock event is represented by a point force on the disk surface (see "Slider force" in Fig.

4.5). The bending moment caused by this slider force about any radial line on the disk surface attains its maximum value about the radial line perpendicular to that across the slider. These two radial lines are shown in Fig. 4.7a along with the center line parallel to the two rails of the disk enclosure (see Fig. 4.3). As shown in Fig. 4.7a, two initial small cracks occur at the inner perimeter of the disk coinciding with the radial line of maximum bending moment caused by the slider point force. This occurs approximately 0.36 ms after the start of the shock event (see Fig. 4.6). As the shock evolves in time the two cracks propagate towards the outer diameter of the disk. Additional cracks form at later time and evolve as shown in Fig. 4.7b. The initial cracks propagate along the radial line of maximum bending moment, i.e., the slider location affects the crack formation of a glass disk.

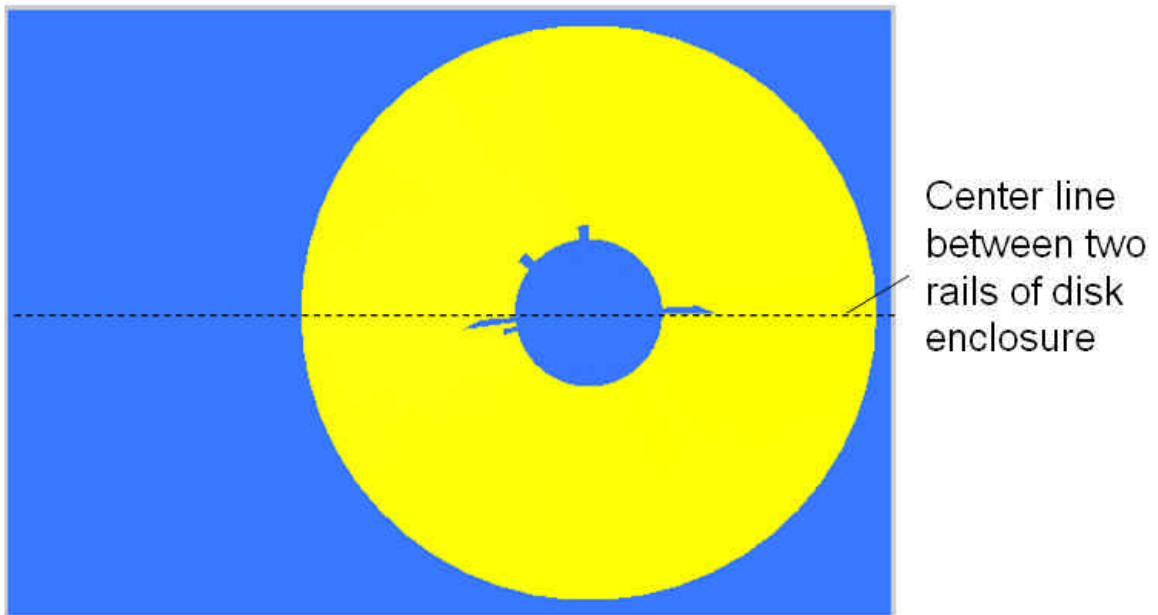


a) Failure inception (slider presence)

Figure 4.7 Crack formation in a glass disk (disk diameter is 27.4 mm)

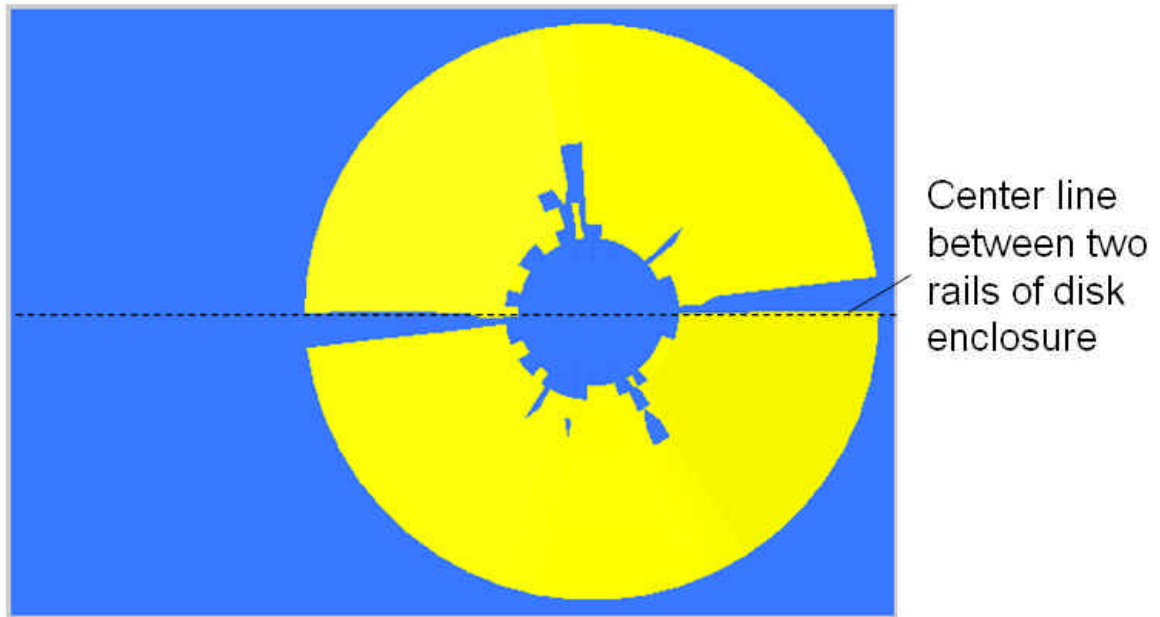


b) Crack propagation (slider presence)



c) Failure inception (no slider presence)

Figure 4.7 (Continued)

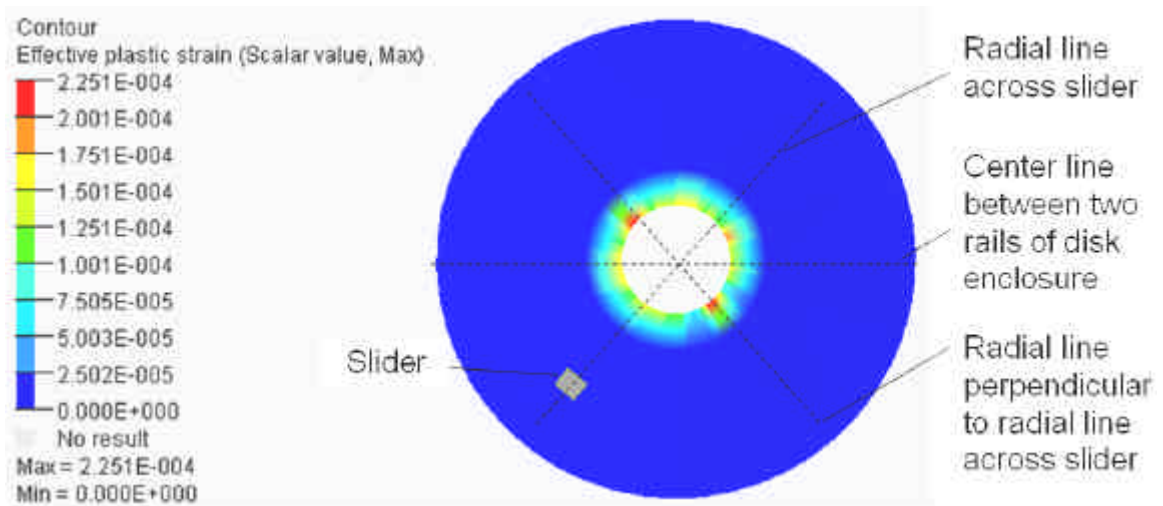


d) Crack propagation (no slider presence)

Figure 4.7 (Continued)

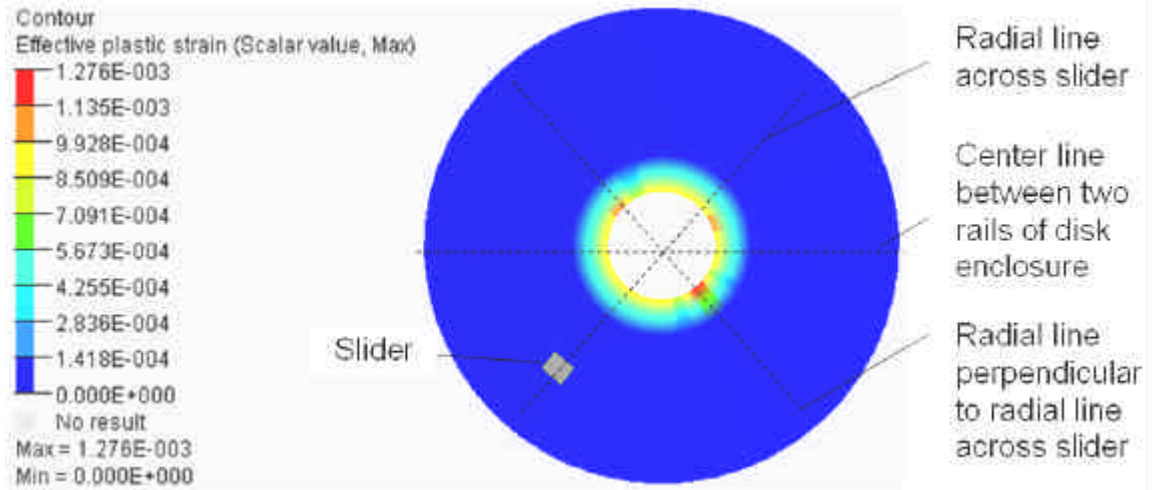
To further investigate the effect of slider position on the formation of cracks (Figs 7a and 7b), an identical shock of 4500 G was applied to the identical glass disk after removal of the slider. In this case the bending moment is caused only by the two rails (see Fig. 4.3) and its maximum value is about their center line. Hence, the failure inception would be expected on this line. Indeed, as shown in Fig. 4.7c, several cracks occur at the inner perimeter of the disk at about 0.37 ms after the start of the shock event, but of all these cracks the two dominant ones are very close to the center line between the two rails. Then, as the shock event progresses, these two dominant cracks expand and propagate towards the outer diameter of the disk along the center line of the rails of the disk enclosure, as shown in Fig. 4.7d.

Very similar failure behavior was observed for ductile disks. Fig. 4.8a shows the plastic yield inception for an aluminum disk of 27.4 mm diameter in response to a shock of 4,500 G (which is 900 G above the critical shock level for this case) in the presence of the slider. The largest effective plastic strain of 2.25×10^{-4} appears at 0.31 ms after the start of the shock, on the inner perimeter of the disk, coinciding with the radial line of maximum bending moment caused by the slider point force. Then, as the shock event progress in time, this plastic strain increases and finally reaches its maximum value of about 1.28×10^{-3} at 0.52 ms, as shown in Fig. 4.8b. Clearly, the maximum plastic strain is on the radial line perpendicular to that across the slider. This indicates that the slider location also affects the plastic deformation of aluminum disks.



a) Failure inception (slider presence)

Figure 4.8 Contours of plastic strain on disk surface (disk diameter is 27.4 mm)



b) Plastic strain at maximum (slider presence)

Figure 4.8 (Continued)

4.6 Design Factors Affecting the Failure of Glass and Aluminum Disks

The failure inception of glass and aluminum disks is affected by a number of design parameters such as disk thickness, disk diameter and clamping conditions. In the following, the effect of these parameters on the critical shock level will be presented. The critical shock levels are normalized by 4,000 G, which is the critical shock level for the standard 27.4 mm glass disk design.

4.6.1 Disk Thickness

In Fig. 4.9, the dependence of the critical shock level on the normalized disk thickness is shown, keeping all the other design parameters constant (clamp outer diameter, clamping force, and disk diameter). The disk thickness is normalized by the “standard” value of 0.381 mm corresponding to the 27.4 mm disk. From Fig. 4.9, we observe that the

critical shock level for failure inception in both glass and aluminum disks increases with disk thickness. Unfortunately, thicker disks which resist shock failure better, are undesirable since increased disk thickness increases the stack-height of the disk drive.

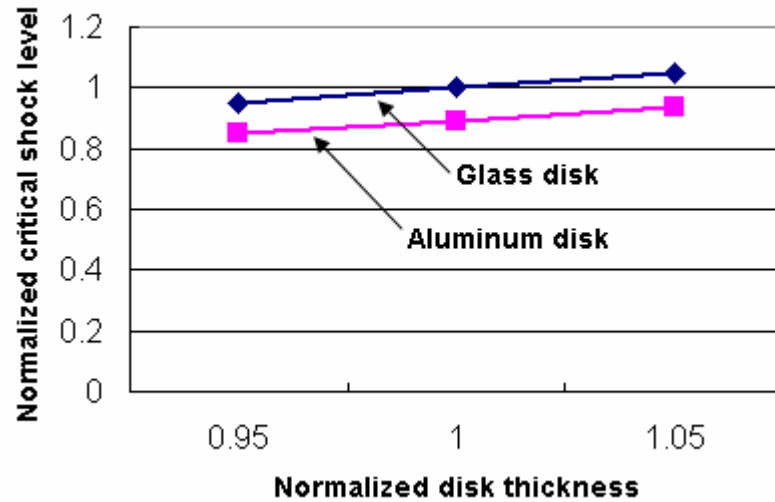


Figure 4.9 Critical shock level vs. disk thickness (other design parameters are kept constant)

4.6.2 Clamping Conditions

The effect of clamping force on the failure inception of glass and aluminum disks is shown in Fig. 4.10. The clamping force is normalized with a “standard” value of 20 N. We observe that the fracture inception in a glass disk is insensitive to the clamping force. However, for aluminum disks, we observe a slight decrease in the critical shock level with an increase in the clamping force.

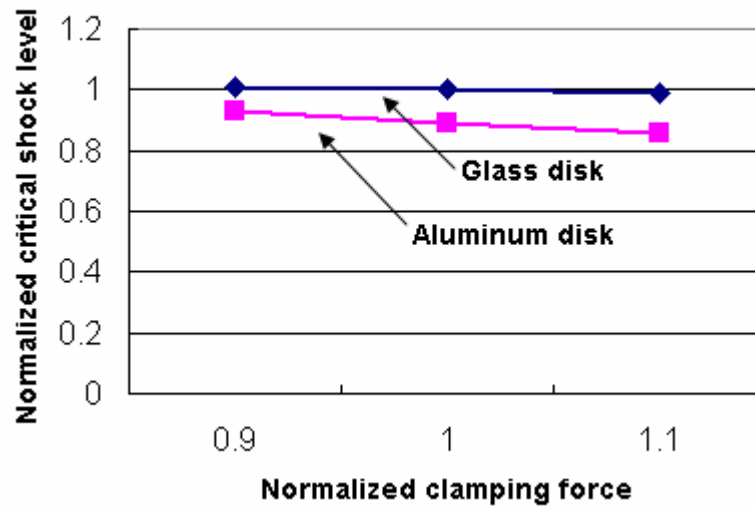


Figure 4.10 Critical shock level vs. clamping force (other design parameters are kept constant)

The effect of the outer diameter of the clamp on the failure inception in glass and aluminum disks is shown in Fig. 4.11. The outer diameter of the clamp is normalized with a “standard” value of 7.62 mm. We observe that the critical shock level for both glass and aluminum disks increases with an increase in the non-dimensional outer diameter of the clamp, i.e., the diameter of the clamp has a strong effect on the critical shock level.

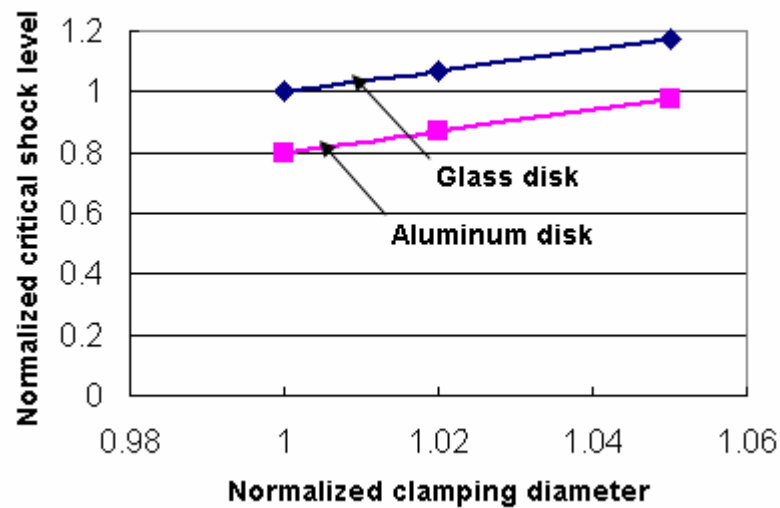


Figure 4.11 Critical shock level vs. clamp outer diameter (other design parameters are kept constant)

4.6.3 Disk Form Factor

Since it is impractical to change the disk diameter alone, without simultaneous structural changes of the entire disk drive system, we will use the form factor rather than the disk diameter as another design parameter. To investigate the effect of the form factor on failure inception we have repeated the previous simulations for a series of form factors (Fig. 4.12). In these simulations we have scaled all geometrical parameters by the same ratio (the form factor). In addition, the clamping force was scaled by the same ratio as the form factor. To leave the flying characteristics unchanged, the load applied by the head gimbal assembly on the slider was kept constant at 25 mN.

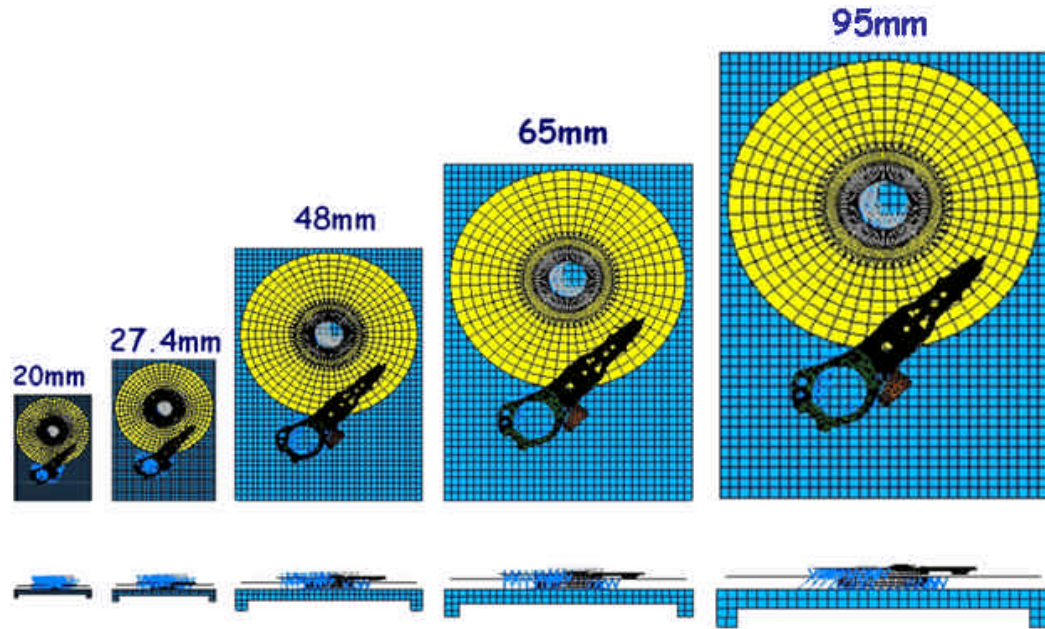


Figure 4.12 Finite element models for hard disk drives with different form factors

The dependence of the critical shock level on the normalized form factor at failure inception of glass and aluminum disks is shown in Fig. 4.13. The disk form factor is normalized with the form factor of the 27.4 mm disk. Clearly, the form factor has the most significant effect on the maximum permissible shock level. The critical shock level increases sharply with a decrease of the disk form factor for both glass and the aluminum disks, i.e., the smaller the disk form factor the higher is its resistance to failure. This would be expected because of the smaller bending moments produced by either the rails of the enclosure or the slider point force.

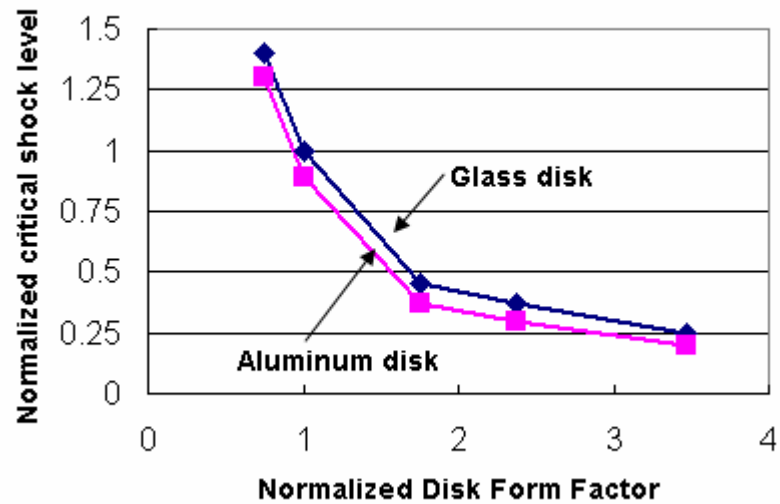


Figure 4.13 Critical shock level vs. disk form factor

4.7 Discussion

From the simulation results shown in the previous section, we observed that the critical shock level increases when the disk thickness or the outer diameter of the clamp increase or when the disk form factor (and thereby the disk diameter) decreases. To explain these results, we examined a somewhat similar case of “bending of a flat circular plate” (see Fig. 4.14). The outer edge of the plate is free but its inner edge is clamped (Ref. [4.30]). Here, the inner radius, b , of the plate represents the outer radius of our disk clamp. The outer radius, a , and the thickness, t , of the plate represent our disk outer radius and thickness, respectively. A load per unit area, q , is applied to the plate.

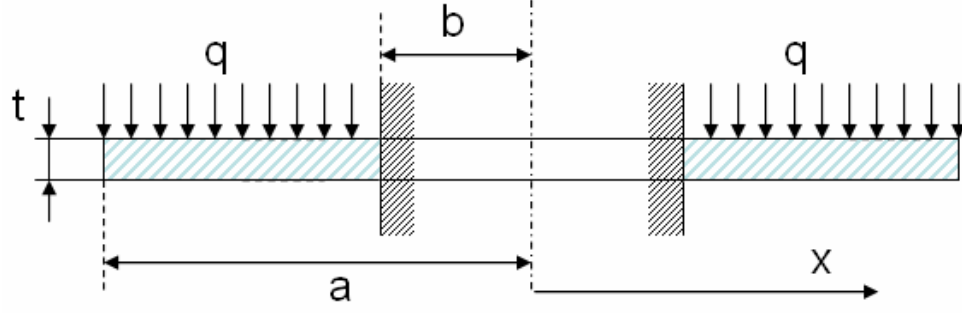


Figure 4.14 Bending of a flat circular plate (outer edge free, inner edge clamped) [30]

The maximum bending moment per unit length at the clamped inner circumference of the disk, at radial location b , is given in [4.30] by

$$M_{rb} = \frac{qa^2}{C_8} \left[\frac{C_9}{2ab} (a^2 - b^2) - L_{17} \right] \quad (4.7)$$

The parameters C_8 , C_9 , and L_{17} in Eq. (4.7) are general functions that are also given in [4.30] and can be expressed at $x = b$ by

$$\begin{cases} C_8 = \frac{1}{2} \left[1+n + (1-n) \left(\frac{b}{a} \right)^2 \right] \\ C_9 = \frac{b}{a} \left\{ \frac{1+n}{2} \ln \frac{a}{b} + \frac{1-n}{4} \left[1 - \left(\frac{b}{a} \right)^2 \right] \right\} \\ L_{17} = \frac{1}{4} \left\{ 1 - \frac{1-n}{4} \left[1 - \left(\frac{b}{a} \right)^4 \right] - \left(\frac{b}{a} \right)^2 \left[1 + (1+n) \ln \frac{a}{b} \right] \right\} \end{cases} \quad (4.8)$$

where ν is the Poisson's ratio. Eq. (4.7) can be rewritten as

$$M_{rb} = qa^2 \left[\frac{C_9}{C_8} \frac{a}{2b} \left(1 - \left(\frac{b}{a} \right)^2 \right) - \frac{L_{17}}{C_8} \right] \quad (4.9)$$

Equation (4.9) can be simplified by noting that typical values of the ratio of the inner to outer diameter of hard disk drives give $b/a \approx 0.2$. Hence, higher orders of the ratio b/a can be neglected in Eq. (4.8) compared to 1. Thus, C_9/C_8 and L_{17}/C_8 can be approximated as

$$\begin{cases} \frac{C_9}{C_8} \approx \frac{\frac{b}{a} \left(\frac{1+n}{2} \ln \frac{a}{b} + \frac{1-n}{4} \right)}{\frac{1}{2}(1+n)} = \frac{2b}{a} \left(\frac{1}{2} \ln \frac{a}{b} + \frac{1-n}{4(1+n)} \right) \\ \frac{L_{17}}{C_8} \approx \frac{\frac{1}{4} \left(1 - \frac{1-n}{4} \right)}{\frac{1}{2}(1+n)} = \frac{3+n}{8(1+n)} \end{cases} \quad (4.10)$$

Inserting Eq. (4.10) into Eq. (4.9), the maximum bending moment per unit length at the inner circumference of a disk is approximated as

$$M_{rb} \approx qa^2 \left[\left(\frac{1}{2} \ln \frac{a}{b} + \frac{1-n}{4(1+n)} \right) - \frac{3+n}{8(1+n)} \right] = qa^2 \left(\frac{1}{2} \ln \frac{a}{b} - \frac{1+3n}{8(1+n)} \right) \quad (4.11)$$

In the absence of any external loading, the load per unit area q is $q = rgt$, where r is the material density of the disk and g is the constant of gravity. Hence, the maximum bending moment per unit length on this plate due to its own mass is calculated as

$$M_{rb} \approx r g t a^2 \left(\frac{1}{2} \ln \frac{a}{b} - \frac{1+3n}{8(1+n)} \right) \quad (4.12)$$

The maximum stress caused by the own mass of the plate can be expressed as (see [4.30])

$$s_{\max} = M_{rb} \frac{6}{t^2} \approx \frac{6 r g a^2}{t} \left(\frac{1}{2} \ln \frac{a}{b} - \frac{1+3n}{8(1+n)} \right) \quad (4.13)$$

Now, if we increase the acceleration (gravity) to a point where $s_{\max}$ becomes equal to the yield strength, we can consider this acceleration as being the critical shock level. Hence, considering only the geometrical parameters in Eq. (4.13), the relationship between the critical shock level and the geometrical parameters can be expressed by

$$G_{crit} \propto \frac{t}{a^2 \left(\ln \frac{a}{b} - \frac{1+3n}{4(1+n)} \right)} \quad (4.14)$$

From expression (4.14), we can see that the critical shock level is linearly proportional to the thickness, t , in agreement with the prediction in Fig. 4.9 where a thicker disk can withstand higher shock levels. Similarly, the critical shock level in expression (4.14) is increased when the inner radius, b , of the plate increases. This trend is in agreement with the results shown in Fig. 4.11 where an increasing clamp diameter results in a higher critical shock level.

From expression (4.14) we can see that keeping a/b and t/a constant, the critical shock level is inversely proportional to the outer radius a . This correlates with the results shown in Fig 13.

4.8 Conclusions

The failure mechanism of small form factor hard disks due to high shock levels was investigated. It was found that failure inception, in both brittle and ductile material disks, occurs at the inside perimeter of the disk on a radial line about which the bending moment is at maximum. When the slider rests on the disk this radial line is perpendicular to that across the slider, otherwise this radial line is the center line of the two rails.

The effect of several design parameters on the critical shock level at which failure incepts was investigated. It was found that an increase in the outer diameter of the clamp and an increase in the disk thickness cause an increase of the critical shock level. The critical shock level is affected strongly by the disk form factor and sharply increases when the disk becomes smaller. The numerical results were compared with a simplified analytical model for static bending of a flat circular plate and were found in good qualitative agreement.

4.9 Acknowledgement

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Chapter 5 Experimental Setup for the Measurement of Slider Dynamics in Five Degrees-of-freedom

The study of the dynamic characteristics of the head/disk interface is important for the reliability of the head/disk interface. In this chapter, we will use a five degrees-of-freedom system to study the dynamic characteristics of the head/disk interface. The set-up of the experimental system and result of the measurement will be discussed.

5.1 Introduction

In order to achieve areal densities above 1 Tb/in², the flying height of a slider has to be reduced to 2.5 nm [5.1]. For such a small spacing, intermittent contacts between slider and disk may frequently occur during disk drive operation. Such contacts lead to slider vibrations, which could cause read/write errors. Therefore, it is very important to study the vibrations of sliders in the so called “proximity contact region”. It is also necessary to understand the head/disk interface at very small flying heights.

An air bearing slider used in hard disk drives has six degrees-of-freedom, as indicated in Fig. 5.1. In particular, it can move up and down (vertical motion), left to right (off-track motion), forward to backward (down-track motion). In addition, it can rotate around three orthogonal axis, i.e., yaw, roll and pitch. Rotation around the vertical axis is denoted by yaw, around the down-track axis by roll and around the off-track axis by pitch.

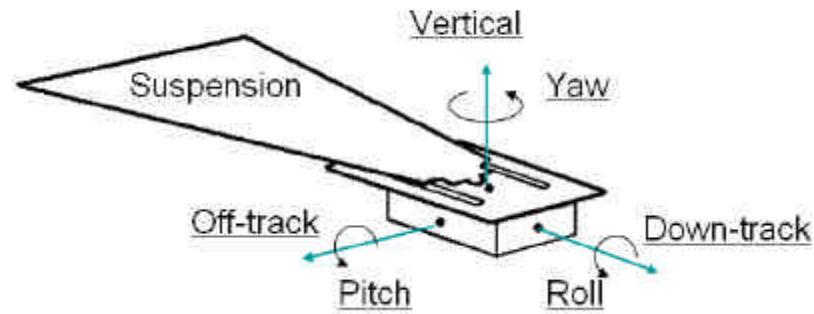


Figure 5.1 Six components of slider vibration

The suspension has a high in-plane stiffness, i.e., the slider vibrations in the yaw direction are very small. Thus, the degree of freedom in the yaw direction is not considered in this thesis, i.e., we are concerned only with five degrees-of-freedom. They are vertical, down-track, off-track, pitch and roll.

5.2 Laser Doppler Vibrometer Used in Measurement of Slider Vibrations

A Laser Doppler vibrometer is an optical instrument that can measure velocity and displacement of vibrating structures [5.2]. It has been widely used to detect slider vibrations when flying on a rotating disk. Miu et al. [5.3] used laser Doppler vibrometer to measure vibrations of sliders. They measured the slider vibrations in the vertical direction, and at different points on the slider. They were also able to calculate the pitch and roll vibrations of the slider. Bogy and Talke [5.4] applied a laser Doppler vibrometer to measure the out-of-plane and in-plane vibrations of a slider. They also applied the laser Doppler vibrometer to measure disk run-out, disk surface topography, as well as suspension and actuator vibrations.

Following the initial experiments described in [5.3, 5.4], laser Doppler vibrometry has been widely used in the hard disk drive industry. Briggs and Talke [5.5] used a laser Doppler vibrometer to measure the dynamics of a “micro” slider. Jeong and Bogy [5.6] used laser Doppler vibrometer to measure slider dynamics during load-unload induced slider/disk contact. Cromwell et al. [5.7] used a laser Doppler vibrometer to investigate surface topography of laser textured disks. Many Researchers [5.8-5.12] investigated the vibration characteristics of low-flying sliders during contact with rotating disks using a laser Doppler vibrometer. Thornton et al. [5.9] and Ono et al. [5.10] used a laser Doppler vibrometer to determine the instability of a slider during its transition from sliding to flying. Recently, Kiely and Hsia [5.12] published an experimental study of slider dynamics using three laser Doppler vibrometers to obtain the motion of a slider in the vertical, off-track, and down-track directions. They also obtained pitch and roll motion from vertical spacing measurements at two separate locations [5.13].

The principle of laser Doppler vibrometry is shown in Fig. 5.2. Light of frequency f from a light source (denoted by S) is pointed towards a reflective object that moves along its normal direction with velocity v . The reflected light from the object surface is observed by a photo detector (denoted by D). In most applications, the light source and photo detectors are integrated. For a giving time Dt , the object moves along its normal direction by a distance vDt . We observe that during time Dt the optical path length is changed by a distance

$$\Delta l = OO' + O'O = 2v\Delta t \quad (5.1)$$

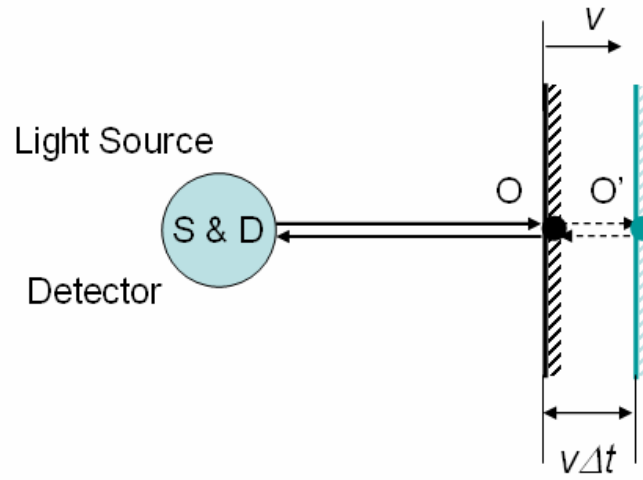


Figure 5.2 Diagram explaining theory of laser Doppler vibrometry

Thus the frequency shift due to the motion of the object is

$$\Delta f_D = \frac{f}{c} \frac{dl}{dt} = \frac{2fv}{c} \quad (5.2)$$

where c is the velocity of light. This frequency shift Δf_D is called the Doppler frequency shift, i.e., the change in frequency of a signal that is reflected from a moving target.

Using $l = \frac{c}{f}$, Eqn. 5.2 can be rewritten as

$$\Delta f_D = \frac{2v}{l} \quad (5.3)$$

Thus, from Eqn. 5.3, the velocity of a moving object is given by

$$v = \frac{l \Delta f_D}{2} \quad (5.4)$$

The Doppler frequency shift Δf_D can be detected by means of a frequency demodulator. As indicated in Eqn. 5.4, by detecting the Doppler frequency shift Δf_D and the angle α (change in direction of the light), the velocity of the moving object can be calculated. The resolution of a laser Doppler vibrometer is limited by the accuracy of the frequency demodulator.

Although laser Doppler vibrometry has been used extensively, no studies have been reported to measure slider vibrations independently and simultaneously in five degrees-of-freedom. Since the dynamics of sliders is very complicated during contact with the rotating disk, a system capable of measuring slider vibrations in five degrees-of-freedom is of great interest.

5.3 Five Degrees-of-freedom Experimental Systems

In this study, an experimental set-up was used consisting of three independent laser Doppler vibrometers and a position sensitive detector. The set-up is capable of measuring slider vibrations in five degrees-of-freedom independently and simultaneously.

5.3.1 Experimental Set-up for Five Degrees-of-freedom Measurement

The experimental set-up for measurement of motions in five degrees-of-freedom is shown schematically in Fig. 5.3. Three independent laser Doppler vibrometers are used to measure the motion of the slider in the vertical, down-track and off-track direction. A position sensitive sensor (PSD) was used to measure the slider motion in pitch and roll direction. Amplifiers and filters were used to amplify and filter signals from the three laser

Doppler vibrometers and the position sensitive detector. A spindown tester was used to load/unload the head/suspension assembly onto the rotating disk. Digital oscilloscopes were used to visualize and capture the output signals from the laser Doppler vibrometers and the position sensitive detector.

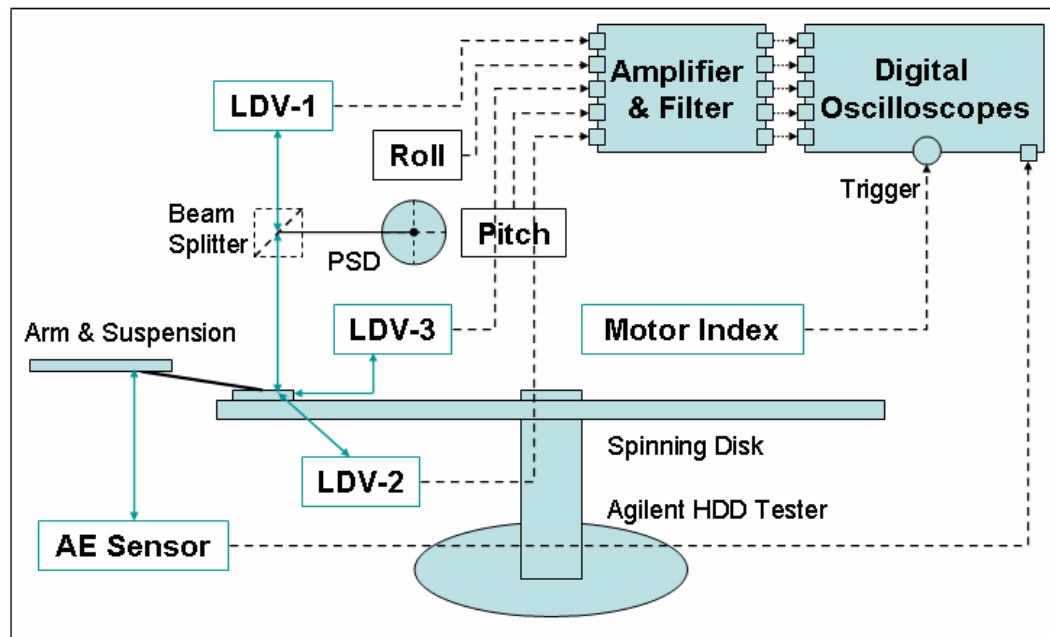


Figure 5.3 System set-up for measurement of slider vibrations for five degrees-of-freedom

A schematic view of the measurement set-up is shown in Fig. 5.4. The vertical motion of the slider is measured by deflecting the laser beam of LDV-1 from a miniaturized metallized silicon mirror that is attached to the backside of the suspension. The mass of such a mirror is less than 5 percent of a slider, and has negligible effect on the slider dynamics. The laser beam is reflected by the mirror and travels to the beam splitter, which splits the signal into two beams: one beam senses the vertical velocity of the slider and the

other beam were measured by the position sensitive detector (PSD) for the pitch and roll motion.

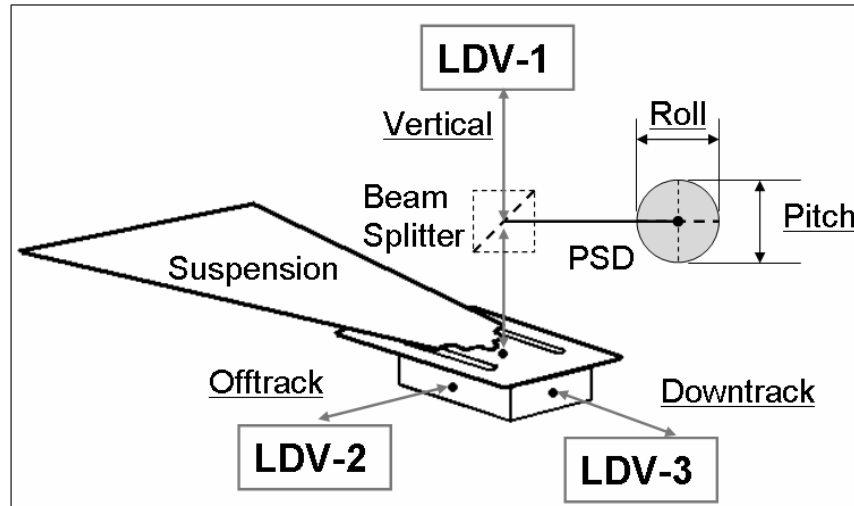


Figure 5.4 Illustration of measurement of slider motion using three independent laser Doppler vibrometers and one position-sensitive detector to detect the slider motion for five degrees of freedom: vertical, down-track, off-track, pitch, and roll

The position sensitive detector measures the angular change of the slider directly. As a result, the position of the laser spot relative to vibration modal lines does not affect the measurement because the angular change of the rigid slider (a solid body) is constant along its entire length. The signal output from the position sensitive detector provides both the pitch and roll angular change of the slider motion. When the laser beam is projected onto the disk without the slider present, the vertical, circumferential slope (pitch), and radial slope (roll) components of disk topography can be obtained simultaneously. Calibration of the position sensitive detector is discussed in appendix A.

The second laser Doppler vibrometer (LDV-2) measures the off-track motion by reflecting the laser beam radially from the side of the slider. Similarly, the laser Doppler vibrometer in the down-track direction (LDV-3) reflects the laser beam from the rear-end surface of the slider pad, thereby measuring the slider motion in the circumferential direction.

In order to synchronize the signals from three laser Doppler vibrometers and the position sensitive detector, a motor index signal was used as triggering signal. The motor index signal excites a sharp rise at every revolution of the disk.

The signals from the three laser Doppler vibrometers and the position sensitive detector are filtered and amplified in order to improve the signal-to-noise ratio. Averaging of the signals is also used for this purpose. In addition, an acoustic emission (AE) sensor is mounted on the loading arm of the head-gimble assembly. The calibrated acoustic emission sensor detects the onset of slider/disk contacts during operating conditions.

5.3.2 Investigation of Slider Vibration in Five Degrees-of-freedom

Using the experimental system described above, we have measured the slider vibrations in five degrees-of-freedom for a slider flying on a rotating disk with and without contacts. The slider design used is shown in Fig. 5.5. The slider has a nominal design flying height of 4 nm at a linear velocity of 13.3 m/s, and a nominal flying height of 8 nm at a linear velocity of 19.1 m/s, both at zero skew. Disks used in this experiment are commercially available thin film disks lubricated with a thin layer of perfluoropolyether

(PFPE) type lubricant at a thickness of 1.4 nm. The disk was glide-height tested with a slider flying at 2.8 nm.

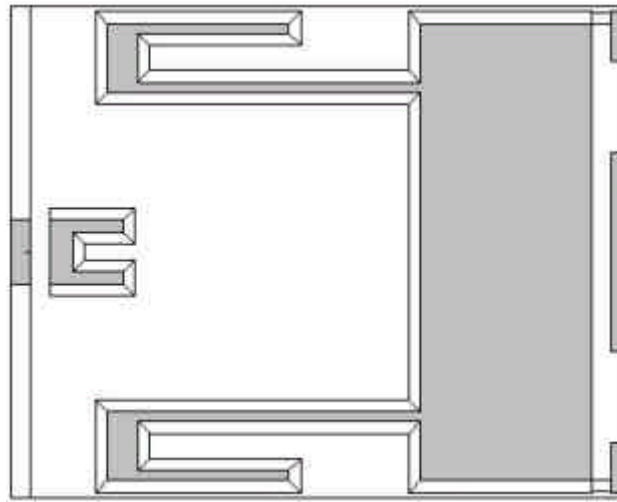


Figure 5.5 Air bearing slider used in the experiment

A plot of the waveforms of the slider motion for the five degrees-of-freedom is shown in Fig. 5.6. The results are shown for the vertical, down-track and off-track motion in nanometers, and for pitch and roll motion in micro radians. The measurements were carried out when the slider was flying on the disk at a linear velocity of 13.3 m/s. The waveforms were filtered by a 10 kHz high-pass filter, averaged 25 times and captured at a sampling frequency of 10 MHz. The waveforms are plotted as amplitude vs. time for 1 ms.

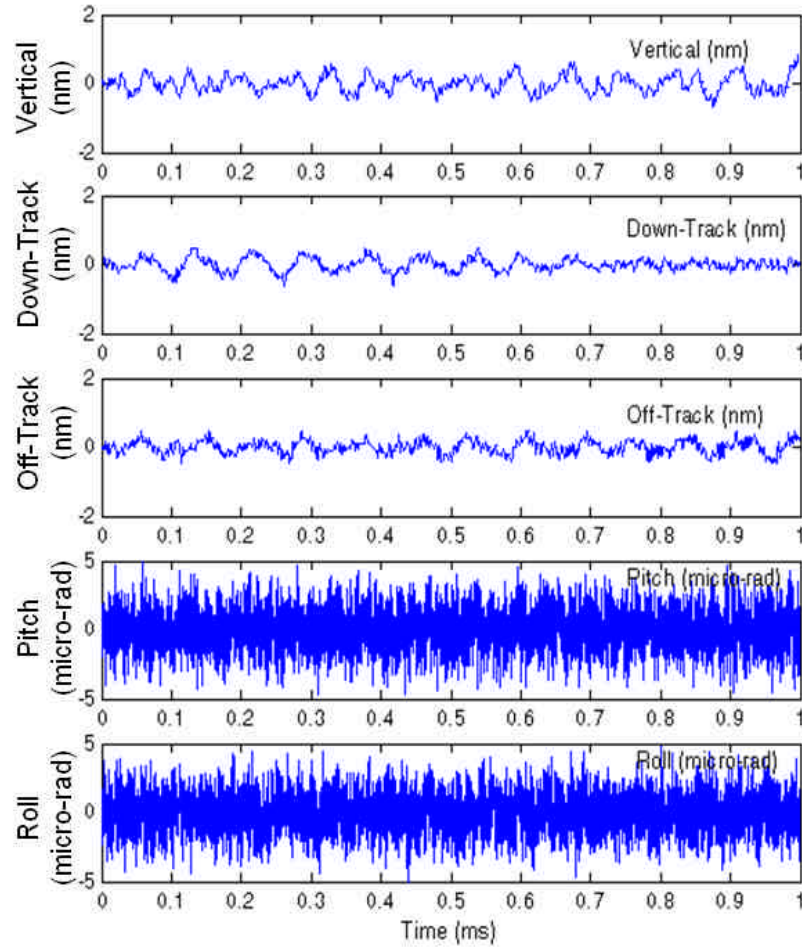


Figure 5.6 Waveforms of slider vibrations for five degrees-of-freedom when flying on a rotating disk. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll

As shown in Fig. 5.6, the slider shows a peak-to-peak vibration amplitude of 0.7 nm in the vertical direction; a peak-to-peak vibration amplitude of 0.8 nm in the down-track direction; and a peak-to-peak vibration amplitude of 0.5 nm in the off-track direction. The slider vibrations for these motions are small compared to the slider flying height. The slider

shows a peak-to-peak vibration amplitude of $7.9 \mu\text{rad}$ in the pitch direction and a peak-to-peak vibration amplitude of $6.4 \mu\text{rad}$ in the roll direction.

In Fig. 5.7, the power spectral density of the waveforms obtained from slider vibrations is shown.

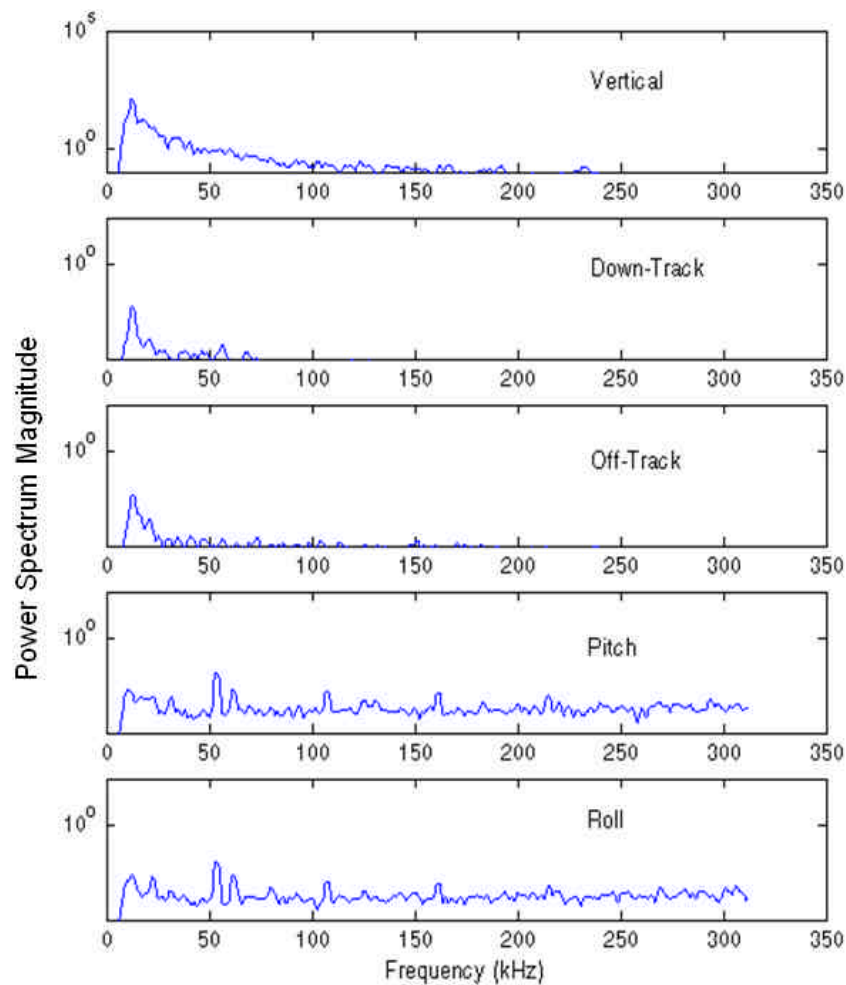


Figure 5.7 Power spectral density of slider motion for five degrees-of-freedom obtained from experimental measurements. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll

We observe that for all five degrees-of-freedom no peaks are present in the frequency range higher than 50 kHz. Thus, we conclude that the slider air bearing mode in the present experiment is not excited.

To measure slider vibrations during contacts with the rotating disk, we reduced the disk rotating speed to a linear velocity of 10.6 m/s. At this speed, the nominal flying height of the slider is approximately 2.4 nm. This flying height is lower than the glide height of the disk, and contacts of the slider with the disk are expected.

A plot of waveforms of the slider vibrations for five degrees-of-freedom is shown in Fig. 5.8. The waveforms are plotted as amplitude vs. time for duration of 1 ms. From top to bottom, the waveforms plots denote the slider vibrations in the vertical direction, the down-track direction, the off-track direction, as well as the pitch and the roll direction.

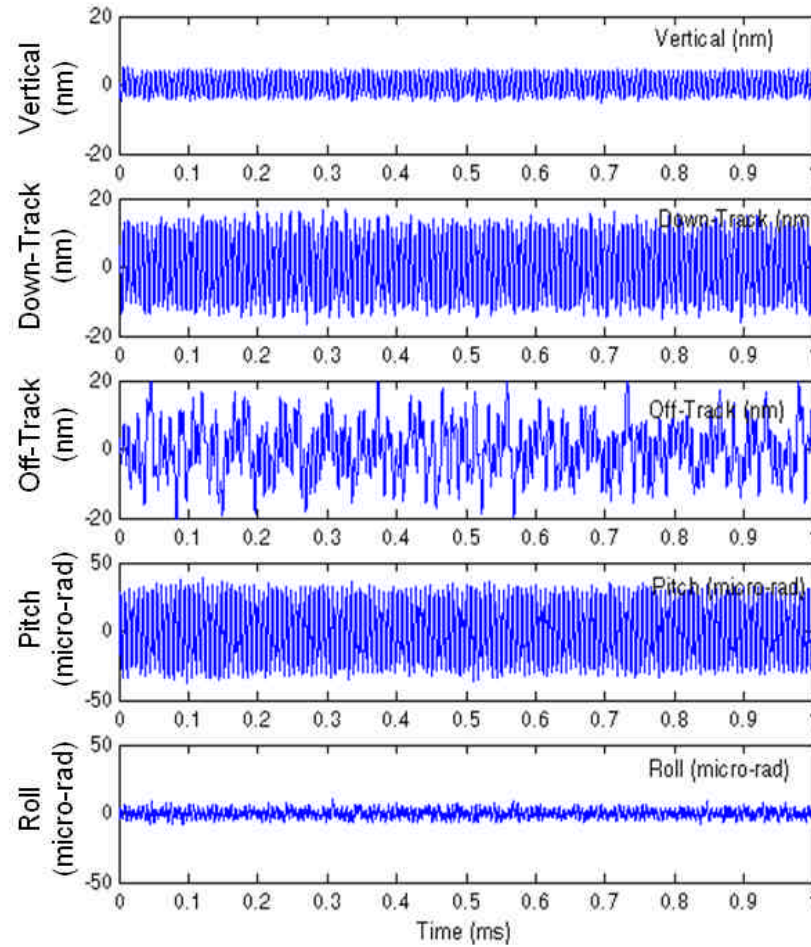


Figure 5.8 Waveforms of slider vibrations for five degrees-of-freedom when contacting with a rotating disk. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll

As shown in Fig. 5.8, the slider shows a peak-to-peak vibration amplitude of 6.9 nm in the vertical direction, a peak-to-peak vibration amplitude of 33.2 nm in the down-track direction, and a peak-to-peak vibration amplitude of 35.0 nm in the off-track direction. In addition, the slider shows peak-to-peak vibration amplitudes of 69.2 μ rad in the pitch

direction and $9.1 \mu\text{rad}$ in the roll direction. It is apparent that the slider is excited by contacts with the disk.

The power spectral density of the waveforms is shown in Fig. 5.9.

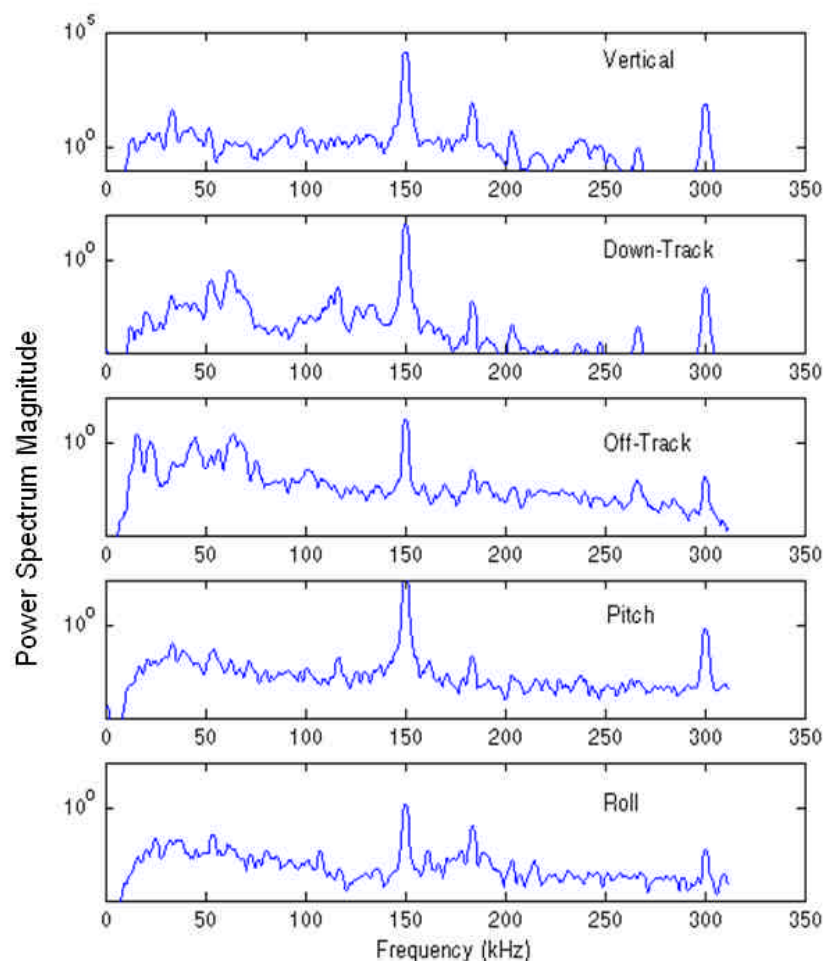


Figure 5.9 Power spectral density of slider motion for five degrees-of-freedom obtained from experimental measurements. From top to bottom is slider motion in vertical, down-track, off-track, pitch and roll

In the power spectral density plots shown in Fig. 5.9, the slider motion in the vertical direction has the highest peak at a frequency of 150.4 kHz. Peaks are found in the

down-track direction at 150.4 kHz, the off-track direction at 150.0 kHz, the pitch direction at 150.4 kHz and the roll direction at 150.0 kHz. The slider motion is dominated by a vibration mode at 150 ± 0.5 kHz for all five degrees-of-freedom. This frequency corresponds to the air bearing frequency (pitch) of the slider, Thus we conclude that the slider air bearing mode is excited during contacts of a slider with a rotating disk.

5.4 Summary

The experimental set-up for detecting slider vibration for five degrees-of-freedom was introduced in this chapter. The set-up enables the detection of vibrations of sliders in five degrees-of-freedom and the determination of the head disk contacts. Slider air bearing frequencies were found to be excited during slider/disk contacts.

5.5 References

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Chapter 6 Experimental Study of Slider Vibrations Using the 5-DOF Measurement System

In this chapter, the five degrees-of-freedom measurement system is used to study the dynamics of an air bearing slider when it is flying on a rotating disk at different conditions. At first, the effect of disk topography on slider dynamics is studied. The gain of the slider, defined as the ratio of the amplitude of slider displacement to disk roughness amplitude, is determined. The dependence of frequency on components of the disk topography is investigated. The dynamic response of the slider as a function of disk topography is analyzed in both the time and the frequency domain.

The measurement system is also used to study the motion resulting from slider-lubricant interactions. The change in slider dynamics is determined by comparing the vibration signals from the slider at time increments of 2 minutes. The effect of slider-lubricant interactions on slider dynamics is studied. The slider motion is also examined using spectral analysis and joint time frequency analysis. The excitation frequency is compared with the air bearing natural frequency.

6.1 Effect of Disk Topography on Slider Dynamics

6.1.1 Introduction

As the flying height of air bearing sliders decreases (currently 8 nm), flying height modulation of the slider becomes an issue of increasing importance. Flying height modulation is caused by the dynamic response of the slider air bearing to the disk

topography. A number of studies have been performed in the past concerning the dynamic response of a slider as a function of disk surface topography, i.e., surface roughness, disk waviness and disk micro-waviness. Suk et al. [6.1] correlated the dynamic response of a slider at a flying height on the order of 100 nm with disk roughness. They concluded that the effect of surface roughness on flying height and air-bearing frequencies was minimal, and that only the slider landing speed was affected by the presence of isolated asperities. However, this conclusion is not valid for sliders with very low flying height. Yao et al. [6.2] used laser Doppler vibrometry (LDV) to measure vibrations of a slider with a flying height of 25 nm and studied the effect of disk waviness on slider vibrations. They defined micro-waviness of a disk as the disk topography in the range of 10% to 80% of the slider length (125 μm to 1 mm for a “pico” slider). They showed that a correlation exists between dynamic micro-waviness and flying height modulations of a slider. Thornton et al. [6.3] investigated effects of disk micro-waviness to sliders with ultra-low flying height ($\sim 5\text{nm}$). They found that a reduced size trailing edge pad improved flying height modulation due to disk micro-waviness. Lee and Polycarpou [6.4] studied dynamics of sliders affected by disk waviness for super smooth disks ($R_a < 0.3 \text{ nm}$). Deoras and Talke [6.5] used system identification method to evaluate the effects of disk micro-waviness on slider flying height modulations. They observed that the flying height modulation of sliders due to disk micro-waviness depends on disk velocity.

The above investigations have helped to clarify the effect of surface topography on slider dynamics. However, none of the previous studies has addressed the effect of circumferential and radial slope of disk topography on slider dynamics. A schematic view

of the slider/disk interface is shown in Fig. 6.1, indicating that the circumferential slope of disk topography may affect the slider vertical and pitch motion (Fig. 6.1a), and that the radial slope of disk topography may affect the slider roll motion (Fig. 6.1b).

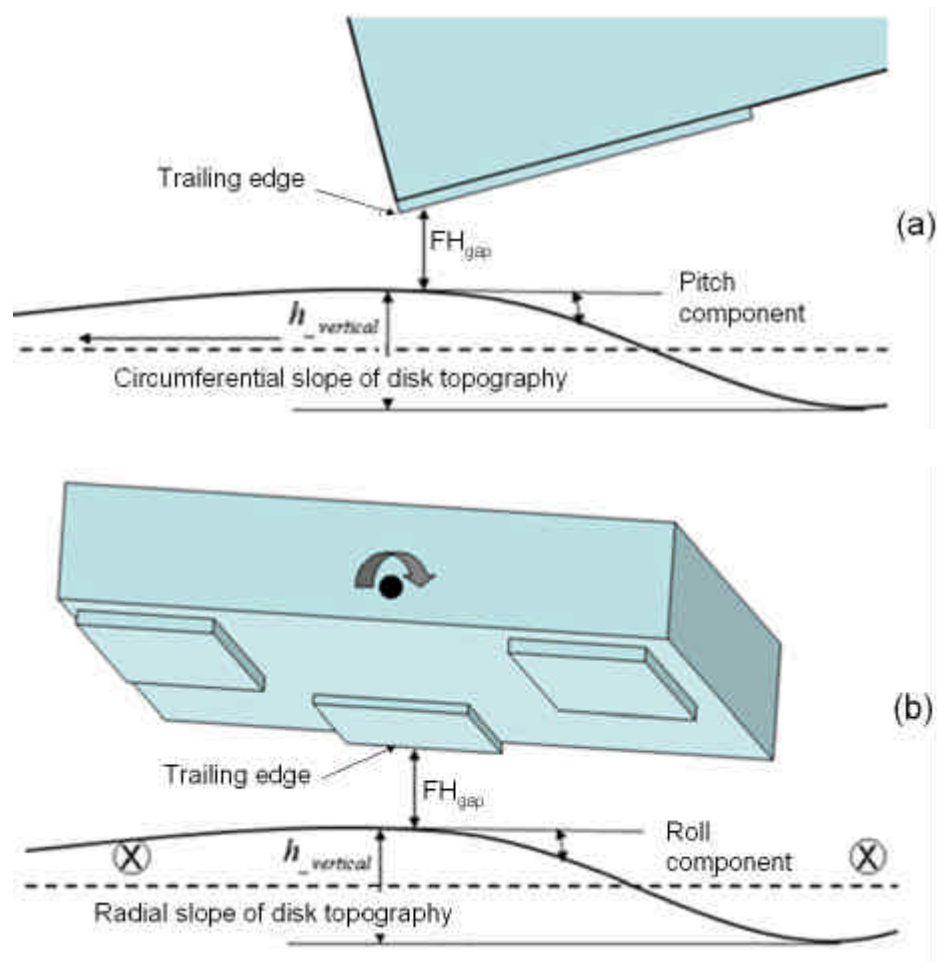


Figure 6.1 Schematic view of the effect of circumferential and radial components of disk topography on the slider vibrations.

In this section, we define the “roll” component of the disk topography as the radial slope of the disk topography, and the “pitch” component of the disk topography as the

circumferential slope of the disk topography. We use the five degrees-of-freedom measurement system described in the previous section to study the effect of disk topography on the vertical, pitch, and roll vibration component of the slider. The gain of the slider, defined as the ratio of slider motion and the disk topography, is determined and the effect of frequency components of the disk topography on slider dynamics is investigated.

6.1.2 Experimental Conditions

Disks used in this experiment were “standard” thin film disks coated with a diamond-like-carbon overcoat for wear protection. The slider in this experiment has a flying height of 8 nm when flying on the radius of 25.4 mm of a disk rotating at 7200 rpm. The air bearing design of the slider is shown in Fig. 6.2. The gray areas of the slider in Fig. 2 are the “top” air bearing surface and the white areas are etched surfaces at a lower level than the “top” air bearing surface.

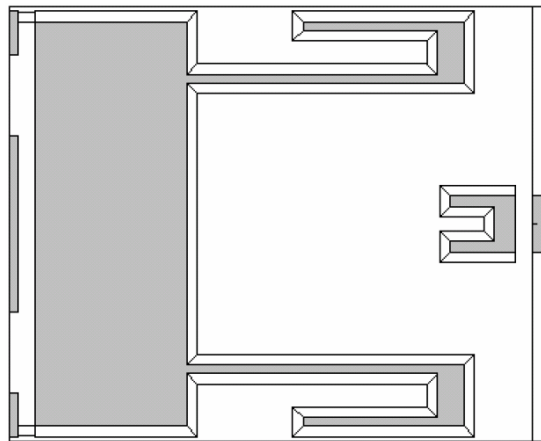


Figure 6.2 The slider air bearing design for the experiment.

As discussed in chapter 5 on the setup of our measurement system, the vertical laser Doppler vibrometer measures the vertical vibration of the slider and the position sensitive detector measures the angular change (pitch and roll) of the slider. If the slider is removed from the disk surface and the laser beam is directed onto the disk surface, the vertical, circumferential slope, and radial slope components of disk topography are obtained simultaneously.

The signals from the vertical laser Doppler vibrometer and the position sensitive detector were triggered using an index pulse from the air bearing spindle. The signals were averaged 25 times to reduce noise. In order to remove the effect caused by disk run-out and suspension vibrations, all signals were high pass filtered at a cut-off frequency of 10 kHz before they were captured. The signals for vertical pitch and roll motion were acquired using three channels of the digital oscilloscope at a sampling frequency at 10 MHz.

Our study is concerned with the excitation of slider air bearing frequencies by disk topography. Typically, air bearing frequencies are in the 50 kHz to 400 kHz range and frequencies below 50 kHz can be neglected in this analysis. Therefore, a high pass filter with a cut-off frequency of 47.9 kHz was applied to the captured signal.

A graph of the wavelength of disk waviness vs. frequency at a linear velocity of 19.1 m/s (slider flying at the radius of 25.4 mm of a disk rotating at 7200 rpm) is shown in Fig. 6.3. As shown in Fig. 6.3, as frequency increases, the wavelength of disk waviness decreases. Since we use a high pass filter with a cut-off frequency of 47.9 kHz, only disk waviness with wavelengths less than 400 μm is detected. The spot size of the laser beam on the disk surface was approximately 15 μm in diameter. As a result, only disk waviness with

wavelengths larger than $15\text{ }\mu\text{m}$ can be detected. Thus, the effective wavelength range of the signals studied in this analysis is from $15\text{ }\mu\text{m}$ to $400\text{ }\mu\text{m}$.

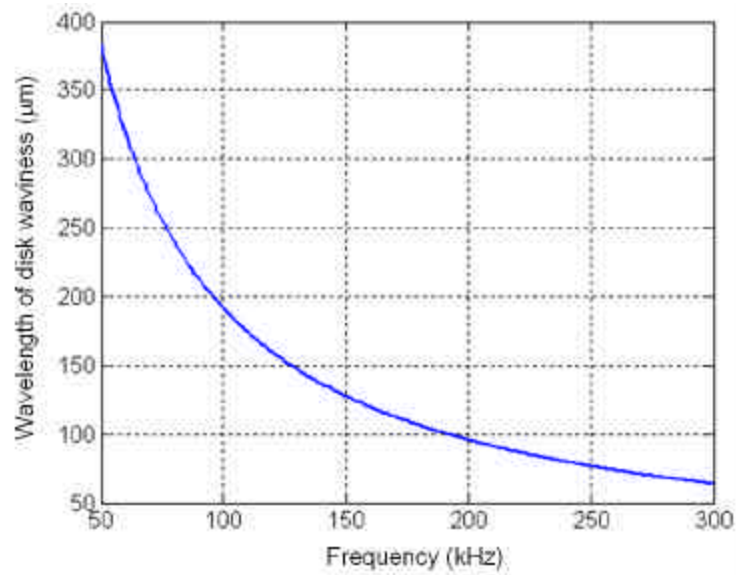


Figure 6.3 Wavelength of disk waviness vs. frequency at radius of 25.4 mm for disk rotating at 7200 rpm

6.1.3 Disk Topography and Slider Response in the Time Domain

In Fig. 6.4, waveforms of vertical, pitch, and roll components of disk topography and slider vibrations are shown for a slider flying on three different disks (denoted 1, 2, 3). In each case, the slider response is shown first on top with the corresponding disk topography shown below. The vertical components are displayed in Fig. 6.4a, the pitch components in Fig. 6.4b and the roll component in Fig. 6.4c.

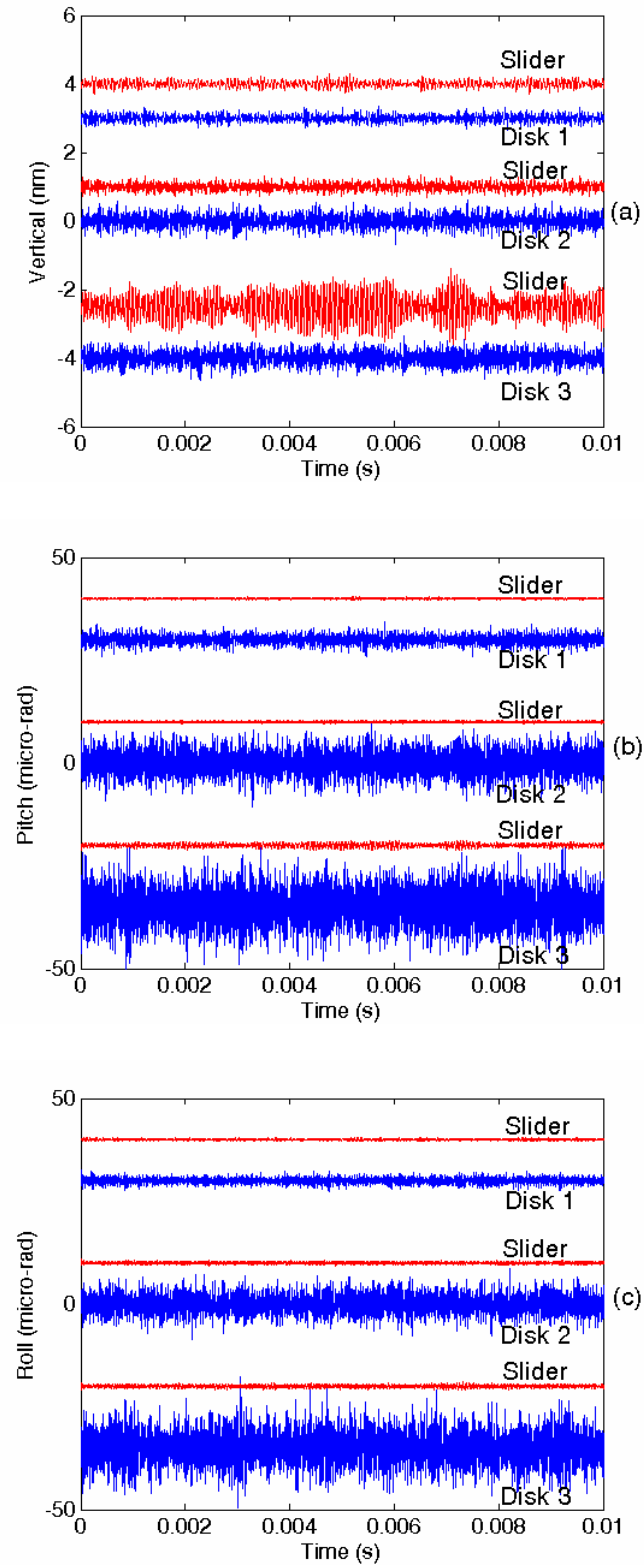


Figure 6.4 Waveforms of slider vibration and disk topography.

As can be observed from Fig. 6.4, the vertical, pitch and roll components of topography are highest for disk 3. On the other hand, the amplitudes for vertical, pitch and roll components of topography are lowest for disk 1. A similar trend was found for the amplitudes of slider vibrations: the highest amplitude was observed when flying on disk 3 and the lowest when flying on disk 1. The amplitudes for the pitch and roll components of slider vibrations were too small to be distinguishable.

Comparing the vertical component of slider vibrations (vertical response of the slider) for disk 3 with the vertical component of disk topography (top two curves of Fig. 6.4a), we observe that the amplitude of the slider response is larger than the amplitude of the disk, i.e. the slider amplifies the disk topography. However, the amplitude of the vertical component of the slider response is smaller than the amplitude of the disk for disk 2 and disk 3, i.e., the slider appears to suppress the effect of disk topography for disk 2 and disk 3. For the pitch component (Fig. 6.4b), the amplitudes of the slider response for all three disks is much smaller than the amplitudes of the disk topography. The same is observed for the roll component of the slider, shown in Fig. 6.4c. The results from Fig. 6.4 demonstrate that the vertical components of the disk topography affect the slider motion in the vertical direction. On the other hand, circumferential slope (“pitch”) and radial slope (“roll”) of the disk topography do not have a strong effect on slider vibrations.

These results can be interpreted as follows: the circumferential and radial slope of the disk topography affect the crown and cross-crown of the air bearing surface. Crown and cross-crown changes do not have as large an effect on slider vibrations as changes in the flying height.

6.1.4 Spectral Analysis of Disk Topography and Slider Response

The power spectral density of the vertical component of the disk topography is shown in Fig. 6.5a for disk 1, 2, and 3. The power spectral density of the pitch component is shown in Fig. 6.5b for disk 1, 2, and 3 and the power spectral density of the roll component is shown in Fig. 6.5c for disk 1, 2, and 3.

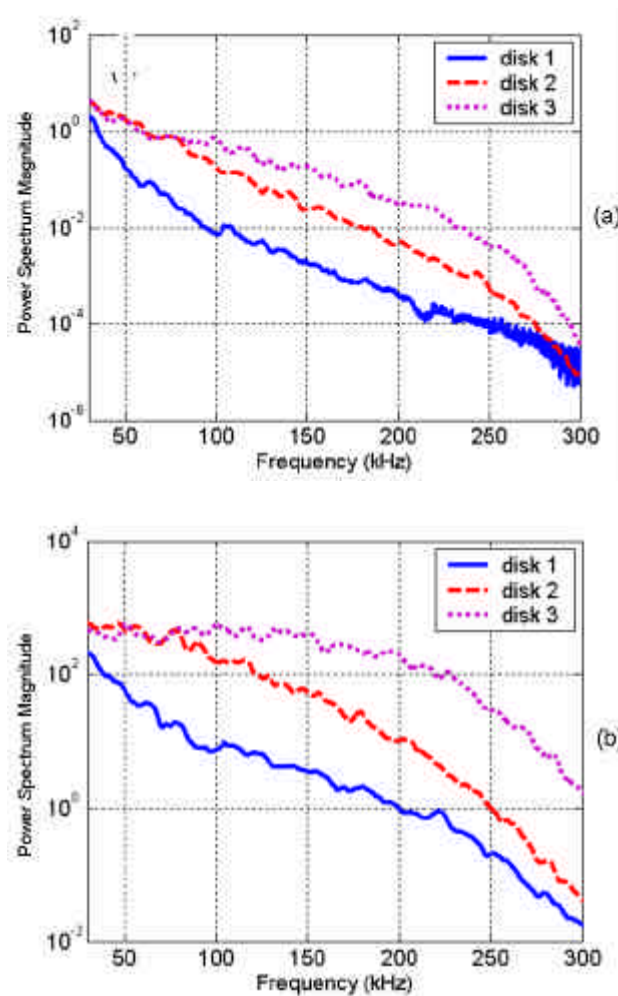


Figure 6.5 Power spectral densities of three components of topography of three different disks: (a) vertical component, (b) pitch component, and (c) roll component

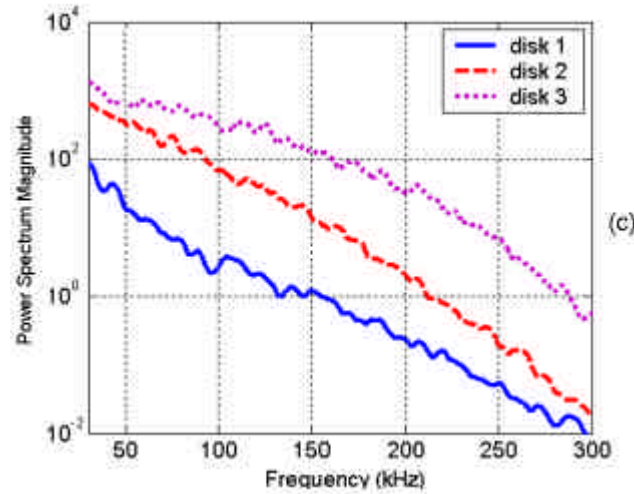


Figure 6.5 (Continued)

As can be observed from Fig. 6.5a, disk 1 exhibits the smallest amplitude throughout the displayed frequency range while disk 3 shows the largest amplitudes. A similar trend can be observed in Fig. 6.5b and Fig. 6.5c for the pitch and roll components of the three disks, respectively, i.e., the power spectral density of the circumferential and radial slope is lowest for disk 1, while it is the highest for disk 3. From these results it is understandable that substantial differences in the response of the slider should be expected.

Defining the gain of the slider as the ratio of the power spectral density of slider motion divided by the power spectral density of disk topography, we have plotted in Fig. 6.6 the gain of the slider for the vertical, pitch, and roll directions, respectively. A gain greater than one indicates that the slider amplifies disk topography, while a gain lower than one indicates the opposite.

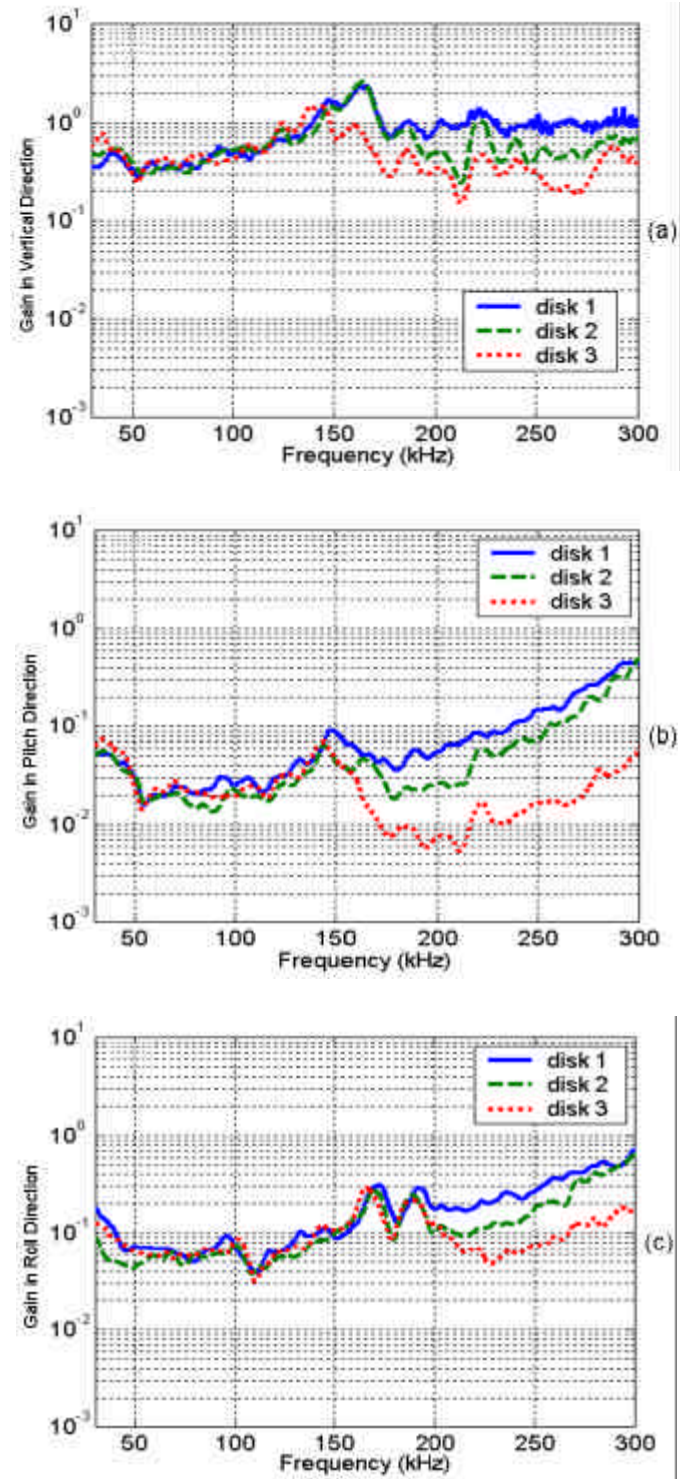


Figure 6.6 Slider gain for (a) vertical, (b) pitch, and (c) roll

Figure 6.6a shows that the gain of the slider in the vertical direction. The gain is close to one for the three disks over most of the frequency range. Two well-defined peaks are present in Fig. 6.6a at frequencies of 150 kHz and 165 kHz, respectively. At frequencies below 150 kHz, the gain of the slider for the three disks is nearly identical. However, at frequencies above 150 kHz, the gain of the slider is different for the three disks. In particular, at frequencies above 150 kHz, the gain of the slider is highest for disk 1 and lowest for disk 3.

Figure 6.6b shows the gain of the slider in the pitch direction for the three disks. We observe that the gain of the slider in the pitch direction does not approach “one” as was observed for the vertical motion. In the frequency range between 50 kHz and 150 kHz, the gain of the slider is very similar for all three disks. On the other hand, above 150 kHz, the gain of the slider is different for the three disks. It is highest for disk 1 and lowest for disk 3. A very similar behavior is found in Fig. 6.6c for the gain of the slider in the roll direction for the three disks.

These gain functions, which are analogous to transfer functions in vibration analysis [6.6], illustrate the role of resonance on the transfer of disk topography to slider motion. Below the first resonance, a single transfer function can accurately predict the slider motion induced by disk topography: i.e., the slider follows the disk. Above this frequency, no single transfer function applies. The physical reason for this is that stiffness and damping of the air bearing are functions of the head/disk separation. As the head/disk separation increases, the air bearing stiffness decreases [6.7]. Thus, if a slider with a given resonance frequency f_0 is subject to broadband excitation, the head/disk spacing fluctuates with

frequency f_0 . Input modulation at a frequency higher than f_0 is attenuated by the slider, i.e., the slider acts as a low-pass filter with a “cut-off” frequency equal to f_0 . Disk topography frequencies higher than f_0 are not “seen” by the slider.

6.1.5 Summary

We have measured three components of slider modulation for three disks with different amplitude profiles of topography. We observe that the slider reacts to the vertical component of disk topography but not to pitch and roll components. Defining gain as the ratio of slider motion over any of the components of disk topography, we find that a single transfer function can describe the slider response in the air bearing frequency range up to the first slider pitch frequency. Above this frequency, we find that the gain is different for disks with different surface topography.

6.2 Dynamics of Ultra Low Flying Sliders during Contact with a Lubricated Disk

6.2.1 Introduction

As the flying height of air bearing sliders is reduced, intermittent contact between flying slider and disk becomes increasingly more frequent. Several studies have been carried out in the past concerning the dynamic characteristics of low-flying sliders. Knigge et al. [6.8] studied slider/disk contact using laser Doppler vibrometer and acoustic emission sensors. They observed nonlinearities of air bearing frequencies using joint time frequency analysis. The resonant frequencies of the air bearing and the slider body were excited when

the slider contacts the disk. Similar results were also found by Harrison et. al [6.9], McMillan et al. [6.10], Kohira et al. [6.11] and Zeng et al. [6.12].

Instability of sliders at low flying heights has been investigated theoretically by Thornton and Boggy [6.13] as a function of van der Waals' forces. Matsuoka et al. [6.14] and Ono et al. [6.15] studied slider instabilities but concentrated on the effect of meniscus forces caused by the lubricant on the disk.

In addition, slider-lubricant transfer has been studied by a number of researchers. Pit et al. [6.16] observed lubricant pickup by a slider after flying on a lubricated disk surface. Ma et al. [6.17] observed so-called "corrugations" of the lubricant thickness in the "wear-track" of the slider. Marchon et al. [6.18] developed a model of lubricant transfer from the disk to the slider at steady-state. Dai et al. [6.19] observed lubricant "moguls" on the disk surface caused by the slider and proposed a model to explain the findings.

In spite of the above contributions to slider dynamics, none of the studies investigated transient effects during interactions of a slider with a lubricated disk. In this study, we will use the five degrees-of-freedom setup from chapter five to study the motion of a slider in the vertical, pitch, roll, down-track, and off-track directions due to slider-lubricant interactions. The transient effect of slider dynamics is analyzed by comparing the evolution of slider vibrations in 2 minute increments.

6.2.2 Experimental Conditions

Standard thin film disks were coated with a 3.5 nm thick diamond-like-carbon film. The disks were then lubricated with a 1.4 nm perfluoropolyether (PFPE) lubricant. The air

bearing slider design is shown in Fig. 6.7. It is the same as the one used in section 6.1.2. The nominal flying height of the slider is 4 nm at a velocity of 13.3 m/s (5000 rpm, radius 25.4 mm). At this speed, intermittent contacts between slider and disk are likely to occur.

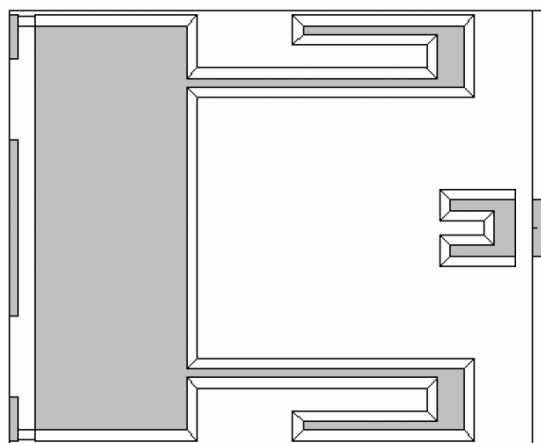


Figure 6.7 The slider air bearing design for the experiment

The LDV signals were triggered using an index pulse from the air-bearing spindle. The data were acquired using a digital oscilloscope at a sampling frequency of 10 MHz. All the data were high-pass filtered and captured for one disk revolution, which takes 12 ms at 5000 rpm.

6.2.3 Slider-lubricant Interactions in the Time Domain

A series of measurements for the vertical displacement of the slider during one revolution is shown in Fig. 6.8. The measurements were done on the same track and at an interval of 2 minutes up to 16 minutes. As shown in Fig. 6.8a, the vertical displacement of the slider at $t = 0$ is excited at five different locations along the track, marked A, B, C, D,

and E. After 2 minutes, only three excitations can be seen (Fig. 6.8b). The duration of each individual excitation is shorter at $t = 2$ minutes than at $t = 0$ minute. At $t = 4$ minutes, three excitations of the previous measurement merge into two and the position of these two excitations along the track has changed (Fig. 6.8c). At $t = 6$ minutes (Fig. 6.8d), the excitation at location B moves to a position further downstream on the track. At $t = 8$ minutes (Fig. 6.8e), the two excitations from the previous measurement are still observed. However, the excitations move even further in the downstream direction. At $t = 10$ minutes (Fig. 6.8f) and $t = 12$ minutes (Fig. 6.8g), only one excitation remains. Similar to the previous cases, the excitation has moved in the downstream direction of the track compared to the previous measurement. At $t = 14$ minutes (Fig. 6.8h) and $t = 16$ minutes (Fig. 6.8i), no excitations can be seen. Similar behaviors of slider-lubricant interactions with time were observed for excitations of the slider in the off-track and down-track directions, as well as the roll and pitch directions.

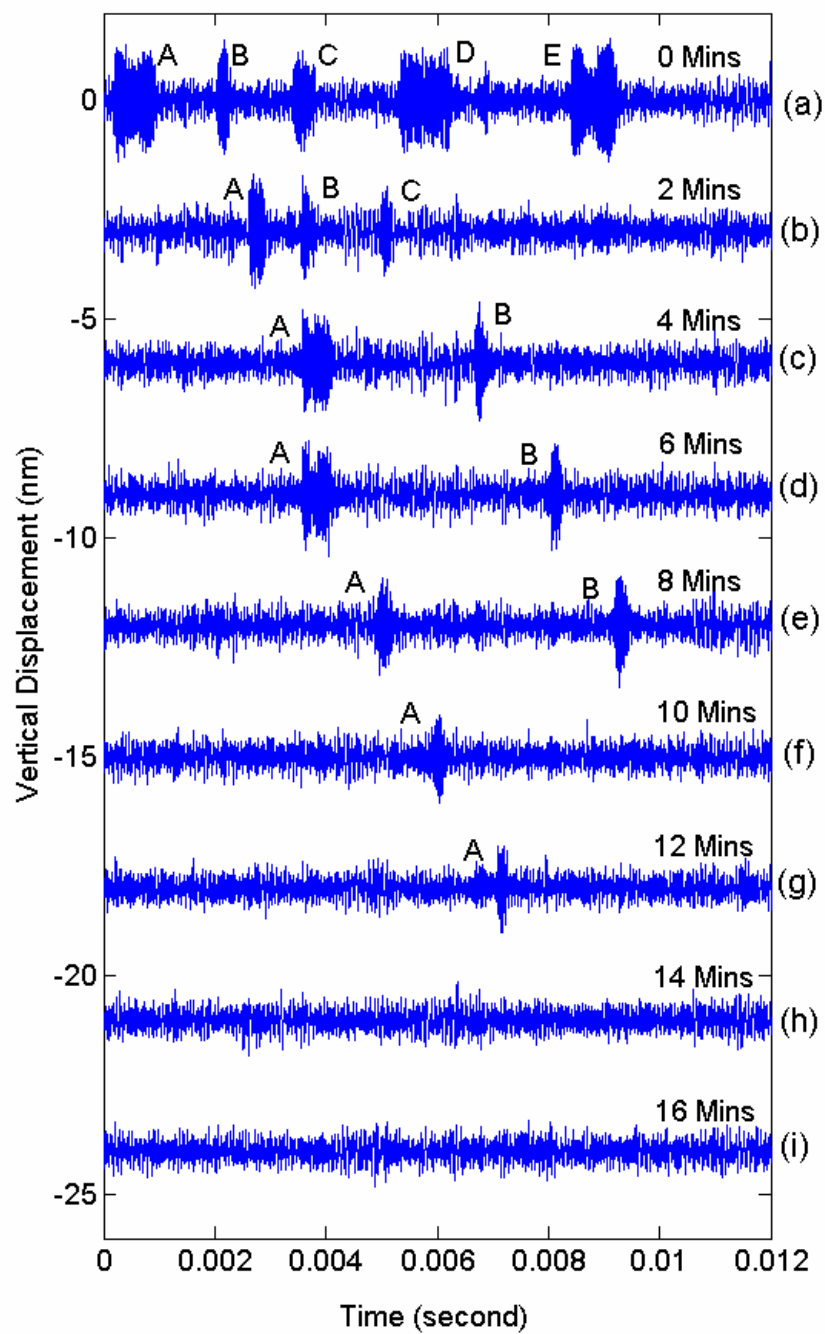


Figure 6.8 Vertical displacement of slider during slider-lubricant interactions at time intervals of $t = 2$ minutes

In order to investigate the tribological characteristics of slider-lubricant contacts, we have examined the disk surface with an optical surface analyzer after each experiment. The results from these measurements are shown in Fig. 6.9.



Figure 6.9 Images of disk after completion of experiment, obtained with optical surface analyzer (image on top is view of P-specula, image on bottom is view of Q-phase)

As shown in Fig. 6.9, the lubricant film thickness is reduced in the “wear-track” of the slider. Visual inspection of the image of the “wear-track” (Fig. 6.10a) with an optical surface analyzer indicates that the lubricant thickness has increased in the area adjacent to the “wear” track and that the carbon overcoat was not damaged. Furthermore, wear particles were absent in the “wear-track”. Inspection of the slider trailing edge pad using a microscope with 20X magnification (Fig. 6.10b) also indicates that no damage occurred to the slider overcoat.

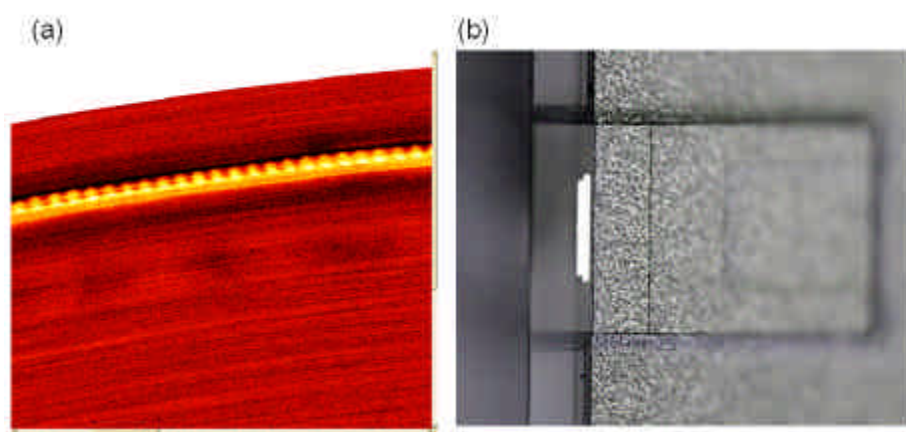


Figure 6.10 Image of the disk “wear-track” after completion of the experiment (left) and the slider trailing edge pad after completion of the experiment (right)

These results suggest that the slider excitations shown in Fig. 6.8 are due to slider-lubricant interactions rather than contacts between slider and disk. Clearly, the loss of lubricant in the slider “wear” track and the accumulation of lubricant at the track edges demonstrate that lubricant was redistributed during the experiment. In addition, a small amount of lubricant was found at the trailing edge pad of the slider when examining the slider air bearing surface after the experiment with a microscope.

6.2.4 Slider-lubricant Interactions in the Frequency Domain

To further investigate the slider-lubricant interactions, we have plotted the power spectral density of the slider displacement in the vertical, down-track, off-track, pitch, and roll directions, respectively, in Fig. 6.11. Measurements are shown for $t = 0$ minutes, $t = 8$ minutes and $t = 16$ minutes.

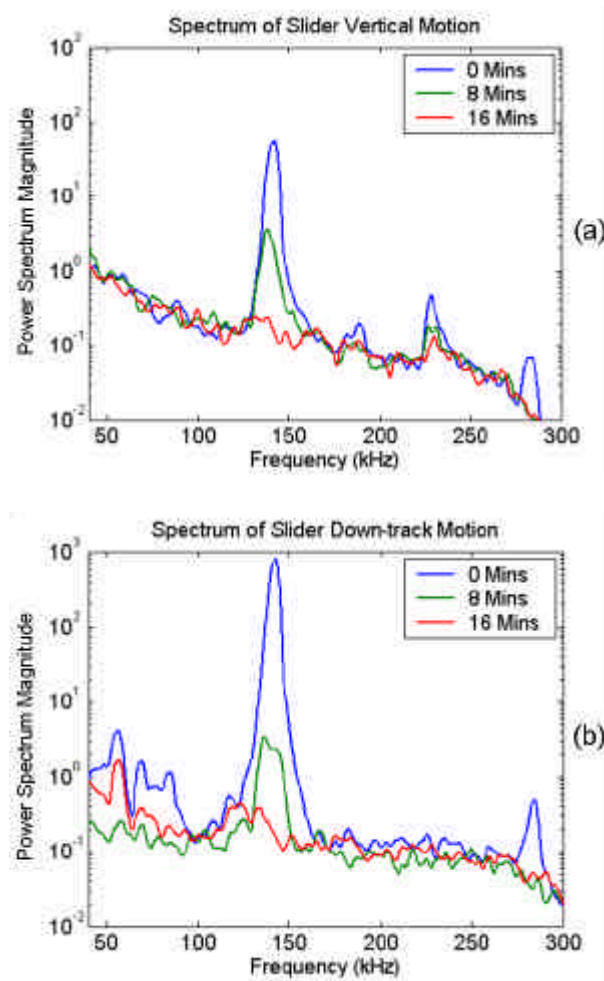


Figure 6.11 Power spectral density of slider displacement in (a) vertical, (b) down-track, (c) off-track, (d) pitch, and (e) roll direction

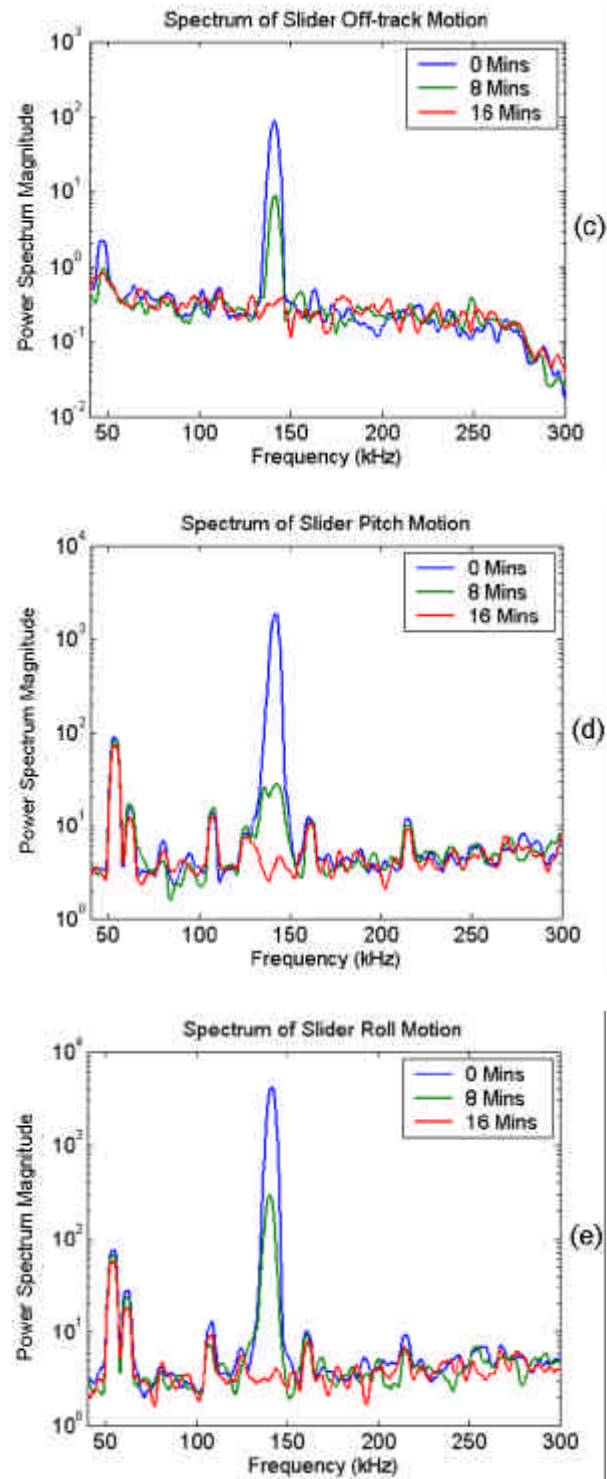


Figure 6.11 (Continued)

As shown in Fig. 6.11, at $t = 0$ minutes, the slider is strongly excited and the peak in the spectrum (142 kHz) is present for all five degrees of freedom, i.e., for the vertical (Fig. 6.11a), the down-track (Fig. 6.11b), the off-track (Fig. 6.11c), the pitch (Fig. 6.11d), and the roll (Fig. 6.11e) directions. It is interesting to note that this peak is at a slightly lower frequency than the numerically predicted pitch mode of the slider air bearing. It is justifiable to suggest that this result is related to meniscus formation at the slider-lubricant interface, causing damping of the free response of vibrations and lowering of the resonant frequency. In the spectrum of the vertical motion, we observe a shoulder on the right side of the 142 kHz peak. This shoulder is at a higher frequency and may correspond to the free excitation frequency (150 kHz); i.e., the peak at 142 kHz is a combination of free and damped vibrations. At $t = 8$ minutes, the amplitude of the peak at 142 kHz is one order of magnitude lower than the amplitude of the peaks at $t = 0$ minutes. Finally, at $t = 16$ minutes, all peaks disappear; i.e., the slider motion is no longer excited by slider-lubricant interactions. This suggests that lubricant in the slider “wear” track has been redistributed along the track.

6.2.5 Slider-lubricant Interactions Using Joint-time-frequency-analysis (JTFA)

Time frequency analysis of the slider displacement in the vertical direction is shown in Fig. 6.12. The measurements shown in Fig. 6.12 are for $t = 0$ minutes (Fig. 6.12a), $t = 8$ minutes (Fig. 6.12b), and $t = 16$ minutes (Fig. 6.12c), respectively. As already shown in Fig. 6.8 and Fig. 6.11, we observe that at $t = 0$ minutes the slider is excited at 142 kHz at five

different positions. At $t = 8$ minutes, we observe that only two peaks exhibit at 142 kHz. Finally, at $t = 16$ minutes, all peaks at 142 kHz have disappeared.

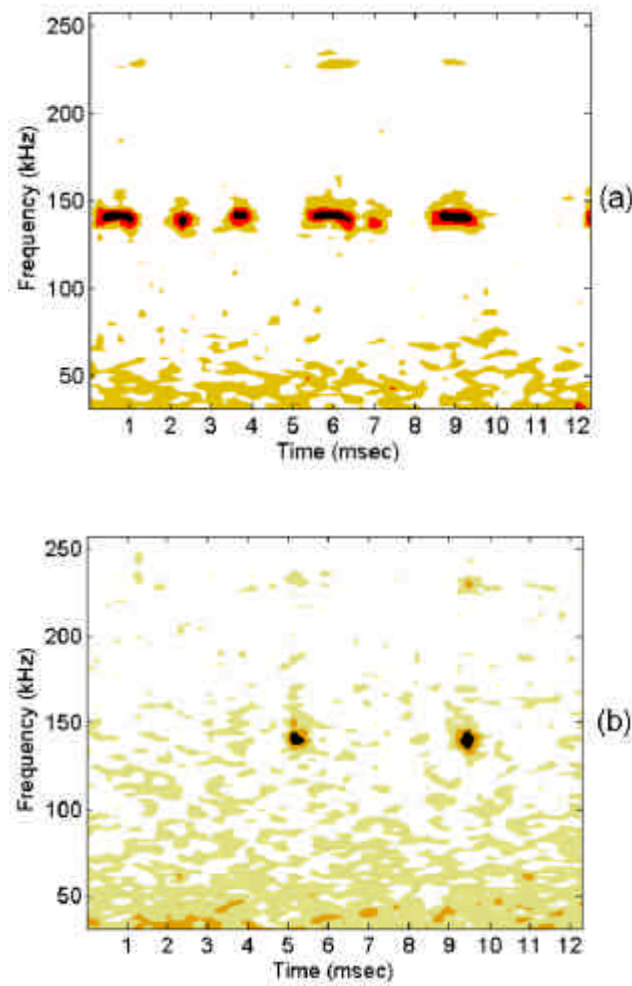


Figure 6.12 Time-frequency plot of slider displacement in vertical direction at time (a) $t = 0$ min, (b) $t = 8$ min, (c) $t = 16$ min

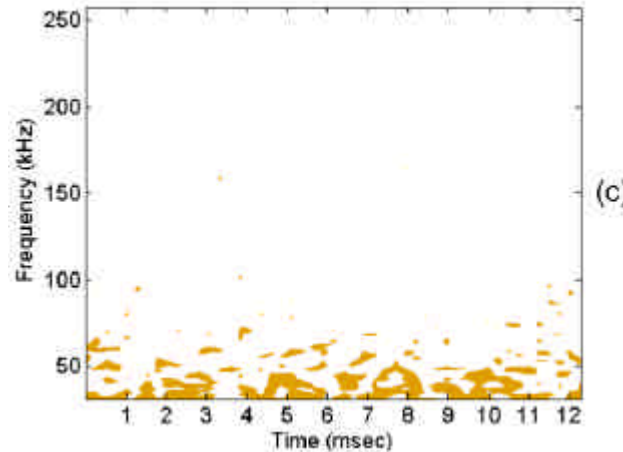


Figure 6.12 (Continued)

The results in Fig. 6.12 indicate that the slider is only excited when it contacts with the lubricant film. After the slider has resumed flying, excitations are absent. The results also show that the frequency of excitation (142 kHz) is at a lower frequency than the free vibration frequency of the slider (150 kHz).

6.2.6 Summary

We have studied the vertical, pitch, roll, off-track, and down-track components of slider motion during slider-lubricant interactions. The data suggest that slider-lubricant interactions at the head/disk interface have a strong effect on slider dynamics. We have observed that lubricant is redistributed along the track by the slider. Spectral analysis of the slider motion indicates that excitation occurs at the same frequency for all degrees of freedom. The excitation frequency is slightly lower than the free vibration frequency, suggesting that lubricant damping plays a role in the excitation of the slider.

6.3 Summary and Conclusion

We have measured three components of slider modulation for three disks with different amplitude profiles of topography. We have observed that the slider reacts to the vertical component of disk topography but not to pitch and roll components. Defining gain as the ratio of slider motion over any of the components of disk topography, we find that a single transfer function can describe the slider response in the air bearing frequency range up to the first slider pitch frequency. Above this frequency, we find that the gain is different for disks with different surface topography. No single transfer function applies across the entire frequency spectrum.

We have studied the vertical, pitch, roll, off-track, and down-track components of slider motion during slider-lubricant interactions. The data suggest that slider-lubricant interactions at the head/disk interface have a strong effect on slider dynamics. We have observed that lubricant is redistributed along the track by the slider. Spectral analysis of the slider motion indicates that excitation occurs at the same frequency for all degrees of freedom. The excitation frequency is slightly lower than the free vibration frequency, suggesting that lubricant damping plays a role in the excitation of the slider.

6.4 Acknowledgement

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Chapter 7 Study of Slider/Disk Contact Due to Current-Induced Thermal Pole Tip Protrusion

Ultra-high areal density for hard disk drives requires a stable head disk interface at a flying height lower than 8 nm. At such a low flying height, small flying height variations may cause slider/disk contacts. Slider/disk contacts can also occur when a write-current is applied to the write coil since the flying height between slider and disk can be affected by the thermal expansion of the pole tip. In general, this type of phenomenon is described as “thermal pole tip protrusion”.

In this chapter, we investigate the vibration characteristics of sliders during thermally induced contacts using laser Doppler vibrometry. We perform a parametric study of contact events using disks with different surface roughness and lubricant thicknesses, and analyze the slider motion statistically. For a given write current, we observe that the slider vibrations increase with disk roughness and lubricant thickness.

In addition, we simulate pole tip protrusion using finite element analysis. The protrusion profile is implemented into the air bearing surface and simulated using the CMRR finite element air bearing solver. Comparison of experimental and numerical results is discussed.

7.1 Introduction

During writing of information on the disk, current flows in the coil of the read/write head. The current generates heat that causes thermal expansion of the pole tip. The

expansion of the pole tip in turn, reduces the mechanical spacing between the slider and the disk. A schematic view of a slider flying on a rotating disk is shown in Fig. 7.1a. In Fig. 7.1b, the protrusion of the pole tip is shown due to the write current. Depending on the magnitude of the protrusion, slider/disk contacts can occur, resulting in pole tip damage and wear of the head/disk interface.

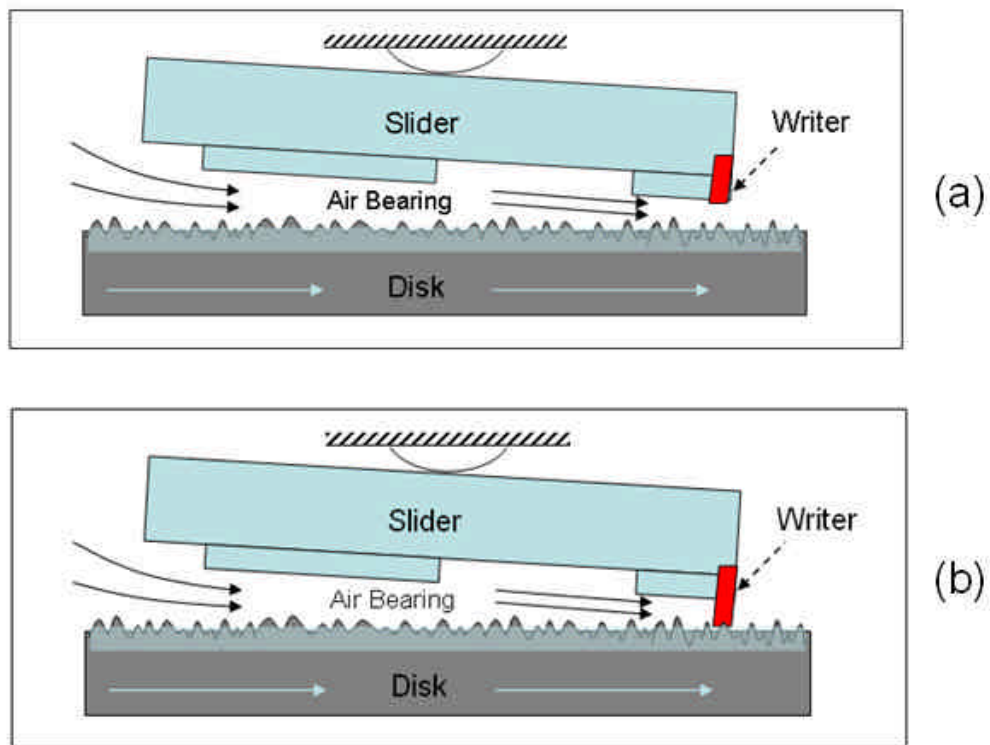


Figure 7.1 Schematic view of slider flying on a rotating disk at conditions a) without thermal pole tip protrusion, and b) with thermal pole tip protrusion

Current-induced thermal pole tip protrusion has been studied by a number of researchers. Tian et al. [7.1] investigated the read back signal disturbance caused by thermally induced flying height fluctuation. They observed that the electrical resistance of

the MR element was temperature dependant. Gupta et al. [7.2] studied the thermal pole tip protrusion with temperature and current in the write coil. They measured thermal pole tip protrusion using an optical profiler and an atomic force microscope and observed that thermal pole tip protrusion was affected by the material and structure of the read/write head. Pust et al. [7.3] simulated thermal pole tip protrusion using three-dimensional finite element analysis and observed that thermal stress in the shields of the head was caused by a mismatch in the coefficient of thermal expansion. They also found that during head operation, the primary heat source was the write coil. Imamura et al. [7.4] and Xu et al. [7.5] investigated the temperature rise in a slider when flying on a transparent disk using infrared microscopy. Kurita et al. [7.6] investigated the flying height reduction of a slider due to thermal pole tip protrusion and found that the protrusion of the pole tip is partially compensated by an increase in the air pressure on the air-bearing surface. Chekanov et al. [7.7] studied the effect of thermal pole tip protrusion on the recording performance. They observed that the overwrite degradation was caused by head-disk interactions due to thermal pole tip protrusion and that appropriate selection of disk lubricants could reduce the effect of thermal pole tip protrusion. Li et al. [7.8] simulated the heat transfer between slider and disk surfaces. They showed that the temperature distribution over the air bearing surface was asymmetrical and was related to the air bearing pressure and heat flux distribution.

In recent hard disk drives, thermal protrusion of the magnetic read/write element has been used to adjust flying height. Meyer et al. [7.9] discussed a heating system in a slider assembly to thermally control the position of the read/write element. Dietzel [7.10]

investigated a method to insert a resistive heater into the substrate of the slider. The expansion of the heater caused a change of the crown of the slider and, consequently, a change in the flying height of the slider. Kurita et al. [7.11] investigated an “active slider” with a nano-thermal actuator near the read/write element. The magnetic spacing of the slider can be controlled *in-situ* during drive operations. Suk et al. [7.12] experimentally verified that the magnetic and mechanical spacing could be controlled by thermally actuating the recording transducer. More recently, Juang and Bogy [7.13] investigated the relationship between air bearing surface design and thermal actuation of a slider. They introduced an efficient spacing adjustment design based on thermal actuation.

Although the above studies have clarified many aspects of current-induced thermal pole tip protrusion, none of them have addressed the dynamics of a slider in five degrees-of-freedom during contact caused by thermal pole tip protrusion. In this chapter, we investigate the dynamics of sliders during contacts caused by thermal pole tip protrusion using the experimental setup discussed in the previous chapter. A parametric study of contact events is performed with disks of different lubricant thickness and surface roughness, and the slider motion is analyzed statistically. The joint time frequency analysis is used to investigate the slider motion and to examine the characteristics of the slider motion due to thermal pole tip protrusion. The thermal pole tip protrusion was also simulated using finite element analysis. The protrusion profile is imported into the air bearing simulator by changing the air bearing surface. The simulation results are compared with experimental results.

7.2 Experimental Conditions

The disks used in this experiment were standard thin-film disks coated with a 3 nm-thick diamond-like-carbon overcoat for wear protection. The sliders used were typical “pico” type sliders with an air bearing surface as shown in Fig. 7.2. The nominal flying height of the slider is 6 nm at a linear velocity of 15.8 m/s and a zero skew angle.

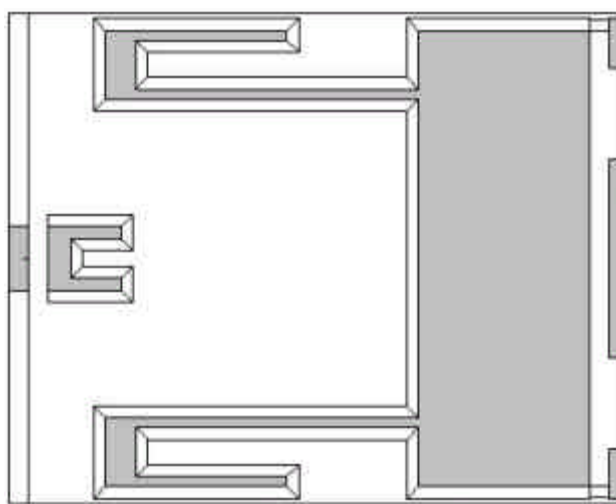


Figure 7.2 Air bearing slider used in the experiment

For all experiments reported in this section, the slider was connected to electronic circuitry with read/write capability. When the write channel is engaged, a current is applied to the slider write element at a pre-determined frequency and amplitude. Due to the resistance of the write element of the slider, a fraction of the electrical energy is converted into thermal energy. This thermal energy, in turn, causes the protrusion of the slider pole tip towards the disk surface. Depending on the design, contact between the slider and the disk

can occur due to the reduced slider/disk clearance. In our experiment, a current of 50 mA at 900 MHz was applied to the write element when the write channel was activated.

The motion of the slider is detected using the five degrees-of-freedom setup discussed in chapter five of this thesis. The signals from all five channels were triggered with the leading edge of the write current signal. We have captured twenty sequences for each measurement using a sampling rate of 5 MHz. The captured slider vibration data were high-pass filtered at 10 kHz to eliminate disk flutter and suspension dynamics effects.

7.3 Experimental Results and Discussion

7.3.1 Vibration characteristics of slider when write channel is engaged

In Fig. 7.3, slider vibrations are shown for the off-track, down-track, vertical, pitch, and roll directions. In addition, the signal from the write gate is shown on the top of the Fig. 7.3, indicating when the write channel is engaged.

From Fig. 7.3 we observe that the vibration amplitudes of the slider increase for all five degrees-of-freedom when the write gate is enabled (marked “On”). A delay of approximately 0.8 ms is seen to present between the engagement of the write channel and the onset of the first slider vibrations. The slider vibrations increase gradually, reaching a maximum value and decreasing slightly thereafter to a constant vibration amplitude. The slider shows a significant increase in vibration amplitude in the down-track (1.6 nm peak-to-peak), vertical (1.4 nm peak-to-peak) and pitch (2.3 μ rad peak-to-peak) degrees-of-freedom. The vibration amplitude in the off-track (0.5 nm peak-to-peak) and roll (0.9 μ rad peak-to-peak) directions is also higher when the write channel is turned “On”

as compared to being “Off”. As soon as the write channel is turned “Off”, the vibration amplitudes of the slider decrease for all five degrees-of-freedom and return to normal “flying” values.

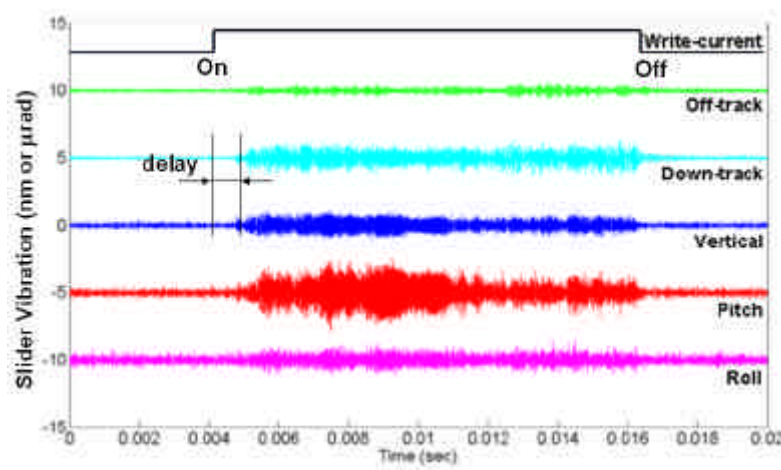


Figure 7.3 Slider vibrations for five degrees-of-freedom

In Fig. 7.4, plots of the joint time frequency results of the slider vibrations of Fig. 7.3 are shown. We observe that the slider is excited at 139 kHz in the down-track, vertical, pitch, and roll direction after the write channel is activated. In addition, distinct slider vibrations are observed at 150 kHz and 100 kHz for all five degrees-of-freedom. It is interesting to note that the largest frequency content is present in the down-track direction. The excited frequencies, between 100 kHz and 150 kHz, are in the range of typical slider air bearing frequencies.

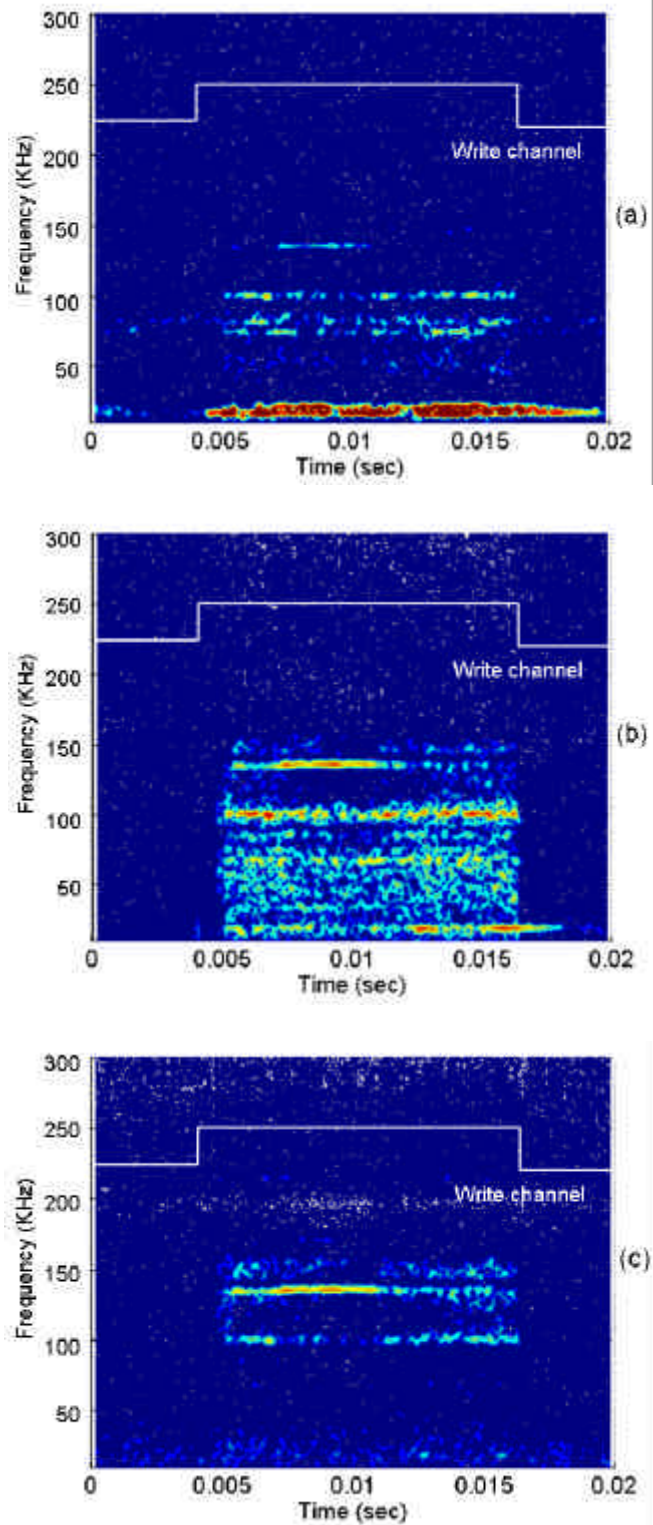


Figure 7.4 Joint time frequency analysis of slider vibrations in five degrees-of-freedom

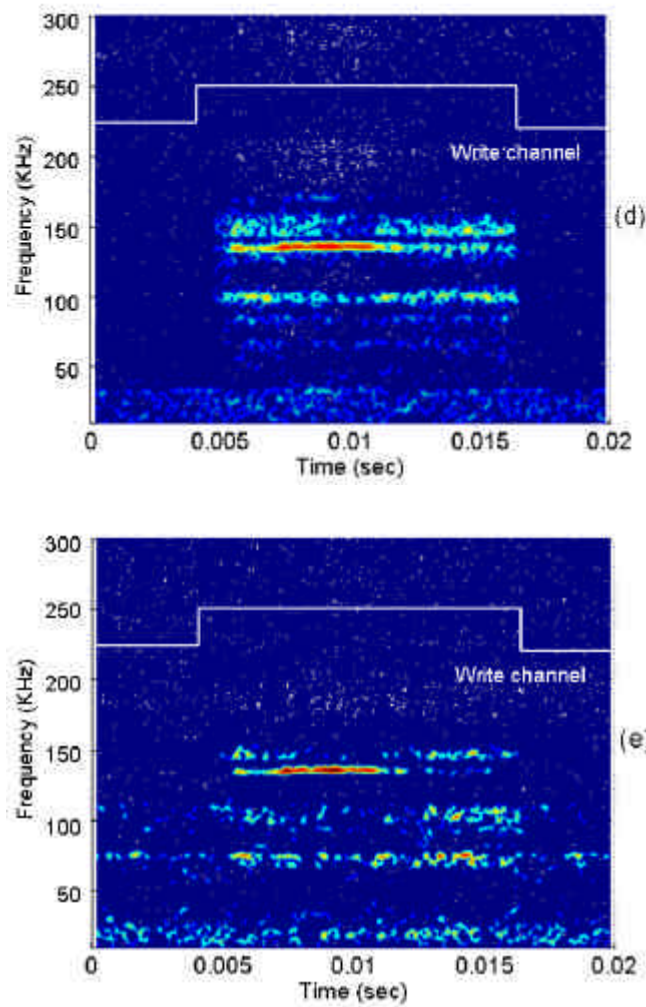


Figure 7.4 (Continued)

7.3.2 “Rough” disk vs. “smooth” disk

To study the effect of disk roughness on thermally induced pole tip contacts, we have repeated the experiments of the previous section with two types of disks that were identical in terms of thickness and lubricant type but were different in terms of surface roughness. As shown in Fig. 7.5, the root-mean-square (RMS) and roughness average (Ra) values of the two disks were 0.32 nm and 0.25 nm for the “smooth” disk, and 0.40 nm and 0.25 nm for the “rough” disk.

0.32 nm for the “rough” disk, respectively. Twelve sliders with the same air bearing design were selected and flown on both disks. The slider dynamics were measured using the previously discussed five degrees-of-freedom measurement system. For each measurement, the slider was positioned to fly on a “new” track and the write channel was engaged so that contact between head and disk occurred. In order to keep a constant linear velocity of 15.8 m/s, the disk rotational speed was adjusted based on the track radius. From each measurement, we have calculated the average peak-to-peak value of slider vibrations for the top ten percent of the total number of amplitude peaks.

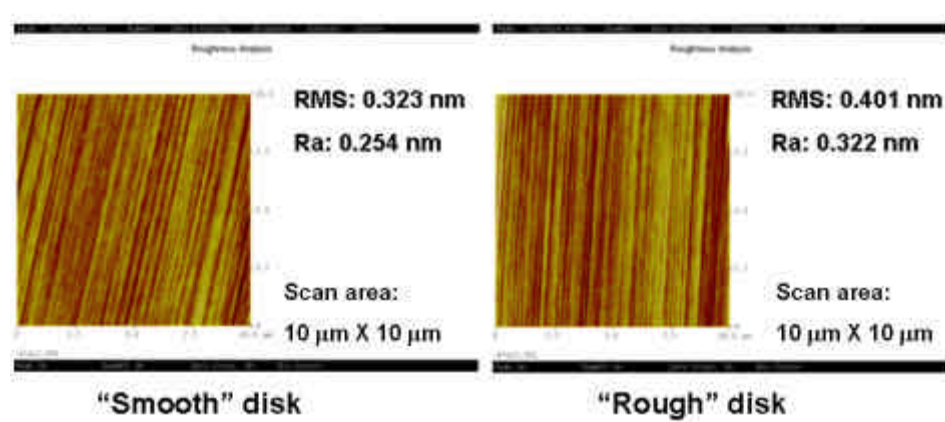


Figure 7.5 Disk surface topography measured using atomic force microscopy

The average peak-to-peak value of the slider vibrations for the two types of disks investigated is shown in Fig. 7.6. We have statistically analyzed the results obtained from the measurements and have calculated a confidence level for each measurement. As indicated in Fig. 7.6, the amplitude of slider vibrations in the down-track, vertical, pitch, and roll directions are higher for the “rough” disk than the “smooth” disk for all vibrations

excited by the same amount of thermal pole tip protrusion. The slider vibrations on the “rough” disk showed increases in amplitude of approximately 60% in the down-track direction, 30% in the vertical direction, 20% in the pitch direction, and 30% in the roll direction compared to the “smooth” disk. Thus, it is apparent that the disk surface roughness has a significant effect on the amplitude of slider vibration, i.e., the higher the surface roughness, the larger the vibration amplitude during slider disk contacts due to the same amount of thermal pole tip protrusion.

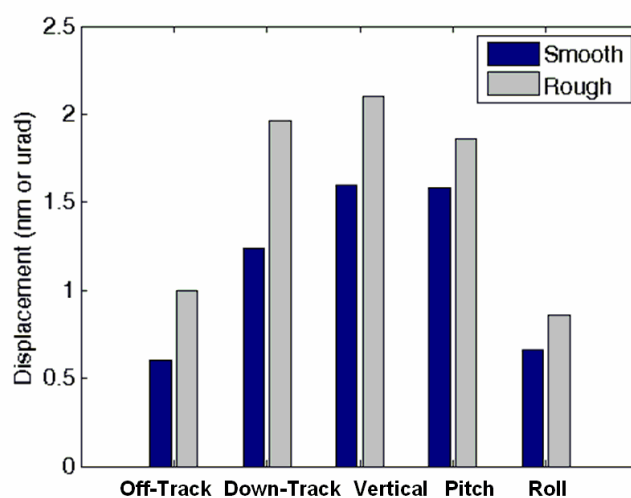


Figure 7.6 Effect of surface roughness on slider vibrations

7.3.3 Lubricated disk vs. de-lubricated disk

In this experiment, we have investigated the dynamic characteristics of contacts induced by thermal pole tip protrusion for lubricated and “de-lubricated” disks. Here, the term “de-lubricated” disks refers to disks that were originally lubricated but were subsequently submerged in a solvent to remove the lubricant layer. The original disk

surfaces were lubricated with a thin layer of perfluoropolyether (PFPE)-type lubricant. After submerging the disk twice into the solvent, the mobile lubricant layer was removed from the disk. The lubricant thickness of all disks was measured using a Fourier transform infrared (FTIR) spectrometer. The change in thickness between lubricated and de-lubricated disks was found to be about 0.9 nm. Twelve sliders with the same air bearing design were selected to be flown on both types of disks.

In Fig. 7.7, the average of slider vibration amplitudes is shown for a set of lubricated and de-lubricated disks. We observe that the amplitudes of lubricated disks are larger by approximately 30% in the off-track direction, 45% in the down-track direction, 20% in the vertical direction, and 50% in the pitch direction than those of the de-lubricated disks. It is apparent from this comparison that lubricant film thickness affects slider vibration amplitudes, i.e., a decrease in the lubricant thickness causes a reduction of vibration amplitudes for slider disk contacts due to thermal pole tip protrusion.

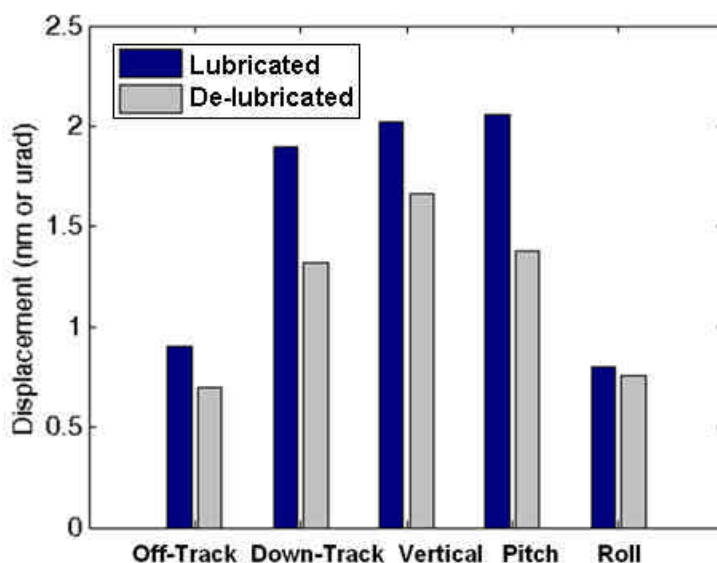


Figure 7.7 Effect of lubricant film thickness on slider vibrations

7.4 Modeling of Thermal Pole Tip Protrusion

7.4.1 Theoretical background

The heat generated by the write current can be expressed as

$$Q = \begin{cases} I_w^2 R_w, & \text{inside the domain of the write head} \\ 0, & \text{outside the domain of the write head} \end{cases} \quad (7.1)$$

where I_w is the current through the write coil, R_w is the resistance of the write coil.

The governing equation for steady heat transfer in a 3-D space is given by

$$\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(k \frac{\partial T}{\partial z} \right) + Q = 0 \quad (7.2)$$

where k is the thermal conductivity and T denotes the temperature as a function of x, y, z .

The temperature distribution of the write head is approached using the finite element method. We consider an element of the write head and approximate the problem as

[7.14]

$$\int_{V^e} N^T \left[\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(k \frac{\partial T}{\partial z} \right) + Q \right] dV = 0 \quad (7.3)$$

where V^e is the volume of the element and N is the shape function of the element. Using the Green-Gauss theorem, Eqn. 7.3 can be written as

$$\int_{S^e} N^T q_n dS - \int_{V^e} \left[\frac{\partial N^T}{\partial x} k \frac{\partial T}{\partial x} + \frac{\partial N^T}{\partial y} k \frac{\partial T}{\partial y} + \frac{\partial N^T}{\partial z} k \frac{\partial T}{\partial z} \right] dV + \int_{V^e} N^T Q dV = 0 \quad (7.4)$$

where S^e is the boundary surface of the element and q_n is the net heat flux from conduction.

q_n is given by

$$q_n = -k \frac{\partial T}{\partial x} n_x - k \frac{\partial T}{\partial y} n_y - k \frac{\partial T}{\partial z} n_z \quad (7.5)$$

where n_x , n_y and n_z the normal of the boundary surface in the x , y , and z directions, respectively. We need to consider the convection term due to air and the head flux at the boundary

$$q_n = q_{cv} - q_s \quad (7.6)$$

where q_{cv} and q_s denotes the convection heat flux and the heat flux at the boundary, respectively.

Thus, Eqn. 7.4 can be rewritten as

$$\int_{S^e} N^T (q_{cv} - q_s) dS - \int_{V^e} \left[\frac{\partial N^T}{\partial x} k \frac{\partial T}{\partial x} + \frac{\partial N^T}{\partial y} k \frac{\partial T}{\partial y} + \frac{\partial N^T}{\partial z} k \frac{\partial T}{\partial z} \right] dV + \int_{V^e} N^T Q dV = 0 \quad (7.7)$$

The temperature within an element can be represented by

$$T = NT^e \quad (7.8)$$

where N is the shape function of the element. Substituting Eqn. 7.8 into Eqn. 7.7, one obtains

$$\int_{V^e} \left[\frac{\partial N^T}{\partial x} k \frac{\partial N}{\partial x} + \frac{\partial N^T}{\partial y} k \frac{\partial N}{\partial y} + \frac{\partial N^T}{\partial z} k \frac{\partial N}{\partial z} \right] dV T^e = \int_{V^e} N^T Q dV + \int_{S^e} N^T (q_{cv} - q_s) dS \quad (7.9)$$

Defining the composite element stiffness matrix K^e and the composite element nodal force vector F^e as

$$K^e = \int_{V^e} \left[\frac{\partial N^T}{\partial x} k \frac{\partial N}{\partial x} + \frac{\partial N^T}{\partial y} k \frac{\partial N}{\partial y} + \frac{\partial N^T}{\partial z} k \frac{\partial N}{\partial z} \right] dV$$

and

$$F^e = \int_{V^e} N^T Q dV + \int_{S^e} N^T (q_{cv} - q_s) dS, \quad (7.10)$$

and substituting Eqn. 7.10 into Eqn. 7.9, one obtains

$$K^e T^e = F^e \quad (7.11)$$

For our case, the convection heat flux due to air is given by

$$q_{cv} = \begin{cases} h(T - T_s) & \text{surface adjacent to air} \\ 0, & \text{surface not adjacent to air} \end{cases} \quad (7.12)$$

where h is the thermal conductivity of air and T_s is the temperature of the air surrounding the head.

A change in temperature DT causes thermal stress σ given by

$$s = b\Delta TE \quad (7.13)$$

where b is the thermal expansion coefficient and E is Young's modulus.

A magnetic head/slider is comprised of different materials. These materials have different thermal expansion coefficients. Using finite element analysis for both the thermal and the structure problem [7.14], one can obtain the thermal protrusion of the pole tip of a magnetic head due to a given write current.

7.4.2 Simulation of Thermal Pole Tip Protrusion

In order to simulate slider vibrations caused by thermal pole tip protrusion, we first have to determine the pole tip protrusion using finite element analysis [7.14]. A schematic view of the model for a write head used in the simulation is shown in Fig. 7.8. The current applied to the coil is 50 mA. The electrical resistance of the write coil is temperature dependent as observed by Tian et al. [7.1]. The relationship between the electrical resistance change DR and the MR temperature rise DT is $DR = aR_0DT$, where R_0 is the initial

resistance ($12.5 \, \Omega$) and α is the temperature coefficient ($0.00239 \, \text{K}^{-1}$) [7.1]. The material for the base of the write head is Al_2O_3 (Young's modulus 390 GPa, thermal conductivity 35 W/mK, thermal expansion coefficient $9.0 \times 10^{-6} \, /\text{K}$). The material for the write coil is Cu (Young's modulus 117 GPa, thermal conductivity 387 W/mK, thermal expansion coefficient $16.5 \times 10^{-6} \, /\text{K}$). The material for the pole is Ni-Fe (Young's modulus 207 GPa, thermal conductivity 35 W/mK, thermal expansion coefficient $14.0 \times 10^{-6} \, /\text{K}$). The thermal conductivity for air is 0.026 W/mK. The temperature is assumed to be 300 K based on the fact that our experiment is performed on a spin-stand in a normal laboratory environment. The material properties listed above have previously been used in the literature [7.2].

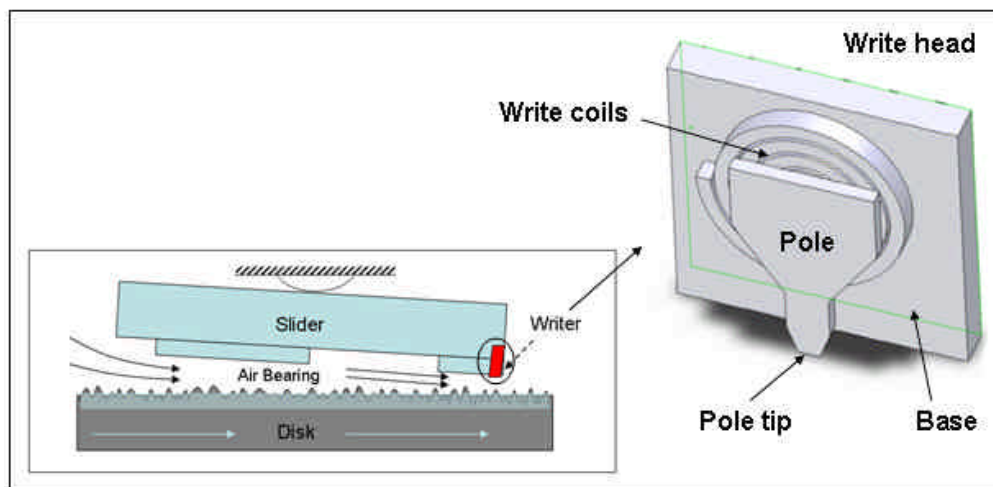


Figure 7.8 Schematic view of the write head

The model was meshed and the temperature and displacement were calculated using a commercially available finite element analysis package (ANSYS). A typical result for the simulated pole tip protrusion is shown in Fig. 7.9. The left figure shows the write

coil, the pole tip, and the head support structure. The figure denotes the finite element mesh over the head. On the right side of Fig. 7.9, an enlarged view of the pole tip protrusion is shown. We observe that the pole tip protrudes below the bottom surface of the head. The dimensions of the pole tip are $4\text{ }\mu\text{m}$ in length and $4\text{ }\mu\text{m}$ in width. For a given current of 50 mA, the pole tip was found to protrude approximately 5 nm.

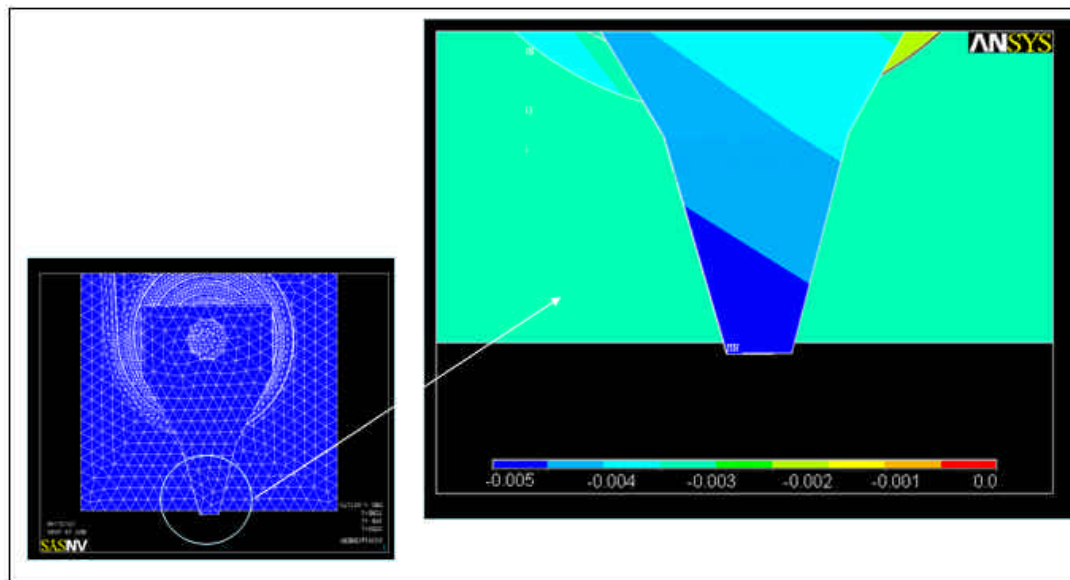


Figure 7.9 Simulation model and simulated pole tip protrusion

A plot of pole tip protrusion versus write current is shown in Fig. 7.10. We observe that an increase in write current resulted in an increase in the pole tip protrusion.

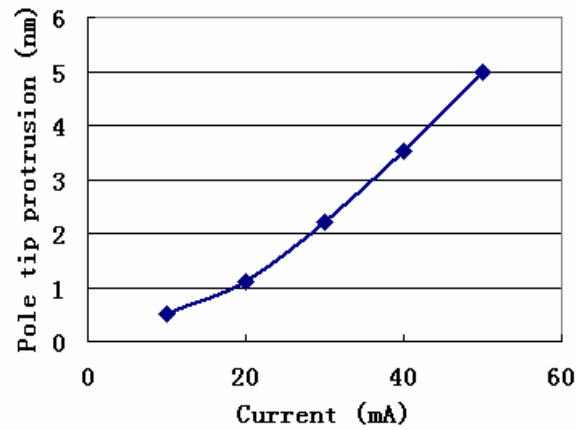


Figure 7.10 Pole tip protrusion versus write current

The protrusion profile of the pole tip due to the applied current was superimposed on the slider air bearing surface and the pressure distribution over the slider was re-calculated using the CMRR air bearing simulator. In Fig. 7.11a, the pressure profile of the slider air bearing is shown, recalculated after pole tip protrusion. The pressure distribution prior to pole tip protrusion is shown in Fig. 7.11b. We observe that a pressure peak is present over the protrusion. The spacing of the pole tip is 3.5 nm to the disk surface.

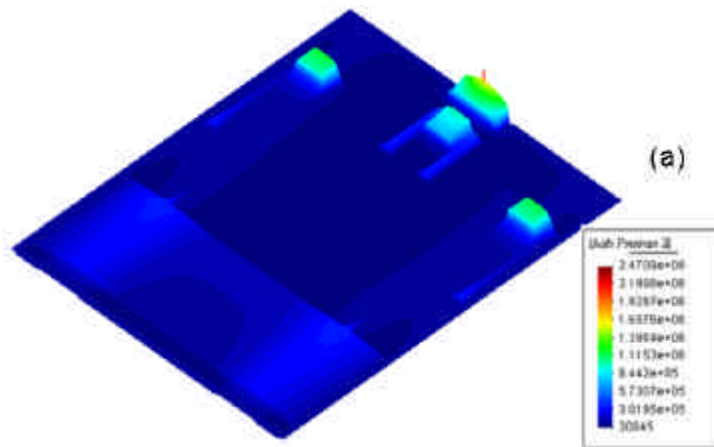


Figure 7.11 Pressure profile of the slider with pole tip protrusion

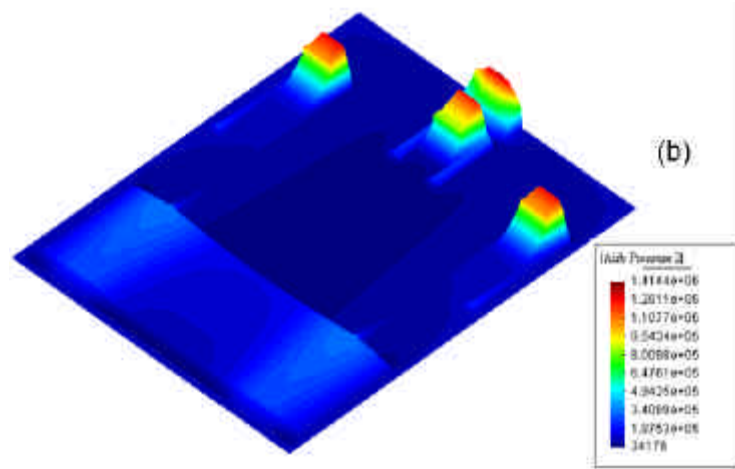


Figure 7.11 (Continued)

7.5 Numerical Simulation of Slider Vibration in Five Degrees-of-freedom

7.5.1 Coordinate System

The coordinate system used for describing the slider motion is shown in Fig. 7.12. The origin of the coordinate system is located at the center of the slider. The z -axis denotes the slider vertical displacement; the x -axis denotes the slider down-track displacement; the y -axis denotes the slider off-track displacement. In addition, q denotes the rotation of the slider around the x -axis (roll) and j denotes the rotation around the y -axis (pitch). Positive values for the quantities are shown in the directions as indicated by the arrows.

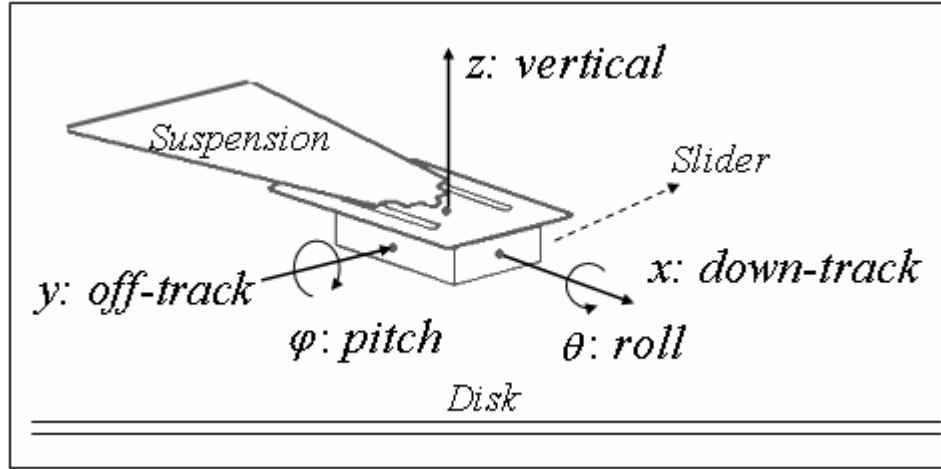


Figure 7.12 Definition of coordinate system for slider vibrations

7.5.2 Equation of Motion for the Slider

The equations representing the slider motion for the five degrees-of-freedom are

$$\begin{cases} M\ddot{x} + C_x\dot{x} + K_x x = f_{x_shear} + f_{x_fric} + 2w_{susp}\dot{x}_c + f_{centripetal} \\ M\ddot{y} + C_y\dot{y} + K_y y = f_{y_shear} + f_{y_fric} + 2w_{susp}\dot{y}_c + f_{frameacceleration} \\ M\ddot{z} + C_z\dot{z} + K_z z = f_{air_bearing} + f_{contact} - f_{gram_load} \\ J_q\ddot{q} + C_q\dot{q} + K_q q = M_{q_air_bearing} + (k_z z + f_{load})(x_f - x_g) + M_{q_shear_cont_fric} \\ J_j\ddot{j} + C_j\dot{j} + K_j j = M_{j_air_bearing} + (k_z z + f_{load})(y_f - y_g) + M_{j_shear_cont_fric} \end{cases} \quad (7.14)$$

where C and K denote damping and stiffness coefficients; M and J denote effective mass and moment of inertia; f_{x_fric} and f_{y_fric} are frictional forces in the down-track and off-track directions. The terms $2w_{susp}\dot{x}_c$ and $2w_{susp}\dot{y}_c$ denote the Coriolis forces in the down-track and off-track direction, respectively. Furthermore, the terms $(k_z z + f_{load})(x_f - x_g)$ and $(k_z z + f_{load})(y_f - y_g)$ are the torque in the roll and pitch direction due to the offset between the load point and the center of gravity of the slider. The

terms $M_{q_air_bearing}$ and $M_{j_air_bearing}$ are the moment in the roll and pitch direction, respectively, due to the air bearing force in the z direction. The terms $M_{q_shear_cont_fric}$ and $M_{j_shear_cont_fric}$ are the moments in the roll and pitch direction, respectively, due to the air bearing shear force and the contact, adhesion and friction forces.

To determine the slider motion, values for the air bearing stiffness, the damping coefficients and the shear force are required. These values can be obtained from a solution of the time-dependent Reynolds equation and slider dynamics equations. Wahl [7.15] solved the time-dependent generalized Reynolds equation using the finite element method. Weissner [7.16] extended the simulation by including a suspension/arm model using beam elements and calculated slider motion for three degrees-of-freedom (vertical, pitch and roll). Jayson et al. [7.17] modeled the slider dynamic response for flying over a bump with the same finite element simulation code. Hu et al. [7.18] developed a control volume scheme to analyze numerically shaped-rail sub-ambient pressure sliders at high speed and ultra-low spacing.

In our simulation code, the numerical solution of the time-dependent Reynolds equation is obtained using a method similar to that developed by Wahl [7.15] and Weissner [7.19]. Their original solution was limited to three degrees-of-freedom. In order to simulate the slider motion for five degrees-of-freedom, the air bearing shear force as well as contact, adhesive and frictional force has to be considered.

7.5.3 Calculation of Air Bearing Shear Forces

In order to calculate the motion of a slider in the down-track (x) and off-track (y) direction, respectively, the air bearing shear force for the x - and y - direction has to be calculated. The shear force is obtained by integration of the shear stress, i.e.,

$$\begin{cases} f_{x_shear} = \iint_{ABS} t_x dA \\ f_{y_shear} = \iint_{ABS} t_y dA \end{cases} \quad (7.15)$$

where ABS denotes the air bearing surface, and t_x and t_y are the shear stresses in the x - and y - direction, respectively.

The shear stress can be calculated from

$$\begin{cases} t_x = \mu \left. \frac{\partial u}{\partial z} \right|_{z=h} \\ t_y = \mu \left. \frac{\partial v}{\partial z} \right|_{z=h} \end{cases} \quad (7.16)$$

where μ is the viscosity, h is the film thickness, and u and v are the velocity components of air film in the x - and y - direction, respectively.

With “no-slip” condition at the fluid-solid interface, the velocity components can be calculated as

$$\begin{cases} u = \frac{1}{2\mu} \frac{\partial p}{\partial x} [z^2 - hz] + U \left[1 - \frac{z}{h} \right] \\ v = \frac{1}{2\mu} \frac{\partial p}{\partial y} [z^2 - hz] + V \left[1 - \frac{z}{h} \right] \end{cases} \quad (7.17)$$

where p is the air bearing pressure, while U and V are the linear speed of the disk surface in the x - and y - direction, respectively.

If the flying height of the slider approaches several nano-meters, rarefaction effects have to be considered. Kang et al. [7.20] calculated the shear stress term including the rarefaction effects at the fluid-solid interface. Their method was verified by Fukui and Yamane [7.21]. The shear stress terms for the x - and y - direction can be expressed as follows

$$\begin{cases} t_x = \left(\frac{h}{2} \frac{\partial p}{\partial x} w_p - \frac{mU}{h} w_c \right) \\ t_y = \left(\frac{h}{2} \frac{\partial p}{\partial y} w_p - \frac{mV}{h} w_c \right) \end{cases} \quad (7.18)$$

where w_p and w_c are the shear stress coefficients introduced to account for rarefaction effects for Couette and Poiseuille flow, respectively. The values of w_p and w_c can be found in Kang [7.22].

Substituting the shear stress terms in Eqn. 7.15 by Eqn 7.18, the shear forces in the x - and y - direction can be obtained from

$$\begin{cases} f_{x_shear} = \iint_{ABS} \left(\frac{h}{2} \frac{\partial p}{\partial x} w_p - \frac{mU}{h} w_c \right) dA \\ f_{y_shear} = \iint_{ABS} \left(\frac{h}{2} \frac{\partial p}{\partial y} w_p - \frac{mV}{h} w_c \right) dA \end{cases} \quad (7.19)$$

Eqn. 7.19 is used in the CMRR air bearing simulation.

7.5.4 Calculation of Contact, Adhesion and Friction Forces

At low flying heights, contacts between slider and disk are likely to occur. A review of the contact models developed by a number of investigators was given in chapter 2. In this thesis a contact model for rough surface developed by Kogut and Etion is used [7.23]. In

order to make this model suitable for the simulation of the head/disk interface, Suh and Polycarpou [7.24] added the effect of lubricant. A brief description of this model is given in Appendix C.

The forces and moments used in the dynamic equation are

$$\left\{ \begin{array}{l} f_{x_fric} = \iint_{ABS} Q(h, dA) \cos(f) dA \\ f_{y_fric} = \iint_{ABS} Q(h, dA) \sin(f) dA \\ f_{contact} = \iint_{ABS} [P(h, dA) - F_s(h, dA)] dA \\ M_{q_shear_cont_fric} = \iint_{ABS} \left\{ [P(h, dA) - F_s(h, dA)](y - y_g) + [Q(h, dA) \sin(f) + t_y] \frac{t}{2} \right\} dA \\ M_{j_shear_cont_fric} = \iint_{ABS} \left\{ [P(h, dA) - F_s(h, dA)](x - x_g) + [Q(h, dA) \cos(f) + t_x] \frac{t}{2} \right\} dA \end{array} \right. \quad (7.20)$$

where Q is the friction force, h is the separation from the element with area dA to the disk surface, f is the skew angle, P is the contact force, F_s is the adhesion force, x_g and y_g are the coordinate of the center of gravity of the slider, and t is the thickness of the slider.

7.5.5 Numerical Solution of the Coupled Slider Dynamics Equation and the Reynolds Equation

In Eqn. 7.7, the displacement and the air bearing pressure are functions of time. Thus, the time-dependent Reynolds equation and the equations of motion of the slider are coupled and need to be solved simultaneously. According to the contact model discussed in Appendix C, the contact force, adhesive force and frictional force are functions of the separation between slider and disk. At each time step, the slider dynamics equation (Eqn 7.7) is solved using the Newmark method [7.25]. The slider vibrations are calculated using

a step size of $1\ \mu\text{s}$ and a convergence criterion of 10^{-3} . A flow chart for the simultaneous solution of the time-dependent Reynolds equation and the equations of motion is shown in Appendix D.

7.6 Simulation Results for Slider Vibrations

Typical simulation results for slider vibrations of a thermally activated slider are shown in Fig. 7.13 for a velocity of $15.8\ \text{m/s}$. The suspension stiffness in the vertical direction was taken as $15.9\ \text{N/m}$, in the down-track direction as $1.4\text{e}4\ \text{N/m}$, and in the off-track direction as $1.5\text{e}3\ \text{N/m}$. The nominal flying height of the slider with pole tip protrusion is $3.5\ \text{nm}$. In order to compare these results with experimental measurements, only stabilized vibrations after $0.45\ \text{ms}$ are shown. For the simulation we assume a lubricant layer of $1\ \text{nm}$ thickness and a disk waviness of $0.32\ \text{nm rms}$. As can be observed from Fig. 7.13b, the peak-to-peak vibration amplitude of the slider is $1.0\ \text{nm}$ in the vertical direction. The peak-to-peak vibration amplitude is $2.2\ \mu\text{rad}$ in the pitch direction (Fig. 7.13c) and the peak-to-peak vibration amplitude in the down-track direction is $1.0\ \text{nm}$ (Fig. 7.13f). The peak-to-peak vibration amplitudes in the roll direction (Fig. 7.13d) is less than $0.1\ \mu\text{rad}$ and in the off-track direction (Fig. 7.13e) is less than $0.1\ \text{nm}$.

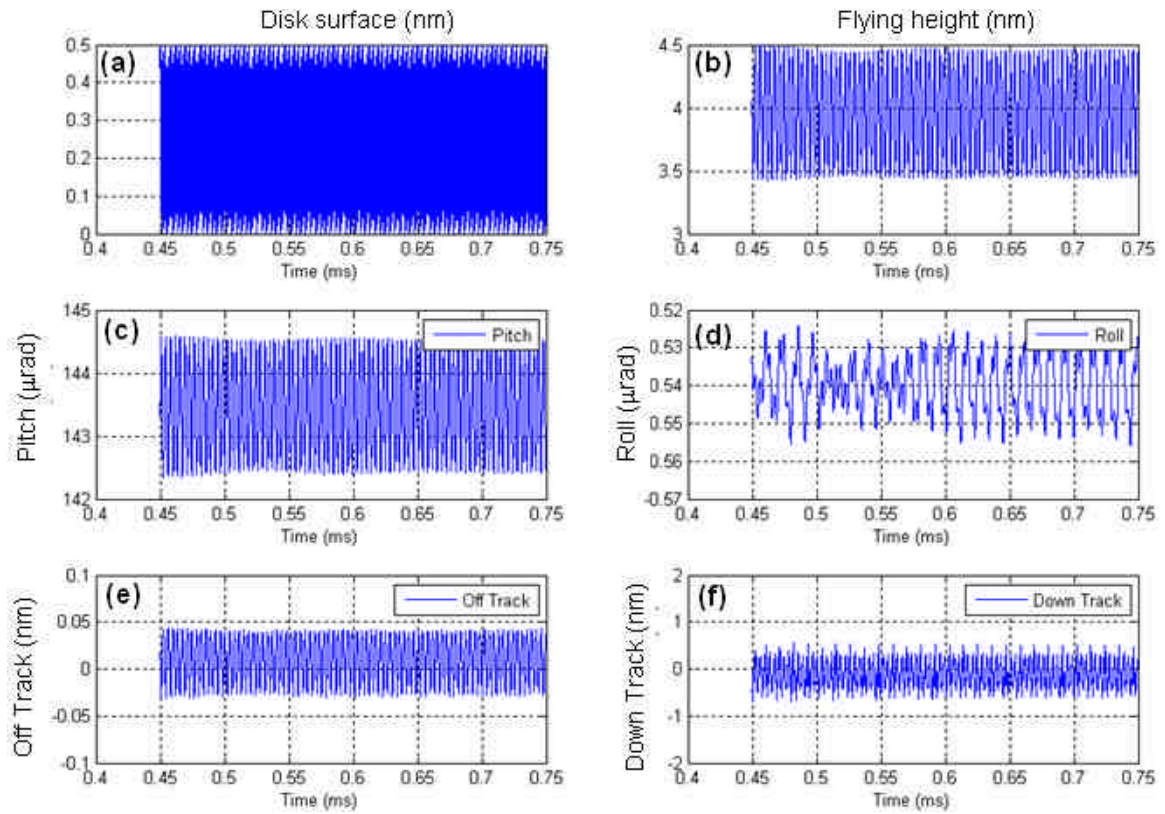


Figure 7.13 Simulation result for slider vibrations due to thermal pole tip protrusion

In order to compare the experimental results with our simulation results, we have simulated the slider vibrations for different surface roughness values. In Fig. 7.14, typical simulation results for slider vibration amplitudes versus surface roughness are shown.

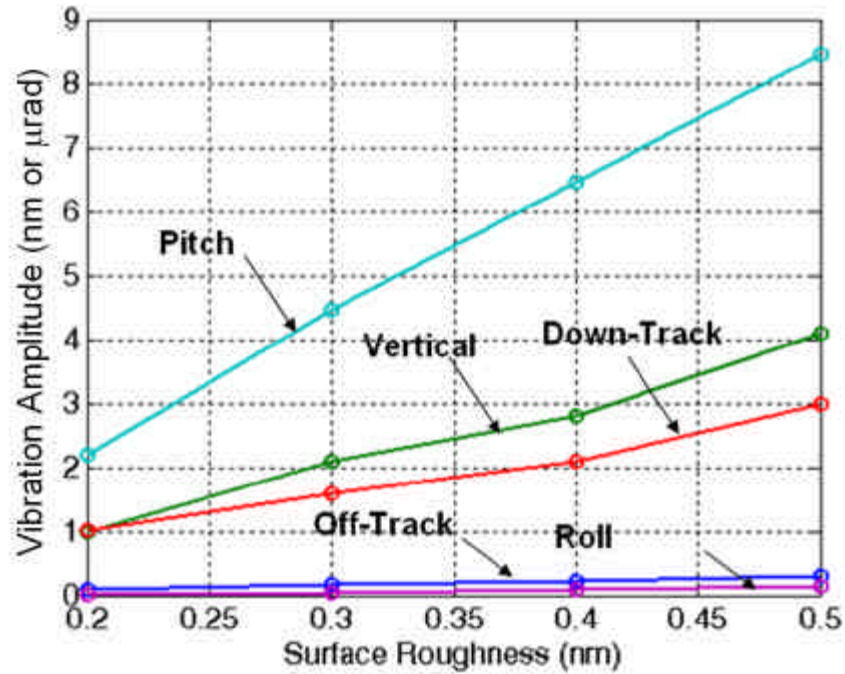


Figure 7.14 Slider vibration amplitudes versus surface roughness

We observe from Fig. 7.14 that the slider vibration amplitude increases with increasing disk surface roughness. In particular, for the disk with surface roughness of 0.4 nm, an increase of 40% was found for the amplitude in the down-track direction, the vertical direction, and the pitch direction as compared to the amplitude on the disk with surface roughness of 0.3 nm. The vibration amplitudes in the off-track and roll direction are much smaller than in the other directions. The simulation results show that disk surface roughness has a large effect on the amplitude of slider vibrations, i.e., the higher the surface roughness, the larger the vibration amplitude during slider disk contacts due to thermal pole tip protrusion.

In Fig. 7.15, we have plotted slider vibration amplitudes versus lubricant thickness.

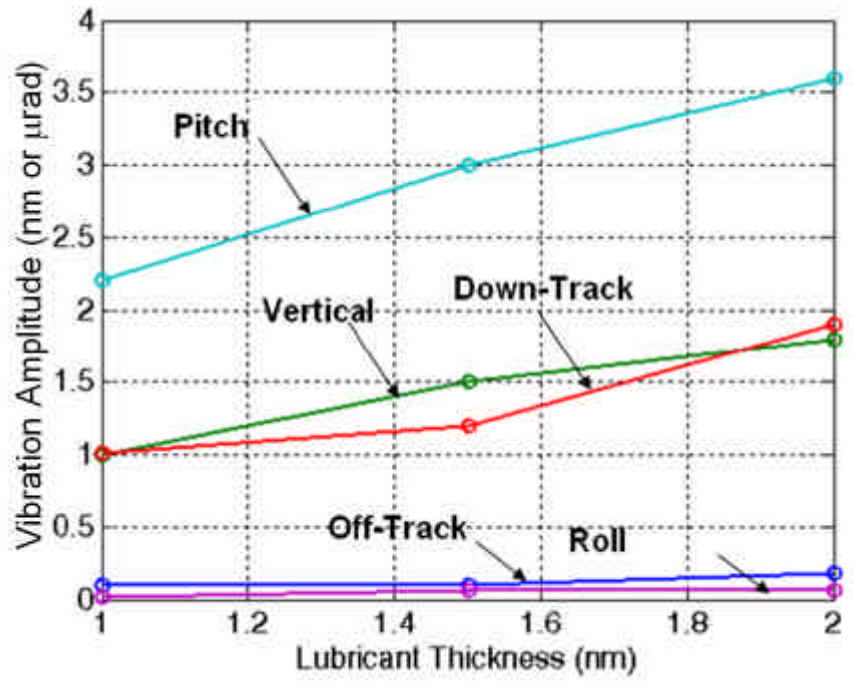


Figure 7.15 Slider vibration amplitudes versus lubricant thickness

We observe from Fig. 7.15 that an increase in lubricant thickness causes an increase in vibration amplitudes in the down-track direction, the vertical direction, and the pitch direction. The vibration amplitudes in the off-track and roll direction are much smaller than in the other directions. Thus, similar to the experimental results, the thickness of the lubricant film affects the slider vibration amplitude. In particular, an increase in the lubricant thickness excites larger vibration amplitudes during slider disk contacts due to thermal pole tip protrusion. Similar results were obtained by Knigge and Talke [7.26] who observed increased slider vibrations with increasing lubricant thickness using laser Doppler vibrometry.

Comparing the numerical result with our experimental results discussed in section 7.3, we observe that the numerical and simulation result shows the same trend. Increased

surface roughness and increased lubricant thickness excite larger vibration amplitudes during slider disk contacts due to thermal pole tip protrusion.

7.7 Summary and Conclusion

In this chapter, we have analyzed the vibration characteristics of sliders during thermally induced head/disk contacts using laser Doppler vibrometry. Slider vibrations were observed in the vertical, pitch, roll, off-track, and down-track directions. An increase in vibration amplitudes in all five degrees-of-freedom was observed when the write gate was engaged. The slider was excited in the range of air-bearing frequencies during head/disk contacts.

In addition, we have simulated thermal pole tip protrusion using finite element analysis and have used the protrusion profile to calculate the air bearing pressure and slider/disk dynamics due to thermal pole tip protrusion. The simulation results were in very good quantitative agreement with the experimental results: slider vibration amplitudes were found to increase for disks with increased surface roughness and for disks with increased lubricant thickness.

7.8 Acknowledgement

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Chapter 8 Magnetic Spacing Measurement Using the Read-back Signal

In the previous chapters, the flying height and magnetic spacing were shown to be important parameters in the characterization of the head/disk interface. In this chapter we investigate the measurement of magnetic spacing using the magnetic read-back signal. In this approach, the magnetic spacing change is calculated using the read-back signal. In addition, the experimental results are compared with the “design” flying height of the slider at the read/write element and the stability of the measurement is assessed.

8.1 Introduction

As the magnetic spacing decreases, improved methods for measuring the spacing between slider and disk are needed. A number of methods have been investigated in the past for measuring the spacing between slider and disk. Lin [8.1] used a capacitive technique to measure the spacing between a slider and a disk. He also discussed an optical interference technique by counting fringes of light to determine the flying height of air bearing sliders. Wahl et al. [8.2] and Lacey et al. [8.3] used optical interference techniques with three light sources at different wavelengths to determine the air bearing spacing. In this manner, the accuracy of the measurement was improved to the order of nanometers. Li and Menon [8.4] discussed factors affecting the accuracy of optical flying height measurement. They found that optical material properties of the slider surface could affect measurement result on the order of nanometers. Groot [8.5] developed a method termed

“polarization interferometry” to measure the flying height of air bearing slider [8.5]. Since the signal-to-noise ratio of interferometric spacing measurement methods decreases with decreasing flying height, the error in the spacing measurement increases as the flying height is reduced [8.6]. New methods for improving the accuracy of spacing measurements are highly desirable.

Several methods for the measurement of the magnetic spacing have been proposed based on the analysis of the spectrum of the read-back voltage [8.7]. For longitudinal recording, the triple harmonic ratio method was proposed to decrease the noise level during the measurement [8.9]. The triple harmonic ratio method relies on the measurement and comparison of two components of the read-back spectrum. Recent studies have shown that the spectrum method can also be used for estimating the magnetic spacing of the head/disk interface for perpendicular recording [8.8].

In this chapter, we will describe the implementation of the triple harmonic ratio method for the measurement of the head disk spacing. In addition, we will study the variation of the magnetic spacing for a number of sliders to investigate the repeatability of this method.

8.2 Theoretical Background

Based on the Wallace model [8.10] of spacing loss presented in chapter 1, the Fourier transform of the read-back voltage from an MR transducer is

$$V = MKe^{\frac{-2pd}{l}} C(g, d, l) \quad (8.1)$$

where V is the harmonic amplitude of the Fourier transform of the read-back voltage,

M is a function of the magnetic field, K is a system parameter, d is the magnetic spacing, l is the recording wavelength, g is the gap length of the recording head, d is the thickness of the magnetic medium and C is a function related to g , d and l .

The triple harmonic ratio method [8.9] uses a waveform with a (111100) pattern. For this pattern, the first harmonic amplitude (V_1) and the third harmonic amplitude (V_3) of the read-back voltage is given by [8.10]

$$\begin{cases} V_1 = MKe^{\frac{-2pd}{l}} C(g, d, l) \\ V_3 = MKe^{\frac{-2pd}{3l}} C(g, d, 3l) \end{cases} \quad (8.2)$$

We can define a harmonic amplitude factor A_{31} as

$$A_{31} = \frac{C(g, d, 3l)}{C(g, d, l)} \quad (8.3)$$

Thus, the ratio between the first and the third harmonic amplitude of the read-back voltage can be rewritten as

$$\frac{V_3}{V_1} = A_{31} \exp \left[\left(-\frac{2pd}{l} \right) - \left(-\frac{2pd}{3l} \right) \right] = A_{31} \exp \left(-\frac{4pd}{3l} \right) \quad (8.4)$$

The magnetic spacing d can be obtained from Eqn. 8.4 as

$$d = -\frac{3l}{4p} \left[\ln \left(\frac{V_3}{V_1} \right) + C_d \right] \quad (8.5)$$

where $C_d = -\ln(A_{31})$ is a constant that depends on various system parameters such as the gap length of the recording head, the thickness of the magnetic medium and the recording wavelength.

If a reference measurement is made at a magnetic spacing d_0 and then another measurement is made at a magnetic spacing d_1 , the equations for the magnetic spacing at each measurement are given by

$$\begin{cases} d_0 = -\frac{3l}{4p} \left[\ln \left(\frac{V_{3,0}}{V_{1,0}} \right) + C_d \right] \\ d_1 = -\frac{3l}{4p} \left[\ln \left(\frac{V_{3,1}}{V_{1,1}} \right) + C_d \right] \end{cases} \quad (8.6)$$

where $V_{3,0}$ and $V_{1,0}$ refer to the reference measurement, and $V_{3,1}$ and $V_{1,1}$ to the second measurement, i.e., the magnetic spacing of interest. Thus, the magnetic spacing change Δd can then be obtained as

$$\Delta d = d_1 - d_0 = -\frac{3l}{4p} \left[\ln \left(\frac{V_{3,1}}{V_{1,1}} \right) - \ln \left(\frac{V_{3,0}}{V_{1,0}} \right) \right] \quad (8.7)$$

As shown in Eqn. 8.7, by measuring the value of $V_{3,0}/V_{1,0}$ and $V_{3,1}/V_{1,1}$, the magnetic spacing change Δd can be determined.

8.3 Experimental Procedure

A schematic of the experimental setup is shown in Fig. 8.1. A commercially available hard disk read/write tester was used to write and read a pre-written data pattern on the disk. A (111100) pattern was written using a wide write head that produced a 10 μm -wide track with a wavelength of $l = 316 \text{ nm}$. The track was sufficiently wide enough so that the read heads could stay on the written track despite variations in the runout of the tracks with disk rotational speed. The read-back signal was captured using a digital

oscilloscope and transmitted to a commercially available “in-situ testing module” (from Data Storage Institute, Singapore).

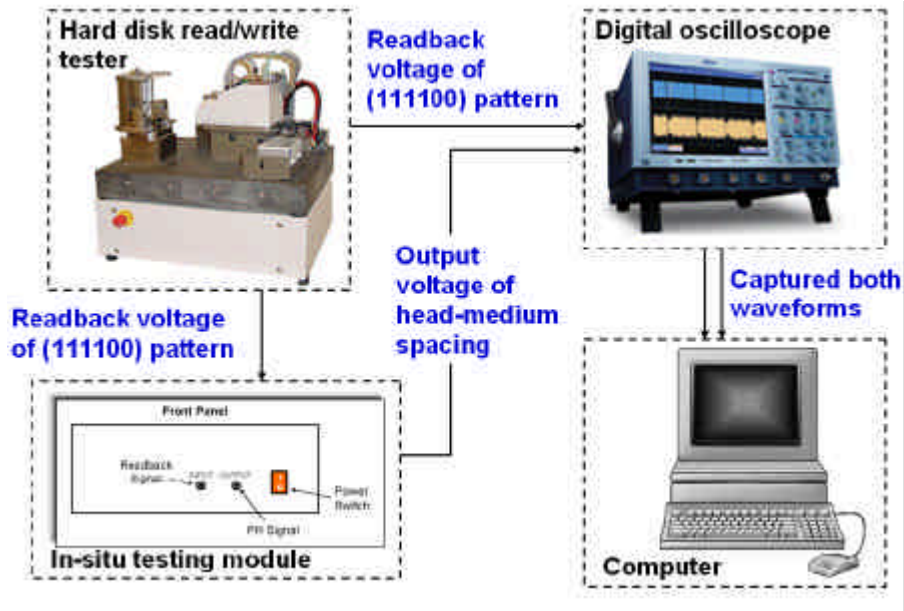


Figure 8.1 Schematic of experimental setup for the magnetic spacing measurement

The in-situ testing module produces an output voltage signal V_{THR} proportional to the triple harmonic ratio

$$V_{THR} = k_1 \ln \left(\frac{V_3}{V_1} \right) \quad (8.8)$$

where k_1 is an constant.

When measured at two different flying heights, say, d_0 and d_1 , we obtain two output voltage signal $V_{THR,0}$ and $V_{THR,1}$, respectively.

$$\begin{cases} V_{THR,0} = k_1 \ln \left(\frac{V_{3,0}}{V_{1,0}} \right) \\ V_{THR,1} = k_1 \ln \left(\frac{V_{3,1}}{V_{1,1}} \right) \end{cases} \quad (8.9)$$

Thus, combining Eqn. 8.9 and Eqn. 8.7, a change in voltage between two measurements can be correlated with a change in spacing, i.e.,

$$\Delta V_{THR} = -k_1 \frac{4p}{3l} \Delta d \quad (8.10)$$

V_{THR} was captured using an oscilloscope and was transmitted to a personal computer for analysis.

In order to produce a spacing change, a velocity-sensitive air bearing slider was used in this experiment. The air bearing design of the slider is shown as insert in Fig. 8.2. It is the same air bearing design used in previous chapters. The variation of pole tip flying height (PTFH) versus disk rotational speed for this slider flying at a radius of 25 mm is shown in Fig. 8.2.

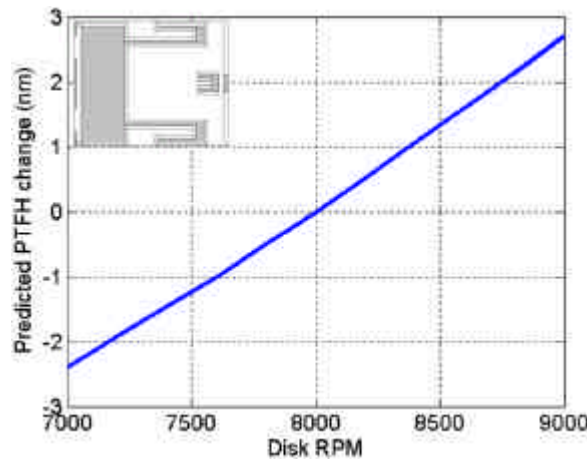


Figure 8.2 Slider design and predicted pole tip flying height change as a function of disk rotational speed (radius at 25 mm)

Figure 8.2 shows that the change in flying height of the slider, Δd , is a linear function of the disk rotational speed (rotations per minute, or RPM)

$$\Delta d = k_2 \Delta RPM \quad (8.11)$$

For the air bearing used in this study, k_2 was approximately 2.5×10^{-3} nm/RPM.

8.4 Experimental Results and Analysis

The variation of V_{THR} for a typical slider is shown in Fig. 8.3 as a function of time in steps of 400 RPM. The radius of the track was 25 mm. As can be seen from Fig. 8.3, the absolute value of the voltage output V_{THR} at 9000 RPM is larger in amplitude and shorter in duration as compared to the value of V_{THR} obtained at 7000 RPM. The magnitude of V_{THR} varies with position as a result of changes in the magnetic spacing. The variation of V_{THR} indicates the flying height modulation caused by disk waviness and clamping distortions [8.11].

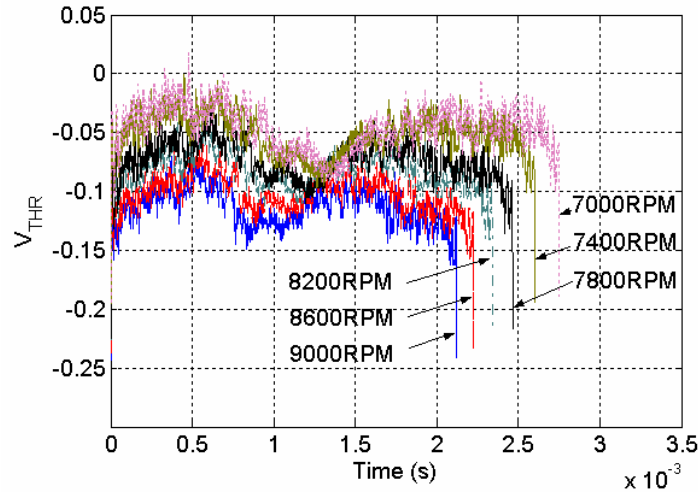


Figure 8.3 V_{THR} as a function of disk position and RPM. The output reproduces variations in the magnetic spacing for a range of rotational speeds

The mean value of V_{THR} for a read-back burst was calculated to yield a single value of V_{THR} for each head and RPM setting. Data from each head were fitted using Eqn. 8.10 to give a value of k_1 using a combination of Eqns. 8.10 and 8.11

$$k_1 = -\frac{3l}{4pk_2} \frac{\Delta V_{THR}}{\Delta RPM} \quad (8.12)$$

Most of the heads used were found to exhibit a nearly linear response as a function of speed (Fig. 8.4). In Fig. 8.4, the expected spacing change (dashed line) is plotted against RPM. V_{THR} was scaled by a factor k_1 to minimize the least-squares error between V_{THR} and the predicted pole tip flying height. The scaled results for V_{THR} were curve-fitted to the pole tip flying height (PTFH). Each head was measured twice and fitted in this manner to give a two-sample average for each head. The measurements were then averaged to yield a mean for k_1 . For the heads used in this study, the mean value for k_1 was found to be 58.92 V with

a standard deviation of 8.95 V.

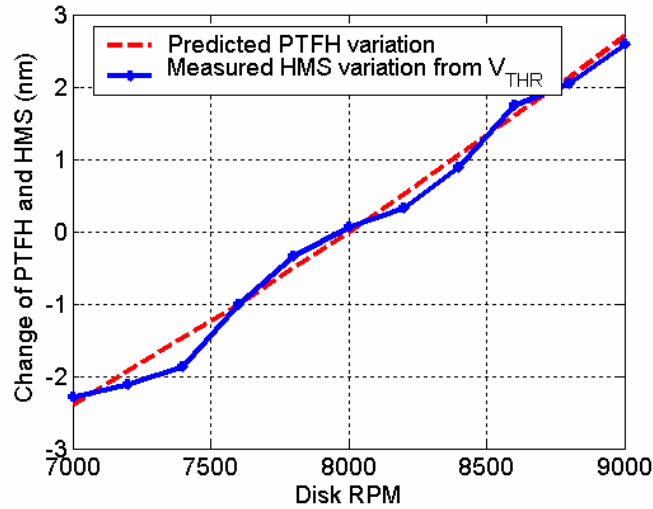


Figure 8.4 Typical plot of scaled V_{THR} vs RPM, demonstrating the linearity in the scaled V_{THR}

Using this global mean value of k_1 , we were able to predict the magnetic spacing variation with RPM for each slider. The results are plotted in Fig. 8.5. For our analysis, we selected the rotational speed of 8000 RPM to be the reference measurement.

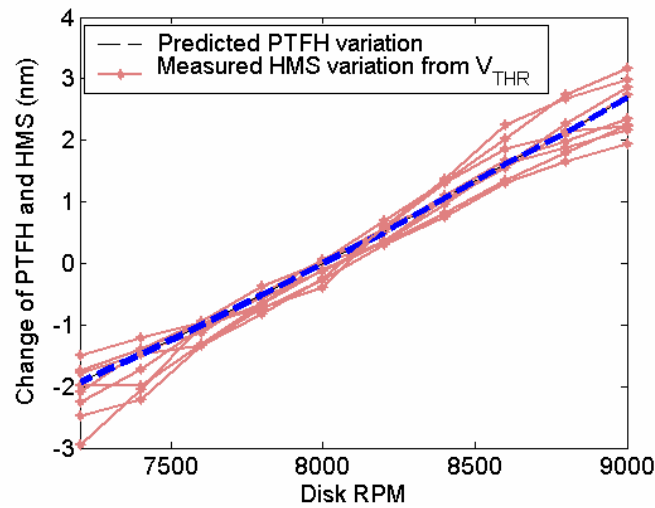


Figure 8.5 Change of PTFH (pole tip flying height) as a function of disk RPM

The data in Fig. 8.5 indicate that the variation of the magnetic spacing is small for the sliders used. Most of the variation arises from scatter at low and high RPM. Signal output variations can also arise from gap length differences of individual read elements and flying height variations of individual heads.

To investigate the stability of the measurements, V_{THR} and the track average amplitude (TAA) of the read-back signal were captured once per minute for 180 minutes. The track average amplitude of the signal and the calculated change in magnetic spacing from V_{THR} are plotted in Fig. 8.6.

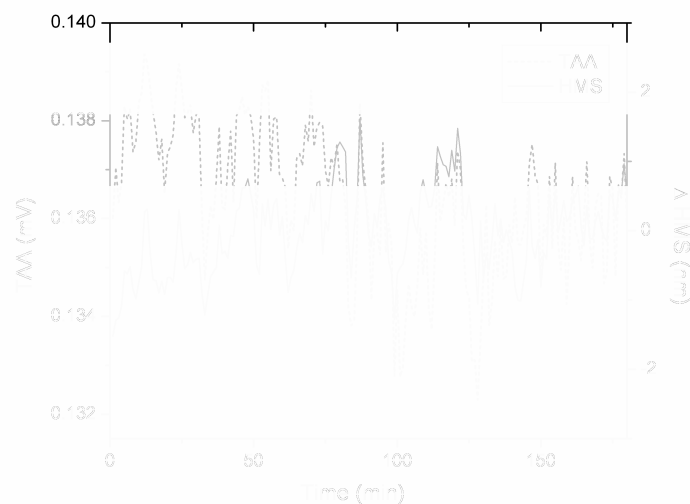


Figure 8.6 Magnetic spacing (HMS) and independently-measured track average amplitude (TAA) as a function of time

The two measurements show differences during the first 75 minutes, indicating that thermal drift may be responsible to the variation in the two measurements. However, after approximately 75 minutes, fluctuations in both spacing and track average amplitude exhibit similar directions and relative magnitudes. Based on the results shown in Fig. 8.6, we conclude that the spacing fluctuates by ± 1 nm over a 3 hour period.

8.5 Summary and Conclusion

We have applied the “triple-harmonic-ratio-method” to measure the magnetic spacing for a series of sliders on a single disk using a velocity-sensitive air bearing. The use of a commercially available testing module made measurement of the spacing change tractable in real time and allowed for identification of minor variations in magnetic spacing

with high sensitivity. We have obtained the change in magnetic spacing for a number of heads and have found that the spacing change with RPM is similar for all sliders. We also found that the change of magnetic spacing closely matched the predicted slider flying height change as a function of disk rotational speed. We have investigated the long-term stability of these measurements and found that the spacing change over a 3-hour period is on the order of ± 1 nm.

8.6 Acknowledgement

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Chapter 9 Conclusion

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

The thesis starts with an overview of the principles of tribology with a view towards applications for the head/disk interface. Contact mechanics is discussed along with boundary and hydrodynamic lubrication. The Reynolds equation was introduced for the analysis of slider air bearings. In addition, various wear mechanisms have been discussed and reviewed.

The influence of size reduction of individual disk drive components on their static and dynamic performance was studied. We observed that all vibration modes increase with decreasing dimensions for both suspensions and disks. Reducing the size of disk drive components is beneficial for both the static and dynamic characteristics of individual components and for increasing the areal density of disk drives.

Finite element analysis using the “radial plasticity approach” has been used to investigate the fracture of small form factor glass disks and the plastic deformation of aluminum disks at high shock levels. For glass disks, cracks were observed to start at the inside perimeter and propagate towards the outer perimeter of the disk. Increases in the clamp diameter and disk thickness were found to result in an increase of the permissible shock levels. Cracks did not form where the slider contacts the glass disk but rather at the inner perimeter. The critical shock levels for fracture of glass disks were found to increase

with decreasing disk diameter. A similar trend was also found for plastic deformation of aluminum disks.

An experimental set-up for measuring slider vibrations for five degrees-of-freedom has been introduced. The set-up enables the detection of vibrations of sliders in five degrees-of-freedom and the determination of the head disk contacts. Slider air bearing frequencies were found to be excited during slider/disk contacts.

We have measured three components of slider vibrations for three disks with different amplitude profiles of topography. We observed that the slider reacts to the vertical component of disk topography but not to pitch and roll components. Defining gain as the ratio of slider motion over any of the components of disk topography, we find that a single transfer function can describe the slider response in the air bearing frequency range up to the first slider pitch frequency. Above this frequency, we find that the gain is different for disks with different surface topography. Thus, surface topography affects the response of a slider at high frequencies, and no single transfer function applies across the entire frequency spectrum.

We have studied the vertical, pitch, roll, off-track, and down-track components of slider motion during slider-lubricant interactions. The data suggest that slider-lubricant interactions at the head/disk interface have a strong effect on slider dynamics. We have observed that lubricant is redistributed along the track by the slider. Spectral analysis of the slider motion indicates that excitation occurs at the same frequency for all degrees of freedom. The excitation frequency is slightly lower than the free vibration frequency, suggesting that lubricant damping plays a role in the excitation of the slider.

We have analyzed the vibration characteristics of sliders during thermally induced head/disk contacts using laser Doppler vibrometry. Slider vibrations were observed in the vertical, pitch, roll, off-track, and down-track directions. An increase in vibration amplitudes in all five directions was observed when the write gate was engaged. The slider was excited in the range of air-bearing frequencies during head/disk contacts.

In addition, we have simulated thermal pole tip protrusion using finite element analysis and have modified the air bearing contour based on the protrusion profile of the read/write element. The simulation results are in good qualitative agreement with the experimental results. The slider vibration amplitude was found to increase with an increase in surface roughness and with an increase in the thickness of the lubricant layer.

Finally, we have applied the triple harmonic ratio method to measure the magnetic spacing for a number of sliders using a velocity-sensitive air bearing to change the flying height. The use of a commercially available testing module made measurement of the spacing change tractable in real time and allowed for identification of minor variations in magnetic spacing with high sensitivity. We have obtained the change in magnetic spacing for a number of heads and have found that the spacing change with RPM was similar for all sliders. We also found that the change of magnetic spacing closely matched the predicted slider flying height change as a function of disk rotational speed. We have investigated the long-term stability of these measurements and found that the spacing change over a 3-hour period is on the order of ± 1 nm.

Appendix A Calibration of the Position-Sensitive Detector

The position sensitive detector (PSD) is a sensor used to monitor the slider/disk vibrations in pitch and roll direction. It detects the laser spot reflected from slider/disk and converts the position of that spot into a voltage signal. A sketch of the principle of operation for a position sensitive detector is shown in Fig. A1. If the laser spot is at the center of the position sensitive detector, both voltage outputs for the x and y direction will maintain at a reference level. If the laser spot moves to the right side of the center along the x-axis, indicated in the figure as position 1, the voltage output for the x direction will increase while the voltage output for the y direction remains unchanged. If the laser spot moves to the left side of the center along the x-axis (position 2), the voltage output for the x direction will decrease while the voltage output for the y direction remains unchanged. Similarly, if the laser spot moves up (position 3) or down (position 4) along the y-axis, the voltage output for the y direction will rise (position 3) or fall (position 4), while the voltage output for the x direction will remain unchanged.

If the laser spot is reflected from slider/disk, the angular vibration of slider/disk is transferred to position change of the laser spot on the position sensitive detector. Thus, by analyzing voltage output of the position sensitive detector and the geometrical relationship between slider/disk and the position sensitive detector, the angular vibration of the slider/disk can be obtained.

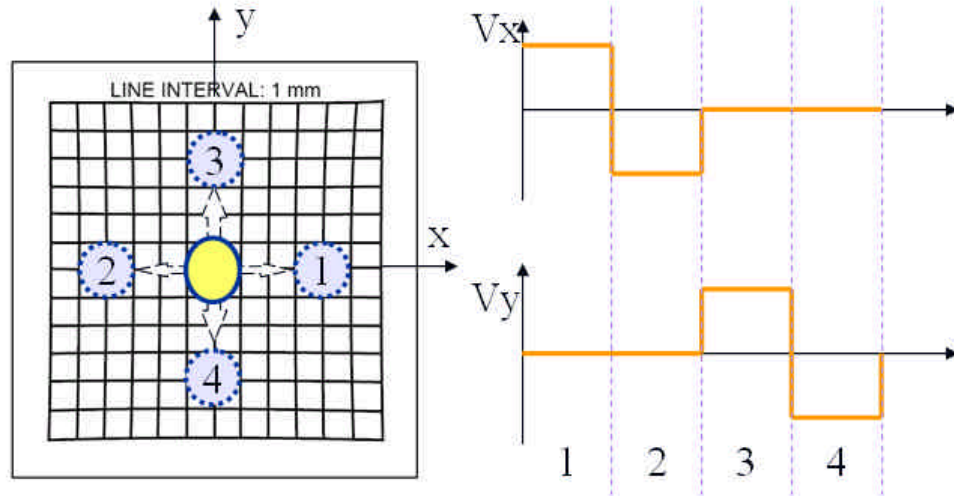


Figure A1 Position-sensitive detector

Calibration of the position sensitive detector typically requires three steps:

- a) Relationship between the voltage output and the displacement of the laser spot

The voltage change (DV) of the position sensitive detector output has the following relation with position change (Dd) of the laser spot

$$\frac{\Delta V}{\Delta d} = \frac{2.257 \text{ sum}}{2R} \quad (\text{A1})$$

where R is the radius of the laser spot on the position sensitive detector and “sum” is a signal output from the position sensitive detector. When the laser spot is moving from one side of the position sensitive detector to the other side, the sum signal varies from zero to the maximum then returns to zero. The dimensions of 90% of the sum signal can be used to estimate the radius (R). A plot of the sum signal versus the position of the laser spot on the position sensitive detector is shown in Fig. A2. The radius of the laser spot on the position sensitive detector is estimated to be 1.225 mm.

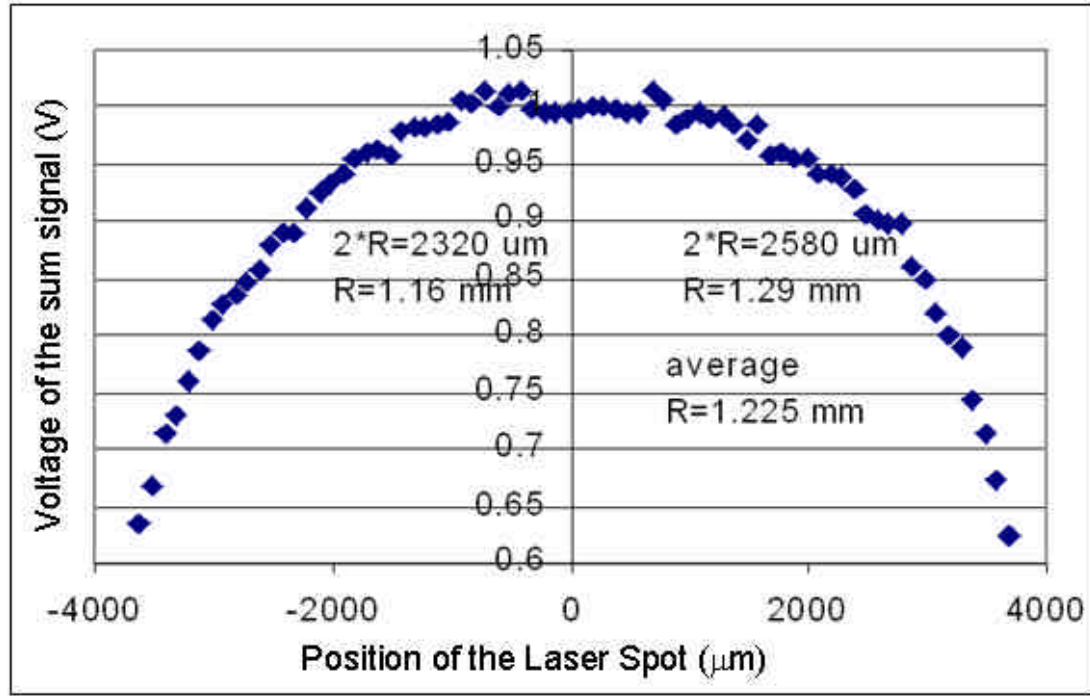


Figure A2 Determination of the radius (R) of the laser spot

After detecting the size of the laser spot, we fixed the position of the laser spot on the slider/disk, moved the position of the position sensitive detector along the horizontal axis from $-150\ \mu\text{m}$ to $150\ \mu\text{m}$ and recorded the pitch, roll and sum voltage at $10\ \mu\text{m}$ intervals. Eqn. A1 predicts the voltage signal of pitch from the sum voltage and the position of the laser spot on the position sensitive detector. We can compare this predicted voltage with the voltage output of the position sensitive detector. The comparison of this predicted voltage and the measured voltage is shown in Fig. A3a and Fig. A3b. The ratio between the predicted pitch voltage and the measured pitch voltage for the laser on the disk surface (Fig. A3a) is 0.98, and for the laser on the slider (Fig. A3b) is 1.01, i.e., the predicted pitch

voltage is in very good agreement with the measured pitch voltage from the position sensitive detector.

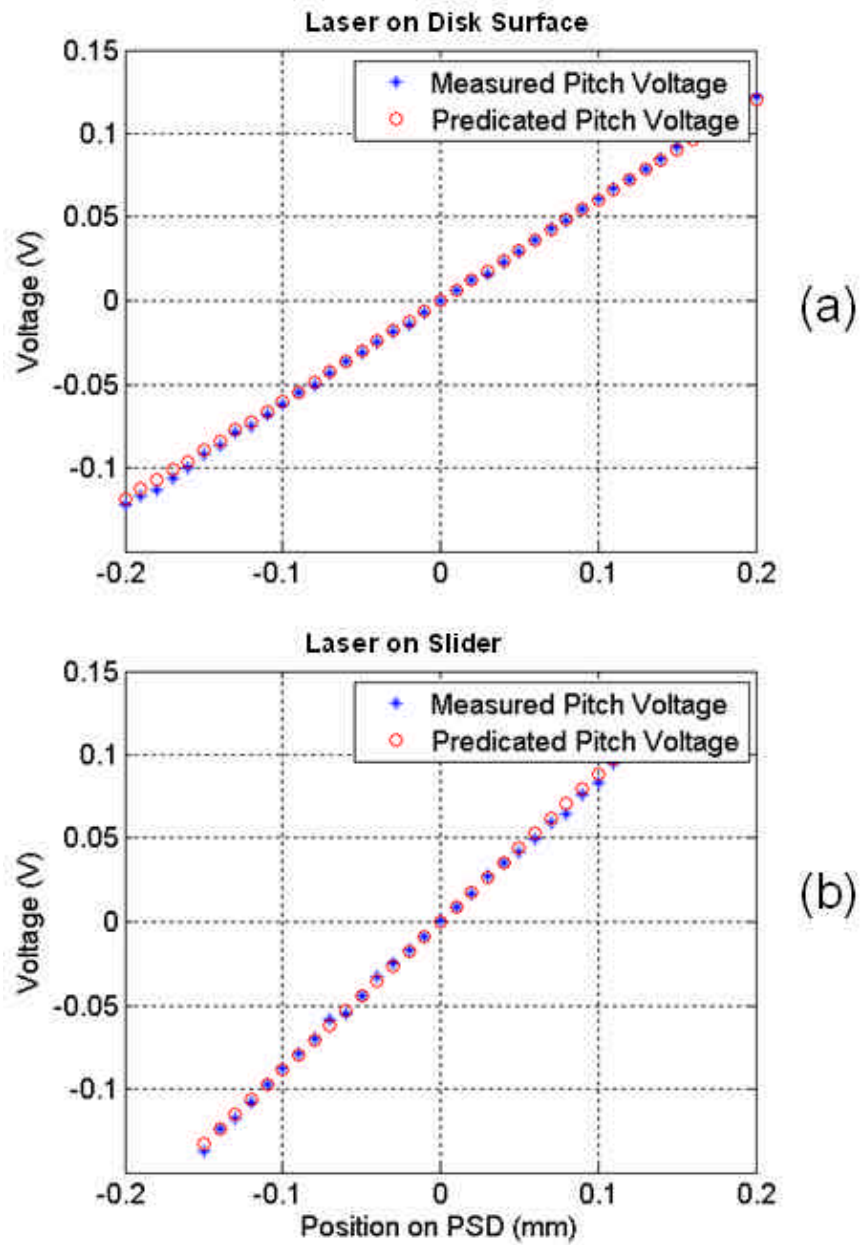


Figure A3 Measured pitch voltage vs. predicted pitch voltage for laser on a) disk surface, b) slider

b) Relationship between the laser spot position and the angular change of the slider

The geometric relationship between the angular change of the slider and the position change of the laser spot on the position sensitive detector is shown in Fig. A4. If the slider has an angular change ($\Delta\theta$), we obtain

$$\begin{cases} \Delta d_1 = L_2 \tan 2\Delta q \\ \Delta d_2 = \frac{L_1 \sin 2\Delta q}{\sin\left(\frac{p}{4} - 2\Delta q\right)} \sin\left(\frac{p}{4}\right) \end{cases} \quad (A2)$$

where L_1 is the distance from the center of the beam splitter to the slider/disk and L_2 is the distance from the center of the beam splitter to the position-sensitive detector. Since $\Delta\theta$ is on the order of sub- μ rad, the above formula can be simplified to

$$\Delta d = \Delta d_1 + \Delta d_2 = 2(L_1 + L_2)\Delta q \quad (A3)$$

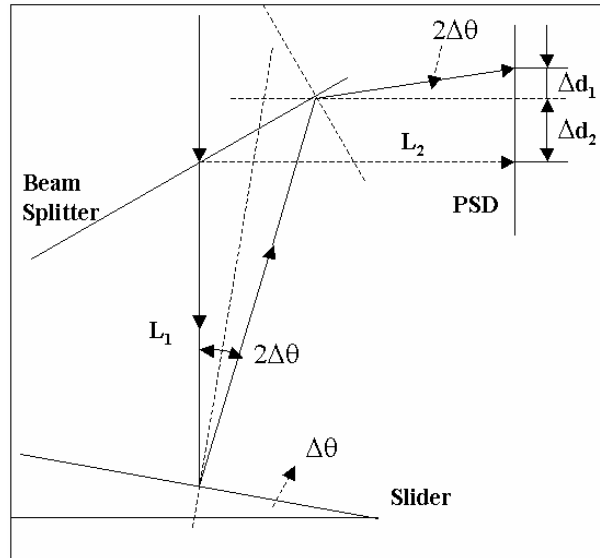


Figure A4 Sketch of the angular change of the slider to the position change of the laser spot

- c) Relationship between the voltage signal and the angular change of the slider

Combining Eqn. A1 and Eqn. A3, we obtain

$$\Delta q = \frac{R * \Delta V}{2.257(L_1 + L_2) * sum} \quad (A4)$$

In our set-up, $L_1 = 315$ mm, $L_2 = 60$ mm and $R = 1.225$ mm. Eqn. A4 can be simplified as follows

$$\Delta q = \frac{\Delta V}{691 * sum} \quad (A5)$$

Capturing the voltage change and obtaining the “sum” signal, we can calculate the angular change from Eqn. A5.

There is another way to directly obtain the pitch angular change from the vertical velocity. This method has been demonstrated by Bogy and Talke [A1] in their measurement of slider dynamics using laser Doppler vibrometry. Two laser Doppler vibrometer beams (LDV-A and LDV-B) were used in their set-up, one as the reference beam and the other as the measurement beam (Fig. A5). If the distance L between the spot and the displacement change Dd between the two laser Doppler vibrometer beams are known, the angular change Dq can be calculated as

$$\Delta q = \arctan \frac{\Delta d}{L} \quad (A6)$$

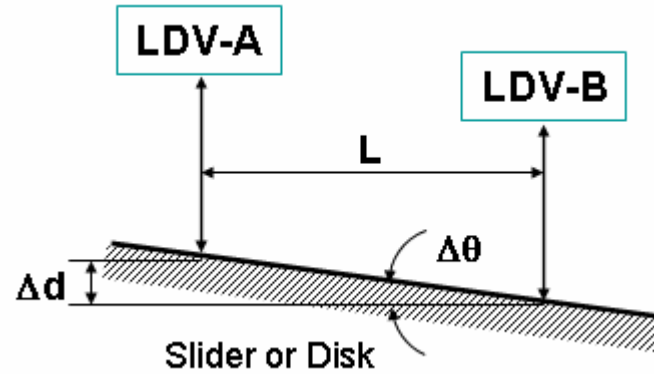


Figure A5 Laser Doppler vibrometer measurement of slider angular vibration

A comparison of the pitch angle obtained using Eqn. A5 (predicted pitch angle) and Eqn. A6 (measured pitch angle) is shown in Fig. A6a and Fig. A6b. The ratios between the predicted pitch angle and the measured pitch angle for the laser on the disk surface (Fig. A6a) and on the slider (Fig. A6b) are 0.91 and 0.95, respectively. The predicted pitch angle is in good agreement with the directly measured pitch angle.

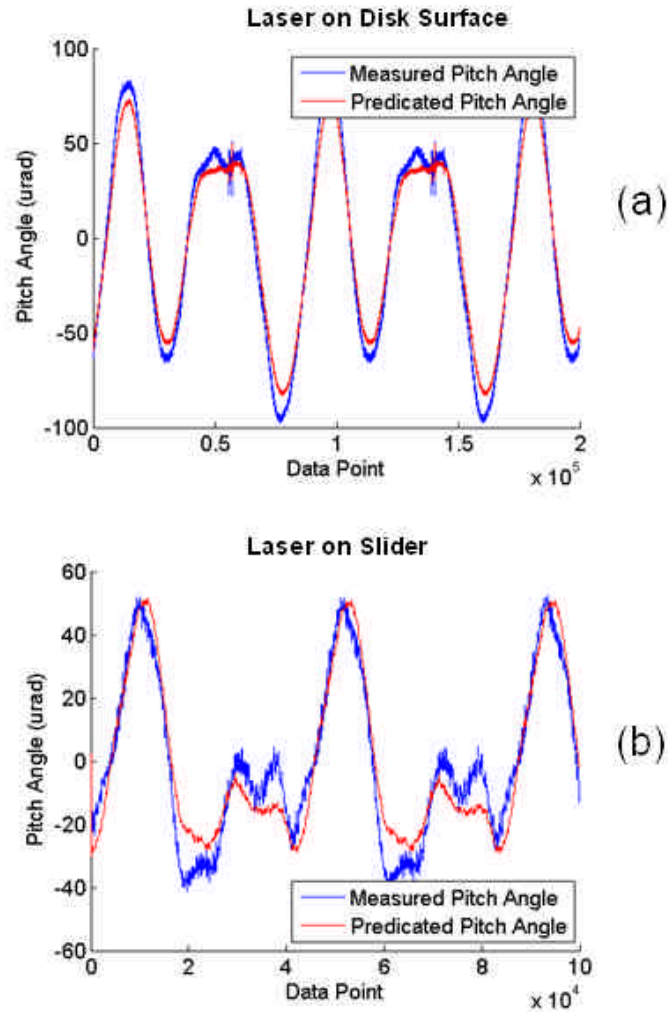


Figure A6 Measured pitch angle vs. predicted pitch angle when laser is on a) disk surface, b) slider

During the calibration process of the position sensitive detector, the important steps for a successful calibration are: 1) the sensitivity of the position sensitive detector affects the accuracy of measurement of the angular change of the slider; 2) the size of the laser spot needs to be closely monitored. The position sensitive detector needs to be recalibrated whenever the focus of the laser beam is modified.

Reference

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Appendix B Measurement Issues and Solutions

Air bearing sliders are designed to follow the disk curvature. The vibrations of a slider are required to be controlled within several nanometers. However, disk flutter and spindle vibrations is on the order of micrometers. To obtain accurate information about slider vibrations, we have to reduce the noise level of the detected signals.

a) Effect of Disk Flutter and Run-out

The vertical vibration of a disk is shown in Fig. B1a measured with a laser Doppler vibrometer. We observe that one revolution of the disk produces vibrations with an amplitude of about 800 nm and a frequency on the order of several kHz. The sliders are designed to follow the disk run-out in order to read/write information from/to the disk. When measuring slider vibrations using laser Doppler vibrometry, the signal of slider vibrations are digitized and recorded to an oscilloscope. During the digitizing process, each data point is represented by 8 bits. The original unfiltered signal could not represent the slider high frequency vibrations at amplitude of sub-10 nm. As a result, all data have to be high-pass filtered in order to remove the effect of disk fluttering and run-out during data sampling process. A high-pass (10 kHz) filtered signal is shown in Fig. B1b. As can be observed from Fig. B1b, the vibration amplitude is in the order of nm and is at a much higher frequency than that in Fig. B1a.

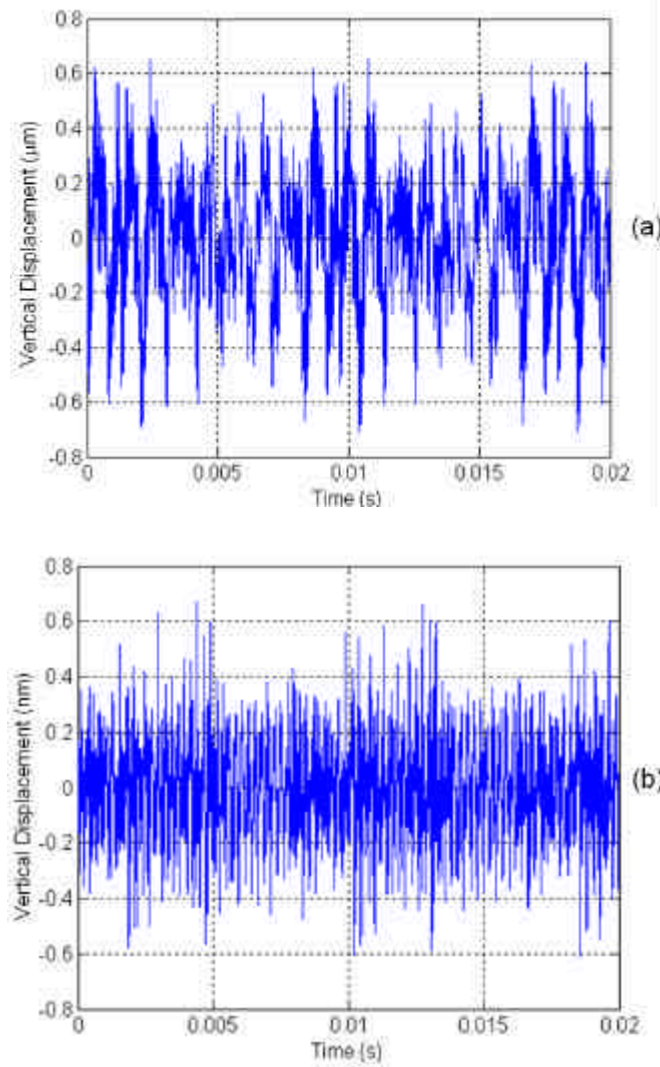


Figure B1 Vertical displacement of a rotating disk, a) signal captured before analog filtering, b) signal captured after analog high-pass filtering at 10 kHz

b) Effect of Signal Averaging

In order to improve the signal-to-noise ratio (SNR), the signals sampled need to be averaged a number of times. A comparison between an original signal captured during slider-disk contact and a signal averaged 25 times is shown in Fig. B2. After averaging, the magnitude of the noise level is reduced by about 70%. In addition, a second peak shows up

at the averaged signal after the noise level been reduced. Averaging is an important method to improve the signal-to-noise ratio (SNR). During our experiments, we have typically captured 20 to 25 sequences of a signal and averaged the signal to improve the SNR.

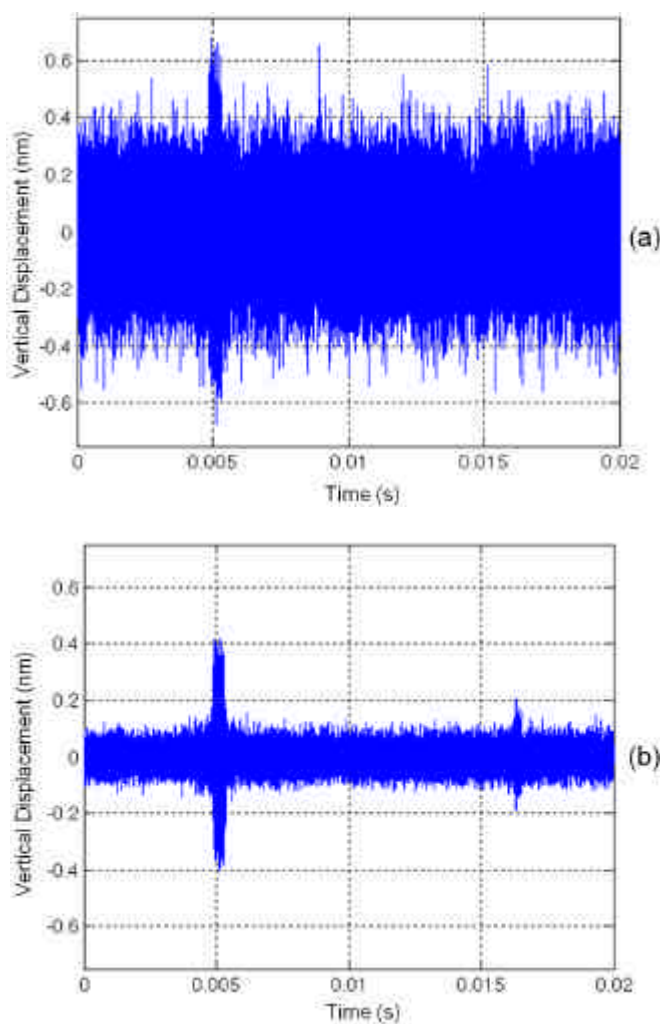


Figure B2 Vertical displacement of a slider, a) single shot signal, b) signal averaged 25 times

Appendix C A Rough Surface Contact/Adhesion/Friction Model

An illustration of a continuum-mechanics-based model developed by Kogut and Etsion [C1] is shown in Fig. C1. Suh and Polycarpou [C2] improved the model by considering the effect of lubricant. The assumption of the model is as follows: 1) The rough surface is isotropic; 2) Asperities are spherical (or at least near their summits) with radius R ; 3) There is no bulk deformation and only asperities deform due to contacts; 4) There is no interaction between asperities. In the formulas shown, parameters marked with “*” are normalized with the standard derivation of surface height s .

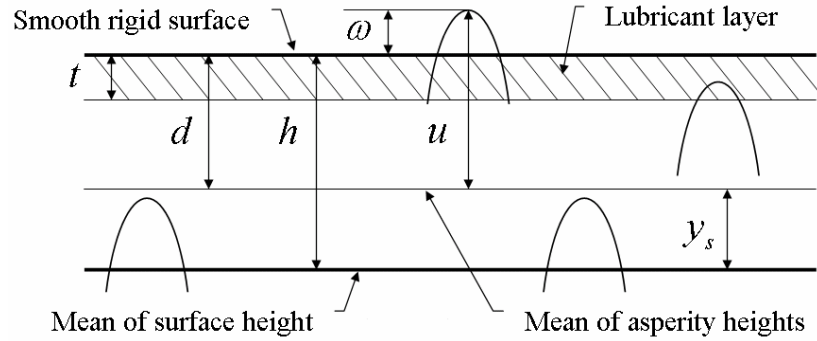


Figure C1 Illustration of continuum mechanics based contact model

In Fig. C1, t denotes the lubricant thickness, d denotes the separation between the reference plane and the mean of asperity height, h denotes the separation between the reference plane and the mean of surface height, u denotes the asperity height, and w denotes the interference height.

The adhesive force F_s is given by

$$F_s = 2phRA_n\Delta g \left\{ \int_{-\infty}^{d^*-t^*} J_{nc} + \int_{d^*-t^*}^{d^*} J_{lc} + 0.98 \int_{d^*}^{d^*+w_c^*} J_{-0.29}^{0.289} + 0.79 \int_{d^*+w_c^*}^{d^*+6w_c^*} J_{-0.321}^{0.356} + 1.19 \int_{d^*+6w_c^*}^{d^*+110w_c^*} J_{-0.332}^{0.093} \right\} \quad (C1)$$

where h is the areal density of asperities and A_n is the nominal surface area. The integral on J_{nc} corresponds to contribution of non-contacting asperities. The integral on J_{lc} corresponds to contribution of lubricant contacting asperities and the integrals on J_c^b corresponds to contribution of contacting asperities. Their expressions are given by

$$J_{nc} = \frac{4}{3} \left[\left(\frac{e^*}{u^* - d^* - t^*} \right)^2 - 0.25 \left(\frac{e^*}{u^* - d^* - t^*} \right)^8 \right] f(u^*) du^* \quad (C2)$$

$$J_{lc} = f(u^*) du^* \quad (C3)$$

$$J_c^b = \left(\frac{u^* - d^*}{w_c^*} \right)^b \left(\frac{e^*}{w_c^*} \right)^c f(u^*) du^* \quad (C4)$$

The normal Gaussian distribution function $f(u^*)$ is given by

$$f(u^*) = \frac{1}{\sqrt{2p}} \left(\frac{s}{s_s} \right) \exp \left(-\frac{1}{2} \frac{s}{s_s} u^{*2} \right) \quad (C5)$$

The critical interference w_c is given by

$$w_c = \left(\frac{pKH}{2E} \right)^2 R \quad (C6)$$

The parameter K , the hardness coefficient and the parameter E , the Hertz elastic modulus

is given by $K = 0.454 + 0.41n_{disk}$ and $\frac{1}{E} = \frac{1-u_{disk}^2}{E_{disk}} + \frac{1-u_{slider}^2}{E_{slider}}$.

The contact force P is given by

$$P = \frac{2}{3}pbKA_nHw_c^* \left\{ \int_{d^*-t^*}^{d^*+w_c^*} I^{1.5} + 1.03 \int_{d^*+w_c^*}^{d^*+6w_c^*} I^{1.425} + 1.4 \int_{d^*+6w_c^*}^{d^*+110w_c^*} I^{1.263} + \frac{3}{K} \int_{d^*+110w_c^*}^{+\infty} I \right\} \quad (C7)$$

where I^b is the general form of the integrand

$$I_b = \left(\frac{u^* - d^*}{w_c^*} \right)^b f^*(u^*) du^* \quad (C8)$$

The model provides the calculation of maximum static friction force $Q_{\max}$. An estimated ratio to transfer the static friction force to dynamic friction force is given

$$Q_{\max} = \frac{2}{3}pbKA_nHw_c^* \left\{ 0.52 \int_{d^*-t^*}^{d^*+w_c^*} I^{0.982} + \int_{d^*+w_c^*}^{d^*+6w_c^*} (-0.01I^{4.425} + 0.09I^{3.425} - 0.4I^{2.425} - 0.85I^{1.425}) \right\} \quad (C9)$$

where I^b is given at Eqn. C8.

The contact force calculated for a nominal contact area as a function of separation is shown in Fig. C2. The surface parameters used are obtained from measurements using atomic force microscopy as shown in Table C1. When the separation between the slider and the disk is more than 2.6 nm, the forces due to contact approach zero. When the separation between slider and disk exceeds 2.6 nm, the effect of contact force, adhesive, and friction force is negligible. If the separation decreases from 2.6 nm to 0.4 nm, the contact force, adhesive force, and friction force increases. However, the adhesive force increases faster than the contact force. If the separation continues to decrease, all three forces increase but the contact force increases faster than the adhesive force. Thus we obtained a ‘net’ adhesive force that is positive for separations less than 0.4 nm, but is negative for separations from 0.4 nm to 2.6 nm.

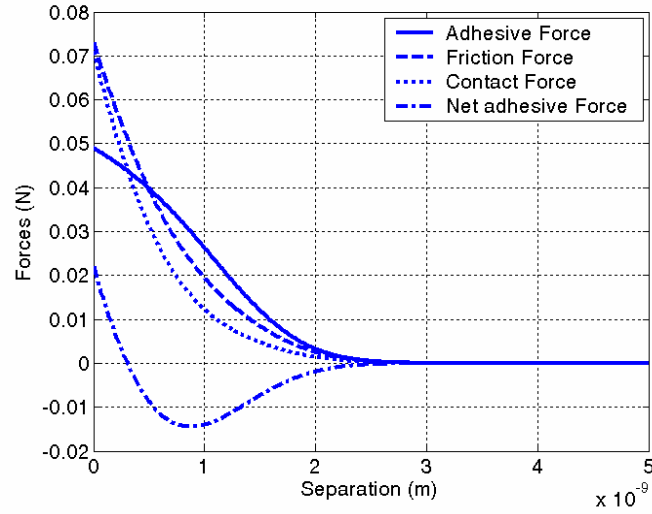


Figure C2 Force calculated using the model for a slider and a smooth disk

Table C1 Parameters for material properties used in contact force calculation

Parameters	Symbol	Unit	Value
Young's moduli	E_{disk}, E_{slider}	GPa	100, 450
Poisson's ratio	ν_{disk}, ν_{slider}		0.30, 0.30
Hardness of the softer material (disk)	H	GPa	3.6
Mobile lubricant thickness	t	nm	0.5
Energy of adhesion	Δg	N/m	0.09
Contact area	A	μm^2	1000
Standard deviation of surface height	s	nm	0.50
Standard deviation of combined asperity height	s_s	nm	0.40
Intermolecular distance	e	nm	0.3~0.5
Area density of asperities	h	μm^{-2}	10.0
Radius of asperity summit	R	μm	7.0
Gap between mean asperity height and mean surface height	y_s	nm	0.70

Reference

- [C1] Kogut, L. and Etsion, I., 2004, "A Static Friction Model for Elastic-Plastic Contacting Rough Surfaces," ASME J. Tribol., 126, 34–40.
- [C2] Suh, A. Y. and Polycarpou, A. A., 2005, "Adhesive Contact Modeling for Sub-5-nm Ultralow Flying Magnetic Storage Head-disk Interfaces including Roughness Effects," J. Appl. Phys., 97, 104328

Appendix D Flowchart for Solving Dynamics Equation

The flowchart for solving the dynamics equation (Eqn. 7.1) as well as the time dependent Reynolds equation is shown in Fig. D1. The Newmark method [D1] is implemented in the numeric simulator (see Fig. D2). The numerical integration of the coupled equations begins with vectors of the calculated forces $\{f\}_i$, displacement $\{d\}_i$, velocity $\{v\}_i$, and acceleration $\{a\}_i$ of the slider from the previous time step. Solving the dynamics equation with the Newmark method, we obtain displacement $\{d\}_{i+1}$, velocity $\{v\}_{i+1}$, and acceleration $\{a\}_{i+1}$ of the slider. For the given displacement $\{d\}_{i+1}$, one can obtain the pressure distribution and air bearing stiffness from solving the Reynolds equation. In addition, one can calculate the air bearing force based on the pressure distribution, contact, adhesive and frictional force based on the rough surface contact model, and integrate them into $\{f\}_{i+1}$.

For a force $\{f\}_{i+1}$, one can solve for a new $\{a\}_{i+1}$. By comparing the new $\{a\}_{i+1}$ and the old $\{a\}_{i+1}$, convergence can be determined. If the convergence criterion is satisfied, the program calculates the values for the next time step until the maximum number of time steps is reached. If convergence is not satisfied, the program will iterate again for new $\{d\}_{i+1}$, velocity $\{v\}_{i+1}$, and acceleration $\{a\}_{i+1}$ until convergence is satisfied.

Reference

[D1] T. J. R. Hughes, "The Finite Element Method: Linear Static and Dynamic Finite Element Analysis," Dover Publications, Inc, Mineola, New York, 2000.

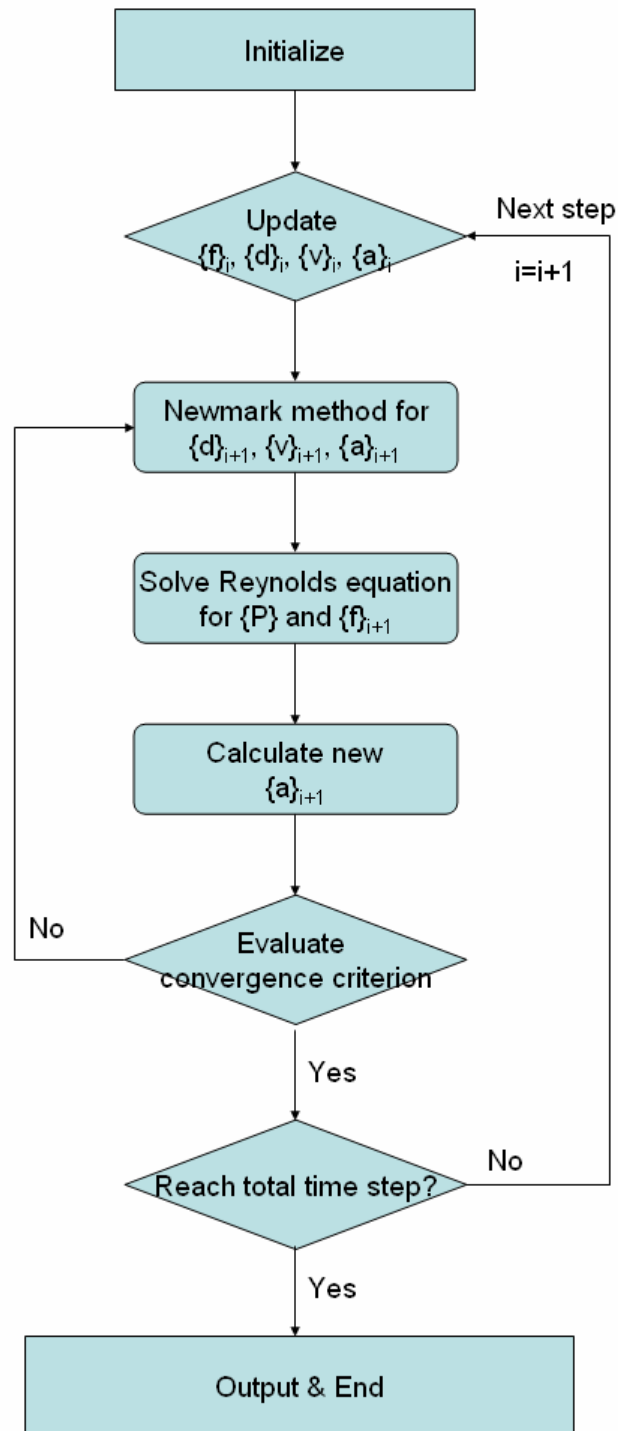


Figure D1 Flowchart on solving the dynamics equation

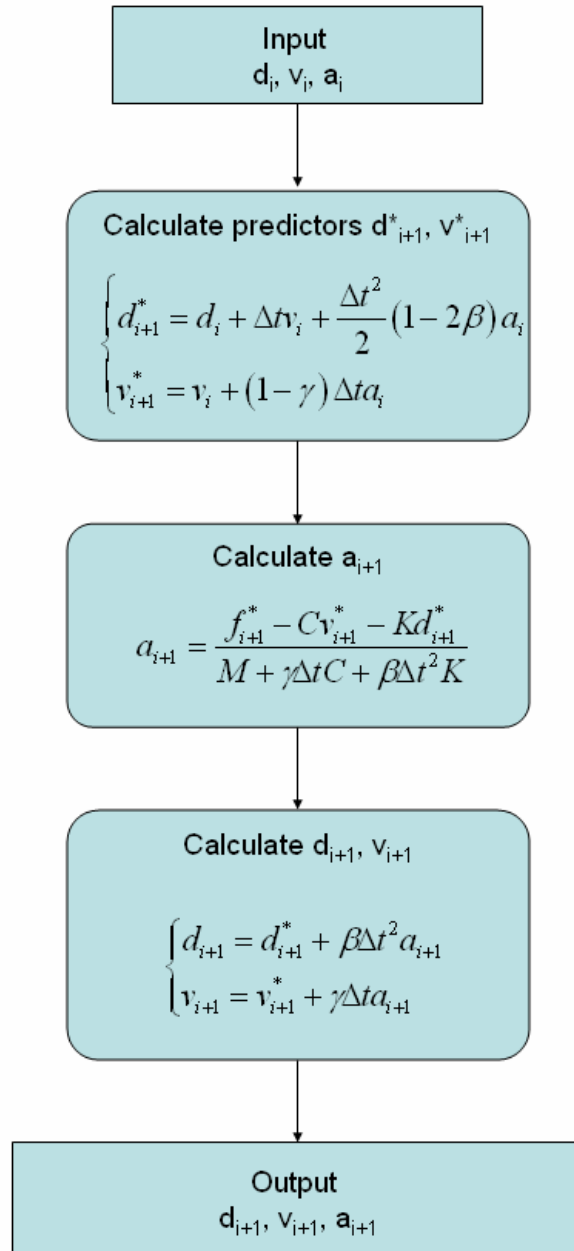


Figure D2 Flowchart for implementation of Newmark method ($\beta=0.25, \gamma=0.5$) [D1]