An integrated gain scheduled control design for an Electrostatic micro-Actuator with aerodynamic effects

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Abstract—In this article an integrated approach regarding the modeling and control design aspects of an electrostatic microactuator (EmA) with aerodynamic effects is presented. The modeling analysis of the squeezed film damping effect is investigated in the case of an EmA composed by a set of two plates. The bottom plate is clamped to the ground, while the moving plate is driven by an electrically induced force which is opposed by the force exerted by a spring element. The damping coefficient that is caused because of the thin film of air, is approximated by a frequency independent nonlinear static term. Under this assumption, the nonlinear model of the EmA is linearized at various operating points, and the feedforward compensator provides the nominal voltage. Subsequently a gain scheduled $H_{\infty}$-controller is used to tune the controller-parameters depending on the EmA’s operating conditions. The controller is designed at various operating points based on the distance between its plates. Simulation results investigate the efficacy of the suggested modeling and control techniques.

I. INTRODUCTION

In the micro-scale squeezed film damping plays an important effect on the performance of MEMS structures [1], [2]. The development of micromachined technologies has resulted in the use of many planar microstructures in microelectromechanical systems (MEMS) such as micro- accelerometers, gyroscopes and RF-switches [3], [4], microphones, micromirrors etc. Squeezed film damping is a term used to describe the primary fluid-structure interactions that impacts the performance of MEMS devices [5]. Squeezed film damping occurs when a plate moves in close proximity to another solid surface, in effect alternately stretching and squeezing any fluid that may be present in the space between the moving plate and the solid surface. This fluid can act as a spring and/or a dashpot, having a significant effect on the dynamics of the moving plate. Therefore, the effects of the surrounding air and most notably the damping force which can be neglected in structures of conventional dimensions, play a critical role with micro-structures with diminutive size.

Silicon microstructures (sensors and actuators) that make use of the capacitive measurement principles [6], or electrostatic driving forces [7], are characterized by very small gaps between their moving surfaces [3] and so the dynamic behavior of movable parts in these electrostatic actuators is largely affected by the air’s presence (i.e. low vacuum conditions for micro- accelerometers [8], ultra thin gas film in magnetic/disk interfaces [9] and tilting micro-mirrors in DLP type projectors [10], [11]). The understanding of the squeezed film damping mechanism in such EmA devices [12] is necessary in order to optimize the controller designs.

The inclusion of the squeezed film damping effects increases the complexity of the dynamics of the EmA plant [13], [14], [15], [16]. The dynamic equations describing both the damper the spring coefficient of the force caused by the parallel plate damper, are nonlinear in terms of the distance between the plates, the voltage applied, and the frequency of the movement of the upper plate. This nonlinearity especially in terms of the frequency, makes it difficult to linearize the system and design the desired controller. In order to avoid the derivation of complicated systems, the coefficient of the force caused by the parallel plate damper is approximated by a nonlinear function frequency independent. The approximation is valid for a range of frequencies for which the system has the same behavior. In this way, simple linear controllers can be designed that are based on linearized models of the system [11].

In the present article a Gain Scheduled $H_{\infty}$-controller [17], [18] is designed for a class of Linear Parameter Varying (LPV) plants characterizing the EmA. The main idea is to separate the control design process into two steps. Firstly the local linear controllers are designed based on the linearizations of the nonlinear system at several operating points [19]. In the sequel, a global controller for the nonlinear plant is obtained by scheduling the gains of the local operating points’ design. The linearized plants’ state space matrices are assumed to depend on a vector of spatial varying parameters. The measured parameters, are fed to the controller to optimize the performance and the robustness of the closed loop system. The resulting controller is inherently “gain scheduled” along the parameter space. The synthesis problem of the controller is fulfilled with the use of Linear Matrix Inequalities (LMIs).

In the rest of this article the modeling of the EmA with squeezed film damping effects is presented in Section II. In Section III the design of the Gain scheduled $H_{\infty}$ control scheme is presented. Simulation results that prove the efficacy of the proposed control architecture are presented in Section IV, while the conclusions are drawn in the last Section V.

II. MODELING OF EMAS WITH SQUEEZED AIR DAMPING

The EmA from a structural point of view corresponds to a micro–capacitor whose one plate is attached to the ground while its other moving plate is floating on the air [20] with the aid of an additional external spring. Figure 1 presents the structure of the EmA. The dynamic nonlinear governing equation of the system [1] is:

$$m\ddot{\eta} + F_d + k\eta = \frac{\varepsilon \ell^2 U^2}{2(\eta_{\text{max}} - \eta)^2} = F_{\text{el}}$$

where $\eta$ is the displacement of the plates from the relaxed position, $m$ is the plate’s mass, $k$ is the spring’s stiffness, $\ell$
is the length of the square plate, $U$ is the applied voltage between the capacitor’s plates, $\eta_{\text{max}}$ is the distance of the plates when the spring is relaxed, $\varepsilon$ is the dielectric constant of the air, $F_d$ is the force caused by the parallel plate damper and $F_{el}$ is the electrically-induced force [21], [22] which is generated by the application of a voltage $U$ between the capacitor’s plates as shown in Figure 2.

![Fig. 1. Electrostatic micro-Actuator structure](image)

**A. Squeezed Film Damping Effect**

The behavior of the gas between the plates is in general governed by both the viscous and inertial effect within the fluid. However, for the very small dimensions encountered in electrostatic devices, the inertial effect is often negligible. In such a case, the behavior of the fluid is governed by the Reynolds equation, a single expression which relates pressure, density and surface velocity for the specific geometry of a bounded film [5], [13], [23], [24].

Under the assumption of isothermal conditions, Blech [25] has derived solutions for the pressure between the gap of two oscillating rectangular plates. The pressure has two components, one in phase with the drive, which represents the spring-like behavior of the gas, and one in phase with the velocity which represents the damping behavior [13] of the air. The integrals of these pressures over the plates provides the expressions for the air spring and damping contributions. For square plates [13], [25] the coefficient of the viscous damping force due to the squeezed film air damping is [26], [27], [28] (under the assumption of a sinusoidal motion of the upper plate with frequency $\omega$):

$$b_1(\eta, \omega) = \frac{64\varepsilon P_a \ell^2}{\omega \pi^6 (\eta_{\text{max}} - \eta)} \sum_{m,n \text{ odd}}(mn)^2 [ (m^2 + n^2)^2 + \sigma^2 / \pi^4]$$

(2)

and the coefficient of elastic damping force is:

$$k_1(\eta, \omega) = \frac{64\varepsilon^2 P_a \ell^2}{\pi^8 (\eta_{\text{max}} - \eta)} \sum_{m,n \text{ odd}}(mn)^2 [ (m^2 + n^2)^2 + \sigma^2 / \pi^4]$$

(3)

where $\sigma = \frac{12\mu u^2 \omega}{\pi P_a (\eta_{\text{max}} - \eta)^2}$ is the dimensionless squeeze number, $P_a$ is the ambient pressure, and $\mu$ is the (air) viscosity coefficient. It should be noticed that for typical gaps encountered in EmAs ($0.1 \mu m \leq \eta_{\text{max}} \leq 40 \mu m$) over an operating frequency range of less than $\omega_{\text{max}} = 10^3 \text{Hz}$, the coefficient $k_1$ increases with the frequency and is significantly smaller than $k$, $(k_1 \ll k, \forall \omega \in [0, \omega_{\text{max}}])$. The force caused by the parallel plate damper $F_d$ presented in Equation (1) is equal to [29]:

$$F_d = b_1(\eta, \omega) \dot{\eta} + k_1(\eta, \omega) \eta.$$  

(4)

**B. Linearized Equations of Motion**

The nonlinear equation of motion for (1) can be rewritten according to (4):

$$m \ddot{\eta} + b_1(\eta, \omega) \dot{\eta} + (k + k_1(\eta, \omega)) \eta = \frac{\varepsilon \ell^2 U^2}{2(\eta_{\text{max}} - \eta)^2}.$$  

(5)

Equation (5) is a nonlinear equation due to the presence of the terms $\omega$, $\eta$, $U$. If $\omega \in [0, 10^3] \text{Hz}$ there is no significant change in the value of $b_1$ in this frequency range. In respect to this observation, the value of the damping factor may be approximated nonlinearly, in terms of $\eta$. The function describing the approximated quantity which is frequency independent is equal to:

$$b_d(\eta) = p_1 \eta^2 + p_2 \eta + p_3 \approx b_1(\eta, \omega), \forall \omega \in [0, \omega_{\text{max}}].$$

(6)

The final nonlinear system with the approximated term is equal to:

$$m \ddot{\eta} + b_d \dot{\eta} + k \eta = \frac{\varepsilon \ell^2 U^2}{2(\eta_{\text{max}} - \eta)^2}.$$  

(7)

In order to design the desired controller, the linearization of the system is demanded. A system’s approximation can be obtained if a certain operating point $\bar{\eta}_i$, $\eta_i^o$, $\omega_i^o$ is chosen in order to achieve a linearized system. All possible "equilibria"-points $\eta_i^o$, $i = 1, \ldots, M$ depend on the applied nominal voltage $U_i^o$. Equation (5) for $\dot{\eta}_i^o = 0$, $\eta_i^o$, $U_i^o$ and for $\omega_i^o \in [0, \omega_{\text{max}}]$, $k_i \approx 0$ (since $k_i \ll k$) yields:

$$k \eta_i^o = \frac{\varepsilon \ell^2 (U_i^o)^2}{2(\eta_{\text{max}} - \eta_i^o)^2}, \text{or}$$

$$U_i^o = \pm \left[ \frac{2k_i \eta_i^o (\eta_{\text{max}} - \eta_i^o)}{\varepsilon \ell^2} \right]^{1/2}.$$  

(8)

(9)

This nominal $U_i^o$—voltage must be applied if the capacitor’s plate is to be maintained at a certain distance $\eta_i^o$ from its un–stretched position.
The system described in Equation (7) is linearized with respect to the parameters $\eta$, $U$ and the approximated linearized equations of motion around the equilibrium points $(U_0^\circ, \eta_0^\circ)$ and $\dot{\eta}_0^\circ = 0$ can be found using standard perturbation theory for the variables $U$ and $\eta$, where $U = U_0^\circ + \delta u$ and $\eta = \eta_0^\circ + \delta \eta$. The linearized equation of motion for the system in (5) is:

$$m\ddot{\eta}_i + \left(b_i|_{\eta = \eta_0^\circ}\right) \delta \eta_i + k_i \delta \eta_i = \frac{\varepsilon \ell^2(U_0^\circ)^2}{(\eta_{\max} - \eta_0^\circ)^3} \delta \eta_i + \frac{\varepsilon \ell^2(U_0^\circ)^2}{(\eta_{\max} - \eta_0^\circ)^2} \delta u$$

(10)

After the substitution of:

$$k_i = k - \frac{\varepsilon \ell^2(U_0^\circ)^2}{(\eta_{\max} - \eta_0^\circ)^2},$$

$$b_i = b_i|_{\eta = \eta_0^\circ},$$

$$\beta_i = \frac{\varepsilon \ell^2(U_0^\circ)^2}{(\eta_{\max} - \eta_0^\circ)^2}, \quad i = 1, \ldots, M$$

(11)

Equation (10) is transformed to:

$$m\ddot{\eta}_i + b_i \dot{\eta}_i + k_i \eta_i + \beta_i \delta \eta_i = \beta_i \delta u.$$  

(12)

Equation (12) is valid only for a specific frequency range $\omega \in (0, \omega_{\max})$. In that case the state space description of the system is equal to:

$$\begin{bmatrix} \delta \eta_i \\ \delta \dot{\eta}_i \end{bmatrix} = \begin{bmatrix} -k_i(\eta_i) & -b_i(\eta_i) \\ 0 & \beta_i(\eta_i) \end{bmatrix} \begin{bmatrix} \delta \eta_i \\ \delta \dot{\eta}_i \end{bmatrix} + \begin{bmatrix} 0 \\ \beta_i(\eta_i) \end{bmatrix} \delta u$$

$$= A_i(\eta_i) \begin{bmatrix} \delta \eta_i \\ \delta \dot{\eta}_i \end{bmatrix} + B_i(\eta_i) \delta u, \quad i = 1, \ldots, M$$

(13)

$$\delta \eta_i = |1 0| \begin{bmatrix} \delta \eta_i \\ \delta \dot{\eta}_i \end{bmatrix} = C \begin{bmatrix} \delta \eta_i \\ \delta \dot{\eta}_i \end{bmatrix}.$$  

III. GAIN SCHEDULED $H_\infty$ CONTROLLER DESIGN

The feedback term is a gain scheduled $H_\infty$ controller, which consists of an LTI controller for each one of the aforementioned subsystems. These controllers switch among them when the operating conditions change. The change of the system matrices depends on the variation of the operating point $\eta_i$. The designed controller is applied to the nonlinear system of the EmA in order to test its efficacy. The controller’s parameters are tuned with the use of LMIs [30], [31].

In order to appropriately weight selected frequency bounds under consideration, the system’s input and output are filtered by filters of transfer functions $W_1(s)$, $W_2(s)$. For the EmA, the low frequency spectrum is of primary importance and low pass filters $W_1(s) = \prod_{i-1}^w \frac{w_{1,i}}{s + w_{1,i}}$, $W_2(s) = \prod_{i=1}^f \frac{w_{2,j}}{s + w_{2,j}}$ are used throughout this frequency-shaping procedure. In the sequel, driven by our application (micropositioning) first order $(q,f=1)$ low pass filters are used. The linearized system’s description using the augmented state vector $\delta \eta_i^T = [\delta \eta_i, \delta \dot{\eta}_i, \delta \eta_f, \delta \eta_q]$ together with the other filter is equal to:

$$\begin{bmatrix} \delta \eta_i \\ \delta \dot{\eta}_i \\ \eta_f \\ \eta_q \\ \delta \eta_i \end{bmatrix} = \begin{bmatrix} A_i^a & B_i^a & B_i^b \\ C_i^a & D_{11} & D_{12} \\ C_i^a & D_{21} & D_{22} \end{bmatrix} \begin{bmatrix} \delta \eta_i \\ \delta \dot{\eta}_i \\ \eta_f \\ \eta_q \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \eta_i \end{bmatrix}$$

(14)

where,

$$A_i^a = \begin{bmatrix} 0 & \frac{v_i(\eta_i)}{m} & 0 \\ \frac{v_i(\eta_i)}{m} & 0 & -w_{2,1} \end{bmatrix},$$

$$B_i^a = \begin{bmatrix} \frac{v_i(\eta_i)}{m} \\ \frac{v_i(\eta_i)}{m} \end{bmatrix},$$

$$C_i^a = \begin{bmatrix} -w_{1,1}c_{11} & 0 \end{bmatrix},$$

$$D_{11} = \begin{bmatrix} 0 & 0 \end{bmatrix},$$

$$D_{21} = \begin{bmatrix} w_{1,1} & 0 \end{bmatrix},$$

$$D_{22} = \begin{bmatrix} w_{1,1} & 0 \end{bmatrix}.$$  

(15)

For the a priori selected operating points $\eta_i^\circ$, the system’s description is within the convex hull of the matrices:

$$\begin{bmatrix} A_i^a(\eta_i) & B_i^a(\eta_i) \\ C_i & D_{11} \\ 0 & 0 \end{bmatrix} \in \text{Co} \left\{ \begin{bmatrix} A_i^a(\eta_i^\circ) & B_i^a(\eta_i^\circ) \\ C_i^\circ & D_{11} \end{bmatrix} : i = 1, \ldots, M \right\}.$$  

(16)

Under the assumption of direct measurements of the state vector $\delta \eta_i^\circ$, the controller adjusts in a gain scheduled approach, its parameters based on the neighborhood of the selected operating point $\eta_i^\circ$.

In this scheme, the plate’s gap space $\eta \in [\eta_{\min}, \eta_{\max}]$ is equidistantly divided into n-segments, defined by the selected operating points:

$$\eta_i = \eta_{\min} + (i - 1) \frac{\eta_{\max} - \eta_{\min}}{n} \equiv \eta_{\min} + (i - 1) \Delta, \quad i = 1, \ldots, n.$$  

(17)

When $\eta_i \in [\eta_i - \frac{\Delta}{2}, \eta_i + \frac{\Delta}{2}]$ the $\delta \eta_i^\circ$ are computed with respect to the $i$th operating point. The gain-scheduled controller provides a dynamic feedback of the form

$$\delta u = K(s, \eta_i) \delta \eta_i^\circ,$$  

(18)

where with a slight abuse of notation, we imply that the $K(s)$-transfer function depends on the operating point $\eta_i$, noted as $K(s, \eta_i)$.

The architecture of the control scheme is presented in Figure 3 [32].

The goal is to create a controller with a form:

$$\dot{x}_c = A_c(\eta_i)x_c + B_c(\eta_i)\delta \eta_i^\circ$$

$$\delta u = C_c(\eta_i)x_c + D_c(\eta_i)\delta \eta_i^\circ$$

that guarantees a quadratic $H_\infty$ performance less than $\gamma$ for the closed loop system, where $x_c \in \mathbb{R}^4$ is the state vector of the controller. With the notation

$$\Omega(\eta_i) := \begin{bmatrix} A_c(\eta_i) & B_c(\eta_i) \\ C_c(\eta_i) & D_c(\eta_i) \end{bmatrix}$$

(19)

the state space matrices of the closed loop system are:
The pairs \((A_i^a, B_i^a)\) and \((A_i^c, C_i^c)\), are quadratically stabilizable and quadratically detectable respectively.

Under the posed assumptions of the LPV-plant, there exists an LPV-controller \([33]\) guaranteeing Quadratic \(H_{\infty}\) performance \(\leq \gamma\) for all state vector trajectories \(\eta(t) \in [\eta_{\min}, \eta_{\max}] = C_0 \{ \eta_i^0, i = 1, \ldots, M \}\) if and only if there exist two symmetric matrices \(R, S\) satisfying the system of LMIs:

\[
\begin{bmatrix}
A^a \eta(t) + B^a \eta(t) & R \eta(t) \\
C_i^a \eta(t) & D_i^a \\
\end{bmatrix} < 0 \tag{21}
\]

where

\[
\hat{N}_R = \begin{bmatrix}
N_R & 0 \\
0 & I \\
\end{bmatrix}, \quad \hat{N}_S = \begin{bmatrix}
N_S & 0 \\
0 & I \\
\end{bmatrix}
\]

and \(N_R\) and \(N_S\) are the orthonormal bases of the null spaces of \([B_i^a]^{T}, D_{12}^a\) and \([C_i^a, D_{21}^a]\) respectively. Moreover there exist \(k\)-th order LPV controllers solving the same LMI problem if and only if \(R, S\) would also satisfy the rank constraint

\[
\text{rank}(I - RS) \leq k. \tag{25}
\]

After the computation of any feasible solution of \(R, S\) matrices the invertible matrices \(\Theta\) and \(\Psi\) can be computed via a SVD as:

\[
\Theta \Psi^T = I - RS. \tag{26}
\]

Having computed \(\Theta\) and \(\Psi\) the matrix \(X_{cl}\) is formed as:

\[
X_{cl} = \begin{bmatrix}
I_{2 \times 2} & S \\
0_{2 \times 2} & \Psi^T \\
\end{bmatrix} \begin{bmatrix}
R & I_{2 \times 2} \\
\Theta^T & 0_{2 \times 2} \\
\end{bmatrix}^{-1}. \tag{27}
\]

Given the matrix \(X_{cl}\) a possible choice of vertex controller:

\[
\Omega_i = \begin{bmatrix}
A_i(\eta_i) & B_i(\eta_i) \\
C_i(\eta_i) & D_i(\eta_i) \\
\end{bmatrix}
\]

is any feasible solution of the LMI problem:

\[
\begin{bmatrix}
A_{cl}(\eta_i)X_{cl} + X_{cl}A_{cl}(\eta_i) & X_{cl}B_{cl}(\eta_i) \\
B_{cl}(\eta_i) & C_{cl}(\eta_i) \\
\end{bmatrix} < 0. \tag{29}
\]

IV. SIMULATION RESULTS

Simulation studies were carried on an EmA’s non–linear model. The parameters of the system unless otherwise stated are equal to those presented in the following Table.

<table>
<thead>
<tr>
<th>parameter (Unit)</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A) ((m^2))</td>
<td>Area of the plates</td>
<td>(10 \times 10^{-6})</td>
</tr>
<tr>
<td>(l) (m)</td>
<td>Plate Length</td>
<td>(100 \times 10^{-6})</td>
</tr>
<tr>
<td>(\mu) (kg (m/\text{sec}^2))</td>
<td>Viscosity Coefficient</td>
<td>(18.5 \times 10^{-6})</td>
</tr>
<tr>
<td>(\rho) (kg/m(^3))</td>
<td>Density</td>
<td>1.155</td>
</tr>
<tr>
<td>(\epsilon) ((\text{cm}^2/\text{N} \cdot \text{m}))</td>
<td>Dielectric constant of the air</td>
<td>(8.85 \times 10^{-12})</td>
</tr>
<tr>
<td>(P_0) (N/m(^2))</td>
<td>Ambient Pressure</td>
<td>(10^5)</td>
</tr>
<tr>
<td>(k) (N/m)</td>
<td>Static stiffness of the spring</td>
<td>0.816</td>
</tr>
<tr>
<td>(\eta_{\max}) (m)</td>
<td>Maximum gap between the plates</td>
<td>(0.2 \times 10^{-6})</td>
</tr>
</tbody>
</table>

Figure 4 shows the magnitude frequency responses for the “linearized” subsystems that are derived from Equation (5) if the above system is linearized with respect to the parameter \(\eta\), and if \(\omega = \omega^c\). The approximated linearized equations of motion around the equilibria points \((U^c_j, \omega^c, \eta^c_i, \eta^c_i = 0)\) can be found using standard perturbation theory for the variables \(U\) and \(\eta\) where \(U = U^c_j + \delta U\) and \(\eta = \eta^c_i + \delta \eta_i\). The linearized equation of motion for the system in (5) is:

\[
m\delta \eta_i + \left( \begin{array}{c}
-\eta_i - \omega_i \omega \delta \eta_i + \frac{k}{\eta_i} \delta \eta_i + \frac{k}{\eta_i} \delta \eta_i + \frac{k}{\eta_i} \delta \eta_i + \frac{k}{\eta_i} \delta \eta_i
\end{array} \right) \delta \eta_i
\]

\[
+ \left( \begin{array}{c}
\frac{k}{\eta_i} \delta \eta_i + \frac{k}{\eta_i} \delta \eta_i
\end{array} \right) \delta \eta_i
\]

\[
= \frac{\varepsilon_0^2 (U_j^c)^2}{2 \left( \eta_{\max} - \eta_i \right)^2} + \frac{\varepsilon_0^2 (U_j^c)^2}{2 \left( \eta_{\max} - \eta_i \right)^2} \delta \eta_i + \frac{\varepsilon_0^2 (U_j^c)^2}{2 \left( \eta_{\max} - \eta_i \right)^2} \delta \eta_i + \frac{\varepsilon_0^2 (U_j^c)^2}{2 \left( \eta_{\max} - \eta_i \right)^2} \delta \eta_i
\]

The responses have been obtained from the above Equation for \(\omega^c \in [0.1, 10^6]\) rad/sec. As it is observed from the plots, the system’s frequency representations are very close considering critical issue points (i.e. system’s dc-gain, resonant frequency, stability issues etc) and also the
systems’ behavior remains the same for a range of frequency $\omega^\circ \in [0, 10^3]$ rad/sec. In this case the approximation of the damping factor with the nonlinear function independent from frequency described in Equation (6) is valid for $\omega \in [0, 10]$ rad/sec.

This system is known to have a bifurcation point at $\eta^\circ_0 = \frac{0.35}{0.466} \mu m$. These are the points where the behavior of the system changes from stable to unstable and vice versa. The effects of thin film damping to the bifurcation points have been presented by the authors in [34]. Since the operating regime is below the well-known bifurcation point ($\eta^\circ_0 = 0.1 \mu m$), the system’s output is expected and verified from the Nyquist plots of these systems shown in Figure 5.

The proposed control scheme was applied in multiple simulation test cases in order to test its efficacy. For simulation purposes, and while the EmA is at rest at $0.1 \mu m$ ($\eta^\circ(0) = 0.1 \mu m$), the EmA’s plate is asked to move in a step-fashion to $0.35 \mu m$. Figure 7 presents the system’s output for the applied reference signal. The measurements are corrupted with noise resulting in a SNR=20dB. Simulation proves the efficacy of the suggested control design.

Another case for a sinusoidal input is also presented in Figure 8 in order to prove the controller’s effectiveness in this case. The measurements are corrupted with noise resulting in a SNR=20dB. From the presented scheme it is obvious that the controller is able to stabilize a system under a step or sinusoidal input. In the case of the sinusoidal input the response seems to be more noisy which is expected as more switchings are caused in the controllers parameters in a shorter term of time than in the case of a step signal.
In this article an integrated design approach and a control technique is presented for a system of an EmA with squeezed film damping effect. The damping factor of the system is approximated in a manner that it is frequency independent regarding the frequency of the movement of the upper plate. The designed controller is a Gain scheduled $H_\infty$-controller tuned via the theory of LMIs. The controller is designed at various operating points and smoothly changes its values as the upper plate of the EmA is moving. The overall control scheme is applied on the EmA's non linear model in order to test its efficacy.

VI. ACKNOWLEDGMENTS

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Fig. 8. EmA’s System’s Output for a sinusoidal input

V. CONCLUSION