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Multiobjective Optimization of Hard Disk Suspension Assemblies: Part I - Structure Design and Sensitivity Analysis

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Abstract

A multiobjective optimization of the suspension assemblies of hard disk drives is demonstrated. The natural frequencies of the suspension assembly are treated as objective functions, subject to side constraints of design variables that describe the shape of the suspension. The gradient information in the optimal design is calculated by finite differences, and the sensitivity analyses are performed numerically with respect to design variables. Preloading and air bearing effects are both considered in the optimal design. Moreover, a final solution is determined by the decision maker with additional knowledge and engineering experience.

1. Introduction

High technologies of the magnetic storage devices are being rapidly developed year in and year out. The hard disk drives have been made small and light-weight, with high-area recording density as well as fast and accurate access. Therefore, the slider is required to be kept extremely close to the rotating disk, and the flying height is retained as small as possible for submicron positioning precisions. Since the flying height is observed always to oscillate but at a very small amplitude, its excessive vibration induced by disturbances must cause a catastrophic damage to the hard disk system. Major sources of disturbances include vibrations from actuator arm, surface roughness of the disk, spindle/bearing inaccuracies (Naruse et al., 1983), rotating flow around the suspension (Tokuyama et al., 1991), and so on.

To reduce the magnitude of the flying height fluctuations, either passive or active designs of the suspension assemblies or servo actuators can be used. It is quite intuitive and natural for the design objective to raise natural frequencies of the suspension assembly so that it will not be excited easily by undesirable disturbances. There exist two strategies of the structure optimization with frequency requirements. Either the weight of the structure is minimized subject to frequency constraints, or the frequencies are maximized subject to the constraint on the shape, weight or frequency distributions. It is very often that a designer minimizes structural weight with frequency constraints (Grandhi, 1993).

It is relatively straightforward to treat natural frequencies as objective functions. Described in Bendsoe et al. (1983-84) was structural optimization with multicriterion objectives, which were expressed in terms of displacement, stress, compliance, eigenvalue, or some other measure of structural performance. Szyszkowski (1991) presented a method of optimization of maximum frequency of free vibrations, handling structures which might experience multimodal eigen-solutions during the solution phase. Szyszkowski and King (1993) derived optimality criteria to maximize a set of frequencies for a structure of given weight. An error norm was proposed and used to determine the values of design variables and Lagrange multipliers at optimum.

2. Modelling of Suspension Assembly

The hard disk suspension assembly consists of a mounting block, suspension beam, flexure and slider, as shown in Figure 1. The finite element model is created with the undeformed state of suspension. After loaded onto the hard disk, the suspension undergos a preloading force between 0.093N and 0.147N (Ruiz and Bogy, 1990), and the angle between the suspension and the base line of the mounting block reduces about 2 degrees. This loading process can be performed by analysis procedures of ABAQUS (a registered trademark of Hibbit, Karlsson and Dorensen, Inc.), where the user simply divides the loading histories into *steps* and comes up with the deformed state of pre-stress for subsequent analyses.

The air bearing that separates the slider from the media during the operation is approximated by 4 linear springs supporting at four corners of the slider. Since the flying height at front corners of the slider is higher than that at rear corners, the stiffness of each front corner is modeled by $10^5 Nt/m$, while the stiffness of each rear corner is $1.5 \times 10^5 Nt/m$. The corresponding flying height is around $0.25\mu m$ under a load force about $0.01kg$, according to the experiments on the measurement of flying height (Pan, 1993) as shown in Fig. 2.

2.1. Finite Element Model

The finite element model for the suspension assembly includes a grand total of 232 elements, 321 nodes, with 1794 degrees of freedom. Two types of elements are used. For the suspension and flexure, we use the quadrilateral thin shell element (S4R5) of 4 nodes, each of which includes 3 orthogonal displacements and two rotations along the shell plane. For the slider, the 8-node linear brick (C3D8H) is used, incompressible and hybrid with 3 displacement degrees of freedom for each node. The mounting block is regarded as a fixed rigid body, to which the suspension beam is connected with weld points.

The mesh generation subroutine defines nodal coordinates as functions of design variables. Four shape parameters are selected as design variables:

- yb: width of the base of the suspension
- xd: length of the suspension base attached on the mounting block
- $x2$: distance between the bending corner and the suspension flaps
- wh: height of the suspension flaps

Other parameters are left constants, including the thickness (th), tip width (yd) and total length (l) of the suspension, the distance from the edge of the mounting block to the line where the suspension width tapers, and the sizes of flexure and slider. The reasons why we do not choose all the above parameters as design variables are explained in Section 4 with sensitivity analyses. The finite element models of the suspension assembly as well as its components are shown in Fig. 3.

Since the large deformation in the loading process of the suspension onto the hard disk is a nonlinear problem, ABAQUS uses Newton's method with iterations to come up with the loaded state of the suspension with pre-stress and pre-stain distributions in the finite element model. This model is used in the subsequent dynamical analysis for each optimization loop.

2.2. Optimization Model and Algorithm

The equations of motion of the suspension assembly for the finite element model of order N have the form

$$M(\mathbf{x})\ddot{\mathbf{q}} + C(\mathbf{x})\dot{\mathbf{q}} + K(\mathbf{x})\mathbf{q} = B\mathbf{u} \qquad (1)$$

where $\mathbf{x}$ is a vector of design variables, $M(\mathbf{x})$ is a positive definite symmetric mass matrix, $K(\mathbf{x})$ is the nonnegative symmetric stiffness matrix, including the equivalent stiffness due to the preload produced pre-stress and pre-strain, and $\mathbf{q}$ is the generalized nodal coordinates. In the modal analysis during the optimization process, the input $\mathbf{u}$, its influence matrix B, and the damping matrix $C(\mathbf{x})$ are neglected while searching for system natural frequencies and natural modes. Once we obtain the optimal design of the suspension assembly, the modal damping ratios are included in the modal equations so that more analyses, such as time responses, vibration controls, etc., can be performed.

The multiobjective optimization problem can be stated as follows:

minimize the cost functionals

$$f_1 = \frac{1}{\omega_1}, \quad f_2 = \frac{1}{\omega_2}, \quad f_3 = \frac{1}{\omega_3 - \omega_2} \qquad (2)$$

subject to the minimum and maximum values of design variables.

The optimization of the suspension assembly is investigated with two techniques: goal programming (Evans, 1984) and compromise programming (Zeleny, 1982). The principles and solution procedures have been described by Tseng et al. (1992). The optimization algorithms are provided by MOST (Multifunctional Optimization System Tool) (Tseng et al., 1993), in which the design variables, initial sizes of the suspension assembly, optimization techniques, objective functions, gradient calculation, and so on, are defined and coded in C language. The ABAQUS is called internally by MOST whenever the structural analysis for eigenvalues and eigenvectors are requested. The designer can either terminate the design loop for any feasible intermediate design, or wait for the final results.

3. ABAQUS/MOST Interface

The optimization is performed by the multiobjective optimization program MOST and the finite element analysis program ABAQUS. An interface between these two programs is developed on a workstation under the UNIX operating system. The sensitivity information is obtained to calculate gradients numerically by finite differences. The optimization of the suspension assembly is then investigated with goal programming and compromise programming techniques provided by MOST. The final design is decided by the decision maker through additional engineering knowledge and experience.

The software package ABAQUS is designed to provide for advanced structure analyses. One of the most attractive features is "step", a portion of the analysis history. This approach is extremely effective for nonlinear problems, since structural dynamical responses may change drastically during an analysis step. The interface between the finite element program ABAQUS and the optimizer MOST is based on the MOST programming structure under the UNIX operating system. In the proposed interface architecture, MOST acts as a master program which internally invokes the execution of command files to perform the finite element analysis in ABAQUS.

4. Sensitivity to Design Variables

In this paper, the sensitivities of natural frequencies of the suspension assembly to the design variables are numerically investigated. In Figs. 4 through 7 the sensitivity curves of the first three natural frequencies are displayed with respect to a single design variable, while other design variables are fixed as the original values of the suspension assembly. We found that in Fig. 4 the first two natural frequencies decrease as the width of the base of the suspension yb increases, but the third natural frequency has a maximum value. Figure 5 shows that the first and third natural frequencies decreases as xd, the length of the suspension base attached on the mounting block, increases. The second natural frequency has a maximum value as xd is around $10mm$ at which, however, the difference between the second and the third natural frequencies seems to be a minimum. There should have a compromise between the design objectives.

In Fig. 6, the natural frequencies tend to decrease as $x2$, the distance between the bending corner and the suspension flaps, increases, while the sensitivity curve of the third natural frequency has a minimum. The maximum difference of the second and the third natural frequencies seems to happen in the region where $x2$ is less than 3 mm. Fig. 7 shows that the first two natural frequencies are increasing smoothly as the height of the suspension flaps wh is increasing, whereas the third natural frequency reaches a maximum as wh is around $0.5mm$ and separates the most from the second natural frequency.

From structural dynamics point of view, it is essential that the longer the suspension is, the smaller its natural frequencies are; and the thicker the suspension beam is, the larger its natural frequencies are. It is obvious that the natural frequencies of the suspension assembly vary nonlinearly with the design variables yb, xd, $x2$, and wh, nevertheless the sensitivity curves are approximately linear with variables l and th. For the subsequent optimal design, l and th are not selected as design variables since their corresponding sensitivity curves are monotonic.

5. Optimization Results

5.1. Preloading

The optimization process is performed with the suspension assembly in its loaded status. The first 6 natural frequencies of the original suspension in the unloaded and loaded positions are compared in Table 1. Due to the preload induced pre-strain and pre-stress in the suspension beam, the natural frequencies of the first bending mode and the first torsional mode become smaller than those of unloaded beam. This preloading process is accomplished by the feature of "step" defined as a portion of analysis history in ABAQUS.

5.2. Optimal Shape Design

The optimal design results of the suspension assembly by the use of goal and compromise programmings for β, γ =1 and 2 are listed in Tables 2 and 3, respectively.

Table 1: Natural frequencies of suspension assembly

	Unloaded		Loaded	
Mode	Freq.(Hz)	Type	Freq.(Hz)	Type
1	2075	bending	2057	bending
2	2713	torsional	2608	torsional
3	5654	torsional	4645	torsional
4	5852	bending	5779	bending
5	7022	torsional	7228	torsional
6	10942	bending	11057	bending

Table 2: Design results with goal programming

	β=1	β=2
Design Variable	(mm)	
yb	7.6683	7.1136
xd	1.7676	2.2518
$x2$	0.4335	0.5181
wh	0.8000	0.8000
Objective		
f_1	3.9489e-4	3.9367e-4
f_2	3.8020e-4	3.6245e-4
f_3	2.5802e-4	2.8014e-4
Freq. (Hz)		
ω_1	2532	2540
ω_2	2630	2759
ω_3	6506	6329

We can conclude the following:

(1) The flap height can be built with its maximum allowable value, which is $0.8mm$ in our design.

(2) The final results on raising natural frequencies are very satisfactory. Without too much change from the original shape of the suspension, the first and second natural frequencies are raised over 400 to 500 Hz. Also, the difference between the second and third natural frequencies increases over 1000 to 2000 Hz.

Table 3: Design results with compromise programming

(α=1)	γ=1	γ=2
Design Variable	(mm)	
yb	6.8513	6.3979
xd	2.4359	2.9607
$x2$	0.3533	0.4663
wh	0.8000	0.8000
Objectives		
f_1	3.8967e-4	4.0530e-4
f_2	3.5747e-4	3.4127e-4
f_3	2.9920e-4	3.4601e-4
Freq. (Hz)		
ω_1	2566	2467
ω_2	2797	2930
ω_3	6140	5820

5.3. Decision Making

The design objective is to raise natural frequencies of the suspension assembly so that it will not be excited easily by undesirable disturbances. This motivates the designer to investigate frequency responses of the optimal suspension shape under the action of disturbances. Assume that the major disturbance comes from the turbulence flow between disks and exerts a vertical force perpendicular to the horizontal surface of the suspension. One may take point a, which is near the centroid of the suspension beam as shown in Fig. 3, as a disturbance input point where the resultant force concentrates, and choose a bottom corner where the head coil locates as the displacement output point. In such cases, the symmetrical torsional modes are uncontrollable. The transfer function of the system with the force input and displacement output can be easily derived by modal equations from the finite element analysis of the resulting optimal suspension assembly.

Three transfer functions are formulated by selecting the first six modes of the corresponding design of the suspension assembly of the original dimensions, goal programming and compromise programming, respectively. Note that modes 2, 3 and 5 are torsional modes, symmetric and hence uncontrollable, and the modal magnitude of mode 6 is much less than those of other bending modes, therefore only two bending modes are present in the Bode plots as shown in Fig. 8. The dashed curve represents the Bode plot of the original suspension system, which has the highest dc gain and resonant peak. Both the dc gain and resonant peak in the Bode plot of the suspension with compromise programming ($\gamma = 1$) design reduce about 5dB, while those with goal programming ($\beta = 1$) design reduce about 9dB. From the frequency response point of view, the decision maker may choose the final design with goal programming techniques.

6. Summary and Conclusions

The optimization of the suspension assembly of hard disk drives by treating the natural frequencies as objective functions has been presented. The design objective is to maximize natural frequencies of the suspension assembly in order not to be easily excited by undesirable disturbances. The design variables are geometrical sizes of the suspension assembly, and are reduced to four by the sensitivity analysis.

The optimization of the suspension assembly is investigated with two multiobjective optimization techniques — goal programming and compromise programming. Different design philosophies yield various optimal design results. Decision makers, therefore, have to choose the best or most proper design based on their engineering knowledge and experience, such as frequency response, transient response and material characteristics, and so on.

Acknowledgment

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APPENDIX A

Original Dimensions of the Suspension Beam

A. Suspension Beam (unit: *mm* and *degrees*)

XL	24.8	X0	1.0	X1	1.5	X2	3.4
YL	6.8	TH1	0.076	ZL	0.65	θ	10.0
A13	0.15	A14	3.005	A15	4.05	A16	0.75

$E = 2.0601 \times 10^{11} N/m^2$ (Young's modulus)
$d = 7.8 \times 10^3 kg/m^3$ (density); $\mu = 0.33$ (Poisson ratio)

APPENDIX B

Goal Programming (Evans, 1984)

The objective of goal programming is to minimize the errors between the optimal solution and the ideal solution in the objective function space. The errors, namely the under-achievement and over-achievement of the i^{th} objective function $f_i(\mathbf{x})$, are defined respectively by

$$d_i^+ = 0.5[|f_i(\mathbf{x}) - T_i^*| + (f_i(\mathbf{x}) - T_i^*)] \quad (3)$$
$$d_i^- = 0.5[|f_i(\mathbf{x}) - T_i^*| - (f_i(\mathbf{x}) - T_i^*)] \quad (4)$$

where T_i^* represents the target or goal set by the decision maker for the i^{th} objective function. Therefore, the general formulation for nonlinear optimization problems can be stated as:

$$\min \quad F(\mathbf{x}) = \left\{ \sum_{i=1}^{q} (d_i^+ + d_i^-)^\beta \right\}^{1/\beta} \quad ; \quad \beta \geq 1 \quad (5)$$

subject to the constraints $h_j(\mathbf{x}) = 0; j = 1, 2, ..., p$ and $g_j(\mathbf{x}) \leq 0; j = 1, 2, ..., m$.

Compromise Programming (Zeleny, 1982)

The distance measure used in compromise programming to evaluate how close the set of nondominated points come to the ideal point is the family of L_γ matrices defined as

$$\min \quad L_\gamma = \left\{ \sum_{i=1}^{q} \alpha_i^\gamma \left| \frac{f_i(\mathbf{x}) - f_i^*}{f_{i,max} - f_i^*} \right|^\gamma \right\}^{1/\gamma} \quad ; \quad 1 \leq \gamma < \infty \quad (6)$$

subject to $\mathbf{x} \in \mathbf{X}$, in which α_i are weights, f_i^* and $f_{i,max}$ are, respectively, the minimum value and the worst value of the i^{th} objective function, $f_i(\mathbf{x})$ is the value of implementing the design variable $\mathbf{x}$ with respect to the i^{th} objective, and $\mathbf{X}$ is the feasible design space.

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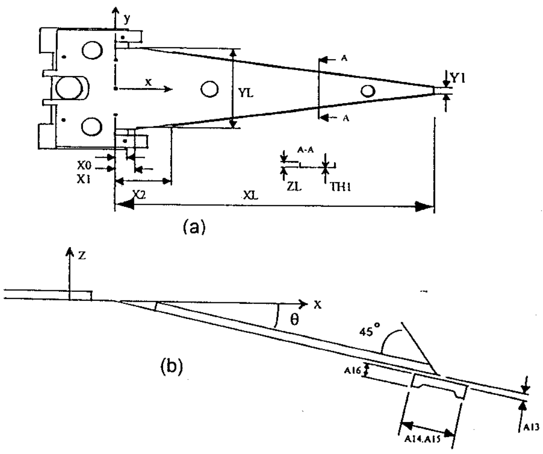

Fig. 1 Suspension assembly

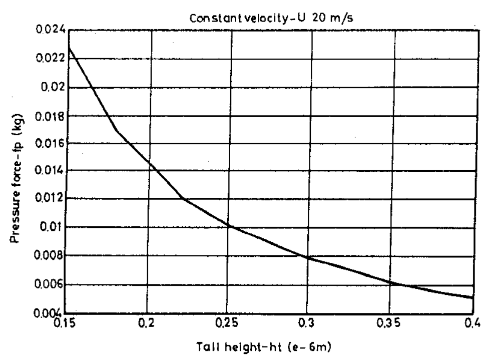

Fig. 2 Resultant air bearing load vs. flying height of slider

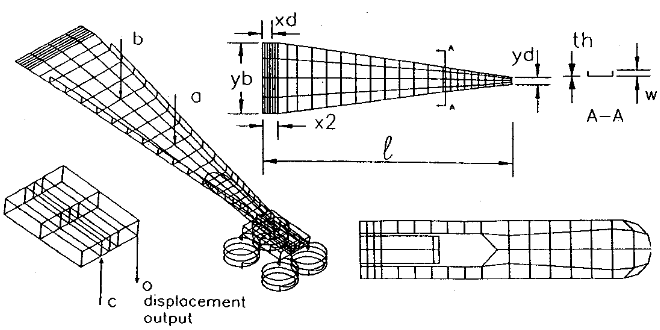

Fig. 3 Finite element model of suspension assembly

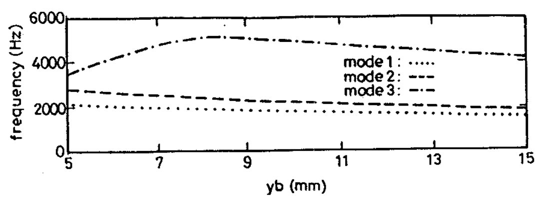

Fig. 4 Sensitivities of natural frequencies with respect to yb

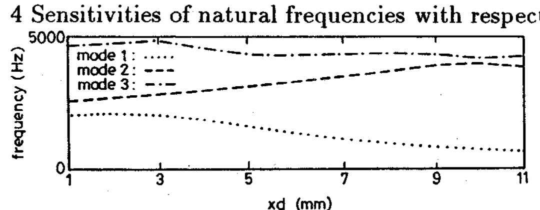

Fig. 5 Sensitivities of natural frequencies with respect to xd

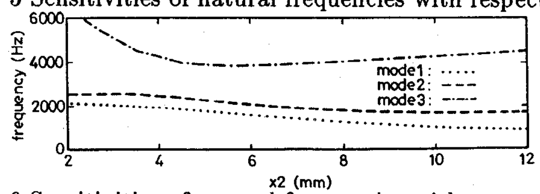

Fig. 6 Sensitivities of natural frequencies with respect to $x2$

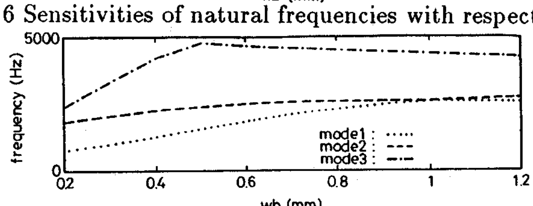

Fig. 7 Sensitivities of natural frequencies with respect to wh

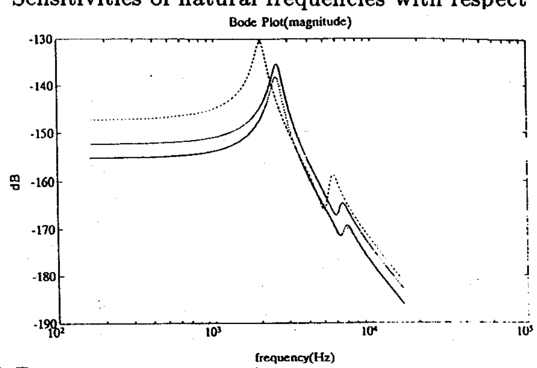

Fig. 8 Frequency response of suspension assemblies (dashed curve: original system, solid curve compromise programming, dotted curve: goal programming)