Identification of Smart Structures with Robust Control under Stochastic Excitation

Amalia J. Moutsopoulou, Georgios E. Stavroulakis, and Tasos D. Pouliezos

Abstract—In light of past research in this field, this paper intends to discuss some innovative approaches in vibration control of smart structures, particularly in the case of structures with embedded piezoelectric materials. In this work, we review the principal problems in mechanical engineering that the structural control engineer has to address when designing robust control laws: structural modeling techniques, uncertainty modeling, and robustness validation under stochastic excitation. Control laws are desirable for systems where guaranteed stability or performance is required despite the presence of multiple sources of uncertainty.

Index Terms—Smart Structures, Robust Control, H Infinity Control, Structural Modeling, Uncertainty.

I. INTRODUCTION

The design and implementation of control strategies for large, flexible smart structures presents challenging problems [1], [2], [3]. One of the difficulties arises in the approximation of high-order finite element models with low-order models. Another difficulty in controller design arises from the presence of unmodeled dynamics and incorrect knowledge of the structural parameter. The study of algorithms for active vibrations control in smart structures became an area of enormous interest, mainly due to the countless demands of an optimal performance of mechanical systems as aircraft and aerospace structures. Smart structures, formed by a structure base, coupled with piezoelectric actuators and sensor are capable to guarantee the conditions demanded through the application of several types of controllers [4], [5], [6]. This article shows some steps that should be followed in the design of a smart structure.

Due to space and weight constraints, smart structures are extensively used in many applications such as aerospace and automotive ones. The lightly damped nature of flexible structures can lead to considerable structural vibrations and cause unpleasant noises, unwanted stresses, malfunctions and even structural failures. In recent years, piezoelectric materials have been widely used as transducers for efficient active vibration control of various flexible structures [10], [31], [32], [33]. Owing to the complex dynamics of piezoelectric flexible structures, their dynamical models are normally obtained with finite element method (FEM) and/or system identification [3], [4], [5]. However, in the presence of random variations in structural properties and/or the errors in the identification process, the obtained dynamical models inevitably have parametric uncertainties. Besides, a dynamic uncertainty is necessary to represent high-frequency neglected dynamics which may lead to the spillover effect [31], [32], [33]. When active control systems are designed based on the reduced nominal models, it is crucial to take into account different inaccuracies in these models to ensure the stability and the performance of the final closed-loop system. Considering the presence of parametric and dynamic uncertainties, phase and gain control policies based $H_\infty$ output feedback control is proposed in Zhang et al. [8], [24] which affords a principle for the weighting function selection in $H_\infty$ control to consider a set of control objectives simultaneously. The beam has length $L$, width $W$ and thickness $h$. The sensors and the actuators have width $b_S$ and $b_A$ and thickness $h_S$ and $h_A$, respectively. The electromechanical parameters of the beam of interest are given in the Table I.

| Parameters | Values |
|------------|--------|
| Beam length, $L$ | 0.8m |
| Beam width, $W$ | 0.08m |
| Beam thickness, $h$ | 0.0093m |
| Beam density, $\rho$ | 1800kg/m$^3$ |
| Young's modulus of the beam, $E$ | 1.5 $\times 10^{11}$ N/m$^2$ |
| Piezoelectric constant, $d_{31}$ | 254 $\times 10^{-12}$ m/N |
| Electric constant, $\varepsilon_{33}$ | 11.5 $\times 10^{-9}$ V m/N |
| Young's modulus of the piezoelectric element | 1.5 $\times 10^{11}$ N/m$^2$ |
| Width of the piezoelectric element | $b_S=ba=0.07m$ |
| Thickness of the piezoelectric element | $h_S=ha=0.0002m$ |

In order to derive the basic equations for piezoelectric sensors and actuators [10], [11], [12], we assume that:

- The piezoelectric sensors actuators (S/A) are bonded perfectly on the host beam;
- The piezoelectric layers are much thinner than the host beam;
- The piezoelectric material is homogeneous, transversely isotropic and linearly elastic [5], [16], [18].

II. THEORETICAL FORMULATION

The dynamical description of the system is given by,

$$Mq(t) + Dq(t) + Kq(t) = f_m(t) + f_e(t)$$  \hspace{1cm} (1)

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where $M$ is the generalized mass matrix, $D$ the viscous damping matrix, $K$ the generalized stiffness matrix, $f_{au}$ the external loading vector and $f_e$ the generalised control force vector produced by electromechanical coupling effects. The independent variable $q(t)$ is composed of transversal deflections $w_i$ and rotations $y_i$, i.e. [20], [23]

Furthermore to express $f_i(t)$ as $B_d(t)$ we write it as $f^{*}_e \mathbf{u}$, where $f^{*}_e$ is the piezoelectric force for a unit applied on the corresponding actuator, and $\mathbf{u}$ represents the voltages on the actuators. Lastly $d(t) = f_{au}(t)$ is the disturbance vector [16], [17].

III. CONTROL DESIGN

We solve a regulator problem for the smart beam with viscous layer. The objective in this section is to determine the optimal vector of active control forces $\mathbf{u}(t)$ subjected to performance criteria and satisfying the dynamical equations of the system, such that to reduce in an optimal way the external excitations. We consider the steady state (infinite time) case, i.e. the optimization horizon is allowed to extend to infinity. We seek a linear state feedback [6], [7], [13]

$$u(t) = -K x(t), \text{ with constant gain } K.$$ 

The control problem is to keep the beam in equilibrium which means zero displacements and rotations in the face of external disturbances, noise and model inaccuracies, using the available measurements (displacement) and controls [14], [15], [18].

IV. H INFINITY CONTROL

To relate the structures used in classical and $H_\infty$ control, let’s look at Fig. 1, Control bloc diagram in the frequency domain [26], [27], [28]

In this diagram are included all inputs and outputs of interest, along with their respective weights $W$, where $W_d$, $W_e$, $W_n$, $W_r$ are the weighs for the disturbances, control, noise, outputs respectively. The exogenous inputs are the noise $n$ and the disturbances $d$. $K(s)$ is the controller, $B$, $G$, $x$, $y$, $C$ define at the relation (6, 7, 8) and $F(s)$ is the transfer function of our system.

To find the necessary transfer functions:

$$y F w = W_j F x = W_j F v = W y f (G W_d d + B u_K)$$
$$u_w = W_n u_K$$
$$n = C F v + W_n n$$
$$= C F (G W_d d + B u_K) + W_n n$$

Combining all these gives,

$$
\begin{bmatrix}
u_w \\
\nu_{y_w}
\end{bmatrix} =
\begin{bmatrix}
0 & 0 & W_e \\
W_d J F W_d & 0 & W J F B \\
C F G W_d & W_e & C F B \end{bmatrix}
\begin{bmatrix}
d \\
n \\
u_k
\end{bmatrix}
$$

Note that the plant transfer function matrix, $F(s)$, is deduced from the suitably reformulated plant equations, $x(t) = A x(t) + B v(t)$

$$y(t) = I x(t)$$

where $v(t) = G d + B u_0$. Hence,

$$F(s) = (s I - A)^{-1}$$

The equivalent two-port diagram in the state space form is Fig. 2 for the close loop, and with more details in Fig. 3,

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$$F(s) = (s I - A)^{-1}$$

The equivalent two-port diagram in the state space form is Fig. 2 for the close loop, and with more details in Fig. 3,
while the closed loop transfer function $M_{cy}(s)$ is,

$$
M_{cy}(s) = P_{cy}(s) + P_{cy}(s)K(s)(I - P_{cy}(s)K(s))^{-1}P_{cy}(s)
$$

or,

$$
z = M_{cy}w = F_z(P, K)w
$$

Equation (9) is the well-known lower LFT for $M_{cy}$.

To express $P$ in state space form, the natural partitioning,

$$
P(s) = \begin{bmatrix}
    A & B_1 & B_2 \\
    C_1 & D_{11} & D_{12} \\
    C_2 & D_{21} & D_{22}
\end{bmatrix}
$$

is used (where the packed form has been used), while the corresponding form for the controller $K$ is [24], [25], [26]

$$
K(s) = \begin{bmatrix}
    A_k & B_k \\
    C_k & D_k
\end{bmatrix}
$$

Equation (11) defines the equations,

$$
\dot{x}(t) = Ax(t) + \begin{bmatrix} B_1 & B_2 \end{bmatrix} \begin{bmatrix} w(t) \\ u(t) \end{bmatrix}
$$

and,

$$
\dot{z}(t) = \begin{bmatrix} C_1 & C_2 \end{bmatrix} \begin{bmatrix} x(t) \\ y(t) \end{bmatrix} + \begin{bmatrix} D_{11} & D_{12} \end{bmatrix} \begin{bmatrix} w(t) \\ u(t) \end{bmatrix}
$$

To find the matrices involved, we break the feedback loop and use the relevant equations:

\begin{align*}
\dot{x}_f &= Ax_f + (Gd_u + Bu), \\
y_f &= C_x x_f
\end{align*}

\begin{align*}
\dot{x}_u &= A_x x_u + B_u u, \\
u_u &= C_x x_u + D_u u
\end{align*}

\begin{align*}
\dot{x}_w &= A_x x_w + B_w y_f, \\
y_w &= C_x x_w + D_x y_f
\end{align*}

\begin{align*}
x_u &= A_x x_u + B_u n, \\
n_u &= C_x x_u + D_u n
\end{align*}

\begin{align*}
x_w &= A_x x_w + B_w n, \\
n_w &= C_x x_w + D_x n
\end{align*}

\begin{align*}
y_u &= C_y y_u + Gd, \\
d_u &= C_y x_y + D_y d
\end{align*}

From (12), we use $d_u$, $n_u$ and $y_w$ and take our initial state in the form of (6, 7, 8),

$$
\begin{bmatrix}
    A_c & 0 & 0 & 0 & Gc_g \\
    0 & A_0 & 0 & 0 & 0 \\
    0 & 0 & A_0 & 0 & 0 \\
    0 & 0 & 0 & A_0 & 0 \\
    0 & 0 & 0 & 0 & A_0
\end{bmatrix}
$$

$$
\begin{bmatrix}
    GD_e & 0 \\
    0 & B_e & 0 \\
    0 & B_e & 0 \\
    0 & B_e & 0 \\
    0 & B_e & 0
\end{bmatrix}
$$

Therefore, the matrices are:

\begin{align*}
A_c &= BC_e A_c + D_e C_e, \\
B_c &= 0 & 0 & 0 & 0 \\
C_c &= D_e C_e + 0 & 0 & 0 & 0
\end{align*}

\begin{align*}
A_0 &= BC_e A_0 + D_e C_0, \\
B_0 &= 0 & 0 & 0 & 0 \\
C_0 &= D_e C_0
\end{align*}

Robust control allows dealing with uncertainty affecting a dynamical system and its environment. In this section, we assume that we have a mathematical model of the dynamical system with uncertainty. We restrict ourselves to linear systems: if the dynamical system we want to control has some nonlinear components (e.g., input saturation), they must be embedded in the uncertainty model. Similarly, we assume that the control system is relatively small scale (low number of states): higher-order dynamics are embedded in the uncertainty model [27], [28], [29], [30].
V. System Uncertainty

The main sources of uncertainty are:
- Nonlinearity and/or dynamic aspects of the system that are ignored at the modeling phase. The error introduced in modal analysis by using only a few significant eigenmodes leads to an uncertainty of the type discussed here.
- Incomplete knowledge of model values and parameters and/or natural fluctuation of those values during system operation.
- Influence of the system’s environment, in the form of disturbances.

Assume uncertainty in the M and K matrices of the form,
\[ K = K_0(I + k_p I_{2nx2n} \delta_K) \]  
\[ M = M_0(I + m_p I_{2nx2n} \delta_M) \]  
(17)
(18)

Also, since, \( D = 0.0005(K + M) \), an appropriate form for \( D \) is,
\[ D = 0.0005[K_k(I + k_p I_{2nx2n} \delta_k) + M_0(I + m_p I_{2nx2n} \delta_M)] = D_0 + 0.0005[K_k k_p I_{2nx2n} \delta_k M_0 m_p I_{2nx2n} \delta_M]. \]  
(19)

Alternatively, by adopting the well-known Rayleigh damping assumption,
\[ D = aK + \beta M \]  
(20)

\( D \) could be expressed similarly to \( K, M \), as,
\[ D = D_0(I + d_p I_{2nx2n} \delta_D) \]  
(21)

In this way we introduce uncertainty in the form of percentage variation in the relevant matrices. Uncertainty is most likely to arise from terms outside the main matrices (since length can be adequately measured).

Here it will be assumed,
\[ \| \Delta \|_{\infty} = \max_{1 \leq j < n} \| A_{nj} \Delta M \|_{\infty} \]  
(22)

hence \( m_p, k_p \) are used to scale the percentage value and the zero subscript denotes nominal values.

(22) is the norm of matrix \( A_{nj} \) through \( \| \Delta \|_{\infty} = \max_{1 \leq j < n} \sum_{i=1}^{n} |a_{ij}| \).

With these definitions (1) becomes,
\[ M_0(I + m_p I_{2nx2n} \delta_M) \ddot{q}(t) + K_0(I + k_p I_{2nx2n} \delta_K) q(t) + \left[ D_0 + 0.0005[K_k k_p I_{2nx2n} \delta_k + M_0 m_p I_{2nx2n} \delta_M] \right] \dot{q}(t) = f_m(t) + f_e(t) \]  
(23)

\[ \Rightarrow \dot{M}_0 \ddot{q}(t) + \dot{K}_0 \ddot{q}(t) + D_0 \dot{q}(t) + K_0 q(t) = \left[ M_0 m_p I_{2nx2n} \delta_M \right] \ddot{q}(t) + K_0 k_p(I + k_p I_{2nx2n} \delta_k) q(t) + 0.0005[K_k k_p I_{2nx2n} \delta_k + M_0 m_p I_{2nx2n} \delta_M] \dot{q}(t) + f_m(t) + f_e(t) \]  
(24)

\[ \Rightarrow \dot{M}_0 \ddot{q}(t) + D_0 \dot{q}(t) + K_0 q(t) = \dot{D} \ddot{q}(t) + f_m(t) + f_e(t) \]  
(25)

Where,
\[ q_u(t) = \begin{bmatrix} \dot{q}(t) \\ q(t) \end{bmatrix} \]  
(26)

\[ \dot{D} = \begin{bmatrix} M_0 m_p & K_0 k_p \\ 0 & D_0 \end{bmatrix} \]  
(27)

Writing in state space form, gives,
\[ \begin{bmatrix} \dot{x}(t) \\ \ddot{q}(t) \end{bmatrix} = \begin{bmatrix} 0_{2n>2n} & I_{2n>2n} \\ -M^{-1}K & -M^{-1}D \end{bmatrix} \begin{bmatrix} x(t) \\ q(t) \end{bmatrix} + \begin{bmatrix} 0_{2n>2n} \\ M^{-1} \end{bmatrix} \begin{bmatrix} f_m(t) \\ f_e(t) \end{bmatrix} \]

In this way we treat uncertainty in the original matrices as an extra uncertainty term.

VI. Inputs-Results

The beam that we use is discretized using one-dimensional finite elements with two degrees of freedom per node. This classical finite element procedure leads to the approximate discretized variational problem. For a finite element the discrete differential equations are obtained by substituting the discretized expressions into the first variation of the kinetic energy and strain energy [6, 15, 22] to evaluate the kinetic and strain energies. Integrating over spatial domains and using the Hamilton’s principle [9], we discretized the model with four finite elements. [18, 19, 21]

First for the mechanical force we use a sinusoidal load 10N on the side of the structure.

Where, \( f_m(t) = 10 \sin(t) \)

![Fig. 4. Displacement with and without control.](image-url)

In Fig. 4 we can see the displacements for the four nodes of the beam with and without control. We can see the displacements with H∞ control theory are almost zeros. This is very important in mechanical structures because we reject the disturbances even for the sinusoidal mechanical displacements.
In Fig. 5 we can see the rotations for the four nodes of the beam with and without control. The results with $H\infty$ control theory are almost zeros. The beam keeps in equilibrium even for sinusoidal disturbances.

Comparison with the open loop response for the same plant shows the good performance of the controller. Results are very good, and the beam remains in equilibrium even under stochastic excitation. Reduction of vibrations is observed, while piezoelectric add-ons produce voltage within their tolerance limits.

Furthermore, we control the structure with variations of the nominal values of the mass matrix $M$, stiffness matrix $K$ and matrices $A$ and $B$. We take into consideration nonlinearities and system dynamics neglected in modeling, incomplete knowledge of disturbances, environment influence in the form of disturbances, and unreliability of system sensor measurements. In Fig. 6 (a), (b) complete vibration reduction is achieved even for variations of matrices $A$ and $B$ up to 70%. Moreover, controller size contains so as to lower energy consumption and maintain piezoelectric materials within operation limits (500 volt). In Fig. 7(a), (b) complete vibration reduction is achieved even for variations of beam mass and stiffness up to 70%. The piezoelectric force is in their endurance limits, less 500 Volt.

Next a typical wind load Fig. 8 acting on the side of the structure. The wind load is a real life wind speed measurements in relevance with time. We transform the wind speed in wind pressure with,

$$f_m(t) = \frac{1}{2} \rho C_u V^2(t)$$

where $V$=velocity, $\rho$=density and $C_u$=1.2.

Moreover, in all simulations, random noise has been introduced to measurements at system output locations within a probability interval of $\pm 1\%$. Due to small displacements of system nodal points, noise amplitude is taken to be small, of the order of $5 \times 10^{-5}$.

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and maintain piezoelectric materials within operation limits (500 volt), Fig. 10(c).

VII. CONCLUSION

The authors are encouraged by the results obtained and it seems that the methodology developed can be extended to other practical systems, for example, smart plates and space truss structures.

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