PIEZOELECTRIC WAFER BASED FEEDBACK IN VIBRATION CONTROL OF LIGHTLY DAMPED BEAMS

TAKÁCS, GERGELY ; ROHAL'-ILKIV, BORIS
Institute of Automation, Measurement and Applied Informatics, Faculty of Mechanical Engineering,
Slovak University of Technology in Bratislava, Námestie slobody 17, 812 31 Bratislava 1, SK,
tel.: +421 2 572 96 111, fax: +421 2 529 25 749,
mail: gergely.takacs@stuba.sk, boris.rohal-ilkiv@stuba.sk

Abstract: The paper presents the experimental identification and validation process of a piezoelectric wafer based deflection position feedback model, utilized in the active vibration attenuation of a lightly damped cantilever beam. An experimental laboratory device is introduced, which utilizes strips of piezoelectric material as structural actuators and sensors to create an active structure emulating the behavior of a class of engineering problems. Detailed account is given on the identification process of the feedback signal provided by the piezoelectric sensors to the system controller. The feedback model is validated both in time and frequency domain, utilizing a linear quadratic controller as a basis of comparison. The cantilever is excited manually and using an electrodynamic shaker, while the piezoelectric sensor based feedback control is compared to direct high precision industrial laser triangulation feedback. Experimental results verify, that a mere second order feedback model based control process covering the first dominant vibration frequency is comparable to control utilizing direct distance readings. Moreover the damping effect exceeds the bandwidth of interest as the single-sided cantilever beam tip deflection amplitude spectra shows no substantial difference between the two feedback methods for higher structural vibration modes.

Keywords: piezoelectric, sensor, feedback, vibration control, mechatronics

1 INTRODUCTION

Employing active vibration attenuation techniques in commercial products is slowly becoming a reality. One might think of aeronautical applications, like the damping of helicopter rotor wing vibrations, active stabilization of large space structures or for example vibration attenuation of antenna masts [Boeing, 2004; Phillips et al., 1990; Blachowski, 2007]. Other active and semi-active methods of vibration damping are taken on by the automotive industry for controlling suspension systems. The possibilities of using advanced materials combined with progressive control algorithms to eliminate undesired vibration effects is practically limitless [Preumont, 2002; Inman, 2006].

While the proven technologies are transferred into practice, there is a constant need to investigate further and widen the boundaries of active vibration damping in research laboratories. A rather important branch of research is focused on advancing the field of active materials; like piezoelectrics, electro-active polymers, magneto-rheologic fluids, shape-memory alloys and others. But no active control system is complete without the proper control algorithm, therefore another essential part of the scientific process is to investigate how already existing technologies can benefit from better control methods.

1.1 Motivation

Acquiring a reliable feedback signal is essential for the efficiency and reliability of all control systems. When considering means of sensing vibration levels, one has to balance between precision, price and the physical interaction with the given mechanical system. Mass produced products need to be cost optimized, what naturally involves tradeoffs in the sensing apparatus and thus also the condition of the feedback signal. But not only the sensors must be simple, quite often it is beneficial to keep the computation load on minimum in order to make optimization based control algorithms a viable option.

Laboratory applications often consider LASER Doppler vibrometry to gain feedback signal to the controller. In addition to great precision, contact-free measurement has an advantage of preventing structural interaction with the controlled system. LASER Doppler vibrometers however are out of question for mass-produced applications, as their placement is very problematic, the sensor heads and processing equipment are heavy, large and very expensive. Industrial grade LASER optical sensors based on triangulation methods are a rather good compromise, since they are smaller and less expensive than their laboratory grade counterparts [Takács and Rohal'-Ilkiv, 2009a]. Physical dimensions and inconvenient placement is still an issue, and because of their price range they can be only recommended for high budget products, like the ones encountered in the aviation industry and military.

The use of accelerometers is very common in academic publications, as they are relatively cheap and provide precise feedback to the control system [Qiu et al., 2009; Dong et al., 2006; Petersen and Pota, 2003]. Real life product integration is also a feasible possibility. Accelerometer miniaturization has come to a point where these devices can be bonded or mounted to the structural surface without significantly altering its mass-stiffness properties.

Similarly common is the utilization of the direct piezoelectric effect to gain vibration level estimates of the controlled mechanical structure [Wills et al., 2008; Kermani et al., 2004; Lin and Nien, 2005; Sloss et al., 2003]. In these feedback control systems a piezoelectric patch is either bonded onto the structural surface, or directly integrated into the structure material. Better mass distribution, very low price and the possibility of direct structural integration is definitely an advantage over accelerometers. Neither accelerometers, nor piezoelectric wafers may sense the D.C. component of vibrations, this is however usually not an issue.

1.2 Problem statement

A lightly damped, cantilever-like active mechanical structure is given with bonded piezoelectric actuators. An additional piezoelectric patch is placed close to the clamped end and acts as a vibration sensor. The task is to create a simple second-order linear model to estimate deflections of the mechanical structure at the beam tip.

The measurement estimates shall be compared to conventionally true deflection levels acquired through a LASER triangulation device both in time and frequency domain. In addition to that, possible damping performance degradation is investigated through comparing damping efficiency of the estimate based feedback with direct feedback control.

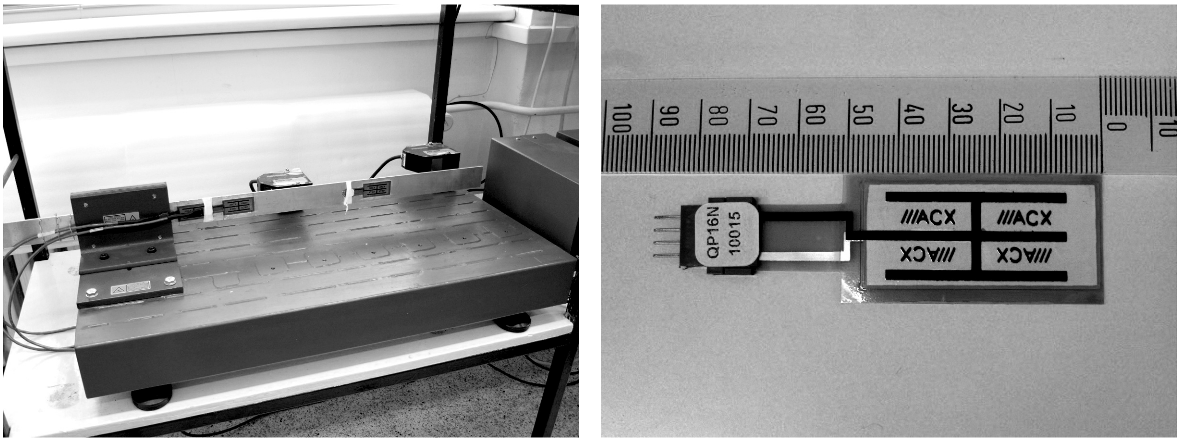
A second order measurement model includes only the first resonant mode of the structure. However the inclusion of the first dominant mode of the lightly damped structure could be sufficient to control the vibrational behavior even at higher frequency excitations. Damping performance of direct and estimate based feedback shall be thus compared through frequency domain measurements, reaching higher resonant modes. This shall answer the question whether a mere second order model is suitable for position estimates, even when the structure is subject to excitations exceeding the bandwidth of the model. Using the simplest possible solution to assess feedback signal may help to keep additional computational load at minimum, which is essential in optimization based control systems with short sampling periods. The paper thus attempts to find a feedback solution which provides hardware and computation time costs at bare minimum.

2 HARDWARE DESCRIPTION

A clamped cantilever beam is given, which may model the general vibrational response of a class of engineering problems. This lightly damped mechanical system behaves similarly to the helicopter rotor beams in flight, manipulation arms or solar panels in outer space and many other real-life structures.

The experimental laboratory setup assumed throughout this work is presented on Figure 1(a). The beam is composed of commercially pure aluminum with the dimensions of $550 \times 40 \times 3$ mm. A heavy base is necessary to prevent mechanical interaction with the outside.

A pair of piezoelectric actuators is mounted close to the clamped end. The actuators are identical, manufactured by MIDÉ having the factory designation mark QuickPack QP16n. The outside dimensions of the actuator are $45.9 \times 20.7 \times 0.25$ mm and it is shown on Figure 1(b). The actuators are connected counter phase, receiving the same high voltage signal through a MIDÉ EL-1225 operational power ampli-



(a) Active structure (b) Piezoelectric patch

Figure 1 – A clamped cantilever beam with piezoelectric sensors and actuators is featured on (a), while detail of the piezoelectric actuators and sensor is shown on (b).

fier. The placement of the actuators has been influenced by the goal of maximizing deflection amplitudes at the first resonant frequency.

A third patch identical to the actuators is bonded onto the structural surface. This piezoelectric patch acts as a sensor, making use of the direct piezoelectric effect. Its optimal placement has not been a subject of this research, however we have to note that Finite Element Modeling (FEM) has been used to avoid anti-resonance nodes at higher frequencies. Other placement criteria included the minimization of mechanical interaction with the structure, by placing the minimum amount of lead wires close to the structural surface. Voltage signal acquired from the sensor is directly connected to the analogue input of a laboratory measurement card, without additional amplification. When the beam is in first resonance, the output voltage levels actually exceed the possible input range of the data acquisition device. A 100 kΩ resistor is installed parallel to the piezo patch to match the voltage levels with the A/D input device[1].

Vibration levels are measured directly through an industrial LASER triangulation sensor, placed at the free end of the cantilever beam. A Keyence LK-G82 sensor with an accuracy of $\pm 0.05\%$ and the resolution of 0.2μm provides direct distance readings in the range of 80 ± 15mm. Measurements are forwarded to a Keyence LK-G3001V central processing unit for filtering and finally a scaled analogue voltage signal is passed onto the measurement card. The distance readings provided by this system are considered as reference throughout this work.

Figure 2 features the simplified schematic representation of the laboratory experimental hardware. The computer marked as xPC Target on the figure serves for implementing the controller and data logging software real-time, on the Mathworks xPC Target rapid software prototyping system. This computer contains a National Instruments DAQ-6030 measurement card with 18bit resolution and amongst others two analogue outputs with ± 10V range, necessary to drive the amplifiers for the piezoelectric actuators.

Controller design and development is taking place on a separate computer, marked as xPC Host on the figure. The development platform used in this setting is Matlab / Simulink, where the block schemes responsible for control and data logging are transferred to the real-time controller via Ethernet, through the TCP/IP protocol.

[1]The A/D device input levels are set to ± 10V.

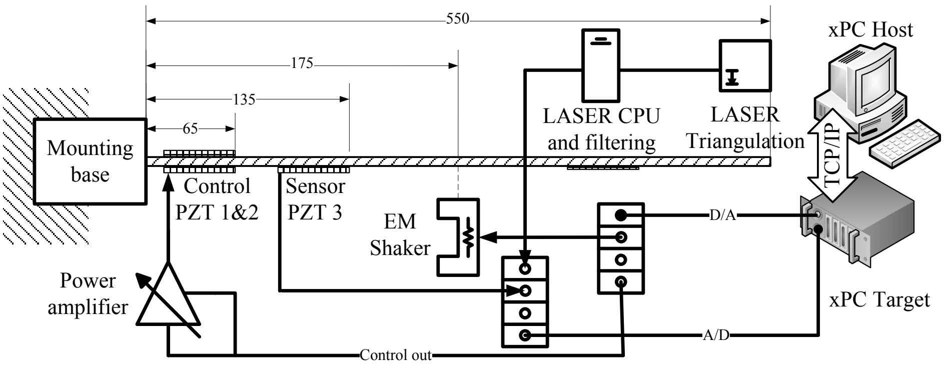


Figure 2 – Simplified schematic representation of the laboratory experimental hardware.

3 SYSTEM AND MEASUREMENT DYNAMICS MODELING

Although it would be possible to create a single dynamic model, describing the input - output relationship of the actuator behavior and the piezoelectric patch; creating separate system and measurement models has its advantages. If constrained model based predictive control (MPC) is considered as the choice of control algorithm, a separate structural model is necessary to enforce constraint limits on the vibration output. Therefore separate system and estimation dynamics is assumed in this paper due to the inspiring possibility to use MPC on lightly damped vibrating structure, posing numerous practical issues upon implementation [Wills et al., 2008; Hassan et al., 2007; Takács and Rohal'-Ilkiv, 2009c].

Linear time-invariant (LTI) state-space systems describe the dynamics of the system and measurement process according to:

$$x_{k+1} = Ax_k + Bu_k \qquad y_k = Cx_k \tag{1}$$

where x is a 2×1 state vector, u is a 1×1 input, y is a 1×1 output. Matrices A,B and C are the transition matrix, input matrix and output matrix. Integer k denotes sampling instances.

3.1 System dynamics

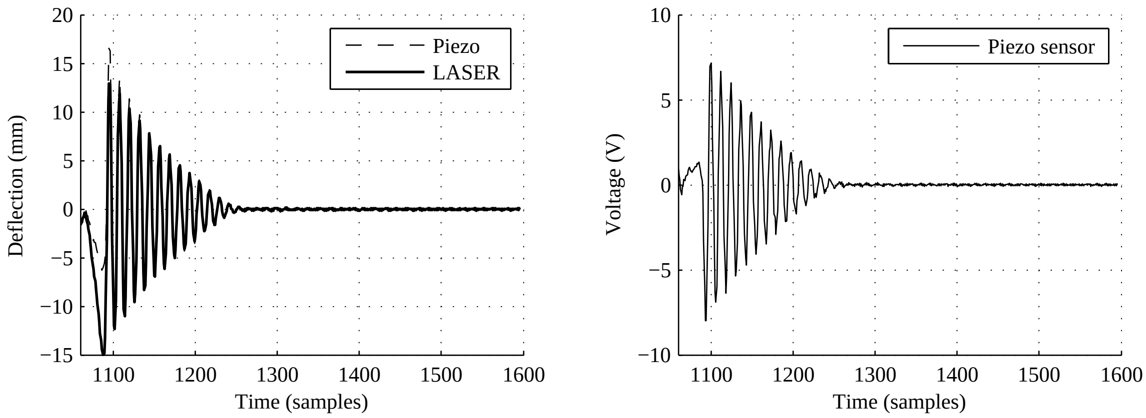
The system dynamics of the experimental device are described by a second-order LTI state-space model, modeling the relationship between voltage input to the piezoelectric actuators and deflections directly measured at the cantilever beam tip in millimeters.

This model has been identified experimentally. A chirp signal in the range of $0-20$Hz, amplified to the polarization limits of the actuators has been supplied, while the tip deflections have been logged with a sampling time of T=0.01 seconds. This sampling period is sufficient for the given model, since it significantly exceeds the first resonant frequency of 8.1 Hz. The filtered and detrended time domain data has been converted into the frequency domain using Fast Fourier Transform (FFT).

The final state-space model has been created utilizing a subspace-iteration method as described in Ljung [1999]. The model described by (2) has been already proven in MPC controlled vibration attenuation, using direct tip deflection readings as reference [Takács and Rohal'-Ilkiv, 2009a,c].

$$A = \begin{bmatrix} 0.867 & 1.119 \\ -0.214 & 0.870 \end{bmatrix} \qquad B = \begin{bmatrix} 9.336E^{-4} \\ 5.309E^{-4} \end{bmatrix} \qquad C = [-0.553 \quad -0.705] \tag{2}$$

Akaike Final Prediction Error (FPE) criterion for this model has been calculated to be 0.0142 (-). The model validation process proved to yield a satisfactory match, while the transient and frequency response of the model was also adequate for the considered bandwidth.



(a) Measured and estimated beam tip deflection

(b) Piezoelectric patch voltage

Figure 3 – Directly measured beam tip deflections are compared to piezoelectric sensor based estimates on (a), while (b) shows the corresponding measured voltage output of the piezoelectric patch .

3.2 Piezoelectric sensor estimation model

The second order LTI state-space model, describing the relationship between the measured output voltage of the piezoelectric sensor and the direct tip deflection readings has been identified experimentally as well.

A pseudo-random manual excitation has been applied to the beam, while the exact tip distance readings using the LASER triangulation system and the voltage output from the sensor has been logged. The resulting data set has been pre-processed to remove trends means and frequencies exceeding the bandwidth of interest. The final state-space model has been calculated using the subspace iteration method featured in Ljung [1999].

After comparing the model output with the validation data, the state-space system described by (3) has been selected as the basis for piezo patch based deflection estimation. Examining the transient and frequency response, model residuals, along with a FPE criterion of 0.0091 (-) indicated a model suitable for further use in this work.

$$A = \begin{bmatrix} 0.987 & 0.144 \\ -0.274 & 0.009 \end{bmatrix} \qquad B = \begin{bmatrix} 3.959E^{-2} \\ 1.851E^{-1} \end{bmatrix} \qquad C = [34.72 \quad -1.359] \tag{3}$$

4 FEEDBACK MODEL VALIDATION

Experiments performed to validate the tip deflection estimation model assume a system model described by (2) and a piezoelectric sensor feedback measurement model according to (3). Estimate model sampling has been set to $T = 0.01$ s, however a high frequency excitation test involved a data logging rate of $T = 0.0002$ in order to capture dynamics above the bandwidth of interest, while leaving model sampling at its default value.

4.1 Time domain position estimates

The beam tip has been deflected 10 mm away from its equilibrium state and released to vibrate under LQ control[2]. After the initial deflection, the cantilever beam has not been subjected to other outside excitation. The results of this experiment are featured on Figure 3.

Figure 3(a) shows measured and estimated tip displacements. As it is expected, the piezoelectric patch based tip deflection estimate is incorrect when the beam is subjected to slow changes, since it is not picking up the D.C. component of a changing signal. After the beam starts to vibrate at sample time

[2] See Section 5 for the controller description.

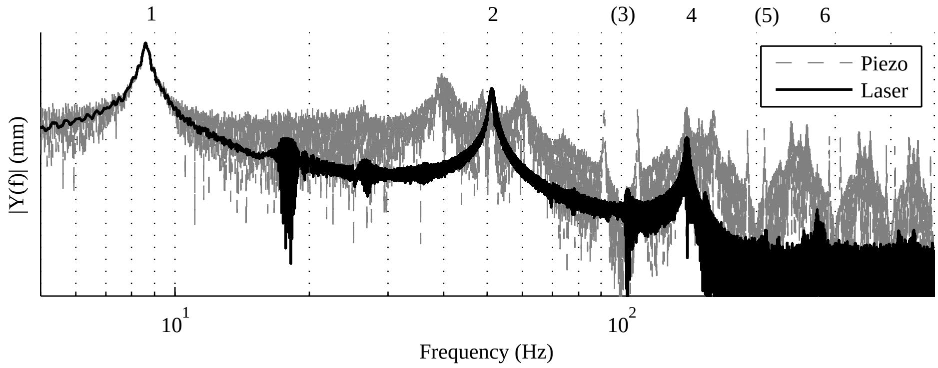


Figure 4 – Measured and piezoelectric sensor feedback estimated beam tip deflections in a single-sided amplitude spectrum. Numbers denote corresponding structural vibration modes.

1100, the model tends to slightly overestimate positive deflections, however at the following periods it gets nearly indistinguishable from the precise LASER reference measurements. The unprocessed direct voltage output of the piezoelectric sensor patch is featured on Figure 3(a).

4.2 Frequency domain position estimates

The frequency domain experiment involved an outside mechanical excitation provided by a Bruel&Kjær Type 4810 electrodynamic shaker. The shaker has been mechanically connected to the beam surface 175 mm away from the clamped beam base [3]. The excitation signal has been passed onto the laboratory shaker through a Bruel&Kjær Type 2718 amplifier. A 200 seconds long chirp signal swept through frequencies of 0-500Hz, exciting the beam through its first five measurable resonant peaks, up to deflections of approximately $\pm$15mm at the first.

The single-sided amplitude spectra of laser measured and piezoelectric sensor based tip displacement estimates are indicated on Figure 4[4]. The black response is measured directly, while the lighter shade indicates the piezoelectric sensor based estimates. Resonant modes are numbered, where modes (3) and (5) are twisting modes which cannot be controlled or directly measured using this hardware configuration [Takács, 2009].

As it is evident from the response, the beam tip deflections are correctly estimated only around the neighborhood of the first resonant mode. Neither the model order nor the native sampling period makes possible to correctly assess tip position above 15 Hz. Spectral leakage, an artifact of the FFT transformation is visible beyond this frequency range for the piezo estimate. This carries no information and is merely a side-effect of performing FFT on a low frequency content data. On the other side, near D.C. position changes and slow vibrations cannot be detected by the piezoelectric sensor.

If the explicit inclusion of higher order dynamics in the estimation model is necessary, the model order can be increased. However this also leads to a need to increase model sampling rate, which could be an issue for computation intensive control algorithms like MPC [Takács and Rohal'-Ilkiv, 2009b].

5 DAMPING PERFORMANCE

To find out whether the use of estimated beam tip deflections in the feedback control loop cause a significant degradation in control performance, a series of experiments were performed both in the time and frequency domain. Deflections have been measured utilizing the LASER triangulation method,

[3]This mechanical connection has been only present when the shaker was needed, time domain tests were performed without this addition.

[4]Note that the input force has not been measured, therefore the featured response is not a transfer function - it only analyzes the nature of the output signal in the frequency domain.

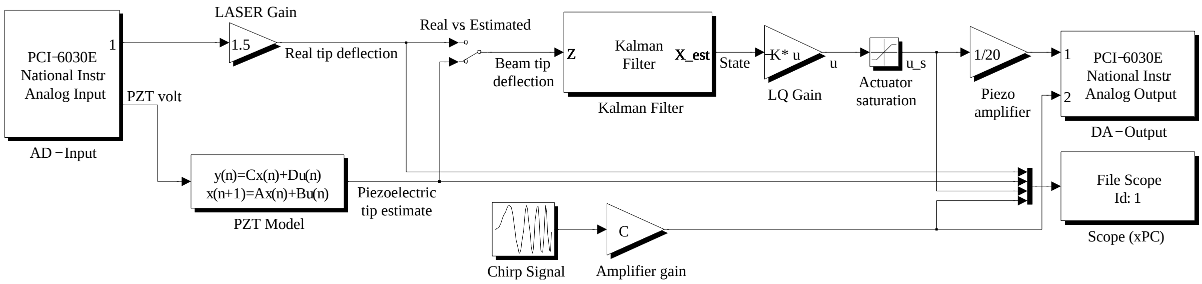


Figure 5 – Simplified block scheme of the controller algorithm.

while the system has been subjected to identical excitation with identical control methods. The only difference was the use of direct measurements or piezoelectric sensor based feedback estimates.

The control strategy considered throughout the damping performance comparison test was a simple linear quadratic (LQ) controller. The output of this controller has been saturated to ± 120 V in order to prevent depolarization of the piezoelectric material [Spangler, 2007; MIDÉ, 2007]. The LQ controller has been calculated using the state space model according to (2), a state penalty matrix $Q = C^T C$ and an input penalty $R = 10^{-4}$. Controller sampling rate has been set to $T = 0.01$s. The input penalty is based on previous experiments, and its value is suitable for the previously mentioned saturation limits [Takács, 2009]. The LQ controller gain K can be expressed according to (4).

$$K = [12.97 \qquad -125.50] \tag{4}$$

Figure 5 indicates the simplified block scheme of the control software implemented on the xPC real-time rapid software prototyping system. The analogue voltage output is acquired through the measurement card, this includes exact measurements from the laser and the voltage signal from the piezoelectric sensor. The piezo sensor voltage is passed through the estimation model, while a switch enables the user to select between direct or estimated position feedback signals. The feedback then passes through a Kalman state estimator, and the resulting state estimates are multiplied by the LQ gain vector. Outputs are saturated and adjusted to the amplification level of the power amplifier. The block scheme also includes input to the electrodynamic shaker and means for data logging.

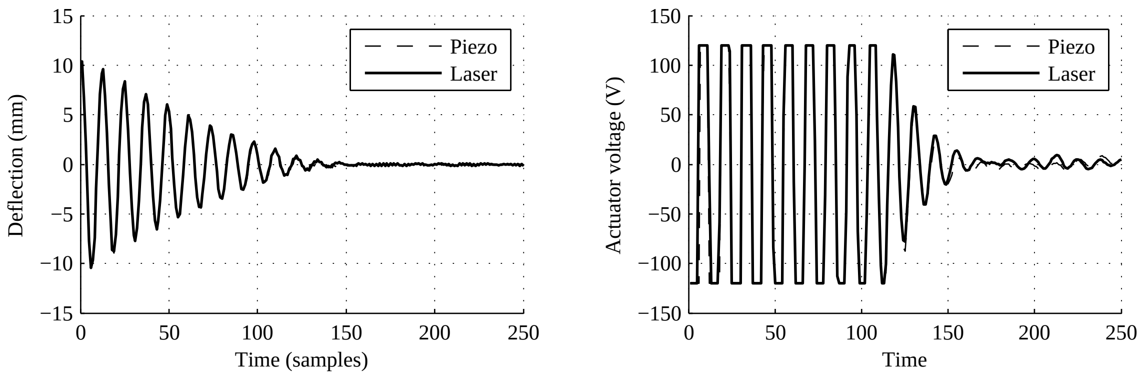
Plots in this section do not indicate free response without control. This is to keep the responses clear and readable. Saturated LQ control with direct feedback provides a very effective damping performance, effectively reducing settling times by an order of magnitude. This is comparable to constrained MPC control featured in [Takács and Rohal'-Ilkiv, 2009b,c; Takács, 2009].

5.1 Time domain test - initial deflection

The beam has been deflected to a position 10 mm away from its equilibrium and then released to vibrate under saturated LQ control. As it has been previously mentioned, the two scenarios differ only in the means of acquiring the feedback signal and all responses are measured directly through the triangulation device.

Beam tip vibrations following the release of the structure are featured on Figure 6(a). As it is evident from this figure, the two responses are indistinguishable, the estimated feedback values do not degrade the damping behavior under saturated LQ control.

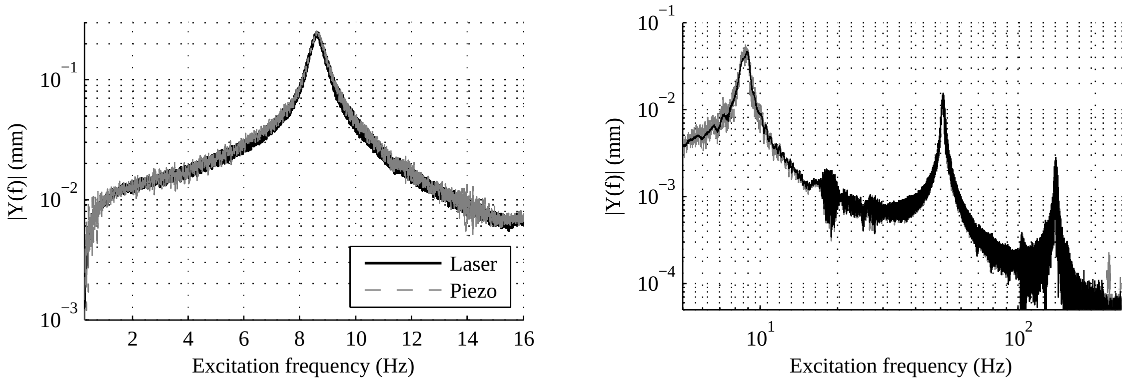
A very slight difference between the two responses is observable on Figure 6(b), showing the voltage signal sent to the piezoelectric actuators. The responses are identical up to the time sample 120. After this point the differences are to be attributed to the fact, that unwanted outside excitations occur in both cases and they are compensated by the controller. These effects are only observable when the beam is near its equilibrium state.



(a) Directly measured beam tip position

(b) Voltage signal to actuators

Figure 6 – Direct position feedback based control compared to piezoelectric sensor estimated feedback in an initial deflection test is indicated on (a), while corresponding controller voltage outputs are presented on (b).



(a) LASER measured frequency response

(b) Wide-band frequency response

Figure 7 – Direct LASER triangulation measured response is indicated on the low frequency single sided amplitude spectra on (a), while (b) shows damping performance for higher structural modes.

5.2 Frequency domain test - electrodynamic shaker

The possible damping performance degradation attributed to the use of estimated tip position feedback has been investigated in the frequency domain as well. The mechanical structure has been excited using the laboratory shaker setup described in 4.2.

Two tests were performed: one for the bandwidth of interest and the other for higher frequencies. The first test involved an excitation through an amplified chirp signal for the bandwidth of $0 - 20$ Hz. All of the discrete sampling periods were set to $T = 0.01$. The second test was aimed to investigate wheter the low order structure and position estimate model provides satisfactory feedback at higher frequencies. For this test the estimation model and controller sampling frequency was left at its original value, however the excitation signal and data logging has been sampled by an increased rate of $T = 0.0002$ seconds.

The directly measured single-sided amplitude spectra of beam tip vibrations for the direct and estimated feedback controlled systems is featured on Figure 7. The two control schemes are practically indistinguishable in the bandwidth of interest, just as it is indicated by 7(a). In the region of the first resonant frequency, both control feedback schemes perform equally well.

According to 4.2 the piezoelectric sensor based low-order feedback models are only useful close to the frequency range around the first structural resonant frequency. Despite of this fact the results featured on Figure 7(b) indicate that the estimated control scheme gives comparable results to the one with access to direct feedback readings. This experiment utilized second order system and measurement models, however it has been excited with frequencies high above its normal bandwidth. As it is indicated by Fig. 7(b) the damping performance of the model estimated feedback scheme is very similar to the one with access to direct deflection measurements.

6 CONCLUSION

A low order position estimation model, based on piezoelectric sensor signals for a lightly damped active structure has been introduced in this paper. Along with the experimental validation of the estimator, the control performance has been evaluated for the direct measurement and estimate based controllers.

The free and controlled vibration response of the beam tip in the time domain is dominated by the first structural resonant mode, thus a second order tip estimation model is sufficient to generate responses closely matching to reference values. The frequency domain experiment clearly indicates limitations of such low order models, along with the physical properties of the piezoelectric sensors: the estimation model provides the best results in the vicinity of the first structural resonance mode.

The controlled vibration verification tests show, that the the controller having access to estimated tip deflections performs nearly identically to the one utilizing direct measurements. This is not only true for the initial deflection test performed in the time domain, but also for the frequency domain. According to these experiments, the damping effect of both control schemes is identical. Additionally, the estimate based feedback controller provides comparable damping performance to the direct measurement feedback counterpart even at excitation frequencies exceeding the original bandwidth of interest.

It may be concluded that a piezoelectric sensor based feedback signal utilized on a lightly damped vibrating structure offers a damping performance comparable to directly measured feedback control, even when using a simple second order feedback estimate model.

6.1 Future works

The inclusion of higher order beam resonance modes in the estimation models may bring a qualitative increase in the precision of tip position estimates. However it still remains a question whether there are any practical advantages to use high order state-space models in model based predictive control of such and similar lightly damped structures. On the other side with larger model orders increased computation times are always to be expected, which may pose practical problems in the implementation of computation heavy optimization based control methods. Future works shall address these questions in more detail.

Further works shall investigate the possibility to utilize other types of sensors in estimating position changes of lightly damped vibrating structures. The use of capacitive proximity sensors in active vibration damping systems is somewhat unusual, however this contact-free measurement method could offer numerous advantages. It is important to study whether the bandwidth limitations of such a sensor permit its use as feedback source in vibration attenuation, or how its dynamic reaction range is influenced by using aluminum instead of standardly assumed steel. An upcoming investigation shall explore this attractive alternative too.

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