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Compensation of mode shapes with a piezo electric actuator for an optical drive

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Abstract

In miniaturized systems (e.g. optical disk drives) much effort is made to design a structure in such a way that the demands on dimensions (height, width) and desired bandwidth are satisfied. Easier, and often cheaper designs can not be used, because they suffer from lower resonance frequencies, which limit the bandwidth. A method is presented for mechatronic structures that compensates unwanted bandwidth limiting resonance frequencies with an additional piezo electric actuator. Advantage of this method is that no extra sensor is used and that the contribution of the piezo is much easier to tune (when compared to a notch filter), since it is part of the structure. Disadvantage is that the piezo influences the dynamics of the original system, which complicates the choice of a suitable piezo actuator.

1 Introduction

When using a mechatronic approach, the mechanical structure and the control system of a device are designed together. This leads to mechanical structures with high internal resonance frequencies, i.e. much higher than the specified closed loop bandwidth of the total system. This means that the mechanical structure has to be stiff but also becomes relatively large and heavy. In our application, the lens actuator of a miniature optical disc drive, the minimal construction space lays a constraint on the size of the actuator structure and thus on its stiffness and resonance frequencies.

If natural frequencies occur below or near the desired bandwidth, notch filters can be used to obtain stability. The disadvantage of such filters are that four (2 frequencies and 2 damping) parameters have to be tuned. Due to mass production tolerances these bandwidth limiting natural frequencies slightly differ and exact compensation with a notch filter is not possible. Moreover, a notch filter is very sensitive to parameter variations during the lifetime of a product.

In this paper we propose a method to compensate the resonances near the desired bandwidth with an extra piezo electric actuator. This method provides a small construction design, a simple controller structure and is easier to tune than a notch filter.

In literature many examples are given of the use of piezo actuators to influence the dynamics of a mechanical structure. For example in [1], [2], [3] and [4] a collocated piezo actuator-sensor pair is used for active damping using integrated force feedback in continuous or flexible structures. Another method of active damping is described in [5], where a two mode piezoelectric shunt circuit is designed and tested to reduce the amplitude of structural modes. Energy is dissipated by a shunt resistor which results in damping. Our method is different in the sense that no piezo electric sensor is used and that the resonances are eliminated instead of damped.

A different and often used application of piezo actuators is found in hard disk drives (see for instance [6], [7], [8]), where the piezo is used as secondary actuator to increase the bandwidth of the system. Most times the piezo is active at high frequencies, while another actuator is active at low frequencies. A single sensor is used to control both actuators. Our method also uses a single sensor to control two actuators, but this controller doesn't drive the two actuators in separate frequency bands.

Summarized our method works as follows. We start with a conventional set-up where a lens is positioned by an electromagnetic actuator. To this we add a piezo electric actuator that excites the mechanical structure at exactly the same resonances as the electromagnetic actuator. By exciting the structure with both actuators, resulting in displacements with appropriate amplitudes but opposite phase, the resonance is cancelled. We will explain this in more detail in section 4.

First we will describe the field of application in section 2. The experimental set-up is described in section 3. This design contains the first (electromagnetic) actuator and shows the dynamic behavior. The effect of resonance compensation is explained and experimental results are shown in section 4. The increase in bandwidth that the proposed method provides in this experiment is shown in section 5. Finally a comparison between the proposed method, a notch filter and input shaping is given in section 6. Some more details of the piezo actuator are given in section 7, including its passive influence on the dynamics. This paper ends with a conclusion and a discussion on directions for further research.

2 Application

In an optical storage disk drive an actuator and a feedback control system are needed to position an objective lens such that the laser spot is kept focused on the centre of the track with high accuracy, despite the presence of disturbances, such as disc skew and eccentricity and external shocks.

In the majority of players a linear tracking actuator is used nowadays, however in miniaturized optical disc players it has certain advantages to apply a rotational tracking actuator. Such a rotational actuator was also used in the first models of Philips CD players [9], but for application in a miniaturized drive one should keep the form factor of rotational actuators as used in hard disk drives in mind [10]. The movements of the lens actuator system are sketched in figure 1. The complete swing arm rotates around a pivot to access and follow the tracks on the disc. Near the lens an electromagnetic actuator generates a vertical force to keep the laser spot focused on the information layer on the disc. As

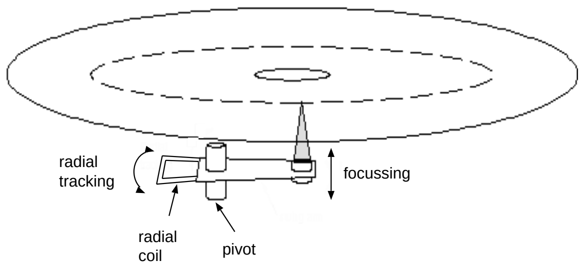

Figure 1: Tracking and focusing movements of a radial lens actuator

can be seen in figure 1 a mechanical structure is needed for straightguiding the focus movement. For this a leaf spring is used that is stiff in the radial direction, but flexible in the focus direction. This flexibility leads to undesired low resonance frequencies. This paper describes a method to compensate these resonances in focus direction. The radial movement is not subject of research here.

3 Experimental set-up

For the experiments a single steel swing arm is used. It is depicted in the photo of figure 2. The arm is actuated in focus direction with an electromagnetic actuator, which exists of a coil placed at the left end of the swing arm and a permanent magnet placed on the fixed world. At the other side the arm is clamped. As said, the radial movement of the arm

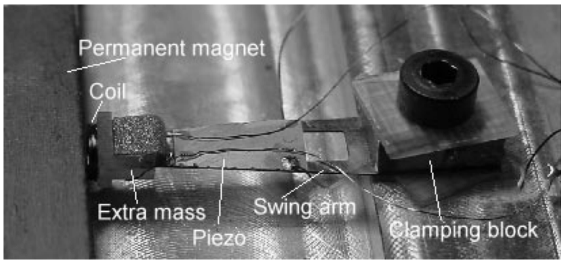

Figure 2: Experimental set-up

will not be examined here. At the small end an additional mass is placed, which represents the lens used in the optical disc drive. In the experiment we measure the vertical displacement of this mass, which is comparable with measuring through the lens as is done in a real player. Here the displacement is measured with a laser Doppler set-up.

The dimensions of the swing arm set-up are as follows. The total length is 12 mm, the width at clamped side is 4 mm. The steel plate has a thickness of 50 μm. The additional piezo electric element has a thickness of 54 μm and is placed in the middle of the swing arm.

The frequency response function of this system (the tip displacement induced by the electromagnetic force) was measured and is given with the dotted line in figure 3. The first resonance frequency occurs near 100 Hz and corresponds to the first bending mode. The second resonance frequency, at 1100 Hz, corresponds to the second bending mode. The occurrence of an anti-resonance just after this resonance results in a phase dip of -180^o. At 4 kHz another resonance (third bending mode), anti-resonance pair occurs. Higher resonances are hardly distinguishable, because their magnitude is comparable to the noise level. Note that we did not optimize the swing arm and that therefore the resonance frequencies are not typical for an optical disc player.

When estimating the maximum bandwidth that can be reached for this system with a lead/lag-controller it turns out that the gain margin at the resonance frequencies is limiting. This analysis results in a bandwidth of about 450 Hz. Not only the second, but also the third bending mode is limiting, because if it had a higher eigenfrequency or a smaller amplitude, it would be possible to place the bandwidth between the second and the third bending resonances. So, if the second bending mode could be compensated, only the third bending mode would be limiting and a bandwidth of approximately 1000 Hz would be achievable.

4 Compensation

To compensate the second bending mode, a second actuator is added to the system, in our case a piezo electric element. The resonance frequencies associated with the transfer function of the piezo actuator to the measured dis-

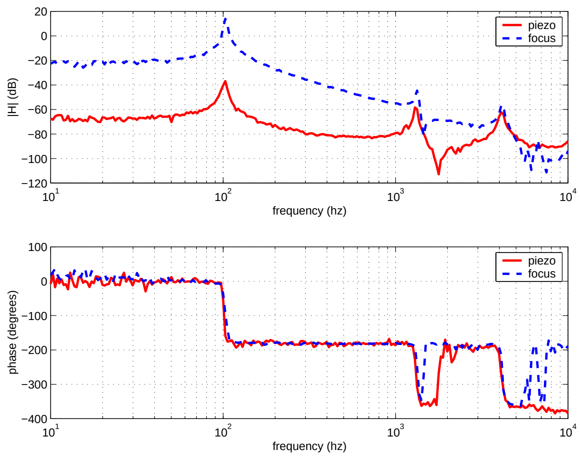

Figure 3: Frequency response of the tip displacement over electromagnetic force and piezo moment

placement are exactly equal to those due to the electromagnetic actuator with a passive piezo, because they only depend on the characteristics of the mechanical structure. The frequency response measurements depicted in figure 3 also show this effect. The anti-resonances have a different value, because they are dependent on the location of sensor and actuators on the swing arm.

Since the tip displacement is determined by the superposition of the contributions of both actuators, the total displacement at the second resonance can be suppressed by giving both contributions appropriate amplitudes but opposite phases. Therefore an additional gain for the piezo actuator is necessary. The block diagram of the total system, which is given in figure 4, shows that the controller structure stays very simple. The optimal gain is determined by the ratio between the displacements due to the electromagnetic and piezo actuator and can be easily tuned in practice.

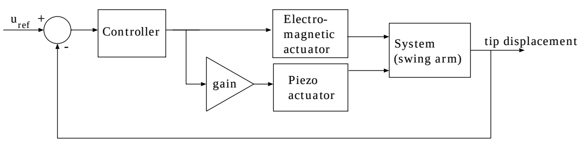

Figure 4: actuation scheme of the swing arm

Theoretically it can be shown that both the second resonance and anti-resonance can be cancelled. For simplicity, two transfer functions with two natural frequencies and no damping are considered. They represent part of the frequency response functions of our system.

$$H_a = \frac{s^2 + z_1^2}{(s^2 + p_1^2)(s^2 + p_2^2)}$$
$$H_b = \frac{s^2 + z_2^2}{(s^2 + p_a^2)(s^2 + p_b^2)} \tag{1}$$

The response functions associated with the piezo and electromagnetic actuator show the same resonances, because they are part of the same system. As a consequence the poles of eq. (1) are corresponding, hence $p_a = p_1$ and $p_b = p_2$.

If the second resonance frequency is compensated, the response function of the total system will only contain the first resonance and a different gain (α). Consequently the corresponding transfer function only contains the poles (s=$\pm jp_1$), as is given in equation (2).

$$H_s = \frac{\alpha}{s^2 + p_1^2} \tag{2}$$

To obtain H_s an additional gain (G) is introduced, which gives the ratio between the input of the actuators. To solve G and α equation (1) and (2) are combined:

$$\frac{s^2 + z_1^2}{(s^2 + p_1^2)(s^2 + p_2^2)} + G\frac{s^2 + z_2^2}{(s^2 + p_1^2)(s^2 + p_2^2)} = \frac{\alpha}{s^2 + p_1^2} \tag{3}$$

Solving this equation for all values of s gives:

$$G = \frac{p_2^2 - z_1^2}{-p_2^2 + z_2^2} \tag{4}$$

$$\alpha = \frac{z_2^2 - z_1^2}{z_2^2 - p_2^2} \tag{5}$$

Thus both the resonance and anti-resonance are cancelled.

The usefulness of this compensation method will be proven with measurements on the experimental set-up, described in section 3. Figure 5 gives the response function of the tip displacement over electromagnetic force with a piezo gain of 2.94. A minor peak stays present in the FRF, due to a small phase shift between piezo and electromagnetic excitation presumably. This is mainly due to a small hysteresis in the piezo electric element and phase shift of the piezo driver. This peak will not result in instability if a controller is used. At the third bending mode the anti-resonance occurs before the resonance, while without active piezo this was opposite. This bending mode is 'overcompensated', therefore a lower piezo gain (G) will result in compensation of this resonance.

To give an idea of robustness of this method, a gain of 5.0 was applied. The resulting frequency response is shown in figure 6. This gain results in overcompensation of both the second and third bending mode. The gain of the third bending mode is larger than in the previous figure. This overcompensation will not lead to instability, due to the phase lead at the resonance frequencies in this case. The resonance frequency remains the same if overcompensation occurs, the anti-resonance frequency is dependent of the used gain. The larger the gain, the larger the influence of the piezo on the system. This results in a lower anti-resonance frequency.

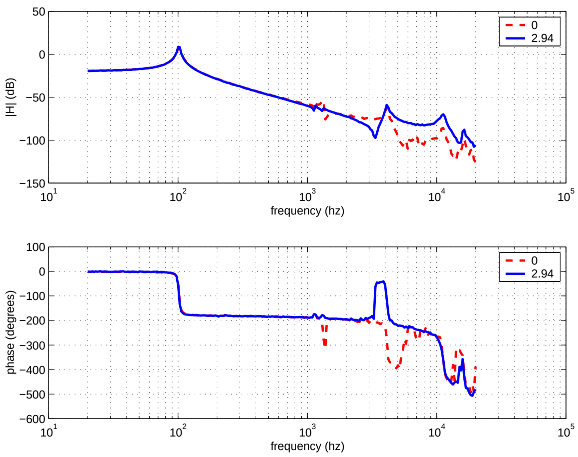

Figure 5: Response of the system with a piezo gain of 2.94

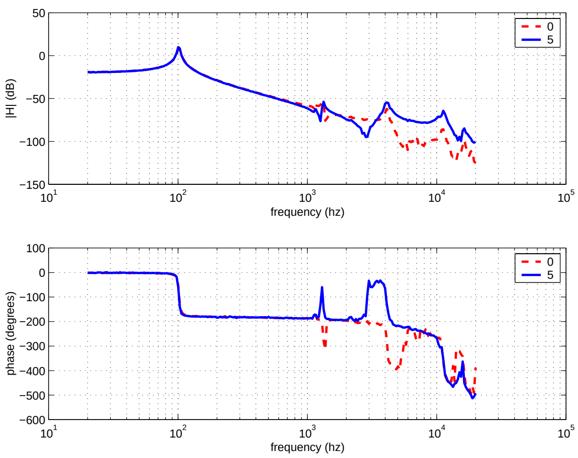

Figure 6: Response of the system with a piezo gain of 5.0

5 Improvements on the bandwidth

A controller for the experimental set-up was designed according to the scheme of figure 4 when the piezo gain is 2.94. A lead/lag controller is used, which stabilizes the system and obtains a large bandwidth. The achieved bandwidth is about 900 Hz, which is twice as much as without piezo actuator (450 Hz) . The open loop behavior is given in figure 7. The time delay caused by the piezo amplifier and the DSP based controller is now limiting for the bandwidth. A phase decrease in the order of 30 degrees is measured just behind the third bending mode, therefore the increase of bandwidth is not much larger than in the situation when the third bending mode is not compensated. Without this time delay a larger bandwidth would be possible, because the fourth bending frequency near 10 kHz would become limiting for the bandwidth. Despite of the problems with time delay the bandwidth is still doubled.

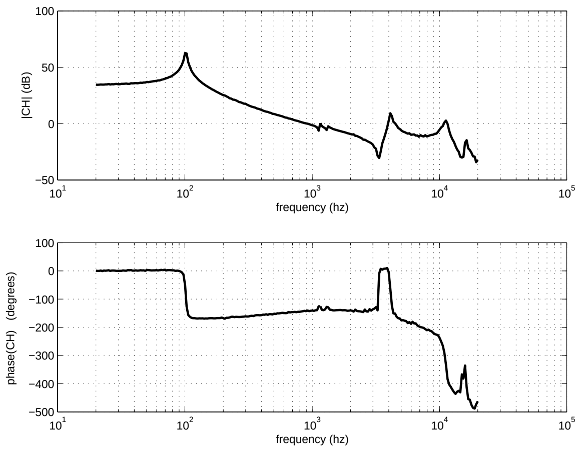

Figure 7: Open loop response with a piezo gain of 2.94

6 Comparison with a notch filter and input shaping

When the measured frequency responses with a passive (see the dotted line in figure 3) and an active correctly tuned ($G = 2.94$) piezo actuator (see figure 5) are compared, a serial filter that is equivalent to the extra actuator can be calculated as follows:

$$H_{filter} = \frac{H_{active}}{H_{passive}} \tag{6}$$

The FRF of the filter is shown in figure 8. The influence of the piezo is comparable to a double (skew) notch filter, around the second and the third bending resonance. The advantage of the piezo actuator is that for this mechanical solution the frequency is automatically correct, while for the notch filter the pole and zero frequencies must be tuned with high accuracy. For the piezo the gain, and for the notch filter the damping parameter are tuned to obtain the desired amplitude of the filter. Moreover a single (skew) notch filter only suppresses the second bending mode (resonance and anti-resonance) and has no influence on the other resonance frequencies. The piezo overcompensates the third bending mode, which results in an anti-resonance/resonance combination. The phase remains larger than -180^{0}, therefore this resonance frequency is not limiting for the bandwidth.

Besides the comparison with a notch filter, the compensation method can be compared with input shaping. Input shaping is used to achieve better settling behavior. This is done by filtering out the resonance frequency of the input signal, to avoid excitation at this resonance. This method does not change the bandwidth of the system. In the method proposed here, the use of the piezo also results in filtering out the resonance frequency to avoid excitation. However, this is now applied inside the control loop at the output of the controller. Therefore this method can result in an increase of the bandwidth, as described in section 5. Advantage of the method is that if resonance frequencies slightly shift during operation, the resonance frequencies of the piezo are equal to the system resonances and only

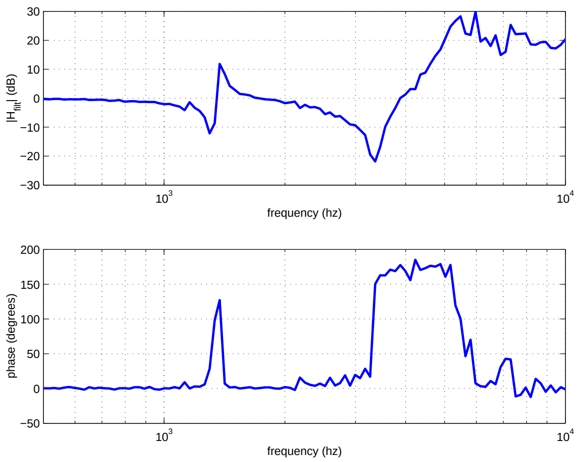

Figure 8: Equivalent filter action of the piezo actuator

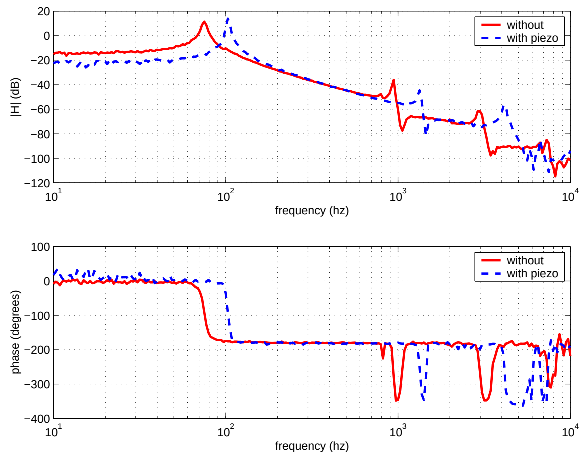

Figure 9: Frequency response function of the swing arm in focus direction with and without passive piezo actuator

a small change in piezo gain may be necessary to maintain compensation. If input shaping is used, the resonance frequencies and amplitudes of the filter may differ from the system dynamics, therefore the resonances may be excited.

7 Piezo actuator

A very thin piezo actuator was used to keep the influence on the dynamics of the swing arm small. A piezo, with a thickness of 54μm of SpemH5D (E_{3c}=2 10^5 V/m, ρ=8150 kg/m^3, d_{31}=290 10^{-12} C/N, Y_{11}=1.675 10^{10} N/m^2) is used, with a length of 6 mm and the width equal to the swing arm. The maximum allowable voltage for this piezo is dependent of the maximum admissible field strength (E_{3c}) and the thickness (t) and is in this case 10 V. The low voltage use of this piezo, makes it applicable in consumer electronics. The piezo actuator is positioned between the hole and the additional mass.

First the response function with a passive piezo was measured (the dotted line in figure 9). By comparing this to the measurement without the piezo actuator, the influence of the piezo on the swing arm is determined. An increase of the resonance frequencies, due to the use of the piezo actuator of approximately 40% is measured. This increase is expected because the total thickness of the structure is increased. Equation (7) (see [11]) can be used to explain this increase.

$$\omega_i = A_i\sqrt{\frac{YI}{\rho A l^4}} \tag{7}$$

where ω_i is the natural frequency associated with mode i, A_i is a constant dependent of the mode number (i) and the boundary conditions but independent of the shapes of the swing arm. Y is the Young's modulus, I the second moment of area, A the surface, l the length and ρ the density. For a beam the surface and second moment of area are defined by:

$$\begin{aligned} A &= bt \\ I &= \frac{bt^3}{12} \end{aligned} \tag{8}$$

Where b is the width and t the thickness.

Combining equation (7) with (8) gives the relation between the thickness and natural frequencies:

$$\omega_i = A_i\sqrt{\frac{Yt^2}{12\rho l^4}} \tag{9}$$

If the total thickness of the stack is doubled, the increase of the resonance frequencies is smaller than a factor two, because the Young's modulus of the piezo is much lower and the density is larger than steel. This stiffness increase results in a larger power consumption of the electromagnetic actuator at low frequencies.

This change in dynamics results in difficulties to determine a suitable piezo actuator. The maximum allowable strain in the piezo actuator should be large enough for compensation. According to [12] these (maximum) strains can be modelled as an equivalent (maximum) bending moment felt by the swing arm. Some adaptations in this model are needed, because the piezo actuator and the swing arm have comparable thicknesses and the piezo actuator is perfectly glued on one side of the swing arm. Therefore it is not valid to assume that the neutral axis of the construction remains. To determine a suitable piezo, simulations are needed which give the needed bending moment of the piezo actuator for compensation. This value should be lower than the maximum allowable bending moment of the piezo actuator. Otherwise new simulations, with another piezo thickness, are executed and compared to the maximum bending moment. This complicates piezo design, because the thickness influences both the needed and maximum allowable bending moment.

8 Conclusion

Resonances of mechanical structures can be compensated by adding an extra actuator to the construction and adding only a gain to the controller. We have proven this with measurements on a model of a swing arm for a miniature optical disc player. The robustness of this method is very good. This gives the possibility to obtain a high bandwidth with a mechanical structure, that has suboptimal dynamics but is lighter and smaller than the optimal structure. The measurements resulted in an increase of the bandwidth with a factor 2.

Only the gain ratio between the piezo actuator and the electromagnetic actuator has to be tuned to achieve this. The frequency where the piezo is active is automatically the same as the resonance frequencies of the complete system, because the piezo is attached to the mechanism. This simplifies tuning, compared to a notch filter.

The piezo influences besides the compensated resonance frequency the other resonance frequencies. The first resonance frequency shows a somewhat larger amplitude. For the third resonance frequency the influence is in this case an advantage, because it is overcompensated and shows now an anti-resonance, resonance. Because overcompensation is possible, the robustness for small changes in resonance frequencies is good if the piezo gain is tuned to slightly overcompensate all cases.

The use of the piezo results in a stiffer swing arm, therefore the resonance frequencies are larger than in the case without piezo. Disadvantage is an increase in power consumption of the electromagnetic actuator. To decrease this the swing arm design can be adapted by using a larger hole.

The increase of the stiffness results also in a different needed bending moment for compensation, which complicates the choice of a suitable piezo actuator.

The used voltage of this piezo actuator is small ($\leq$ 10 V), which makes it useful for consumer electronics.

9 Final Remarks

The influence of the position of the piezo on the piezo gain will be investigated in further research. Mode shapes of the system and the influence of the piezo on these shapes will be simulated and measured. This should lead to an optimal piezo position, with respect to minimal power consumption.

Besides compensation of a bending mode, further research should determine if it is possible to compensate torsion modes by using at least two piezo actuators.

Different measurement and amplifier systems should result in less time delay, which gives the possibility to increase the bandwidth up to the fourth resonance frequency.

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