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ROBUST DESIGN FOR DYNAMIC PERFORMANCE: OPTICAL PICK-UP EXAMPLE

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ABSTRACT

To achieve high quality and reliability in mechatronics products, thorough consideration of dynamic performance in structural design is essential. Typical examples include the design of optical pick-ups for disk related products, such as Compact Disc (CD) drives, and Magneto-optical Disc (MO) drives. Many researchers have developed design methodologies to optimize the nominal dynamic characteristics. Recently, robustness of dynamic performance has become an important issue in addition to optimizing nominal performance. Engineers seek to improve the quality of products by minimizing the effects of numerous variations that arise during the product's manufacturing and life-cycle operation. This paper reports on the application of the robust design concept to the dynamic design of an optical pick-up actuator focusing on shape synthesis using computer models and design of experiments. By comparing this approach to analytical sensitivity solution, the authors provide a rationale for using the relatively simple statistical approach with orthogonal arrays.

1. INTRODUCTION

1.1 Robust Design Challenge in Optical Disk Related Products

In many development projects, engineers end-up interacting late in the development cycle to stabilize the performance of mass produced products into a desired range. This process is often time-consuming and depends heavily on the engineers' empirical know-how. For the design of optical disk products, such as computer data storage drives and portable music CD players, high quality and reliability are required for product competitiveness. Engineers need a new systematic design methodology to enhance the robustness of performance under variations in manufacturing and operational conditions. The methodology must apply to the early design stage so that engineers can reduce time-to-market by eliminating the late iterations. Our research addresses the practical and effective shape optimization for optical disc related products. This paper focuses on the dynamic design of a pick-up actuator and describes the robust design of actuator structural elements under manufacturing variations.

1.2 Previous Work

Regarding the dynamic design of pick-ups, Ichihara, et al. (1991) described a procedure for raising the higher order resonance frequency of leaf springs to achieve high controllability for actuator tracking servo system. Saekusa et al. (1991) discussed “cross action,” the secondary movement of pick-up in the tracking direction when actuated to move in the focus direction. They related cross action to the elastic resonance of the leaf spring and assembly errors and found optimum values of rough suspension configuration that reduce cross action. As for moving parts of the pick-up such as the lens, the lens holder, and the bobbin, Nakamura et al. (1995) introduced their analysis process to adjust lens holder natural frequency using FEM. Seki (1996) applied the design optimization method for modification of natural mode shape to eliminate the bobbin elastic vibration effect on pick-up track servo system. These studies focused on the dynamic performance and did not consider performance variations that arise from manufacturing. As a recording density of optical discs increases and required precision becomes tighter, design engineers must consider these manufacturing variations during the early design stages.

There are many successful applications of robustness concepts in mechanical design. Rao et al. (1990) improved the robustness of actively controlled structures through structural modification. They focused on the integrated structural and control design problems. Sundaresan et al. (1992) presented a methodology to incorporate manufacturing and operational variances in the design optimization of beverage cans to achieve robust and optimal static performance. Kazmer et al. (1996) developed a systematic procedure for assessing the design and manufacturing robustness of injection molded plastics using Monte Carlo simulation. Their method addressed the detailed design of the parts and demonstrated improvement in dimensional control. Esterman et al.(1996) applied robust design principles with design of experiments to fatigue life evaluation using shot peening examples. Koch et al. (1996) discussed a new approach and computer infrastructure for configuring turbine system using robust concept for early design stage. There are many other reports on robust design for static performance. However, there are few studies that address robust dynamic performance in actual products.

1.3 Research Approach

Engineers have already applied conventional deterministic optimization methods to geometry synthesis problems such as pick-up actuator design to enhance the dynamic performance at minimum cost. This paper applies the robust design method to achieve statistical optimization of the pick-up actuator. As discussed before, consideration of natural frequencies and dynamic response is essential in design of a mechanical actuator. This study seeks to confirm the validity of the proposed robust design method in the practical development of mechatronics products.

The following section presents the simple analysis model of the example pick-up actuator using the finite element method (FEM) that approximates the dynamic characteristics. We choose the pertinent set of uncontrollable (noise) and controllable (design) parameters by investigating past design records. By selecting the shape parameters of an actuator leaf spring as a controllable factor, we aim for the optimum shape of an actuator leaf spring that is insensitive to variations including assembly errors, part fabrication errors, and material property variation. We also calculate the analytical sensitivity of frequency response gain to the shape parameters and compare it to the S/N ratio of the gain with respect to noise of the shape parameter, i.e., the robust sensitivity. These robust sensitivities lead to the synthesis of a new design with enhanced robustness in dynamic performance. Figure 1 summarizes the step in this case study for robust design of dynamic performance.

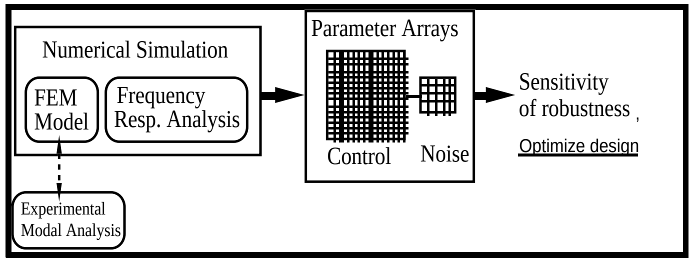


Figure 1: Robust dynamic design step

Section 2 summarizes the design process of pick-up actuators and the required dynamic characteristics. Section 3 describes the robust design procedure for pick-up actuators. Section 4 discusses the results of the robustness analysis and presents an improved design of the pick-up actuator, while section 5 provides the conclusions and outlines future directions of research.

2. MECHANICAL DESIGN PROCESS FOR OPTICAL PICK-UP

2.1 Servo systems in Optical disc drives

The pick-up is a key device in Optical disc drives. Figure 2 shows a typical configuration of the pick-up actuator mechanism. An actuator is a major mechanical structure in the pick-up that controls the position of the objective lens. In general, disc drives use four types of servo systems for reading digital signals on the disc using pick-up actuators (c.f. figure 3). In compact disc players, the servo types are:

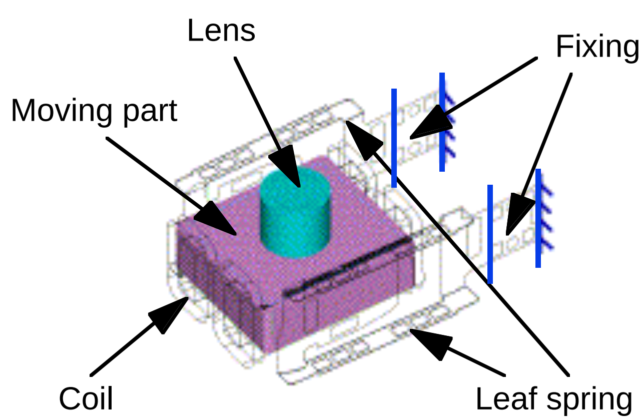


Figure 2: Pick-up actuator configuration (2-axis device)

Focus servo system controls vertical movement of the actuator and guarantees that the focal point of the laser beam is precisely on the surface of the compact disc.

Tracing servo system controls the horizontal movement of the actuator and makes the laser beam follow the tracks on the compact disc.

Sled servo system drives the sled motor which moves the optical block across the compact disc.

Spindle motor servo system controls the speed of disc motor, guaranteeing that the optical pick-up follows the compact disc track at a constant linear velocity.

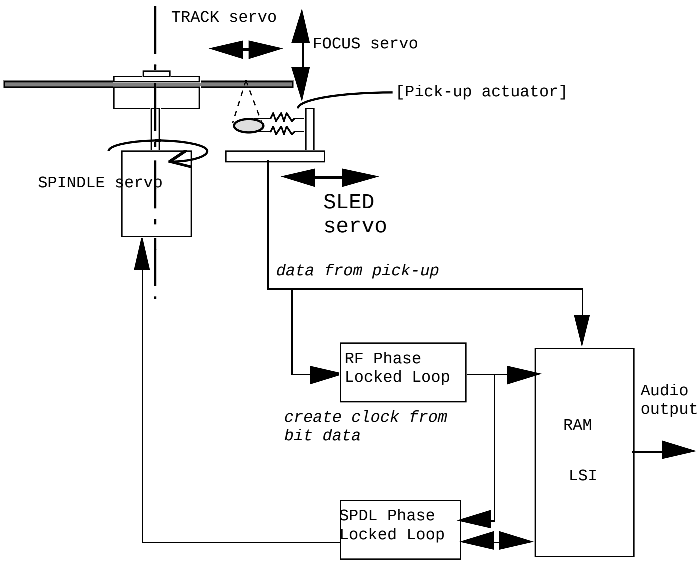


Figure 3: Servo systems in a CD player

2.2 The Design of Actuators in Optical disc drives

To achieve the above-mentioned servo functionality, the actuator structure should satisfy the following mechanical requirements.

(a) Should be as small and light as possible

(b) Should have independent movement in focus and track directions

(c) Should maintain the direction of laser beam during actuator movement

(d) Should place vibration modes in high frequency except for the first vertical and the first lateral bending modes

Especially, important are **(c)** and **(d)** in pick-up dynamic performance design. The ideal characteristics of the structural transfer function (direct compliance at the lens) is the "one degree-of-freedom" type response curves.

2.3 The Design of Pick-up Actuators

As stated before, the actuator is designed to achieve a one degree-of-freedom type frequency response function to provide controllability of the positioning servo system. Theoretically, one can attain the required dynamic characteristics by following appropriate design procedures: setting a driving force to the gravity center of a moving part, justifying elastic principle axis and mass inertia axis, etc. Engineers design the actuator to suppress the effect of resonance modes in a certain frequency range for focus and track direction. However, manufacturing variations often cause undesirable peaks in this frequency range. Figure 4 shows one example of undesirable peaks due to actuator torsional mode caused by some variations described in the next section.

The following sections examine this phenomenon as a particular example for robust design investigation. We seek the optimum actuator leaf spring specifications which achieve the required dynamic performance, even though some manufacturing variations exist. Figure 5 shows the configuration of the sample actuator for this research.

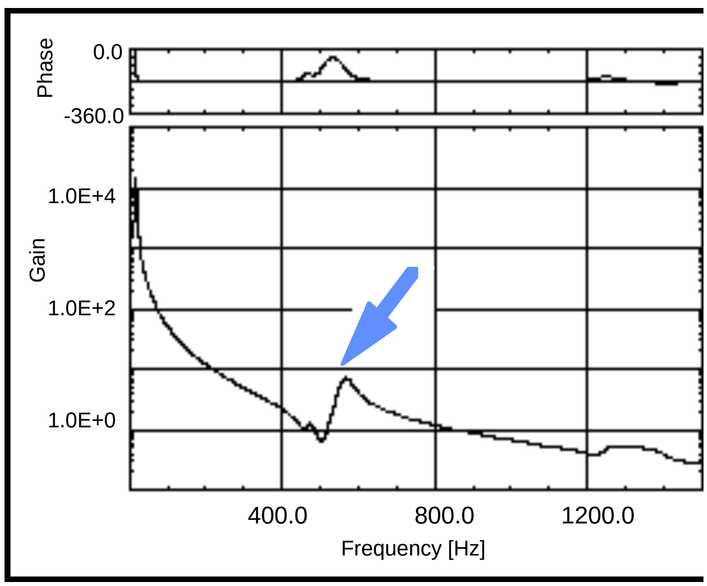


Figure 4: Frequency response function (lens, focus direction (FEM))

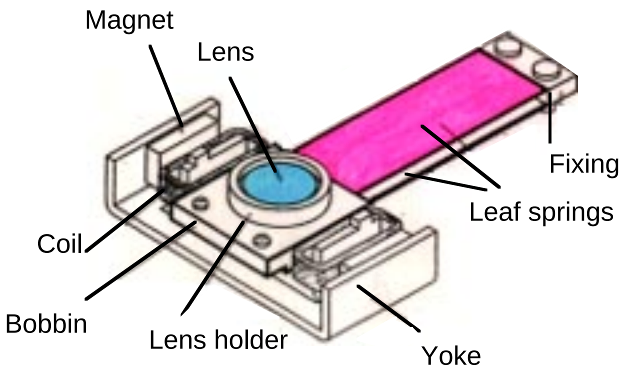


Figure 5: Sample pick-up actuator (1-axis device)

2.4 Pertinent factors that cause variation in dynamic performance

There are many factors that affect the dynamic response as shown in figure 6. In the case of actuator torsional mode, an undesirable peak in the response is due to:

- Eccentric magnetic force due to assembly errors of coil, magnet, and spring
- Static coupling of the suspension system due to leaf spring misalignment
- Dynamic coupling of the suspension system due to moving part misalignment

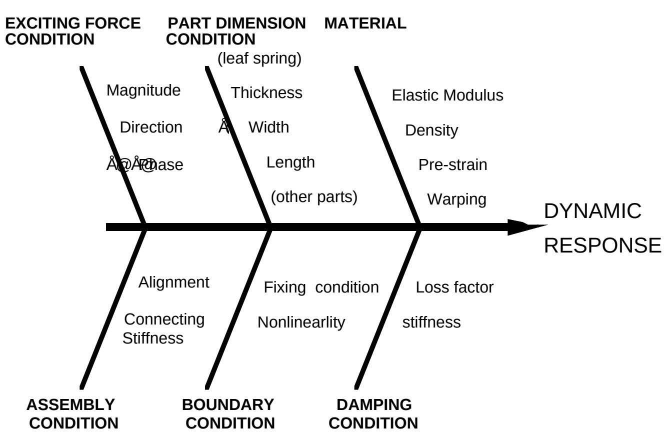


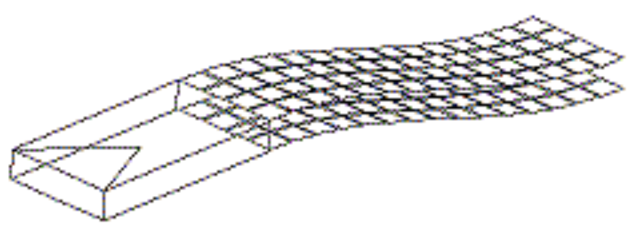
Figure 6: Cause and effect diagram for pick-up dynamic variation

3. ROBUST SENSITIVITY ANALYSIS

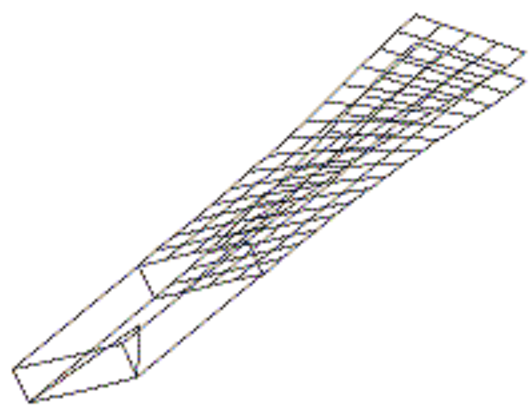
3.1 Analysis Model

To facilitate parameter sensitivity study, this section introduces the simple analysis model of an actuator vibration system using the finite element method (FEM). This model approximates the leaf spring with beam elements and represents the moving part using a concentrated mass and characteristic inertia. To validate this simple model, we compared it to a more detailed leaf spring model using the shell-elements. The results from both models agree well with experimental data. Hence, we adopt the simple approximated model for our robust design study.

Figure 7 shows the shape of the first bending mode that is used for positioning control, and the first torsional mode that often induces an undesirable peak. The upper and lower actuator leaf springs deform with the combinatorial deformation modes of torsion and lateral bending in the first torsional mode. The following robust sensitivity analysis is conducted focusing on these two modes.



[first bending mode (focusing direction)]



[first torsional mode]

Figure 7: Mode shape (FEM calculation)

3.2 Quality Characteristics

To address the torsional mode, we choose the response gain at the resonance frequency of this mode as the measure of vibration energy at the undesired peak. This quantity has the following mathematical form:

$$\{y_i\} = \sum_{r=1}^{k} \frac{\{\phi_r\}^T \{f\} \{\phi_r\}}{k_r + jc_r\omega - m_r\omega^2} \tag{1}$$

where

$\{y_i\}$ $\equiv$ response gain of the physical degree - of - freedom i

$\{\phi_r\}$ $\equiv$ natural mode vector of r th mode

$\{f\}$ $\equiv$ external force

k_r $\equiv$ modal stiffness of r th mode

c_r $\equiv$ modal damping of r th mode

m_r $\equiv$ modal mass of r th mode

ω $\equiv$ excitation frequency

For the first bending mode, we focus on its natural frequency, , which is one of the pick-up specifications, as a quality characteristic. We adopt the "signal-to-noise ratio (S/N ratio)" as a metric to characterize the robustness of the system. Regarding the torsional mode, the goal is to minimize the mean and variance of response gain simultaneously. For this reason, we adopt the "lower-the-better" S/N ratio described as :

$$\begin{aligned} S/N &= -10 \log \left[\frac{1}{n} \sum_{i=1}^{n} y_i^2 \right] \\ &= -10 \log \left[S^2 + \bar{y}^2 \right] \end{aligned} \tag{2}$$

where

y_i $\equiv$ dynamic response in equation (1)

n $\equiv$ total number of individual data points

S^2 $\equiv$ sample variance of response

$\bar{y}$ $\equiv$ sample mean of response

For the first bending mode, the "nominal-the-best" S/N ratio is adopted since the pick-up requires a particular natural frequency during the design of a particular actuator.

$$S/N = 10 \log \left[\frac{\bar{\omega}_1^{\,2}}{S^2} \right] \tag{3}$$

where

ω_1 $\equiv$ natural frequency

S^2 $\equiv$ sample variance of natural frequency

3.3 Parameter Settings

Table 1 shows the settings of the controllable parameters for robust sensitivity analysis. We divide the leaf spring into eight regions and choose each width dimension as a controllable parameter. In the analysis model, each parameter corresponds to a width dimension of the beam section profile. The robust choice of each parameter leads to the optimum robust shape. The robustness analysis uses the following three noise parameters as having a pertinent effect on the variations in dynamic response (figure 5). Table 2 summarizes the ranges for the noise parameters which are set for this particular simulation.

In this study, we assume that the variations in width of spring is much smaller than that of thickness. Therefore, we adopt the spring width only as control parameters to seek robust optimum spring shape.

Table 1 Controllable Parameter Setting

Control Parameter	Level 1	Level 2	Level 3
Width #1 -#8 (mm)	3.9	5.2	6.5

Table 2 Noise Parameters

Noise Parameter	Range
magnetic force imbalance	small --- large
thickness of leaf spring	-5% --- +5%
modulus of elasticity	-5% --- +5%

3.4 Design of Experiments

We adopted the L18 (three-level) Control Factor Array for exploring the design space of the shape synthesis of leaf springs. Figure 8 shows the L4 (two-level) Noise Factor Array used in our study to simulate the manufacturing variations. Our study adopted the inner-outer array strategy proposed by Taguchi (Fowlkes,W.Y., 1995).

The analysis model described in Section 3.1 and FEM-based eigenvalue and frequency response analysis code served as our simulation model and predicted the dynamic response for the 18*4 = 72 runs according to equation (1).

									Noise Factor				
								NF1	L	L	H	H	
								NF2	L	H	L	H	
	Control Factor							NF3	L	H	H	L	
Run	CF1	CF2	CF3	CF4	CF5	CF6	CF7	CF8	1	2	3	4	S/N
1	1	1	1	1	1	1	1	1	0.83	0.68	2.00	2.07	-3.73
2	1	1	2	2	2	2	2	2	0.58	0.42	1.33	1.28	0.08
3	1	1	3	3	3	3	3	3	0.24	0.26	0.54	0.73	6.20
4	1	2	1	1	2	2	3	3	0.39	0.35	0.94	0.97	2.79
5	1	2	2	2	3	3	1	1	0.64	0.54	1.57	1.60	-1.55
6	1	2	3	3	1	1	2	2	0.55	0.41	1.23	1.24	0.56
7	1	3	1	2	1	3	2	3	0.44	0.41	1.04	1.11	1.75
8	1	3	2	3	2	1	3	1	0.64	0.44	1.42	1.34	-0.42
9	1	3	3	1	3	2	1	2	0.64	0.44	1.42	1.37	-0.51
10	2	1	1	3	3	2	2	1	0.56	0.46	1.39	1.40	-0.43
11	2	1	2	1	1	3	3	2	0.41	0.38	0.98	1.05	2.25
12	2	1	3	2	2	1	1	3	0.57	0.47	1.40	1.42	-0.53
13	2	2	1	2	3	1	3	2	0.44	0.36	1.02	1.06	2.06
14	2	2	2	3	1	2	1	3	0.53	0.38	1.20	1.15	0.99
15	2	2	3	1	2	3	2	1	0.48	0.44	1.11	1.20	1.12
16	2	3	1	3	2	3	1	2	0.56	0.38	1.24	1.17	0.76
17	2	3	2	1	3	1	2	3	0.39	0.37	0.91	1.01	2.71
18	2	3	3	2	1	2	3	1	0.48	0.43	1.09	1.18	1.26

Figure 8: Cross array layout for the parameter optimization experiment

3.5 Analytical Sensitivity Calculation

In addition to the Taguchi statistical approach, we also conducted the conventional analytical sensitivity analysis between control parameters and quality measure, i.e., the response gain. Since we are focusing on a specific lower order vibration mode in this particular case, reducing the undesirable peak gain due to torsional mode may lead to decreasing of its scatter at the mode. Therefore, the analytical sensitivity of mean frequency response gain could be used to validate the robust sensitivity obtained by statistical approach to a certain extent. The analytical sensitivity is derived from the following equations.

A frequency response at point l on the pick-up actuator that is excited at point i on the actuator by the force can be described by the following equation:

$$X_l = \sum_{r=1}^{N} \frac{\phi_{ri}\ \phi_{rl}\ F_i}{k_r\ (1-\beta_r^{\,2})} \qquad (4)$$

where

X_l ≡ response at point (dof) l

ϕ_{ri} ≡ i th component in natural mode vector of r th mode

ϕ_{rl} ≡ l th component in natural mode vector of r th mode

F_i ≡ i th component in external force vector

k_r ≡ modal stiffness of r th mode

$\beta_r^{\,2}$ ≡ ω / ω_r

ω ≡ excitation frequency, ω_r ≡ natural frequency

From equation (4), we obtain the "equivalent stiffness Kr" that expresses the dynamic stiffness at a particular point,

$$K_r = \frac{k_r}{\phi_{ri}\ \phi_{rl}} = \frac{\omega_r^{\,2}}{\phi_{ri}\ \phi_{rl}} \qquad (5)$$

Sensitivity S, that is for mean dynamic response, is obtained by differentiating "Fi / Kr" with respect to the control factor t,

$$S = \frac{F_i}{\omega_r^{\,2}} \left(\frac{\partial \phi_{ri}}{\partial t}\ \phi_{rl} + \frac{\partial \phi_{rl}}{\partial t}\ \phi_{ri} - \frac{2\ \phi_{ri}\ \phi_{rl}}{\omega_r}\ \frac{\partial \omega_r}{\partial t} \right) \qquad (6)$$

Thus, equation (6) can provide the sensitivity S for the frequency response gain using the sensitivity values of natural frequency and natural mode shape that are calculated separately.

3.6 Results of Robust and Analytical Sensitivity Analysis

Figure 8 summarizes the results of Robust sensitivity calculations using the L16 inner array with the L4 outer noise array. Figure 9 illustrates the control factors adapted, i.e., width of each segment of the cantilevers. Note that in manufacturing the shape profile will be smoothed. Figure 10 shows the sensitivity plot of the signal-to-noise ratio for each control parameter. Note that maximum sensitivity occurs near the fixed end of the leaf springs. Also, the moving part of the spring has higher sensitivity, which reflects the strain energy distribution in its vibration mode of combinatorial leaf spring deformation of torsion and lateral bending. Figure 11 shows the analytical sensitivity of the frequency response at the torsional mode resonance frequency.

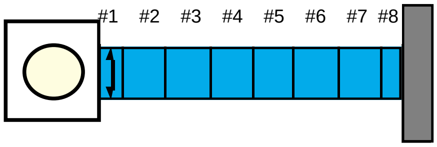


Figure 9: The Control Factors (width of each segment)

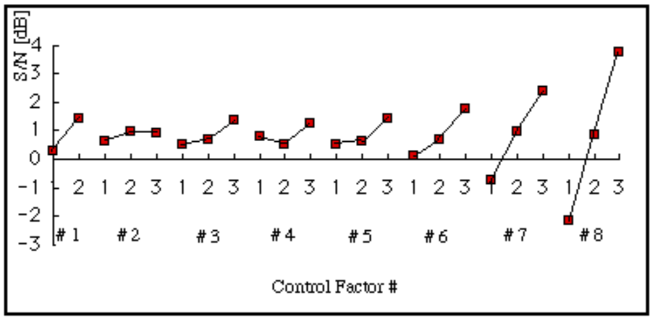


Figure 10: S/N sensitivity (2nd mode response gain)

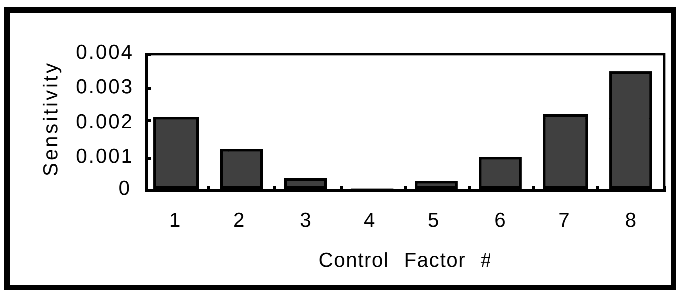


Figure 11: Analysis sensitivity of frequency gain (2nd mode)

Figure 12 shows the sensitivity plot of the mean value of the first bending resonance frequency. The variation in the current control factors have only a small effect on the S/N sensitivity of first mode resonance frequency.

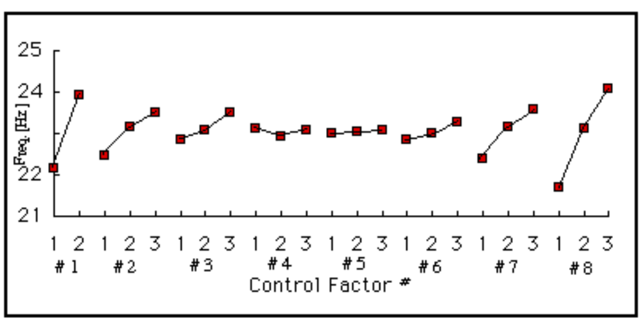


Figure 12: Mean sensitivity (1st mode frequency)

4. DISCUSSION AND SYNTHESIS OF A NEW CANDIDATE DESIGN

4.1 Comparing Robust and Analytical Sensitivity

The comparison of the robust sensitivity calculation results (figure 10) and analytical sensitivity of frequency response (figure 11) reveals that the S/N sensitivities trend is similar and the statistical analysis can accurately estimate the S/N sensitivities by simply using the performance evaluation program and simple arithmetic.

Figure 13 illustrates the frequency response function for the original set of control factors (which is the uniform leaf spring width) under four noise conditions. The robust sensitivities indicate a possibility of reducing the undesirable peak and its scatter by changing the width of the leaf spring along its length. The following section discusses the new design of the leaf spring shape aiming at improving the robustness of frequency response.

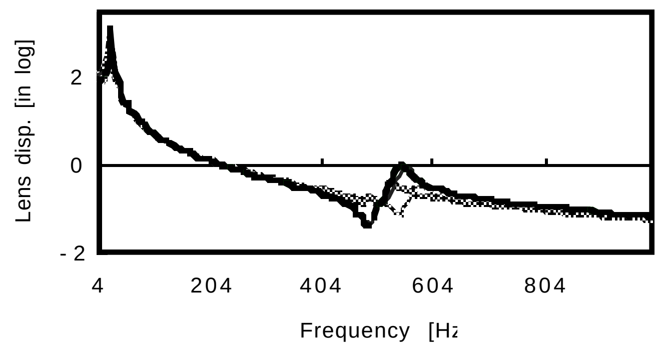


Figure 13: Frequency Response

4.2 Candidate design (A) : to reduce the peak response gain

We seek to reduce the undesirable peak gain and its scatter, but also to avoid weight increase. Thus, we propose spring width modifications in proportion to sensitivity magnitude in control factors #1, 6, 7 and 8, which have comparatively high sensitivity. Table 3 shows the new width values derived to enhance dynamic response robustness.

Table 3 Spring width modification (A)

	# 1	# 2	# 3	# 4	# 5	# 6	# 7	# 8
width	6.78	5.2	5.2	5.2	5.2	6.78	7.36	9.6

Figure 14 and 15 summarizes the frequency response function of the original model and that of candidate design (A). These figures show that the new design significantly suppresses the peak gain and reduces the difference of gain in four curves corresponding to each noise condition.

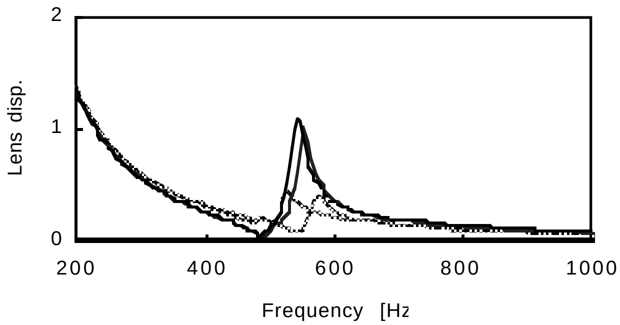


Figure 14: FRF of original model

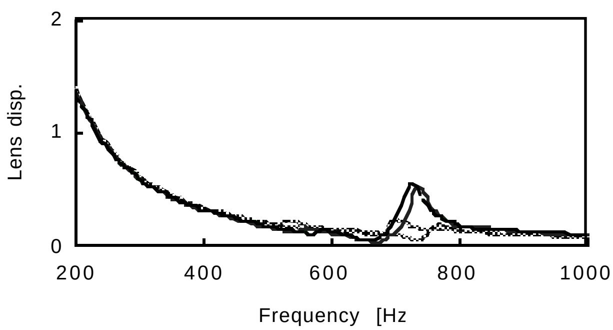


Figure 15: FRF of candidate design (Modification A)

4.3 Candidate design (B) : to adjust first mode natural frequency

Table 4 shows the change in the S/N ratio of the undesirable peak gain and the first mode natural frequency. The modification (A) moved the first bending mode frequency off the design target despite the significant improvement in S/N ratio. Therefore, in order to adjust the frequency, we propose the following alternative designs shown in table 5 and figure 16. Candidate (B) modifies its width in control factors # 2 and 3 that have high sensitivity in the mean frequency of the first mode and low S/N sensitivity in the peak gain.

Table 4 Change in peak gain and natural frequency

	"S/N ratio" of 2nd mode Gain	"mean frequency" of 1st mode
Original	1.95	23.5 Hz
Target	larger-the-better	around 24 Hz
Modification (A)	7.85	26.6 Hz
Modification (B)	5.84	25.1 Hz

Table 5 Spring width modification (B)

	# 1	# 2	# 3	# 4	# 5	# 6	# 7	# 8
width	6.78	3.4	4.0	5.2	5.2	6.78	7.36	9.6

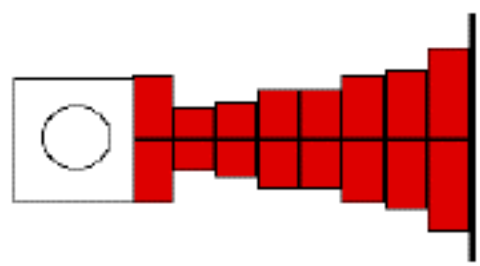


Figure 16: Proposed profile of the new leaf spring

Figure 17 shows the frequency response function of candidate design (B). Table 4 shows the shift in the first mode natural frequency closer to the design target and that the S/N ratio for the undesirable peak gain still remains high compared to the original design condition.

Through the robust sensitivity analysis and redesign iterations, we find an appropriate leaf spring shape which achieves robustness in dynamic performance. The Candidate design (B) would release us from the burden of making the assembly tolerance strictly tight, which decreases the eccentric driving forces.

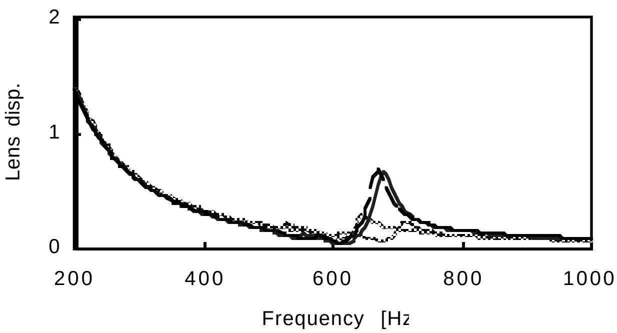


Figure 17: FRF of candidate design (Modification B)

5 . CONCLUSIONS AND FUTURE WORK

This paper demonstrated the application of robust design to the shape synthesis for the dynamic design of a pick-up actuator. The sensitivities derived from the design of experiments using an orthogonal array have sufficient accuracy for engineering design. The sensitivity analysis can lead to the synthesis of a new design with enhanced robustness in dynamic performance. This robust shape-optimization procedure is well suited for the dynamic design of an optical pick-up actuator.

This study considered the excitation force imbalance, the thickness of the leaf spring, and the material property of the leaf spring as the noise factors that affect the dynamic performance. Our study showed that, among these factors, the assembly errors that cause the eccentric magnetic force have the most significant effect on the undesirable peak and its variations. However, the robust design results revealed that a properly shaped leaf spring can reduce considerably the response gain and the variations due to the noise factors. The authors intend to include additional noise factors in the future, such as misalignment of the moving part and leaf spring. These factors may represent more accurately the actual noise conditions and the performance variations.

This case study confirmed the validity of the proposed robust design method in the practical development of mechatronics products. The methodology for robust dynamic design can be summarized as follows.

1) Define the quality characteristics that express the dynamic performance and its variations directly using a signal-to-noise ratio expression

2) Choose the significant factors in manufacturing variation as the noise factors

3) Set the shape parameters as control factors

4) Prepare the math model and execute simulations using design of experiments technique

5) Evaluate the robust sensitivity for each control parameter

6) Optimize the structural shape to maximize the quality characteristics considering the design constraints

This study also performed conventional analytical sensitivity calculations between control parameters and the mean frequency response gain. We confirmed the validity of statistic sensitivity derived from the DOE approach through the comparison of both sensitivity results. However, we could not detect the distinct characteristics in robust sensitivity trend through the investigation of this particular pick-up problem.
If we can establish some relationship between peak optimum and statistical optimum in terms of structural shape, the relationship may lead to a general guideline for robust shape synthesis.

In this actuator design example, manual iterations of the design specifications lead to the multi-objective design optimization of the response gain and the natural frequency. In the future, the authors intend to broaden this design methodology with a numerical design search method for seeking the robust solution that also satisfies multiple design constraints. Pertinent constraints include total structural weight, natural frequency specifications, and part dimension limits.

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