Simulation of Head-Disk Interface and Ramp/Lift-Tab Interfaces During Load/Unload Process of Hard Disk Drive*

Hongrui AO**, Haodong WEI** and Hongyuan JIANG**

**School of Mechatronics Engineering, Harbin Institute of Technology
92 West Dazhi Street, Nangang Dist., Harbin 150001, P.R. China
E-mail: hongrui_ao@yahoo.com

Abstract

A 2-D dynamic model was proposed to describe the dynamic behavior of the ramp/lift-tab interface of a one-inch hard disk drive with consideration of the air bearing force and contact stages during the unloading process. The contribution of the geometric parameters of the ramp/lift-tab interface on the dynamic performance of the suspension lift-tab was studied through simulation. A modified Reynolds equation and an iterative formulation for numerical scheme were presented considering continuum Poiseuille flows. The dynamic model developed for the lift-tab motion was extended to simulate the contact and separation stages of head-disk interface. The contact forces and deformation at the ramp/lift-tab contact interface were calculated. The effect of the horizontal velocity of the suspension lift-tab on the dynamic performance of the slider was analyzed. The results show that the dynamic characteristics of slider were independent on the geometric parameters of the ramp and the velocity of the suspension lift-tab. The calculation method to solve the Reynolds Equation proposed in the paper is with enough precision and high efficiency.

Key words: Hard Disk Drive, Load/Unload, Ultra-thin Film, Lubrication, Contact

1. Introduction

Contact-start-stop (CSS) technology has been the mainstream design in hard disk drive industry since the introduction of Winchester drives, in which the air bearing slider is parked at a landing zone when the disk drive is in non-operation state. The slider stays at the landing zone during spinning up and spinning down, and moves to the data zone only when the rotational speed of disk reaches operational condition. However, this mechanism may produce high contact force, and generate wear debris [1]. For this reason, a dynamic loading/unloading (L/UL) approach has gained attention in recent years. The L/UL technology can eliminate the head-disk stiction and improve the shock performance effectively. A schematic representation of a ramp loading/unloading disk drive device is shown in Fig. 1(a). The trajectory of the suspension lift-tab on a ramp surface is shown in Fig. 1(b). The air bearing slider rests at point E on a ramp when the disk drive is not in use. When the read-write head needs to access data, the voice coil motor will move the head-suspension-assembly to the disk along the route E-D-C-B-A (see the points on the ramp surface shown in Fig.1(b)) till the lift-tab separates from slope part of ramp surface.

[Received 1 Aug., 2009 (No. 09-0413)] [DOI: 10.1299/jamdsm.4.23]

Copyright © 2010 by JSME
A few researchers have studied robustness and reliability of the ramp loading process both experimentally and theoretically. Zeng and Bogy developed a 4-DOF model for dynamic L/UL simulation [2]. Ao et al proposed a model to describe the interaction between lift-tab and ramp during an unloading process [3]. Thornton et al studied the dynamic characteristics of a slider at ultra-low level by using nonlinear dynamic model [4]. With the introduction of L/UL technology into HDD, the study contents of HDI had been changed. For instance, Zeng and Bogy built a dynamic model to describe the L/UL performance and extended to the study of suspension movement [5]. Furthermore, some experiments have been approached to investigate the dynamic performance of hard disk drives with L/UL technology. The research of Wang et al indicated that when the system was with a very high unloading velocity, the possibility of occurrence of air bearing breaking might increase while the unloading distance might increase with increase of rotating velocity of suspension [6]. Yeager supposed that the wear at the ramp surface would induce more particles at the interface and accumulate in the drive [7]. Those efforts have been focused on dynamic behavior at the slider head and disk interface, in which a few of them were concerned with the design of ramp profile and the selection of ramp materials.

For the theoretical analysis of the L/UL processes, several models were proposed based on different assumptions and hypothesis. Jeong and Bogy used a finite element model to describe the suspension [8]. In the reference [9], the suspension was modeled to 3 pairs of spring-damper systems, and interaction between the lift-tab and ramp was considered as ‘de-gramming’ rate. In the 4-DOF model proposed by Zeng and Bogy [10], the inertial effect of the suspension was considered in the model in order to improve the simulation results. Based on this model, Wei and Liu proposed a 9-DOF model [11], in which a complicated consideration of many parameters was added for the purpose of accurate description of L/UL behavior.

It is noted that the variation of air bearing force during L/UL process was not considered because of the complexity of solving the modified Reynolds equation for ultra-thin film lubrication between disk and slider. Normally, the classic Reynolds Equation are used to obtain the air bearing force at the air bearing surface. However, it is not suitable for the description of gas film when the flying height is reduced to about 10nm. Researchers had proposed many modified models to solve this problem. Fukui and Kanko proposed a so-called F-K model [12] which was used widely in the following studies of head disk analysis. White obtained the solution of Reynolds equation for compressible gas lubrication by using factored implicit scheme [13]. Huang et al [14] proposed a finite difference method which was suitable to improve the iteration precision level of ultra-thin film lubrication with large bearing number.

In the present study, a recent dynamic model [3] for the lift-tab motion was extended to simulate the dynamic behavior of ramp/lift-tab interface with consideration of the effect of air bearing force on the dynamic behavior of the slider. A new approach to solve the Reynolds equation was presented to obtain the air bearing force. Both loading and unloading processes were investigated theoretically.
2. Modeling of the L/UL processes

In order to investigate the effect of the loading and unloading processes on the dynamic performance of the slider suspension, it is essential to understand the interaction and related tribological and dynamic issues between the suspension lift-tab and ramp, as well as interaction between the suspension and the disk.

(1) Modeling of the L/UL system

Figure 2(a) shows the suspension assembly, including suspension lift-tab, gimbal arm, dimple, slider, read/write head and disk. Based on this structure and their interaction during loading and unloading processes, a dynamic model shown in Fig. 2(b) is established. In this model, the function of the limiter which is used to protect the slider when the slider is over-loaded is considered. The equation of motion of this system is written as

\[ \begin{align*}
    m_1 \ddot{z}_1 &= -k_1(z_1 - z_{10}) - k_2[(z_1 - z_{10}) - (z_2 - z_{20})] - c_1 \dot{z}_1 - c_2(\dot{z}_1 - \dot{z}_2) \\
    &\quad + k_{req}(h_x(t) - z_1)^{1/2} + c_r(v_r(t) - \dot{z}_1) \\
    m_2 \ddot{z}_2 &= F_a - F_{pro} + F_c + k_1[(z_1 - z_{10}) - (z_2 - z_{20})] + c_2(\dot{z}_1 - \dot{z}_2)
\end{align*} \]

where \( m_1 \) and \( m_2 \) are equivalent masses of the suspension lift-tab and the gimbal-slider assembly, \( k_1 \) and \( k_2 \) are the equivalent stiffness of the suspension lift-tab and slider, respectively; \( k_r \) is the stiffness of the ramp contact surface, \( c_r \) is the damping coefficient of the ramp surface, \( c_1 \) and \( c_2 \) are the equivalent damping coefficients of the suspension lift-tab and slider, respectively; \( F_{pro} \) is the preloading of the suspension, \( F_a \) is the air bearing force, \( F_c \) is the contact force at the slider-disk interface, \( z_{10} \) is the initial displacement of the suspension at the steady state, \( z_{20} \) is the initial displacement of the slider at the steady state, \( h \) and \( v_r \) are the height and the velocity, \( z_1 \) and \( z_2 \) are the height of the suspension lift-tab and the slider relative to the disk surface, respectively.

It is noted that in the Equation (1), the contact stiffness between \( m_1 \) and \( m_2 \) is dependent on displacement between the dimple and the slider. Figure 3 shows the dependency, that is, when there is a contact between the dimple and slider, the stiffness \( k_2 \) equals to \( k_a \), the slope of line \( a \); when the dimple separates from slider, \( k_2 \) equals to \( k_a \). When the limiter works, the stiffness \( k_2 \) becomes \( k_{lim} \).

(2) Determination of the equivalent stiffness and masses

The finite element method is used to determine the parameters \( k_1, m_1, k_2 \) and \( m_2 \) in the equation (1). Figure 4 shows the finite element model of the suspension and the flexure with loads and boundary conditions. The displacement \( \Delta x \) caused by the load \( F \) can be obtained from this model. From the modal analysis, the first resonant frequency, \( \omega_1 \), of the system also can be calculated. Then, the equivalent stiffness and mass can be calculated.
In the present study, the calculated results for the equivalent stiffness and mass for a commercial hard disk drive product with one inch form factor are as follows: \( k_1 = 21.4 \text{N/m}, \) \( m_1 = 3.64 \times 10^{-6} \text{kg}, \) \( k_2 = 14.1 \text{N/m}, \) \( m_2 = 1.74 \times 10^{-6} \text{kg}. \)

(3) Contact force between the slider and disk

The contact between the slider and disk is simulated as the contact between a smooth plane and plate with multiple spherical asperities with the same radii \( R_0 \) and height \( z, \) as shown in Fig. 5, for the simplification of calculation process. Assume the number of asperities is \( N. \) When the contact surface is loaded in the normal direction, an approach occurs, expresses as \( \delta = z - d, \) where \( d \) is the distance between the smooth plane and the reference surface. Each asperity had the same deformation under this load, so the load shared by each asperity, \( W_{SDi}, \) can be calculated based on Hertz contact theory as

\[
W_{SDi} = \frac{4}{3} E' R_0^{1/2} \delta^{3/2}
\]  

(4)

where \( E' = (1 - v_1^2) / E_1 + (1 - v_2^2) / E_2. \)

Then, the total load can be obtained

\[
F_e = \sum_{i=1}^{N} W_{SDi} = \frac{4}{3} N E' R_0^{1/2} \delta^{3/2}
\]  

(5)

Assume the contact occurs at a small area with a length \( m \) and width \( n, \) then the number of contact can be calculated as \( N = m n \eta, \) where \( \eta \) is the aerial density of asperities.
(4) Contact between the suspension lift-tab and the ramp

The contact at the lift-tab/ramp interface is simplified as the contact between two elliptical cylinders, as shown in Fig. 6. At the contact area, the lift-tab and ramp have radii \( R_1 \) and \( R_2 \), respectively. When loaded with force \( F_r \), there will be elastic deformation with elliptic shape, and a very large contact stress. Based on Hertz contact theory,

\[
F_r = \frac{4}{3} E^* \left( \frac{(R_1 + R_2)}{E} \right) \delta^{3/2} = k_{req} \delta^{3/2}
\]

where \( E^* \) is the equivalent modulus of elasticity, \( E^* = (1 - v_i^2)/E_i + (1 - v_j^2)/E_j \), \( E_1 \) is modulus of elasticity, \( v \) is Poisson’s Ratio; \( k_{req} = 4E^* [(R_1 + R_2)/(R_1 R_2)]^{3/2}/3 \); \( R \) is the radius of the contact surface, the footnote 1 and 2 indicate the ramp and lift-tab, respectively. In the present study, the ramp material is PTFE; the material of suspension lift-tab is steel 302; \( R_1 \) and \( R_2 \) are 0.46mm and 0.1mm, respectively. Then, \( k_{req} = 4.02 \times 10^6 \) N/m.

(5) Air bearing force

The classical Reynolds equation is suitable for the description of the lubrication dynamics when the flying height of the slider is less than the free distance of gas molecular (\( \lambda = 65 \)nm). The steady state distribution of pressure in the air film between the slider and the head is described by nonlinear generalized Reynolds equation, written in dimensionless form as

\[
\frac{\partial}{\partial X} \left( Q(D)H^* P \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Y} \left( Q(D)H^* P \frac{\partial P}{\partial Y} \right) = \nabla^2 (PH)
\]

where \( Q = Q_{con} \), \( H \) is the dimensionless air film thickness, \( \Lambda = 6 \mu U/L \), \( p_0 \) is the ambient pressure, \( h_0 \) is the minimum film thickness, \( \mu \) is the viscosity of gas, \( D \) is the inverse Knudson number, \( Q_{con} \) is coefficient of continuum Poiseuille flow, \( Q_{con} = D/6 \), \( Q_p \) is obtained from the flow rate database proposed by Fukui and Kaneko [12], using local interpolation method.

To solve Equation (7), an iterative formulation for numerical scheme was presented considering continuum Poiseuille flows. The equation was discretized with an upwind finite difference scheme for the convective term on the right, and a centered difference scheme for the diffusion terms on the left. The discretization is focused on the slip flow in the equation. The detailed upwind finite difference method can be found from the paper by D. Srida and N. Balakrishnan [15].

This treatment is good for the convergence at the sharp change of the film thickness.
The iteration scheme is as follows

\[
\hat{P}_{i,j} = \left[ 2\Delta\hat{P}_{i,j} H_{i,j} / \Delta\chi_i + (\hat{q}_{i+1,j} - \hat{q}_{i-1,j}) / \Delta\chi_i + (\hat{q}_{i,j+1} - \hat{q}_{i,j-1}) / \Delta\chi_j \right] \\
\frac{1}{2H_{i,j}} \left[ \Delta\chi_i + \Delta\chi_j \right] 
\]

(8)

where \( \hat{q} = O(\delta)x^2 \), '~' represents the previous iteration results, and '^' is for the current iteration results, i.e.,

\[
\hat{P}_{i,j} = \beta\hat{P}_{i,j} + (1 - \beta)\hat{P}_{i,j}
\]

(9)

where \( \beta \) is a factor.

The calculation of the grid independency for suppressing the numerical smearing of finite difference scheme can be referenced to [14].

The air bearing force is obtained by using integration of pressure distribution, expressed as

\[
F_a = \iint (p - p_{\infty})dA
\]

(10)

It is noted that the pitch and roll angles are set to constants for simplification.

6. Solution of the differential equation

After the parameters mentioned above were determined, they were substituted into Equation (1). A 4-5 order Runge-Kutta method was adapted to solve the equation of motion. It was programmed in MATLAB software.

3. Results and discussion

Figure 7 shows the pressure distribution at the air bearing surface of slider. \( P(0, Y) = 1, P(1, Y) = 1, P(X, 0) = 1, P(X, B/L) = 1 \). Table 1 shows the comparison of the calculated results with the results from reference [4]. It is shown from the results that the calculation method proposed in the present study is with enough precision and higher efficiency.

![Fig. 7 The pressure distribution at the air bearing surface (flying height 10 nm)](image)

<table>
<thead>
<tr>
<th>( h_0 (\text{nm}) )</th>
<th>( F_a (10^2 \text{mg}) )</th>
<th>( F_a ) from Ref.[4]</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>1.588</td>
<td>1.571</td>
<td>1.08%</td>
</tr>
<tr>
<td>15</td>
<td>2.794</td>
<td>2.753</td>
<td>1.49%</td>
</tr>
<tr>
<td>10</td>
<td>4.113</td>
<td>4.028</td>
<td>2.11%</td>
</tr>
<tr>
<td>5</td>
<td>7.012</td>
<td>6.950</td>
<td>0.89%</td>
</tr>
<tr>
<td>2</td>
<td>12.29</td>
<td>12.355</td>
<td>1.07%</td>
</tr>
</tbody>
</table>

The contact forces between the ramp and the suspension lift-tab during loading and unloading processes are shown in Fig. 8. For an unloading process, when the suspension lift-tab approaches to the ramp surface with a ramp angle, a contact occurs between the lift-tab and ramp which causes the vibration of the lift-tab and part of the ramp surface, but this vibration will stop in a very short time because of damping effect at the contact interface. It can be predicted that the vibration will occur at the point where the slope of the ramp surface changes sharply. The vibrating energy will dissipate shortly after the contact begins. These phenomena can be found during either unloading or loading process.
In order to investigate the effect of the horizontal velocity, $V_a$, of the suspension lift-tab on the dynamic characteristics of the slider, the contact force and deformation at the interface for different loading and unloading velocities are compared, as shown in Fig. 9. It can be seen from Fig. 9 that the increase of the velocity $V_a$ causes the increase of the amplitude of contact forces and the deformation of the ramp surface, but reduces the contact frequency in a certain time. This means that the decrease of the loading or unloading velocity will be beneficial to keep a steady working status for suspension lift-tab and slider. However, a low loading or unloading velocity may cause low working efficiency for a hard disk drive in an operational condition. Therefore, it should be helpful to optimize the horizontal velocity of the suspension lift-tab during loading and unloading process.

It is known that the loading and unloading operations in a hard disk drive are cyclic when the hard disk drive is reading and writing data, so it is necessary to evaluate the energy consumption during these processes. The energy dissipated by the friction force, $W$, for an unloading process is calculated as

$$W = \int_{x_e}^{x_A} \mu F_t \, dx$$  \hspace{1cm} (11)$$

where $\mu$ is coefficient of friction, $F_t$ is friction force, $x_e$ and $x_A$ are the positions of the lift-tab at the ramp surface during an unloading process, respectively. Figure 10 shows the comparison of the energy consumed at the ramp.lift-tab interface. It can be concluded that the ramp angle plays an important role during these processes, that is, a small ramp angle...
will be good to save energy.

Fig. 10 Energy consumed by friction force during L/UL process

4. Conclusion

A simplified model was developed to simulate the motion of suspension lift-tab over the ramp surface with consideration of the air bearing force variation during an unloading process. A new solution method for modified Reynolds equation for the description of ultra-thin film lubrication was proposed. The contact forces and deformation at the ramp/lift-tab contact interface during loading and unloading processes were calculated. The effect of the horizontal velocity of the suspension lift-tab on the dynamic performance of the slider was analyzed. It is concluded that the dynamic characteristics of slider were independent on the geometric parameters of the ramp and the velocity of the suspension lift-tab. The calculation method proposed in the paper to solve the Reynolds Equation is with enough precision and high efficiency.

Acknowledgement

The study is part of the project supported by the Development Program for Outstanding Young Teachers in Harbin Institute of Technology (grant No. HITQNJS. 2008.009).

References


