Structural Optimization Design of H-type Piezoelectric Actuator Based on APDL

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Abstract. Aiming at the problem of low efficiency in optimizing the H-type piezoelectric actuator with the “trial method”, this article uses a program written in the APDL (ANSYS Parametric Design Language) to optimize the structure of the piezoelectric actuator. First, a parametric finite element model is established based on the initial data values of the mover structure. Modal analysis and sensitivity analysis are performed to determine the design variables. Then, the objective function is determined based on the frequency consistency of the two-phase working modes. Finally, based on the optimization tool proposed in this paper, the structure of the H-type piezoelectric actuator is optimized. The optimization results show that the optimization program based on APDL can achieve the optimal design requirements of the H-type piezoelectric actuator and shorten the time required for structural optimization design.

1. Introduction
Piezoelectric actuator has the advantages of faster response speed, higher positioning accuracy and smaller size [1]. It has important application value in optical fiber communication, precision machining, aerospace, military field, etc. [2-7] However, the electromechanical coupling efficiency of the patch piezoelectric actuator in the d³¹ mode is low. Coupled with the influence of the fatigue life of the adhesive layer, there are certain difficulties in improving the output performance of the patch piezoelectric actuator. The H-type piezoelectric actuator in this paper is to embed the piezoelectric stack in a metal elastomer without the need to paste a piezoelectric ceramic sheet. The electromechanical coupling efficiency in the d¹₃ working mode is higher, and the driving effect is better. Therefore, the H-type piezoelectric actuator is beneficial to broaden application areas.

The H-type piezoelectric actuator uses the first-order symmetric longitudinal vibration and the second-order symmetric bending vibration to work. The frequency consistency of the two-phase working modes is the basic requirement of the design. In the past, the commonly used “trial method” required manual modification of the structural dimensions for finite element analysis. The steps were cumbersome, and the design cycle was relatively long. The optimization results may not meet the expected requirements. For this problem, this paper discusses the optimization design method of the mover structure based on APDL.

2. Design Requirements of the H-type Piezoelectric Actuator
The H-type piezoelectric actuator discussed in this paper is mainly composed of a metal elastomer, piezoelectric stacks, wedges, and a parallel base. The basic structure is shown in figure 1. The H-type piezoelectric actuator needs to excite the first-order symmetric longitudinal vibration and the
second-order symmetric bending vibration of the elastic body at the same frequency to make the particle on the driving foot move in a standard elliptical trajectory. According to the driving principle of the H-type piezoelectric actuator, the excitation signal, the arrangement position of the piezoelectric stack and the frequency consistency of the two-phase working modes all affect the output performance of the actuator. Therefore, in order to ensure the good mechanical output performance of the H-type piezoelectric actuator, the following requirements should be met when optimizing the design:

- Two sets of same frequency alternating voltages with phase difference $\pi/2$ are applied to the corresponding piezoelectric stacks respectively.
- The four driving feet should be located at the peaks or troughs of the second-order symmetric bending vibration respectively.
- The piezoelectric stacks should be located at the maximum strain of the two-phase working modes.
- The working frequency of first-order symmetric longitudinal vibration and second-order symmetric bending vibration should be adjusted to the same frequency as far as possible. The frequency difference should be less than 200Hz.

Among them, adjusting the frequency difference of two-phase working modes is the key issue of the optimal design of the mover structure. The optimized design of the mover structure is also a key part of the design of the H-type piezoelectric actuator.

![Figure 1. Basic structure of H-type piezoelectric actuator.](image1.png)

![Figure 2. The flow of establishing an optimized design program.](image2.png)

3. Optimal Design

It is very difficult to use the analytic method to calculate the exact solution of the mover’s modal frequency under free-free boundary conditions. But the finite element method can deal with the complex geometry and boundary conditions, and the solution accuracy is high. In addition to this, a multi-physics coupling analysis can also be performed. ANSYS finite element simulation software can better reflect the electromechanical coupling characteristics of the piezoelectric stack. ANSYS is suitable for the optimal design of the mover in the H-type piezoelectric actuator. To improve the speed of optimization design, APDL is used to write a program for the optimization design of the mover structure. Figure 2 is the flow of establishing the optimal design program of the mover structure.

3.1. Establish a Parametric Finite Element Model of the Mover

The material of the metal elastomer in the H-type piezoelectric actuator is phosphor bronze, and the material of the wedge is aluminum alloy. The piezoelectric stack is produced by Suzhou Pant
Piezoelectric Tech Co., Ltd. The size is 2×2×2 mm., The initial size of the elastomer is determined based on the design experience. Figure 3 is the structure diagram of the mover with structural parameters. Table 1 is the initial value of structural parameters of the mover. There are 15 basic structure parameters. Size B is the metal elastomer thickness. Then, the SOLID186 element is used to divide the grid of the metal elastomer. Because of having piezoelectric coupling characteristics, the SOLID226 element is selected to divide the grid of the piezoelectric stacks. And the parametric finite element model of the mover is established on the basis of APDL, as shown in figure 4. The finite element model contains 7842 elements and 13610 nodes.

**Table 1. The initial value of structural parameters of the mover.**

| Structural parameter | L  | L₁ | L₂ | L₃ | L₄ | L₅ | L₆ | H₁ | H₂ | H₃ | H₄ | H₅ | B  | R  |
|----------------------|----|----|----|----|----|----|----|----|----|----|----|----|----|----|
| Initial value (mm)   | 20 | 7.4| 5.4| 1.1| 2  | 4  | 5  | 4  | 8.9| 1  | 4.8| 0.9| 2  | 4  | 0.5|

3.2. *Sensitivity Analysis*

Under the condition that the material and structure of the mover are determined, the structure size of the mover determines the vibration mode. In order to accelerate the convergence speed of optimization design and clear the direction of modifying the structure size, it is necessary to analyze the sensitivity of working modal frequencies to each structural parameter after establishing the parametric finite element model of the mover. The first-order symmetric longitudinal vibration and second-order symmetric bending vibration are working modes of the mover. The structural parameters with high sensitivity are extracted as optimization design variables.

Due to the size limitation of the applied system, the structural parameters, including L, R, L₅, L₆ and B in figure 3, are fixed. If the position of the driving feet, the position of the piezoelectric stack, and the specification of the piezoelectric stack are fixed, L₁, L₂, H₃ and L₇ are unchanged. In order to ensure the strength of the connection between the crossbeam and the vertical beam in figure 3, and to reduce the influence of the vertical beam on the mode of the crossbeam, the size of L₄ and H₂ are fixed. The remaining parameters, including L₃, H₁, H₃ and H₄, are variables. L₃, H₁, H₃ and H₄ are represented by pᵢ (i=1,2,3,4) in sequence. Modal analysis is carried out on the basis of the initial model, and the variation of the structural parameter pᵢ is 0.1mm. As in equation (1) [8], sensitivity analysis is made for the four structural parameters respectively, including L₃, H₁, H₃ and H₄. The results of the sensitivity analysis are shown in figure 5.
\[
\begin{align*}
S_{bj} &= \left( \frac{\partial f_b}{\partial p_j} \right)^{-1} \left( f_{bv} - f_{b0} \right) \left( \Delta p_j \right)^{-1} \\
S_{lj} &= \left( \frac{\partial f_l}{\partial p_j} \right)^{-1} \left( f_{lv} - f_{l0} \right) \left( \Delta p_j \right)^{-1}
\end{align*}
\]

As in equation (1), \( \Delta p_j \) is the variation of the structural parameter \( p_j \), \( f_b \) is the modal frequency of the second-order symmetric bending vibration of the mover, \( f_{b0} \) is the modal frequency of the second-order symmetric bending vibration of the initial mover, and \( f_{bv} \) is the modal frequency of the second-order symmetric bending vibration of the modified mover. \( f_l \) is the modal frequency of the first-order symmetric longitudinal vibration of the mover, \( f_{l0} \) is the modal frequency of the first-order symmetric longitudinal vibration of the mover's initial structure, and \( f_{lv} \) is the modal frequency of the first-order symmetric longitudinal vibration of the mover's modified structure.

According to the analysis results of figure 5, the modal frequency of the first-order symmetric longitudinal vibration is relatively sensitive to \( H_3 \) and \( H_4 \). If the structural parameters \( H_3 \) or \( H_4 \) are modified, the modal frequency of the first-order symmetric longitudinal vibration is relatively easy to be changed. The second-order symmetric bending vibration has the highest sensitivity to \( H_3 \). Therefore, in the optimization design, \( L_3 \), \( H_1 \), \( H_3 \) and \( H_4 \) are selected as design variables. \( L_3 \) is the second design variable. The processing accuracy requirements of the structural parameters of \( H_1 \), \( H_3 \) and \( H_4 \) are improved to reduce the processing error because the processing error can bring adverse effects on the frequency consistency of the two-phase working modes.

**Figure 5.** Sensitivity of the first-order symmetric longitudinal vibration and the second-order symmetric bending vibration to structural parameters.

### 3.3 Automatic Identification of Modes

The modal analysis based on the finite element method requires a discrete vibration structure, and the element characteristic matrix is calculated inside the element. The working mode shape cannot be directly obtained. Modal analysis extracts a series of modes, not only the first-order symmetric longitudinal vibration and second-order symmetric bending vibration. In addition, in the process of adjusting the consistency of the two-phase working modal frequency, it is necessary to modify the structural parameters many times. The size of the working modal frequency and the modal order will also change with the size of the structure.

Therefore, it is necessary to use the modal assurance criteria (MAC) [9-10] to write a program using APDL. In this program, the ANSYS software can automatically identify the first-order symmetric longitudinal vibration and second-order symmetric bending vibration. And it can also automatically extract its modal frequency during the optimization analysis. The calculation formula of the mode correlation coefficient (the value of MAC) is

\[
(MAC)_i = \left( \phi_i^T \phi_{ref} \right)^2 \left( \phi_i^T \phi_i \right)^{-1} \left( \phi_{ref}^T \phi_{ref} \right)^{-1}
\]
As in equation (2), $\phi_i$ is the mode shape to be identified, $\phi_{ref}^i$ is the reference mode shape, and $i$ is the corresponding order of the mode.

The value of MAC ranges from 0 to 1. The larger the MAC value, the higher the degree of similarity between the mode to be identified and the reference mode. By comparing the value of MAC, modal identification can be achieved. Taking the modal identification process of the H-type piezoelectric actuator in this paper as an example, the modal analysis of the finite element model based on the initial structural data is first performed, and the first-order symmetric longitudinal vibration and the second-order symmetric bending vibration are manually found as reference mode shapes. Figure 6 and figure 7 are the corresponding mode shapes. Then, 9 uniformly symmetrically distributed nodes are selected on each of the 8 horizontal outer contour lines of the two crossbeams. There are 72 effective reference nodes in total. Then extract the X (Y) axial modal displacement value of each reference node to form a matrix as the reference mode shape of the first-order symmetric longitudinal vibration (second-order symmetric bending vibration). The mode shape is normalized by mass. Similarly, the mode shape to be identified also needs to extract the equal number of effective nodes at the same position and the X (Y) axial modal displacement values of the effective nodes. Calculate the MAC values of a series of mode shapes to be identified. The mode shape to be identified, which has the largest vale of MAC, is the required first-order symmetric longitudinal vibration (second-order symmetric bending vibration). Figure 8 is a flowchart of modal recognition.

**Figure 6.** First-order symmetric longitudinal vibration mode.
**Figure 7.** Second-order symmetric bending vibration mode.

**Figure 8.** Flowchart of modal recognition.

**Figure 9.** Iteration curve.

### 3.4. Objective Function

The optimal design requires the consistent frequency of the two-phase working modes. The ANSYS software is to find the minimum value of the objective function. Therefore, the absolute value of the frequency difference between the first-order symmetric longitudinal vibration mode and the second-order symmetric bending vibration mode is determined as the objective function, which is

$$f_{obj}(x) = |f_1(x) - f_2(x)|$$  \hspace{1cm} (3)
3.5. Optimization Results
The optimization calculation is started according to the initial values of the structural parameters in table 1. The convergence error of each design variable is $1 \times 10^{-6}$. The convergence error of the objective function is 50, and the zero-order optimization method provided by the ANSYS software's optimization module is used. The results are shown in figure 9 and table 2. It can be seen that the frequency difference of the two-phase working modes is 410.73Hz at the beginning. After structural optimization, it is reduced to 148.21Hz, which meets the design requirements. The optimization effect is obvious, and the optimization efficiency is improved.

| Status            | L3 (mm) | H1 (mm) | H3 (mm) | H4 (mm) | First-order symmetric longitudinal vibration frequency (Hz) | Second-order symmetric bending vibration frequency (Hz) | Frequency difference (Hz) |
|-------------------|---------|---------|---------|---------|----------------------------------------------------------|--------------------------------------------------------|--------------------------|
| Before optimization | 1.1     | 8.9     | 4.8     | 0.9     | 90573.41                                                 | 90162.68                                                | 410.73                   |
| After optimization | 1.41    | 8.84    | 4.88    | 1.1     | 90165.91                                                 | 90314.12                                                | 148.21                   |

4. Conclusion
The key requirement for optimizing the H-type piezoelectric actuator is frequency consistency. The optimization results show that the structure designed by this optimization method based on APDL meets the design requirements. The structure optimization program based on APDL can quickly establish the finite element model and facilitate the later modification of the finite element model, which is suitable for batch operations. This structural optimization method avoids the blindness of "trial". The method shortens the design cycle. It is accurate and convenient.

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6. References
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