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A new optical pickup suspension design

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Abstract

A new beam-type optical pickup suspension of a CD player or a CD-ROM driver is developed. Simply using the sensitivity analysis rather than sophisticated optimization tools, we show how a new good pickup suspension can be designed to satisfy certain conditions. Unlike existing suspension systems, the present suspension structure consisting of four beams has two rectangular bends in both ends of each beam. The present choice of the rectangular bend with optimally selected thickness and width of the beams provides very good low-frequency dynamic characteristics of the pickup; the natural frequencies in two major motion directions are within the target range, and unwanted natural frequencies are made as high as possible.

The development of the present suspension structure consists of two parts. The initial suspension system with the uniform width is determined based on the simplified analytic pickup model proposed in this work. Then by performing the sensitivity analysis for the resonance frequencies, based on the detailed finite element model of the initial suspension system, an optimal suspension width variation is determined. It is shown that the sensitivity analysis even without sophisticated optimization algorithms is extremely useful in actual engineering practice with a tight development schedule constraint. The strain energy is utilized to find the optimal treatment location of damping bonds which can control the quality factors at major resonances within the allowance limit. The experiments performed on the present sample model indeed yielded satisfactory results and its mass production is in progress. © 1999 Elsevier Science Ltd. All rights reserved.

1. Introduction

Multimedia products such as CD, CD-ROM and DVD drivers are becoming more popular these days because of their large storage capacity. In these products, stored data in optical discs are retrieved by optical pickups. Well-designed pickup structures as well as fast and efficient position controllers are critical in pickup design. Among several design aspects of a pickup, we are mainly concerned with its structural design, in particular, the optimal structural design of the

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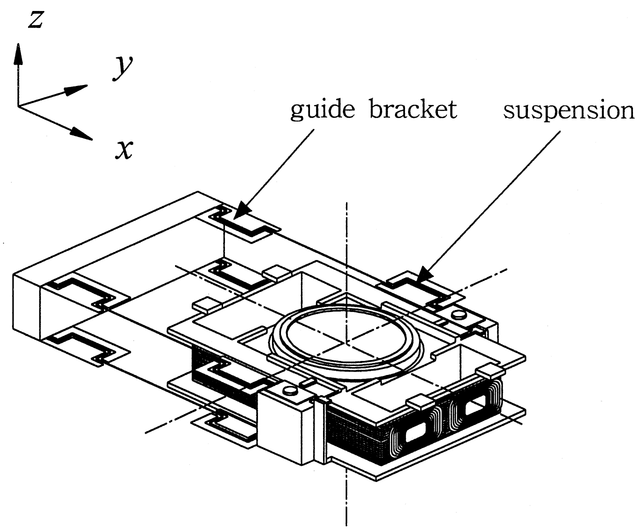


Fig. 1. A typical optical actuator pickup.

pickup suspension system which can provide good low-frequency dynamic behaviors. To discuss the specific design goal, it is worth examining the mechanics involved in the pickup motion dynamics.

A typical pickup consists of a bobbin serving to hold a lens, a suspension system, and other parts (see Fig. 1). It is known that the suspension system, among others, affects most significantly the dynamic characteristics particularly in the low frequency range (say, below 100 Hz). Since pickups must position accurately both in the tracking and focusing directions in order to read data accurately from optical discs, a well controlled pickup having proper focusing and tracking resonance frequencies is very important. In addition to satisfying these frequencies, the effect of higher natural frequencies must be minimized; an ideal pickup should behave as a system of a single degree of freedom in each moving direction. Therefore, the structure of the suspension must be designed in such a way that the deviation from the single degree of freedom motion due to the higher modes should be minimized. Furthermore, the magnitude of the system response at the resonance peak should not be larger than an allowable level (say 20 dB). This requirement can be fulfilled by proper damping treatments at optimally selected locations.

In designing the present suspension system model, the major constraint was that the size and shape of a pickup other than suspension systems are pre-assigned because of the manufacturing cost constraint. In addition, the pre-selected controller requires that the first natural frequencies of the pickup in the focusing and tracking directions lie around 20 Hz for the best performance. Therefore, the specific goal in the present work is to determine the optimal size and shape of the suspension system satisfying these frequency constraints and that the effects of the unwanted resonances are made as small as possible in the system response. In addition, the shape of the suspension system must allow easy damping treatments.

To review the research done in the related area, Khot (1985) investigated the problem of designing a structure with multiple frequency constraints by using an optimality criterion method.

He examined the problem of finding the cross-sectional areas of truss members for the minimum weight of the structure. Grandhi and Venkayya (1988) presented a design optimization algorithm for structural weight minimization with multiple frequency constraints and discussed the optimality algorithm, resizing procedure and scaling technique. They also examined large structures with hundreds of design variables.

Kajiwara and Nagamatsu (1993a) have recently studied a structure design technique using the sensitivity analysis in order to eliminate the resonance peak from the frequency response function. Later they apply their technique in the pickup bobbin design (Kajiwara and Nagamatsu, 1993b); they try to move the anti-resonance frequency toward the neighboring resonance frequency. However, the effect of those natural frequencies may not be negligible if the response points are not located exactly at the computationally predicted points, say, due to manufacturing errors. It is noted that no technical paper on an optimal design of suspension systems is documented in the view point of the optimal suspension structural design, although several types of suspensions are actually developed by many manufacturers.

In the first part of the present work, a method to determine the shape and initial dimension of a new beam-type suspension system is presented, based on a simplified concentrated mass-spring-beam model proposed in the present paper. In the second part, the sensitivity analysis of natural frequencies, using the finite element model based on the initial design, are conducted to determine the optimal width and thickness of the suspension beams, which will serve to achieve the good dynamic characteristics while satisfying given constraints.

In the present investigation, new beam-type suspension systems with rectangular bends at the end are proposed for which the cost-effective etching process is applicable. The key feature of the present model is to use rectangular bends at both ends, with which better dynamic characteristics can be obtained without increasing the manufacturing cost in comparison with existing systems.

Once the initial shape and size is determined, the refined optimization can be performed. In the present development, the widths of the suspension beam segments are selected as the design variables. The sensitivity analysis of the natural frequencies of the bobbin-suspension system is conducted to determine the optimal widths using commercial finite element analysis software. Although there have been many researches on optimal structural design, most methods are not easy to apply to realistic problems having many constraints; the sensitivity analysis alone may be sufficient for many practical problems including the present one. One objective of this paper is to show that relatively simple sensitivity analysis alone is extremely useful in actual engineering design practice with a very tight development schedule constraint.

In order to determine the optimal damping treatment locations for controlling the magnitude of the response at the resonance frequencies within certain levels, the use of the sensitivity analysis of the strain energy stored in the beam is proposed. Based on the present analysis and design, a new suspension system is finally developed, and several experiments conducted on the sample products confirmed that the present design is very satisfactory.

2. Determination of initial shape and dimensions

In order to determine the initial shape and dimensions of the present pickup, we first develop a simplified mechanical model for the pickup as shown in Fig. 2. A bobbin (serving to hold a lens),

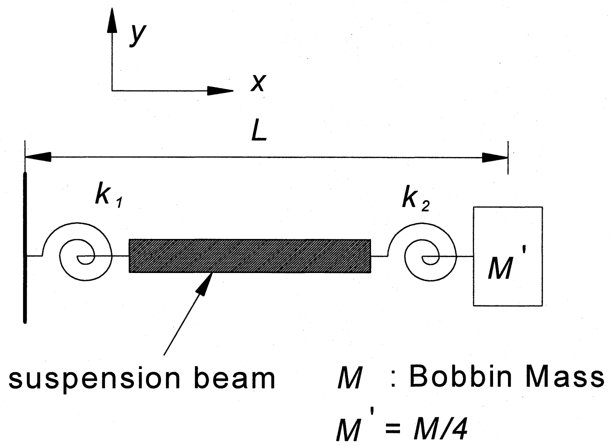


Fig. 2. The proposed simplified mechanical model of the suspension-bobbin structure.

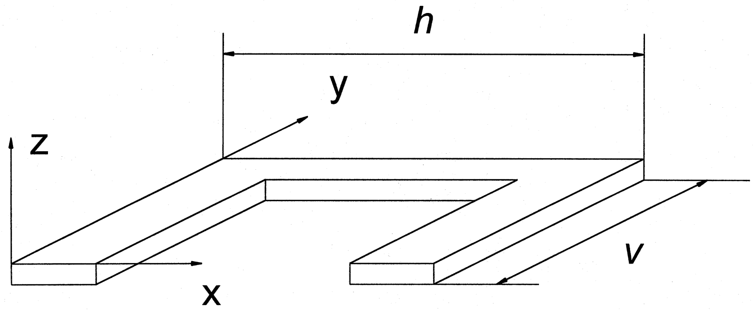


Fig. 3. The rectangular bend which will be modelled as a rotational spring.

straight sections and bends of the suspension are modelled as a concentrated mass, uniform beams and rotational springs (k_1 and k_2 of which subscripts indicate the positions of both ends), respectively. Compare Fig. 1 and Fig. 2. One of the important aspects in the present simplified model is the idealization of the rectangular bend shown in Fig. 3 by means of the rotation springs. Although rectangular bends are considered in Fig. 3, the bend shapes are not of a major concern at this step, as we shall show that the rectangular bends are very effective shapes in comparison with other shapes in meeting the design constraints.

As the next step, one may solve the following equation and boundary conditions using the simplified model in order to predict the dynamic response of the suspension system:

$$EI\frac{\partial^4 u_z(x,t)}{\partial x^4} = \rho A\frac{\partial^2 u_z(x,t)}{\partial t^2} \tag{1}$$

$$\left.\begin{aligned} u_z(0,t) &= 0 \\ k_1\frac{\partial u_z(0,t)}{\partial x} &= EI\frac{\partial^2 u_z(0,t)}{\partial x^2} \end{aligned}\right\} \quad \text{at } x = 0 \tag{2}$$

$$\left.\begin{aligned} k_2\frac{\partial u_z(L,t)}{\partial x} &= EI\frac{\partial^2 u_z(L,t)}{\partial x^2} \\ M'\frac{\partial^2 u_z(L,t)}{\partial t^2} &= EI\frac{\partial^3 u_z(L,t)}{\partial x^3} \\ M' &= M/4 \end{aligned}\right\} \quad \text{at } x = L \tag{3}$$

In eqns (1)–(3), ρ and A are the density and cross-sectional area of the straight beam, and M is the concentrated mass. Since four suspension beams support the bobbin of mass M, we assume that each suspension beam carries the quarter of the bobbin mass. Although the derived governing differential equations are not new, we believe that the present simplified model of the suspension-bobbin system is proposed here for the first time. In particular, the use of the rotational springs to model the rectangular bends is critical for a simplified and effective analysis.

The general solution to eqn (1) takes the following form:

$$u_z(x,t) = \sum_{n=1}^{\infty}\left(A_n\cos\alpha_n\frac{x}{L} + B_n\sin\alpha_n\frac{x}{L} + C_n\cosh\alpha_n\frac{x}{L} + D_n\sinh\alpha_n\frac{x}{L}\right)\sin\omega_n t \tag{4}$$

where

$$\omega_n = \alpha_n^2\sqrt{\frac{EI}{\rho AL^4}}$$

We introduce a normalized natural frequency, α_n which is determined as the solution of the characteristic equation. This equation is obtained by substituting eqn (4) into eqns (2) and (3) for a nontrivial solution. This characteristic equation may be put into the following form

$$f(\eta_1, \eta_2, \zeta, \alpha_n) = 0 \tag{5}$$

where the natural frequency α_n is expressed as a function of the dimensionless parameters η_1, η_2, and ζ such that

$$\begin{aligned} \eta_1 &= \frac{k_1 L}{EI}, \\ \eta_2 &= \frac{k_2 L}{EI}, \\ \zeta &= \frac{M'}{\rho AL} \end{aligned} \tag{6}$$

These parameters are indeed the key design variables which are used to determine the initial shape and size of the suspension systems. It should be noted that the same equations, from eqn (2) through eqn (6), apply for both tracking and focusing motions; obviously, different values of EI, k_1, k_2 (thus, η_1, η_2) must be used depending on the motion direction.

The straight suspension system without any bend is easier and less expensive to manufacture than that with bends. Any suspension system without sufficient damping can not be used as a servo-mechanical system and thus rubber-like bulks at the end of the straight beams have been used. However, the use of bends with damping treatment is known to be more cost-effective. In addition, the suspension system having bends works much better in designing a pickup satisfying prescribed focusing and tracking resonances. Furthermore, the system with bends has higher resistance to the rotation about the x axis and is therefore more stable than those without any bend.

For the present development, we propose the suspension with the rectangular bends rather than the circular bends adopted in some commercially available models because of the following reasons. For the circular bend, the dimensionless rotational spring parameters η_1 and η_2 in focusing and tracking directions, can be shown to be:

$$\eta_1|_{\mathrm{FC}} = \frac{2L}{\pi r_1(1+\lambda)}, \quad \eta_2|_{\mathrm{FC}} = \frac{2L}{\pi r_2(1+\lambda)} \tag{7}$$

$$\eta_1|_{\mathrm{TR}} = \frac{L}{\pi r_1}, \quad \eta_2|_{\mathrm{TR}} = \frac{L}{\pi r_2} \tag{8}$$

where

$$\lambda = \frac{EI}{GJ}$$

and r_1 and r_2 are radii of the circular bends at both ends. The subscripts FC and TR stand for focusing and tracking in eqns (7), (8), and GJ and EI are the bending and torsional rigidities. If symmetrical suspension configuration is adopted (i.e., $r_1 = r_2 = r$), which is actually being used in existing models, it is evident from eqns (7) and (8) that one cannot adjust the focusing and tracking frequencies independently only with varying r.

Now consider the rectangular bends proposed in Fig. 3. One can derive the dimensionless parameters η_{FC} and η_{TR} for the symmetric configuration, as

$$\eta_{\mathrm{FC}} = \frac{L}{2\lambda v + h} \tag{9}$$

$$\eta_{\mathrm{TR}} = \frac{L}{2v + h} \tag{10}$$

Different target resonance frequencies in tracking and focusing directions can be easily realized now by adjusting both h and v. Although the result appears simple, the significance of this observation should not be underestimated in the actual suspension design.

To determine the initial values of h and v for which required resonance frequencies are obtained, we have plotted the curves for constant focusing (and tracking) frequencies as functions of h and

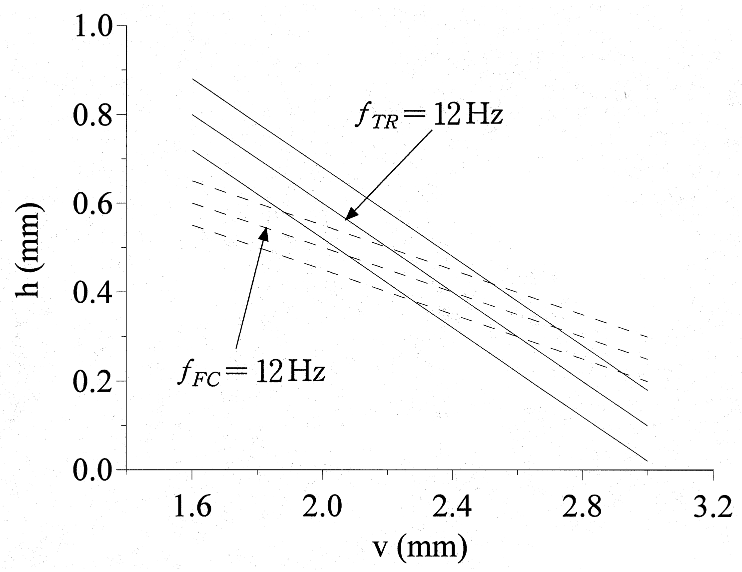


Fig. 4. Curves of constant frequencies of the focusing and tracking directions. (See Fig. 3 for the definition of h and v.)

v. Figure 4 shows that the dimensions of h and v are uniquely determined for given tracking and focusing natural frequencies. For instance, if both f_{TR} and f_{FC} are 12 Hz, then h and v are obtained as 0.4 and 2.4 mm, respectively. Although natural frequencies around 20 Hz are usually required, we start with lower natural frequencies because it is easier to allow natural frequency increase during the optimization process; this will be discussed in the subsequent sections in more details.

To verify the validity of the present simplified model, we analyzed the whole bobbin-suspension system (i.e., the bobbin and four suspension beams) using finite elements. The bobbin is modelled as a concentrated mass and mass moments of inertias ($m = 2.3$ g, $I_{xx} = 21.1$ g mm^4, $I_{yy} = 26.4$ g mm^4, $I_{zz} = 40.0$ g mm^4). Detailed geometries of the suspension including rectangular bends were modelled in IDEAS (1993) using beam elements.

Table 1 shows the first four natural frequencies obtained from the finite element and present analysis for the model shown in Fig. 5 ($h = 0.4$ mm, $v = 2.4$ mm). The corresponding modes are depicted in Fig. 6. The first two natural frequencies from the finite element analysis agree well with

Table 1
Natural frequencies of the initial model (unit: Hz)

	Focusing	Tracking	Torsional	Stretching
Finite element analysis	12.22	12.37	24.55	132.28
Present analysis	12.00	12.00	—	—

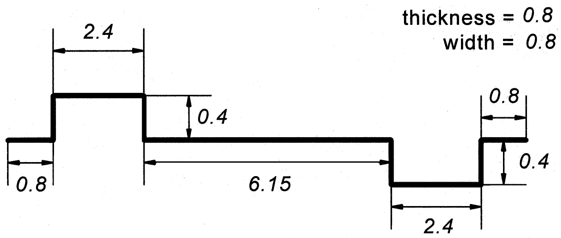


Fig. 5. The dimensions used in the initial suspension (unit: mm).

those obtained from the proposed simplified model, and the torsional and stretching frequencies cannot be predicted from the present simplified model. However, this simplified model has been shown to be very useful in determining the initial size satisfying the given focusing and tracking resonance frequencies.

3. Design improvement

In the previous section, we determined the initial shape and dimensions of a new suspension system which can satisfy the desired natural frequencies in the tracking and focusing directions. Based on this model, we now carry out structural modification to minimize the effect of sub-resonance frequencies (e.g., the torsional frequency) on the system response. One should also realize the importance of locating optimal damping treatment positions in order to keep the quality factor at the resonance within a certain level (say, 20 dB). Otherwise, it is difficult or infeasible to maintain good controllability of the pickup. This damping treatment also reduces the effect of the sub-resonances. To achieve these two goals, we have carried out the sensitivity analysis, which is described below.

3.1. Structural modification of suspension system

One way to reduce the effect of higher models (i.e., torsional and stretching modes) on the system response in the tracking and focusing directions is to push the corresponding natural frequencies as high as possible. It appears that the easiest way is to increase the width of the beam uniformly, but this will also increase the first fundamental tracking and focusing frequencies almost by the equal amount, which is not desirable. Therefore, we divide the suspension beam into several segments whose widths can vary independently: see Fig. 7. Then we investigate the effect of the segment width change on the increase of the natural frequencies, i.e., the sensitivity. As pointed out in Introduction, more elaborate optimization algorithms may be utilized for optimal design, but one objective of this work is to show that the sensitivity analysis alone is very useful in everyday engineering practice with straight development period constraint.

The results of the sensitivity analysis are plotted in Fig. 8, and the 30% increase of the segment width is used to compute numerically the sensitivities shown in Fig. 8. Rather a large amount for

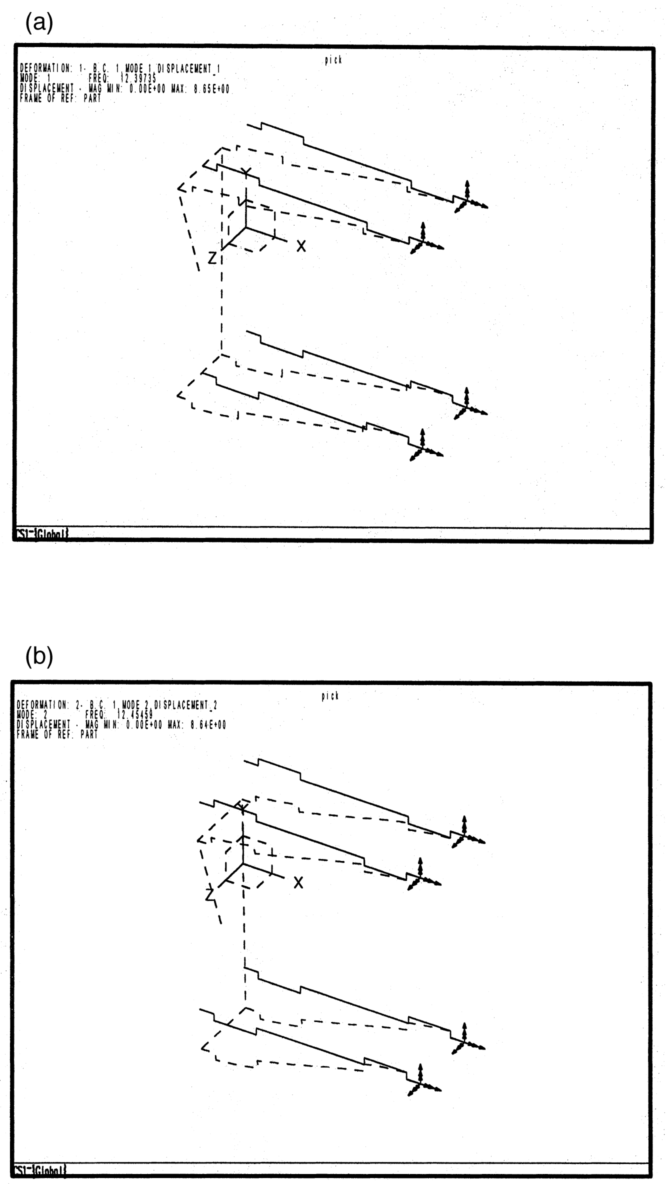
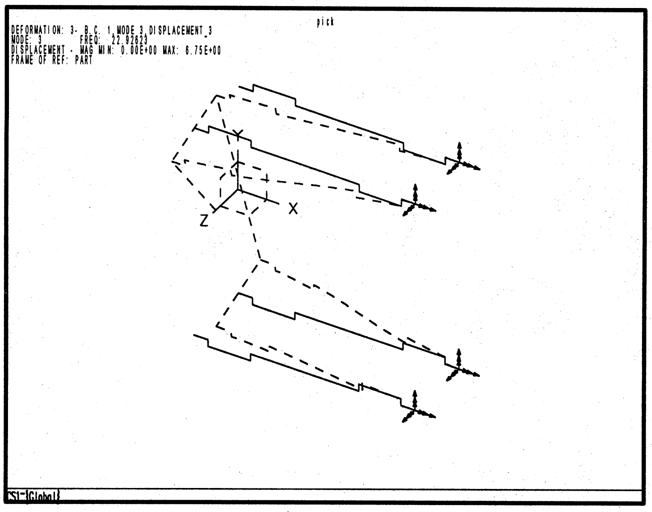


Fig. 6. The first four mode shapes of the pickup having the suspension beam depicted in Fig. 5, (a) focusing, (b) tracking, (c) torsional and (d) stretching modes are shown.

the sensitivity calculation is chosen because only moderate changes can be realized in the real situation. Figure 8 indicates that the width increase of c_1 will make the tracking natural frequency even higher than the torsional frequency and thus its increase is not desirable, whereas the width

(c)



(d)

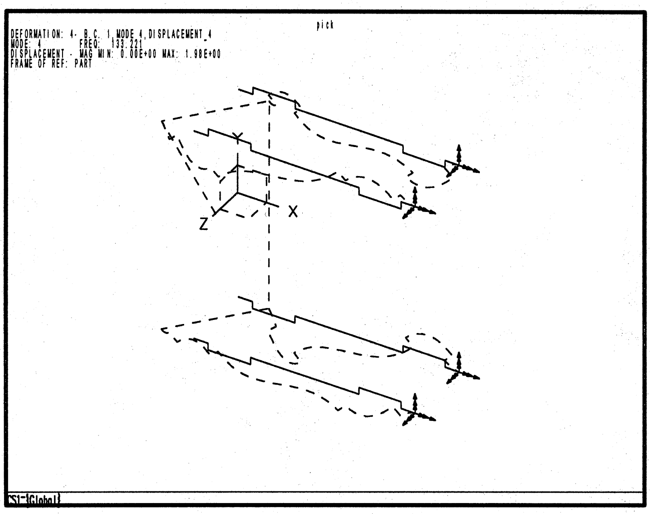


Fig. 6.—*Continued.*

increases of e_1 and e_2 are desirable. By taking stepwise structural modification,* we finally obtain the improved suspension beam geometry as tabulated in Table 2. The resonance frequencies of

* One may use more systematic methods such as the pseudo-inverse method to determine optimal segment width as done by Kajiwara and Nagamatsu (1993b), but it is also very intuitive and effective to take this stepwise modification in practical situations.

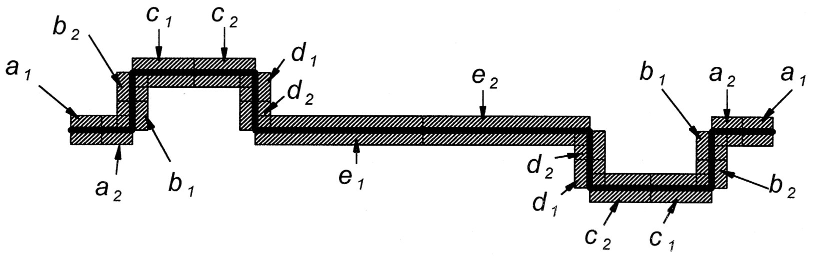


Fig. 7. Design variables, which are the segment widths, are designated by a_1, b_1, etc.

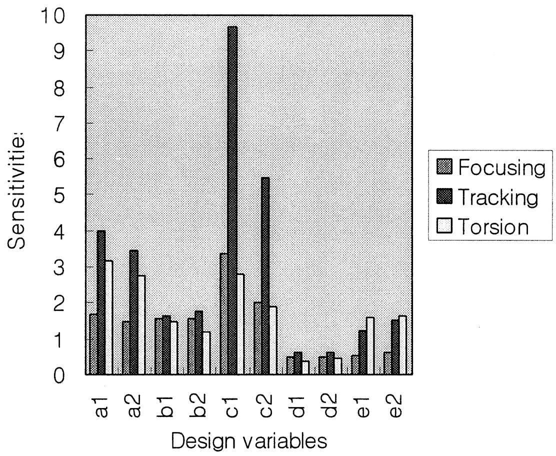


Fig. 8. Sensitivities of the natural frequencies with respect to each width.

Table 2
The width (unit: mm) of the improved suspension beam (the nominal width is 0.08 mm)

a_1	a_2	b_1	b_2	c_1	c_2	d_1	d_2	e_1	e_2
2.0	0.15	0.08	0.08	0.08	0.08	0.16	0.16	0.16	0.16

this improved model are compared with those of the initial model in Table 3. The increase in the unwanted frequencies, torsional and stretching, is remarkable and the increase of the tracking and focusing frequencies is marginal.

3.2. Determination of the damping bond treatment location

In designing a pickup, it is also important to keep the first resonance peak within a certain level. In order to satisfy this requirement, several techniques have been used in industry, but the use of damping bonds is preferred in the present development because of its cost-effectiveness.

Table 3
Natural frequencies of the initial and improved models (unit: Hz)

	Focusing	Tracking	Torsion	Stretching
Initial (analysis)	12.22	12.37	24.55	132.28
Improved (analysis)	14.09	15.19	32.76	213.11
Frequency increase	1.87	2.82	8.21	80.83

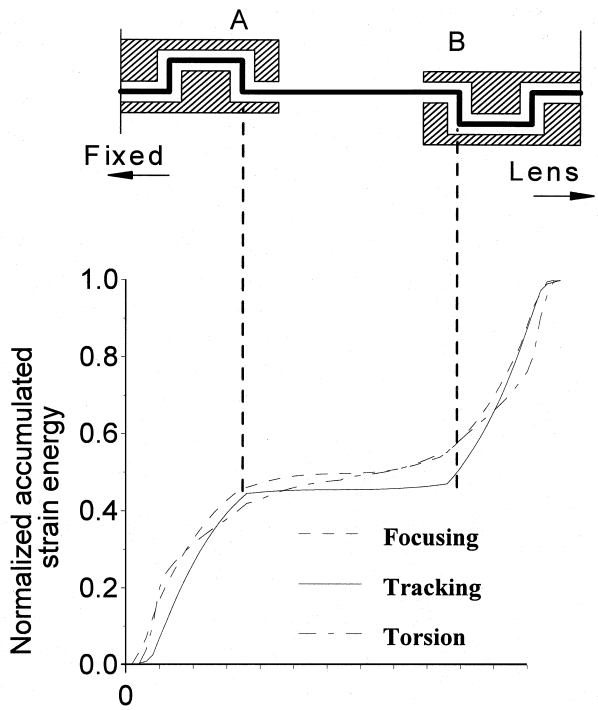


Fig. 9. The accumulated strain energy sum along the length of the suspension.

Since the suspension beams are very narrow, pasting the damping bond on the top surfaces of the beam is neither effective nor practical. Thus we have arranged guide brackets near the rectangular bends as shown in Fig. 1 (see also Fig. 10) so that the damping bonds apply to the guide brackets as well as the suspension beams. Once damping bonds are actually applied, the attached guide brackets tend to stiffen the beam system quite significantly and this is why we have started from relative low resonance frequencies at the initial design step.

Since there appears no criterion established or reported in the literature to determine the appropriate damping bond treatment location for the present situation, we have examined several physical quantities such as the distribution of strain energy density, its sensitivity, accumulation of the strain energy, and others.

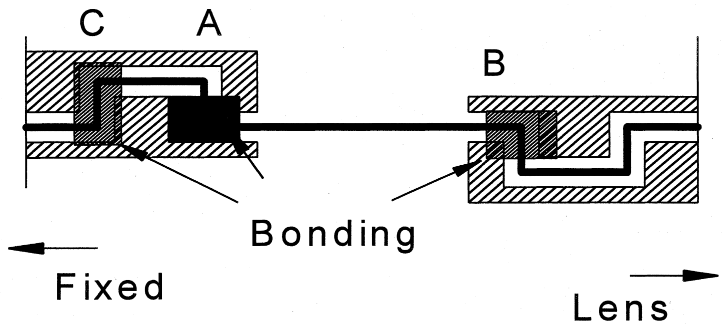


Fig. 10. The proposed optimum damping position and other candidate position.

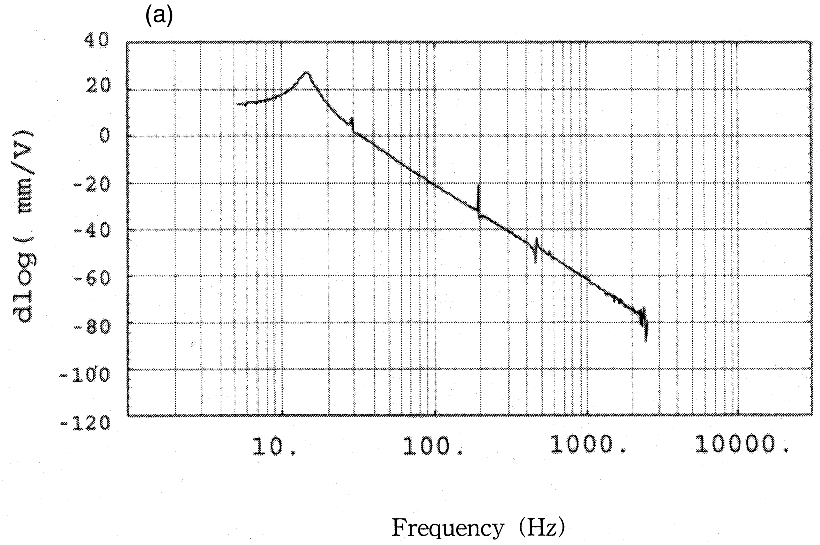
Figure 9 shows the accumulated strain energy from one end to the other end of the beam. Between A and B, the strain energy sum does not change much and reaches at a certain level for all three modes of vibration considered. In order for the damping bonds to be useful, they should be applied at the locations where certain levels of strain energy are stored, namely between A and B. Thus, locations A and B are the candidate locations as suggested in Fig. 10.

Several test samples are manufactured through an etching process, and experiments are performed. In experiments, we have actually applied damping bonds at various locations, say, locations between A and the fixed ends. For these experiments, we have found that the presently selected damping locations A and B are indeed effective, and further observed that the damping treatment only at A as suggested in Fig. 10 is sufficient to obtain the desirable performance. For instance Figures 11 and 12 show the frequency response functions (FRF) from the test sample manufactured, before and after the damping bond treatment at location A and C, respectively. It is clear that the damping treatment at C does not reduce the quality factor at the first resonance significantly. However, the damping treatment at A reduces the quality factor to as low as 13 dB. This result indicates how important the proper selection of the damping bond treatment location is in designing a cost-effective suspension system.

4. Summary and discussions

We have developed a new suspension type for an optical pickup, which can be designed easily and manufactured inexpensively through the etching process. The advantages of the present model may be summarized as follows:

(1) The present suspension system with rectangular bends has relatively a higher torsional frequency than other existing systems, and the use of the rectangular bends is shown to be very effective.
(2) The required resonance frequencies in both focusing and tracking directions are easily materialized by changing the variables v, h, namely the vertical and horizontal dimensions of the bend.
(3) The present suspension system made of Beryllium-Copper can be cost-effectively manufactured through the chemical etching process.



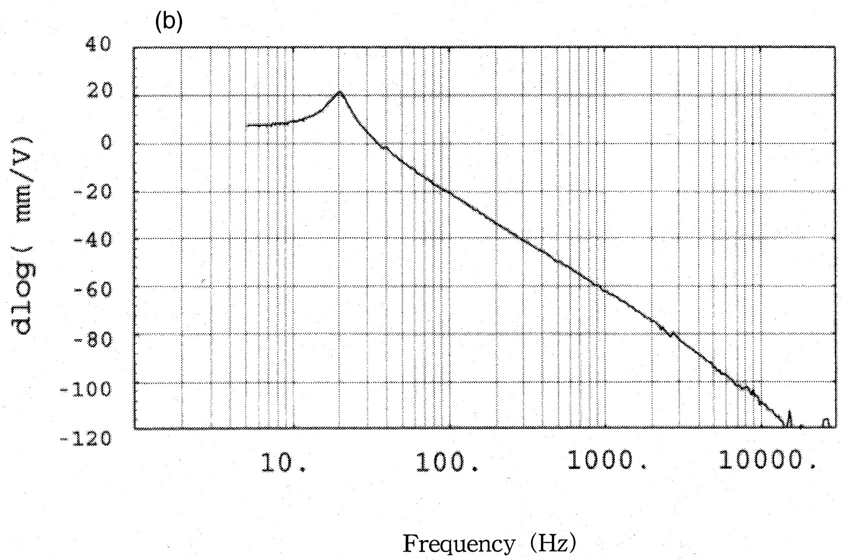


Fig. 11. The frequency response function in the focusing direction (a) before and (b) after the damping bond treatment at location A.

To design the initial geometry and size of the suspension beams, we proposed a simplified bobbin-suspension system model. The rectangular bends are modelled as rotational springs. Based on the initial model, structural modification was performed to push the unwanted natural frequencies as high as possible. Furthermore, the quality factor at the dominant resonance peak is controlled within a certain level by the optimally positioned damping bond treatment. For the optimal positioning, the concept of the accumulated strain energy was utilized. The experiments

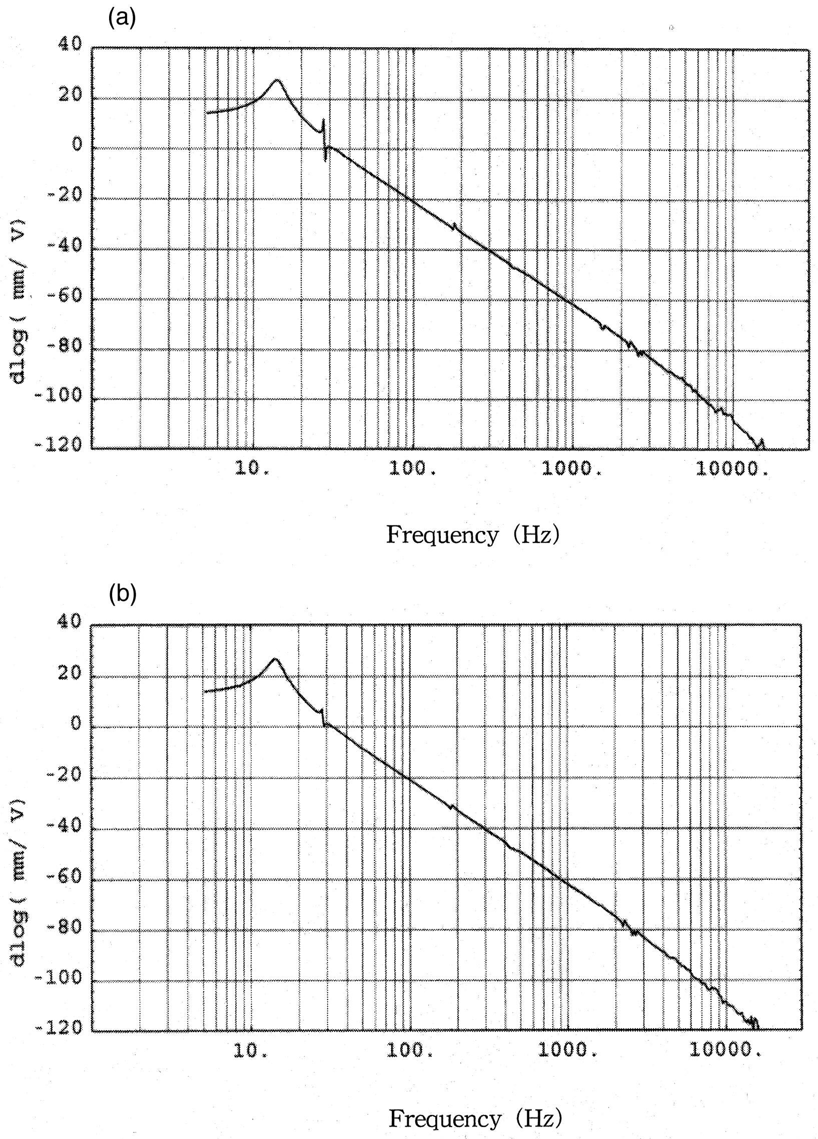


Fig. 12. The frequency response function in the focusing direction (a) before and (b) after the damping bond treatment at location C.

on test samples confirmed the very satisfactory performance of the present new pickup suspension model.

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