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Configuration Optimization of Two-Rail Sliders on Dynamic Characteristic*

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Abstract: An optimal design methodology for the configuration of two-rail slider was proposed to get better dynamic performance. The taper length, taper height and the rail width of the reading/writing head are considered as design variables. The complex geometry method is utilized as the search scheme in the optimization process. Optimization results show that the new slider has better dynamic characteristics and is more stable than the original designed slider. The optimization process also demonstrates that the optimum model and optimum method is effective.

Key words: optimum design, complex geometry method, slider, configuration design

The flying height between the magnetic recording slider and the disk surface has consistently diminished to increase the recording density in hard disk drives. The head-disk spacing in new design is expected to reach sub-25 nm level soon. At such a low spacing, the slider tends to strike the disk more easily when the disk is at the start and stop operation or when the slider encounters wave bumps on the disk surface and the spacing fluctuation results in output modulation. Thus the dynamic performance of the slider is worth considering and it's necessary to improve the air film's bearing characteristics. The design and application of new slider configuration is one way to meet the demand and achieve high read-write performance. And when the slider is disturbed, it should return to the equilibrium position as soon as possible.

Some authors have devoted to the configuration optimization of the slider. Their work focused on enhancing flying characteristics. Yoon Sang-Joon and Choi Doog-Hoon introduced a design methodology for determining configuration of slider by meeting the desired flying characteristics over the entire recording band[1-3]. Matthew A. O'Hara designed a shaped rails slider by maintaining nearly uniform flying height profile at specified fly height and roll minimization[4]. Sha Lu studied near contact recording slider, the cost measure in the research is defined as the maximum difference in flying height among three points, which are located at the inner, center and outer radius of the disk, with corresponding skews plus the difference of the mean value of these three flying height and the target flying height[5]. Hiromn Hashimoto studied the improvement of the static and dynamic characteristics of slider by optimal design[6]. In their optimal design model, the flying height fluctuation of the reading/writing head over the entire radius of the disk is considered as the optimal objective. However, all of these works were based on static characteristics to design the configuration of a slider subjective to keep the static flying height close to a target value across the recording band or keep the static flying height fluctuation as low as possible.

The central aim of this paper is to present a concept of slider configuration design based on dynamics analysis. In contrast to the static optimum, the design method in this paper is to utilize the dynamics coefficient of the air bearing as cost measure instead of the flying height fluctuation in the steady state condition. The modified Reynolds equation and the perturbation theory are employed to set up the static and dynamic model of the head disk system and the complex geometry method is used to solve the optimum problem. To investigate the effectiveness of our optimal algorithm, the numerical simulations of both original slider and optimized slider are completed and some concluding remarks are given.

1 Optimization Method

1.1 Cost function

The diagram of two-rail slider is shown in Fig.1. The slider is composed of two rails separated by a recess region. Each rail has a flat taper. Since air film bearing pressure is established only by the rails for the

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two-rail slider, we aim to design the shape of the rail and choose taper length l_T, taper height h_T and rail width b as design variables.

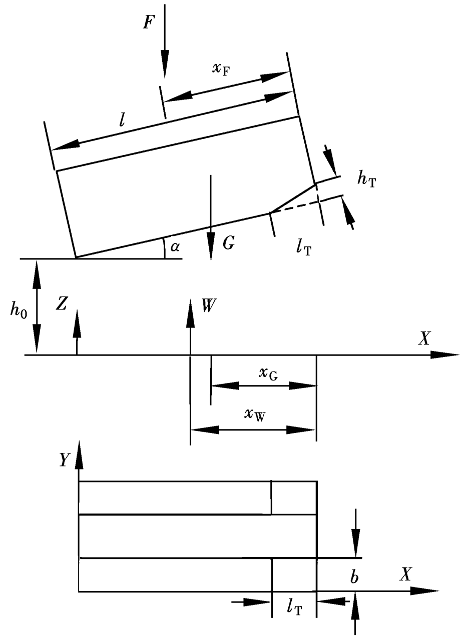


Fig.1 Slider geometry

The flying stability of the slider depends on the dynamic characteristics of the air film bearing, which are expressed by the dynamic coefficients, such as stiffness and damping coefficients. It is believed that the stiffer the air film bearing is, the smaller flying height fluctuation the slider experiences. On the other hand, translation stiffness has more effect on flying characteristic than the other stiffness, such as pitching stiffness. So denoting the vector of the design variables as $\boldsymbol{X} = [x_1, x_2, x_3]^T = [l_T, h_T, b]^T$, we can formulate the optimization problem as follows: Find the design variable vector $\boldsymbol{X}$ to maximize

$$F(\boldsymbol{X}) = K_{11} \tag{1}$$

For a two-rail slider flying above a rotating disk, Reynolds equation is used to describe the air lubrication problem. By assuming no roll, and considering gas compressibility and rarefaction, the modified Reynolds equation for the slider is written in the nondimensional form as

$$\frac{\partial}{\partial X}\left[\hat{Q}PH^3\frac{\partial P}{\partial X} - \Lambda_x PH\right] + \frac{\partial}{\partial Y}\left[\hat{Q}PH^3\frac{\partial P}{\partial Y}\right] = \sigma\frac{\partial}{\partial T}[PH] \tag{2}$$

where H is the nondimensional film thickness; X and Y are nondimensional coordinate variables of the slider along length direction and width direction; P is the nondimensional pressure distribution in the head disk surface; T is nondimension time; $\hat{Q}$ is flow factor; Λ_x is bearing number and σ is squeeze number.

Using perturbation theory in Eq.(2), we can get the static pressure equation

$$\frac{\partial}{\partial X}\left[P_0H_0^3(\hat{Q})_0\frac{\partial P_0}{\partial X}\right] + \frac{\partial}{\partial Y}\left[P_0H_0^3(\hat{Q})_0\frac{\partial P_0}{\partial Y}\right] = \Lambda_x\frac{\partial}{\partial X}[P_0H_0] \tag{3}$$

and dynamic pressure equation

$$\begin{aligned}&\frac{\partial}{\partial X}\left[(\hat{Q})_0(3P_0H_0^2\Delta H + \Delta PH_0^3)\frac{\partial P_0}{\partial X} + P_0H_0^3 \times \left(\frac{\partial\hat{Q}}{\partial P}\cdot\Delta P + \frac{\partial\hat{Q}}{\partial H}\cdot\Delta H\right)\frac{\partial P_0}{\partial X} + \left(P_0H_0^3(\hat{Q})_0\frac{\partial\Delta P}{\partial X}\right)\right]\\&+\frac{\partial}{\partial Y}\left[(\hat{Q})_0(3P_0H_0^2\Delta H + \Delta PH_0^3)\frac{\partial P_0}{\partial Y} + P_0H_0^3 \times \left(\frac{\partial\hat{Q}}{\partial P}\cdot\Delta P + \frac{\partial\hat{Q}}{\partial H}\cdot\Delta H\right)\frac{\partial P_0}{\partial Y} + \left(P_0H_0^3(\hat{Q})_0\frac{\partial\Delta P}{\partial Y}\right)\right]\\&= \Lambda_x\frac{\partial}{\partial X}(P_0\cdot\Delta H + H_0\cdot\Delta P) + \sigma\frac{\partial}{\partial T}(P_0\cdot\Delta H + H_0\cdot\Delta P)\end{aligned} \tag{4}$$

where P_0 and H_0 are the static pressure and film thickness; ΔP and ΔH are the dynamic pressure and film thickness. From Eq. (4), the dynamic pressure distribution can be obtained and then the cost value and other dynamic coefficient can be calculated.

The work of calculating dynamic coefficient has been finished and has been compared with publications results to verify the solution precision[7].

The dynamic characteristics of the slider are analyzed using a two-degree-of-freedom model. Neglecting the suspension stiffness for translation and pitching motion, which are negligible compared with the air film stiffness, the dynamic equations are

$$\left.\begin{aligned}M\ddot{Z} + C_{11}\dot{Z} + C_{12}\dot{\Theta} + K_{11}Z + K_{12}\Theta = 0\\ I\ddot{\Theta} + C_{21}\dot{Z} + C_{22}\dot{\Theta} + K_{21}Z + K_{22}\Theta = 0\end{aligned}\right\} \tag{5}$$

where K_{ij} is nondimensional air film stiffness; C_{ij} is nondimensional air film damping coefficient, $i = j = 1$, translation mode; $i = j = 2$, pitching mode; $i \neq j$, couple mode. By solving Eq.(5), we can simulate the dynamic response of the slider.

1.2 Constraints and optimum method

There is a performance requirement, static flying state, which must be satisfied for the configuration optimization of the two-rail slider. Through satisfying this constraint, we get flying attitude parameters of the slider: flying height h_0 and pitch angle α.

Fig.1 presents all forces acting on the slider that consists of suspension force F, slider gravity G and net air bearing load W, along with their locations. The residuals of forces and moments, r_1 and r_2, are defined as

$$\left.\begin{aligned} r_1 &= F + mg - W \\ r_2 &= x_F \cdot F + x_G \cdot mg - x_W \cdot W \end{aligned}\right\} \tag{6}$$

where

$$\left.\begin{aligned} W &= \iint (p - p_a)\,\mathrm{d}x\mathrm{d}y \\ x_W &= \iint x \cdot (p - p_a)\,\mathrm{d}x\mathrm{d}y / W \end{aligned}\right\} \tag{7}$$

$r_1 = r_2 = 0$ means that the slider is in the steady state. Performance constraint can be formulated as follows: find h_0 and α to minimize

$$\psi(X) = R_1{}^2 + R_2{}^2 \tag{8}$$

where

$$\left.\begin{aligned} R_1 &= \frac{r_1}{F + G} = \frac{F + G - W}{F + G} \\ R_2 &= \frac{r_2}{M_F + M_G} = \frac{M_F + M_G - M_W}{M_F + M_G} \end{aligned}\right\} \tag{9}$$

Air pressure distribution p in Eq. (7) can be calculated by solving static pressure Eq. (3) using line-by-line method. This work has also been finished in our previous work[8]. The flying state parameters were calculated with optimization technology, detail solution procedure was explained in our previous work[9].

The side constraints are

$$x_i^L \leqslant x_i \leqslant x_i^U \qquad x_i = 1,2,3 \tag{10}$$

Eq.(10) is imposed to explicitly bound the values of the design variable and the flying attitude parameter within practically allowable ranges, where lower and upper bounds are denoted by superscripts "L" and "U", respectively.

Based on the problem formulation, the complex geometry method is used to solve the optimization model. The overall procedure of the proposed approach for configuration optimization is illustrated in Fig.2.

At first, the initial values of the design variables are assumed, which define the configuration of the two-rail slider. Given the slider's configuration, the flying attitude parameter values can be calculated in the static analysis module. Using these values, the cost and constraint value can be obtained in the cost/constraint evaluation module. These values of the current design are fed back into the optimization module, and then the design variables are updated by using an optimization technique. The whole process is repeated until the convergence criteria are met. In the optimization, we always use static module and cost/constraints evaluation module whenever the design variables are modified.

In the static analysis, the air bearing pressure W is calculated by solving the lubrication equation with the Fukui-Kaneko model considering high Knudsen numbers[10]. The finite difference technique and control volume formulation are used to discrete the lubrication equation to get the numerical solution. The air bearing stiffness coefficients are obtained by the use of the perturbation technique. In the perturbation techni-

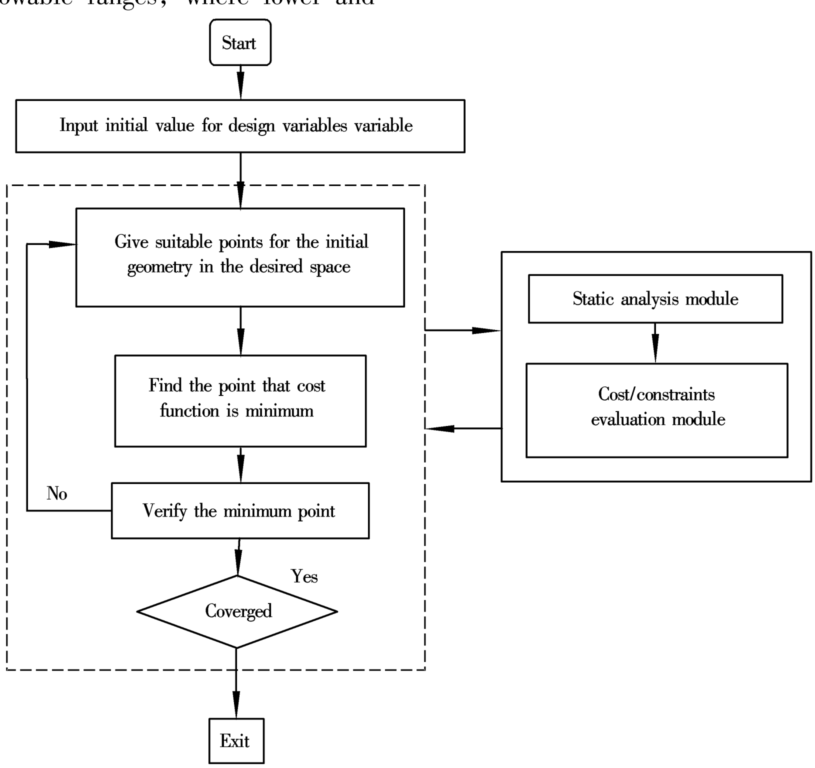


Fig.2 Flowchart of configuration optimization que, the modified Reynolds equation is divided into two set equations after the Laplace transformation. One is steady equation for steady state pressure, and the other is for dynamic pressure. The two set equations can be solved with the same numerical procedure as for modified Reynolds equation. And thus, dynamic coefficients are obtained from the results of dynamic pressure.

2 Results and Discussion

The configuration optimal design of two-rail slider is performed. The design parameters of the slider used in this study are shown as follows:

$L = 500\,\mu\text{m}$, $b = 51.5\,\mu\text{m}$, $l_T = 50\,\mu\text{m}$, $h_T = 435.6\,\text{nm}$, $\Lambda_b = 12$, $D_0 = 0.69$, $F = 0.3\,\text{mN}$, $M = 0.3\,\text{mg}$, $x_F = 265.5\,\mu\text{m}$, $x_G = 255\,\mu\text{m}$.

The lower and upper bounds of the design variables and the flying attitude parameter are

$l_T^L = 25\,\mu\text{m}, l_T^U = 70\,\mu\text{m}$, ${h_T}^L = 0.2\,\mu\text{m}$

$h_T^U = 0.7\,\mu\text{m}$, $h_0^L = 5\,\text{nm}$, $h_0^U = 90\,\text{nm}$

$\alpha^L = 10\,\mu\text{rad}, \alpha^U = 800\,\mu\text{rad}$

The final optimized results are listed in Tab.1. Compared with original design, the taper height and rail width of the optimum design increase, while the taper length decreases. Tab.2 shows the nondimensional air film stiffness and damping coefficient of both Z direction and pitching direction. From Tab.2, we can find that the translational stiffness, pitch stiffness, translational damping coefficient and pitch damping coefficient of the optimum designed slider all increase.

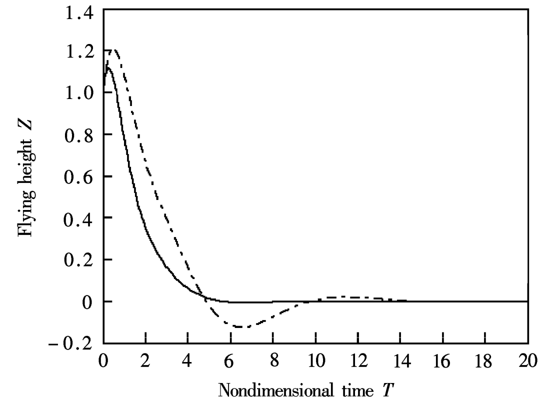
Tab.1 Optimum results and flying attitude parameters

Slider	l_T/μm	h_T/nm	b/μm	h_0/nm	α/μrad
Original	50.0	435.6	51.5	50	202
Optimum	27.5	700.0	80.0	54	119

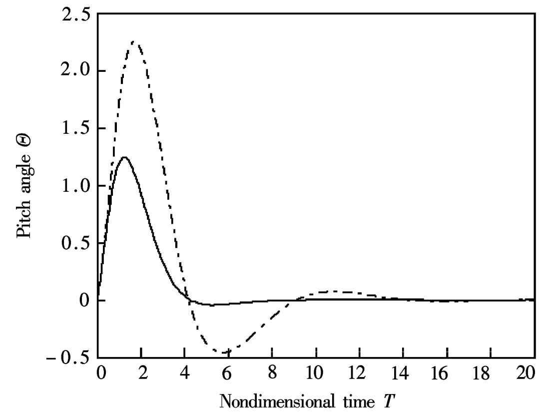
Tab.2 Nondimensional stiffness and damping coefficients of the optimum design and the original design

Slider	K_{11}	C_{11}	K_{22}	C_{22}
Original	0.035 9	0.035 5	0.003 52	0.002 35
Optimum	0.064 7	0.072 0	0.004 96	0.004 50

Fig.3 presents the dynamic response of the two sliders. Fig.3(a) gives the transient response along Z direction. In Fig.3 the horizontal axis is nondimensional time T and the vertical axis is the transient response along Z direction. In initial excitation the slider was lift to a position with twice the minimum space of the steady flying height at $T = 0$. In Fig.3(a) the optimum slider reaches equilibrium position at $T = 6$, while the original slider reaches equilibrium position at $T = 16$, and the vibration amplitude of optimum slider is lower than original slider. The same conclusion we can make in Fig.3(b) that presents the transient response along pitch angle direction.



(a)



(b)

Fig.3 Transient response of the optimum design (solid) and the original design (dashed). (a) Z direction; (b) Pitch angle direction

Fig.4 gives the transient response on different track radius of the slider. In Fig.4 there are four curves labeled 1, 2, 3, 4 which stand for double track radius (original slider), double track radius (optimum slider), half track radius (original slider), half track radius (optimum slider), respectively. From Fig.4, we can find that the response discrepancy between original slider and optimum slider is obvious for different track radius. Optimum slider has better transient response along Z direction and the pitch angle direction than original slider no matter how varies the track radius does. Fig.3 and Fig.4 can prove that the shape of the slider has an important influence on the dynamic response of the head/disk system and also prove that the optimum slider's dynamic performance is better than original slider.

3 Conclusion

A configuration optimization method is proposed in this paper for the design of two-rail slider. A multi-

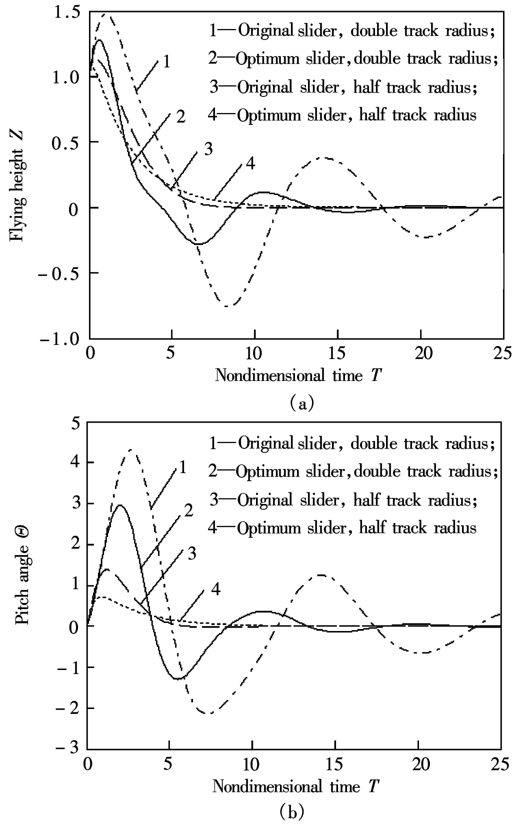


Fig.4 Transient response of optimum design and original design on different track radius. (a) Z direction; (b) Pitch angle direction

criteria optimization problem was formulated to maximize the translational stiffness, along with boundary constraints and steady state constraints. A numerical procedure of the complex geometry method was established to find the optimal values of taper length, taper height and rail width. Then a computer program implementing the numerical procedure was developed and applied to the design of a two-rail slider.

The optimal configuration to meet the design requirement was presented. Simulation results demonstrated the effectiveness of the suggested optimization design scheme by showing that the dynamic characteristics of the optimally designed slider fulfill the performance requirement mentioned in this study. Even though the design methodology only concentrated on the shape of the rail, it is believed that the same methodology can choose more parameters to act as designing variable, and can be applied to the design of other slider types.

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磁头形状的动态优化设计

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摘　要　提出了一种磁头形状的动态优化设计方法,优化设计的目标是使系统有更好的动力学性能.优化设计变量取磁头的斜台长度,斜台高度和磁头单轨轨道宽度,对优化模型采用复合形法求解.初始磁头和优化磁头的动力学仿真结果表明,优化磁头有更好的动力学特性,且更加稳定.结果也表明本文的优化模型和优化算法是有效的.

关键词　优化设计, 复合形法, 磁头, 形状设计

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