Dynamics of a Partial Contact Head Disk Interface

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ABSTRACT

A nonlinear dynamics model is developed to analyze the bouncing vibration of partial contact air bearing sliders, which are designed for the areal density of 1 Tbit/in^2. In this model the air bearing with contact is modeled using the generalized Reynolds equation modified with the Fukui-Kaneko slip correction and a recent slip correction for the contact situation. The adhesion, contact and friction between the slider and the disk are also considered in this model. The adhesion force is calculated using a modified intermolecular force model; the contact force is obtained through an elastic quasi-static contact model that considers the slider and disk roughnesses. Realistic measured roughnesses of the slider and disk are used in the simulation. It is found that minimizing the trailing pad size can reduce the slider's bouncing and crash likelihood. The surface roughness and adhesion have a strong effect on the slider's bouncing vibration, while the friction between the slider and disk is found here to have less effect. The slider's bouncing can be decreased without much increase in contact force, to some extent, through increasing the preload.

1. INTRODUCTION

Reducing the flying height (FH) of sliders is a requirement to achieve higher recording densities in hard disk drives. The Wallace spacing loss equation reveals that the magnetic signal increases exponentially as the distance decreases between the magnetic media and the read/write transducer. Reducing the flying height of the slider is necessary, since the protective layers- e.g. the slider and disk DLC and lubricant- must have certain minimum thicknesses for their performance. The maximum magnetic signal can be obtained at a spacing of zero, but this requires a contact recording interface.

There are several contact interface designs under consideration for the planned magnetic recording density of 1 Tbit/in^2 in Hard Disk Drives (HDD): "wear in", "proximity", and "full contact". By a "wear-in" interface we mean an air bearing slider that initially flies with its trailing pad contacting the tallest disk asperities. After an initial service period it is expected that these asperities will be slightly worn so that the contact is lost. By "proximity" we mean an air bearing in which there continues to be intermittent or continuous contact between the trailing pad and the disk, while "full contact" means a contact interface without an air bearing. It is expected that all of these technologies, except possibly the last one, will rely on an air bearing to support most of the suspension load, while the trailing pad of the slider is in contact with the disk at the beginning, frequently or continuously. In this sense the HDI has partial contact.

In this paper we develop a nonlinear dynamics model to analyze the bouncing vibration and contact of partial contact sliders. In the model the partial air bearing is obtained through the generalized Reynolds equation modified with the Fukui-Kaneko slip correction and a recent slip correction for the contact condition [1]. The adhesion, contact and friction between the slider and disk are also considered in this model. Realistic measured roughnesses of the slider and the disk are used in the simulation. It is found that minimizing the trailing pad size can reduce the slider's bouncing and tendency to crash. The surface roughnesses and adhesion between the slider and disk have a strong effect on the slider's bouncing vibration, while the friction between the slider and disk has less effect. The slider's bouncing can be decreased without much increase in contact force, to some extent, through increasing the preload.

2. DYNAMICS, ADHESION AND CONTACT MODELS

The generalized time-dependent Reynolds equation is used to model the air bearing between the partial contact slider and the disk. The Reynolds equation is modified using the Fukui-Kaneko (FK) slip correction [2] to account for the rarefaction of the ultra thin air film within the slider/disk spacing. As indicated in Wu and Bogy [1], the FK correction has an unbounded contact pressure singularity for the air bearing with contact. They proposed a new second order slip model without the pressure singularity, which predicts results not far from the FK correction when the modified inverse Knudsen number is small. For the contact region in an air bearing, Huang and Bogy [3] adopted in their Monte Carlo method a no-fly-zone condition, which assumes that air molecules can not enter a gap smaller than themselves. Here we combine the FK model and the new second order slip model. When the air film thickness is larger than 0.3 nm, close to the diameter of an oxygen or nitrogen atom, we use the FK model; when it is less than 0.3nm, we use the new second order slip model to avoid the pressure singularity.

The impact between the partial contact slider and the disk is quasi-static and can be modeled with an elastic contact model based on the static influence coefficient matrix. The

CML slider dynamic simulation shows that the impact speed of the slider is on the order of 10^{-1}m/s. The sliding speed of the slider with respect to the disk, which is proportional to the disk rotation speed and the radial position of the slider, is on the order of 10^{1}m/s. Both speeds are much less than the elastic wave speeds in the disk media. So the slider-disk impact is quasi-static, which means that the deformation is restricted to the vicinity of the contact area and can be obtained through use of static contact theory. Johnson [4] described a contact model based on influence coefficients from an elasticity analysis of loading on an elastic half-space. This model can be incorporated with the approach that approximates the contact between two rough surfaces as that between a rigid flat surface and an equivalent elastic rough surface. We use this model instead of asperity-based contact models, such as that in the CEB model [5], because those models are only valid when bulk deformation and interactions between asperities are negligible. For a partial contact HDI, the flying height at some parts of the air bearing surface (ABS) might be negative, which means that the distance between those parts of the slider and the undeformed disk surface is less than zero. Under this condition, bulk deformation and interactions between asperities are not negligible.

Adhesion is calculated through the modified intermolecular force model [6], which does not suffer from an infinite repulsion pressure when the slider and disk are in contact. The effect of the lubricant is included through the value of the surface energy difference before and after contact. This model is used instead of the asperity-based adhesion models, such as the CEB model, also because of the non-negligible bulk deformation and interactions between asperities.

As to the friction between the slider and disk, we use coulomb's law, the product of the normal contract force and a friction coefficient. Asperity-based friction force models, such as the CEB friction force model [7], are only valid for static friction with negligible bulk deformation and interactions between asperities. They are not suitable for the dynamic simulation of the partial contact HDI.

All of these models were implemented in the CML slider dynamic air bearing program. The ABS is discretized into small grids, which are approximately parallel to the disk surface with various flying heights. The modified Reynolds equation is then discretized using Patankar's control volume method, and the final discretization equations are solved using the alternating direction line sweep method combined with the full multi-grid algorithm. Then the modified intermolecular force model and the elastic contact model are applied to each grid. The suspension is approximated here using three springs and three dampers in the vertical, pitch and roll directions. The dynamic program uses the Newark Beta method to solve the slider dynamics equations.

3. SIMULATION RESULTS AND DISCUSSION

Using the models described above, we analyze the dynamics of a partial contact HDI. We employ micro-trailing pad sliders in the simulations. As was found in [8], in the contact regime a slider with a minimized trailing pad incurs smaller short range attractive forces between the slider and disk as well as less contact force. The ABS design of the slider is shown in Fig. 1, and the related slider, disk and suspension parameters are listed in Table I. In the dynamic simulation we analyze the effect of the trailing pad width, disk roughness, change of surface energy $\Delta\gamma$, friction coefficient and suspension preload on the slider's bouncing vibration and the contact force. These parameters have values of different levels and those with an upper asterisk are the default values used in the simulations. A partial contact HDI should have less slider bouncing and a smaller contact force. Less bouncing keeps a stable head media spacing; smaller contact force does not incur serious wear, and therefore gives a more stable and reliable HDI.

A. Disk roughness and surface energy change

We use one ideally smooth disk and two real disks with measured track profiles in the simulation. The RMS values of the two surface profiles are 0.2 nm and 0.6 nm, respectively. Fig. 2 shows the history of the flying height, pitch angle, roll angle and contact force, as well as the frequency analyses of the flying height, pitch angle and roll angle of the slider on the disk surface with RMS roughness equal to 0.6 nm. It is seen that the slider continuously bounces on the rougher surface with two frequency components; one is around 150 KHz and the other is about 900 KHz. The slider's pitch motion also has these two frequency components. The situation is the same on the disk surface with RMS roughness equal to 0.2 nm. On the flat disk the slider flies on the disk with a slight contact. And its vibration does not have the 900 KHz frequency component. This higher frequency component is evidently associated with the slider-disk contact. The elastic contact between the slider and the disk has a much larger contact stiffness than the rear air bearing, and this evidently causes the higher pitch frequency. The roughness of the disk can excite this high frequency component.

Fig. 3 shows the 3σ of slider bouncing displacement vibration and the mean contact force on disk surfaces with various RMS roughness and $\Delta\gamma$ values. It is seen that disk roughness is the main cause of the slider's bouncing. On the ideally smooth disk the slider achieves steady state without bouncing, while the slider's bouncing increases as the disk surface becomes rougher. Since $\Delta\gamma$ is proportional to the Hamaker constant [6], increased $\Delta\gamma$

means increased slider-disk adhesion. Increased slider-disk adhesion incurs more slider bouncing and larger contact force.

Usually the smoother the disk surfaces are, the larger is the adhesion between the slider and the disk [6]. Our dynamic model separates the roughness factor and the adhesion factor. But we still can see that a partial contact HDI needs to balance the surface roughness and the adhesion.

B. Friction coefficient

The friction coefficient of today's DLC coated disks with lube is less than 1. In our simulation we use three different values, 0, 0.3 and 0.6. Fig. 4 shows the corresponding 3σ of slider bouncing displacement vibration and mean contact force on disk surfaces with two values of $\Delta\gamma$. It shows no dramatic difference between these cases with different friction coefficients. This partially contradicts the analysis of Ono and Yamane [9] on the effect of a wide range of friction coefficients. They asserted that friction excites the slider vibration. Actually the adhesion and contact force exerts larger torques than the friction force with respect to the slider's mass center, since the pitch angle is on the order of several hundred micro radians and the friction force is almost parallel to the ABS. So the effect of the friction force upon slider bouncing might be important when the friction coefficient is larger than 1 and the slider's pitch angle is also very large. In our cases we do not expect major effects of the friction coefficient on the partial contact slider dynamics.

C. Micro trailing pad width

Here we analyze the dynamics of partial contact sliders with the trailing pad widths of 120 μm, 100 μm to 80 μm. Fig. 5 shows the 3σ of the slider bouncing displacement vibration and mean contact force of each design. Obviously the decrease of trailing pad size from 120 μm to 100 μm causes a decrease of the slider's bouncing and contact. However, the further decrease from 100 μm to 80 μm only slightly reduces the mean contact force. Decreasing the slider's trailing pad width can lessen the slider's bouncing and contact to some extent. A partial contact HDI relies on a trailing pad contact to support part of the suspension load. Also the read/write structure needs to be embedded in the trailing pad. So the tailing pad width can not be decreased beyond a certain value. For a stable HDI the micro trailing pad needs to be optimized as part of the ABS design.

D. Preload

The effect of suspension preload on the slider dynamics is analyzed with a simple spring-damper model for the suspension. Three levels of preload are used in the simulation, 0.1 gm, 0.8gm and 1.6gm, where 0.8gm is a typical preload for a femto slider. Fig. 6 shows the 3σ of the slider bouncing displacement vibration and mean contact force under different values of preload. The result illustrates the high nonlinearity of air bearing. The increased preload changes the slider altitude, causing the air bearing force to increase. Hence the mean contact force does not increase as much as the preload. As the preload increases, the mean contact force increases, while the slider's bouncing vibration decreases. However, the slope of the decreasing bouncing vibration is much steeper than that of the increasing mean contact force. This means that the slider's bouncing can be suppressed without much increase in the mean contact force, to some extent, through increasing the preload.

4. CONCLUSIONS

Our nonlinear dynamics model, which includes the generalized Reynolds equations for the air bearing, an elastic contact model for the slider-disk impact and the modified intermolecular force model for the slider-disk adhesion can simulate the partial contact head disk interface. In the dynamic simulations we found that,

(1) As the slider-disk adhesion increases, the slider's bouncing amplitude is increased and the contact force is also increased.

(2) Disk roughness is a main factor of slider bouncing.

(3) The friction coefficient of the disk surface has a slight effect on the slider's bouncing.

(4) Minimizing the trailing pad width can decrease the slider's bouncing and slider-disk contact to some extent.

(5) The slider's bouncing can be suppressed without much increase in the mean contact force through increasing the preload.

ACKNOWLEDGMENT

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TABLE I SLIDER, DISK AND SUSPESNION SPECIFICATIONS

Slider	Trailing pad width: 120 μm, 100 μm*, 80 μm; Slider Size: 0.85×0.7×0.23mm^3; Crown: 18 nm; Camber: 2.5nm; Twist: 0.0nm.
Disk	RMS: 0.0 nm, 0.2 nm*, 0.6nm; Change of surface energy: 0.008 J/m^2 *, 0.08 J/m^2; Friction coefficient: 0, 0.3*, 0.6; Disk RPM: 10000; Slider skew angle: 6.65°.
Suspension	Preload: 0.1 gm, 0.8gm* and 1.6 gm.

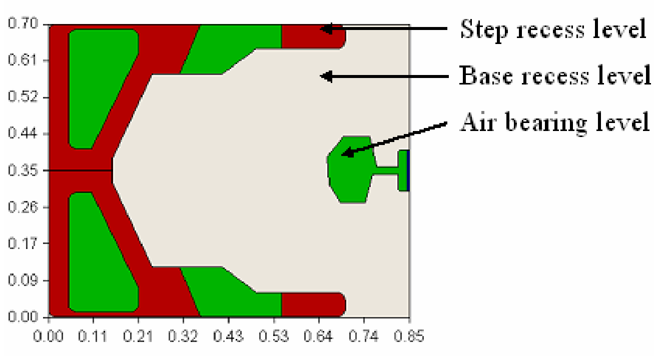

Fig. 1. Air bearing surface design (unit: mm).

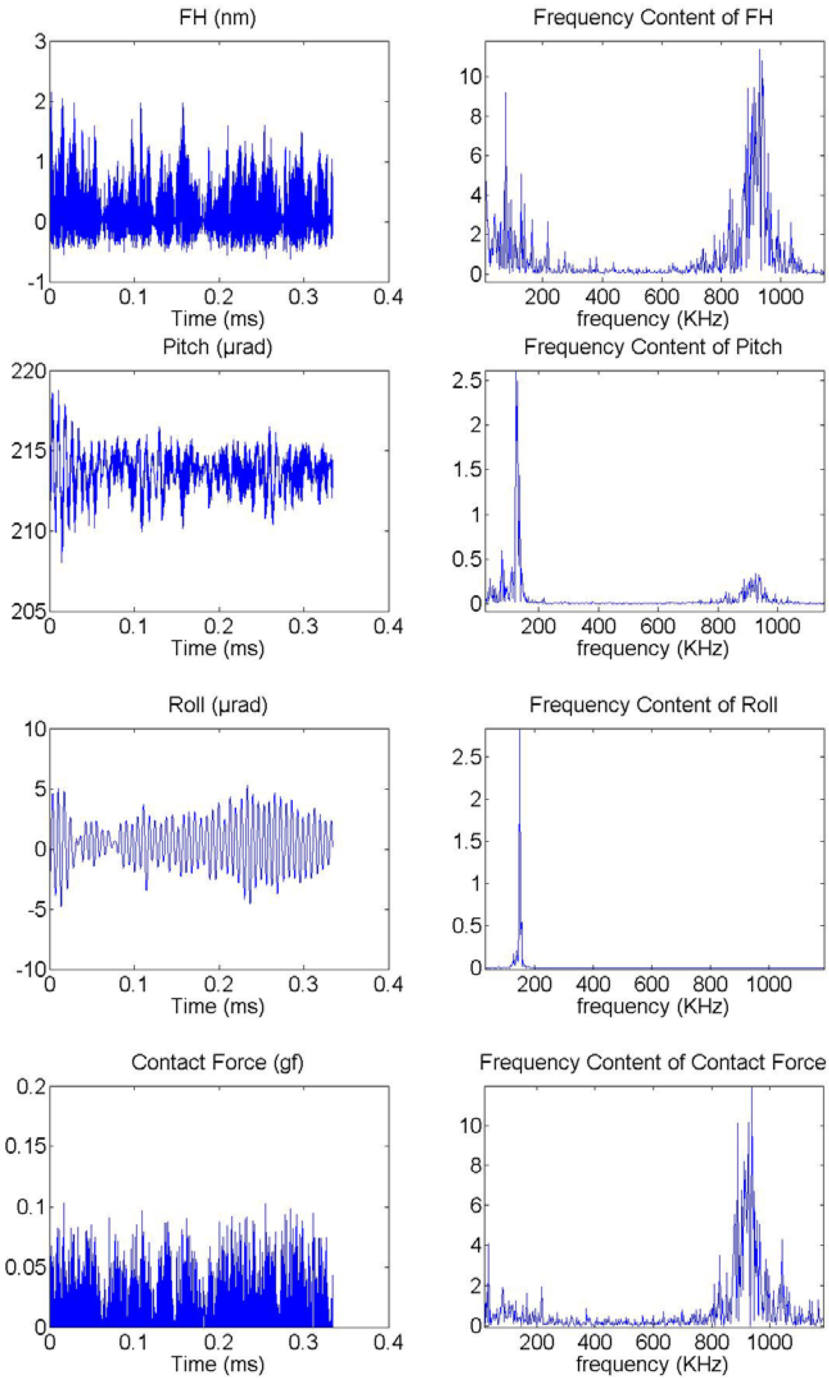

Fig. 2. History of the FH, pitch, roll and contact force and their frequency analyses on the disk surface with RMS 0.6 nm.

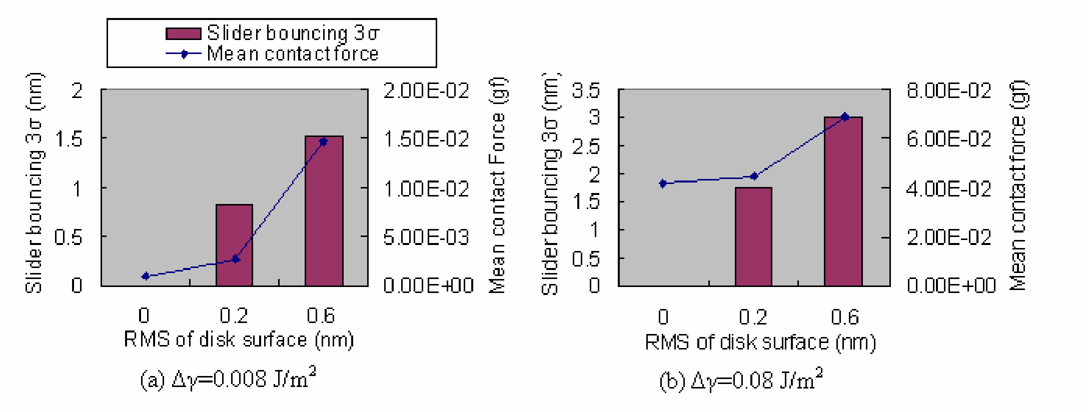

Fig. 3. Slider bouncing 3σ and contact force on disk surfaces with various RMS and Δγ values.

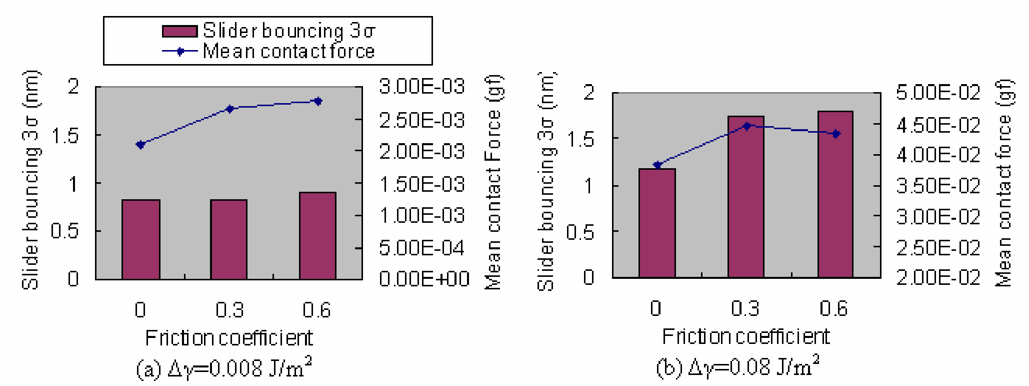

Fig. 4. Slider bouncing 3σ and contact force on disk surfaces with various friction coefficients and Δγ values.

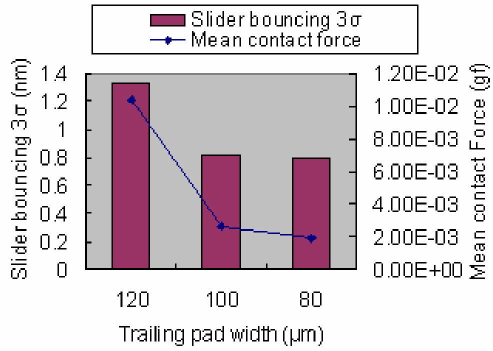

Fig. 5. Slider bouncing 3σ and contact force of micro trailing pad sliders with various pad widths.

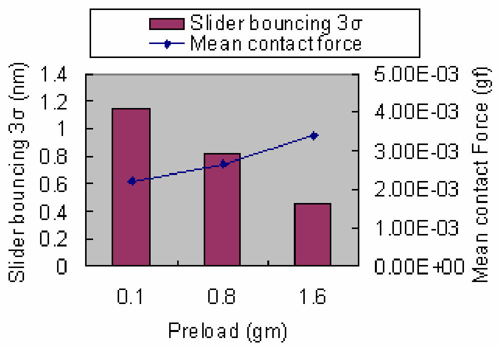

Fig.6. Slider bouncing 3σ and contact force of micro trailing pad sliders with various values of preload.