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DYNAMIC MODELLING AND EXPERIMENTAL IDENTIFICATION OF MICRO/MACRO HYBRID MANIPULATOR SYSTEM

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Abstract. *The objective of this work is to describe the design and the implementation of an experiment to study the dynamics, the experimental identification and the active vibration control of a Flexible Structure Mounted Manipulator System (FSMS). The system consists of a three degree of freedom cylindrical manipulator system with a flexible link on its tip. A two-degree of freedom polar rigid manipulator is mounted on the flexible macro manipulator. The dynamic modeling and experimental modal analysis identification in the frequency domain are being applied to design active digital control strategies for the micro-manipulator system to damp the mechanical vibrations of the flexible structure on the tip of the macro-manipulator system.*

Keywords: *Robotics, Modelling, Flexible Structure, Micro/Macro Manipulator*

1. INTRODUCTION

Investigations on dynamic modeling and control of flexible structure mounted manipulator systems (FSMS) have attracted a great deal of interest due to special applications in the control of large flexible manipulator systems, with large work-space and simultaneous requirements of great precision at its end-effector positioning (Lew, & Trudnowski, 1996). In these applications the active vibration control of the macro flexible manipulator requires considerable control energy, and a possible solution involves the concept of a micro rigid manipulator mounted on the macro-flexible system. This concept is generally referred as a micro/macro hybrid manipulator system (Nenchev, Yoshida & Uchiyama, 1996).

The FSMS system was assembled at the Mechatronics Laboratory of the University of Taubaté (UNITAU), and it consists of a micro manipulator system with two polar rotational joints, mounted on a flexible macro-manipulator system. The flexible structure is a cantilever beam mounted on the tip of a three degree of freedom cylindrical (r, θ, z), rigid manipulator system. A schematic view of the system is shown in Figure 1. The flexible beam has a low stiffness in the horizontal direction, in the same plane of motion of the polar micro manipulator. In this work we describe the analytical modeling together with model validation studies carried out through experimental modal testing and parametric system identification studies in the frequency domain. A nonlinear control law is proposed and tested to actively damp the mechanical vibrations of the flexible link.

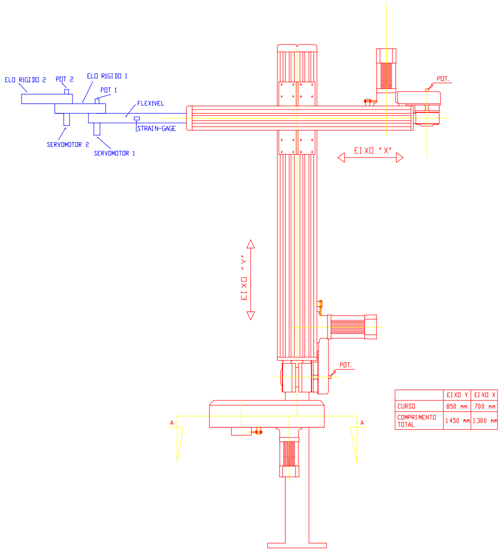

Figure1 - Schematic view of the FSMS

2. DYNAMIC MODELING OF THE FSMS

The analytical model of the system is derived from Hamilton’s extended principle and the assumed mode method. In this work we have considered only a one-degree of freedom micro manipulator mounted on the tip of a flexible Euler-Bernouille beam. The motion of the cylindrical macro-manipulator is used only to position the FSMS on its workspace, and excite the vibracional degrees of freedom of this system. A generalized flexible homogeneous transformation and symbolic manipulation are applied to derive the equations of motion, (Junkins, 1993) and (Soares, Góes & Gadelha,1996). The generalized coordinates are the angular displacement of the micro manipulator joint, $q=q_r(t)$, and the modal coordinate, $q_f(t)$ associated with the elastic displacement of the beam, $w(x,t)$.The generalized force include the joint torque of the micro manipulator system, $u(t)$.
Considering that the macro-manipulator have assumed a known position in its work space, the Lagrangian of the FSMS system, is written as, $L = T_{ap} + T_f - V_f$,

$$T_f = \frac{1}{2}\int_0^{L_f} \rho \left(\frac{\partial w(x,t)}{\partial t}\right)^2 dx \qquad\qquad V_f = \frac{1}{2}\int_0^{L_f} EI\left(\frac{\partial^2 w(x,t)}{\partial x^2}\right)^2 dx \tag{1}$$

and T_{ap} is the kinetic energy of the micro manipulator system, with one degree of freedom,

$$T_{ap} = \frac{1}{2} m_r v_r^2 = \frac{1}{2} m_r \dot{w}^2 + \frac{1}{2} m_r v_t^2 + m_r \dot{w}\, v_t \cos q_r \tag{2}$$

where $v_t = {}^{L_r}\!/_2\, \dot{q}_r$ e $\dot{w} = \dot{w}(L_f, t)$, L_f is the length of the flexible link and L_r is the rigid micro manipulator link length.

Applying the assumed mode method, we write the elastic displacement as, $w(x,t)=\psi(x).q_f(t)$, where the admissible function is given as (Junkins, 1993).

$$\psi_i(x) = 1 - \cos(\frac{i\pi x}{L_f}) + 0{,}5(-1)^{i+1}(\frac{i\pi x}{L_f})^2 \tag{3}$$

which upon substitution on equations (1) and (2) above, gives (Lew & Trudnowsky, 1996) :

$$T_f = \frac{1}{2}\int_0^{L_f} \rho \left(\frac{\partial \psi(x) q_f(t)}{\partial t}\right)^2 dx = \frac{1}{2}\int_0^{L_f} \rho \psi^2(x) dx \quad \dot{q}_f^2(t) \tag{4}$$

$$V_f = \frac{1}{2}\int_0^{L_f} EI \left(\frac{\partial^2 \psi(x) q_f(t)}{\partial x^2}\right)^2 dx = \frac{1}{2}\int_0^{L_f} EI \psi''^2(x) dx \quad q_f^2(t) \tag{5}$$

$$T_{ap} = \frac{1}{2} m_r \psi^2(L_f) \dot{q}_f^2(t) + \frac{1}{2} m_r \frac{L_r^2}{4} \dot{q}_r^2 + m_r \psi(L_f) \dot{q}_f(t) \frac{L_r}{2} \dot{q}_r \cos q_r \tag{6}$$

The virtual work of non-conservative forces is written as,

$$\delta W = u \;\delta q_r + u \frac{\partial \psi(L_f)}{\partial x} \delta q_f \tag{7}$$

and applying Lagrange's equations with only one assumed mode, the equations of motion can be written in a matrix form as below,

$$\begin{bmatrix} M_{rr} & M_{rf} \\ M_{fr} & M_{ff} \end{bmatrix} \begin{Bmatrix} \ddot{q}_r \\ \ddot{q}_f \end{Bmatrix} + \begin{bmatrix} 0 & 0 \\ N_{fr} & 0 \end{bmatrix} \begin{Bmatrix} \dot{q}_r \\ \dot{q}_f \end{Bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & K_{ff} \end{bmatrix} \begin{Bmatrix} q_r \\ q_f \end{Bmatrix} = \begin{Bmatrix} u \\ bu \end{Bmatrix} \tag{8}$$

where :

$$M_{rr} = m_r \frac{L_r^2}{4} \tag{9.a}$$

$$M_{ff} = \int_0^{L_f} \rho \psi^2(x) dx + m_r \psi^2(L_f) \tag{9.b}$$

$$M_{rf} = M_{fr} = m_r \psi(L_f) \frac{L_r}{2} \cos q_r \tag{9.c}$$

$$N_{fr} = -m_r \psi(L_f) \frac{L_r}{2} \dot{q}_r \operatorname{sen} q_r \tag{9.d}$$

$$K_{rr} = \int_0^{L_f} EI \psi''^2(x) dx \tag{9.e}$$

$$b = \psi'(L_f) \tag{9.f}$$

The off-diagonal terms in the mass matrix, M_{fr}, represent the inertial coupling between the micro manipulator motion, q_r, and the dynamics of the supporting flexible structure, q_f.

The unforced, undamped motion of the flexible structure, is excited by the dynamics of the micro manipulator system, according to the following equation:

$$M_{ff}\,\ddot{q}_f + K_{ff}\,q_f = -\left(M_{fr}\,\ddot{q}_r + N_{fr}\,\dot{q}_r\right) = -\frac{\partial}{\partial t}(M_{fr}\,\dot{q}_r) \tag{10}$$

The coupling force is expressed as the time derivative of the *coupling momentum* of the FSMS, $\Im = \mathrm{M}_{\mathrm{fr}}\dot{\mathrm{q}}_{\mathrm{r}}$. Zero reaction on the flexible structure can be achieved with a joint velocity command given by, $\mathrm{d}\phi/\mathrm{dt} = \mathrm{M}_{\mathrm{fr}}^{-1}\Im + \mathrm{cte}$, or a joint acceleration, $\mathrm{d}^2\phi/\mathrm{dt}^2 = -\mathrm{M}_{\mathrm{fr}}^{-1}\mathrm{N}_{\mathrm{fr}}\mathrm{q}_{\mathrm{r}}$ (Nenchev, 1996).

In the following sections we describe the experimental procedures used to validate the FSMS dynamics, through non-parametric identification in the frequency domain. The micro manipulator is driven by a velocity servo loop and both a piezoelectric accelerometer and a full strain gage bridge are used to estimate the modal coordinate, q_f associated with the elastic deformation of the beam. The identified model is used to design a non-linear control strategy to actively damp the vibrations induced in the flexible structure.

3. MODEL IDENTIFICATION AND CONTROL OF THE FSMS

The theoretical transfer functions, between the sensors output (potentiometer, accelerometer and extensometric sensor) and the velocity servocontrol reference signal, are determined by neglecting the non-linear terms in equation (8). To illustrate this procedure let us consider the open-loop dynamics of the flexible structure, obtained by eliminating the joint acceleration from equation (9) :

$$\left(\frac{\mathrm{M}_{\mathrm{ff}}\mathrm{M}_{\mathrm{rr}}}{\mathrm{M}_{\mathrm{rr}}} - \frac{\mathrm{M}_{\mathrm{rf}}\mathrm{M}_{\mathrm{fr}}}{\mathrm{M}_{\mathrm{rr}}}\right)\ddot{\mathrm{q}}_{\mathrm{f}} + \frac{\mathrm{K}_{\mathrm{ff}}\mathrm{M}_{\mathrm{rr}}}{\mathrm{M}_{\mathrm{rr}}}\mathrm{q}_{\mathrm{f}} = \left(\frac{\mathrm{bM}_{\mathrm{rr}}}{\mathrm{M}_{\mathrm{rr}}} - \frac{\mathrm{M}_{\mathrm{fr}}}{\mathrm{M}_{\mathrm{rr}}}\right)\mathrm{u(t)} \tag{11}$$

The output signal is derived from a strain gage bridge signal, which is proportional to $\varepsilon(L_f)$, the strain at the position L_f of the gage. This signal is related to the elastic displacement of the beam by the following equation :

$$\varepsilon(\mathrm{L_f}) = (\frac{\mathrm{eb}}{2}).\frac{\partial^2 \mathrm{w}_{\mathrm{x,t}}}{\partial \mathrm{x}^2}\bigg|_{\mathrm{x=L_f}} = \mathrm{K}_{\mathrm{form}}.\mathrm{q_f(t)} \tag{12}$$

Substituting the above equations (11) and (12) in the closed loop servocontrol, we calculate the following analytical transfer function,

$$\frac{Q_f}{V_a} = \frac{sK_vK_f(a_2s^2 + a_1s + a_0)}{[s(a_2s^2 + a_1s + a_01) + K_vK_b(b_2s^2 + b_1s + b_0)]\left(c_2s^2 + c_1s + c_0\right)} \tag{13}$$

where the coefficients, a, b, c, are given in terms of system parameters. In a similar manner we obtain the analytical transfer function between the joint motion, q_r, and the servo reference signal, V_a,

$$\frac{sQ_r}{V_a} = \frac{\eta_x K_v\left(b_2s^2 + b_1s + b_0\right)}{\left[s\left[a_2s^2 + a_1s + a_0\right] + K_vK_b\left(b_2s^2 + b_1s + b_0\right)\right]} \tag{14}$$

here the system parameters are given as, $K_v=K_m/R_a$, $b_2=M_{ff}$ $-bM_{rf}$, $b_1=N_{rf}$, $b_0=K_{ff}$,

$a_2 = M_{ff}M_{rr} - M_{rf}M_{rf}$, $a_1 = N_{rf}M_{rr}$, $a_0 = K_{ff}M_{rr}$, K, R_a is the motor armature resistance, K_m is the open loop gain and η is the motor gear transmission ratio.

The above derived analytical transfer functions are used to determine the FSMS system structure in the experimental model validation studies as shown in the next section.
The experimental control of the FSMS slew motion and the active vibration control of the flexible system is achieved with digital control strategies based on vibration dynamic compensation (Lew & Trudnowski, 1996), and null space reaction control (Nenchev, 1996)

A nonlinear control law for the joint motion can be easily derived to actively damp the vibrations of the flexible structure. The flexible dynamics, given by equation (11) above, can be written as a standard damped, second order system, if the control torque is given by:

$$u(t) = -\left(\frac{bM_{rr}}{M_{rr}} - \frac{M_{fr}}{M_{rr}}\right)^{-1} .\tilde{G}_{ff}\ \dot{q}_f \tag{15}$$

leading to the closed-loop equation,

$$\tilde{M}_{ff}\ \ddot{q}_f + \tilde{G}_{ff}\ \dot{q}_f + \tilde{K}_{ff}\ q_f = 0 \tag{16}$$

This equation leads to a standard eigenvalue problem to obtain the eigenvalues and eigenvector of the closed loop system. Considering that the open-loop system is completely observable and controllable, the linearized control law implements a standard pole placement technique. The control gains are determined by assigning the closed-loop eigenvalues.

4. SIMULATION RESULTS

In this section, is presented a scheme for the implementation of the proposed control law, stated in equation 15. The figure 2 shows the model simulations, equation (8). The physical parameters of the micro/macro system are in the table 1.

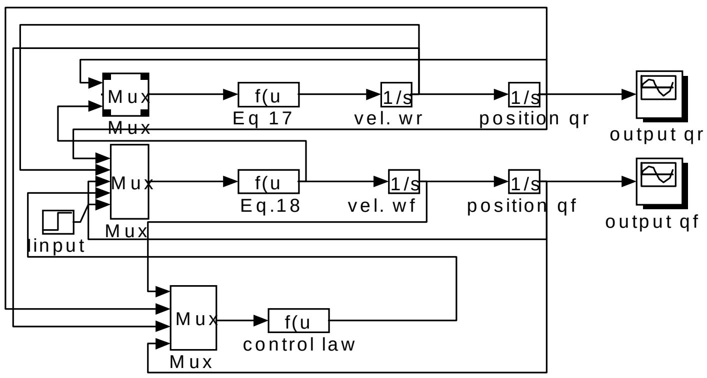

Figure 2 – Simulation scheme

Substituting the table 1 parameters in equations (8) and (15), we reach at the equations (17), (18) and (19):

$$0.01125\ddot{q}_r - 0.26775.\cos(q_r)\ddot{q}_f = 0 \tag{17}$$

$$6.4138\ddot{q}_f + 1192.85q_f - 0.26775.\sin(q_r).\dot{q}_r\dot{q}_r = -M_{fr}\ddot{q}_r \tag{18}$$

$$\ddot{q}_r = M_{fr}^{+}(G_f\dot{q}_f - (-0.26775.Sin(q_r).\dot{q}_r)\dot{q}_r) \tag{19}$$

The figure 2, shows the schematic diagram used in the simulation results. In figures 3 and 4, are shown the simulation results. Figure 3 shows the modal coordinate signal q_f, where the oscillation is dumped because the control law term $G_f \dot{q}_f$, equation (16).

Table 1 . Physical parameters

Parameter	name	value	Unit
Aluminum Young's modulus	E	7.1e10	N/m
Flexible link moment of inertia	I	2.97e-10	m^4
Flexible link shape function	Fi	3.57	m
Rigid link mass	m_r	0.45	Kg
Rigid link length	L_r	0.30	m
Flexible link length	L_f	0.81	m
Flexible link density per unit length	ρ	0.26/0.81	Kg/m
Flexible link thickness	e_b	5.20	mm
Flexible link width	h_b	25.4	mm

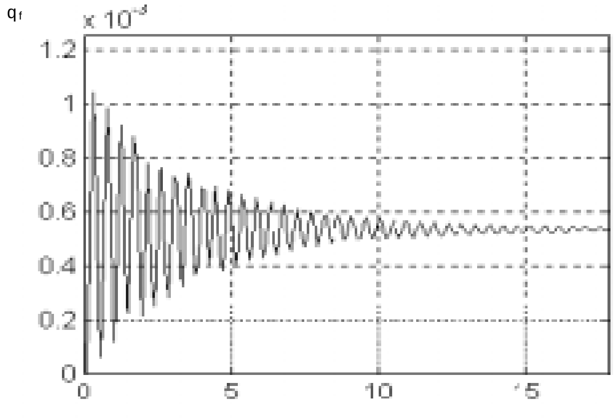

Figure 3 – Flexible link tip displacement, $q_f(t)$

Figure 4 shows the angular displacement of the micro manipulator revolute joint coordinate, $q_r(t)$.

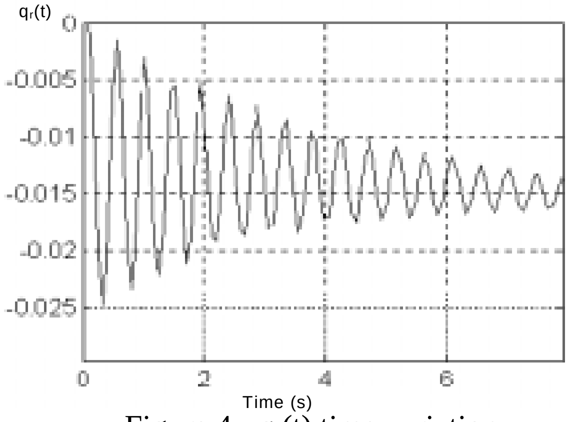

Figure 4 - $q_r(t)$ time variation

5. EXPERIMENTAL RESULTS

The model validation is carried out through experimental modal analysis and the estimation of the experimental transfer functions from the sensor signals. A piezoelectric accelerometer conditioned by a charge amplifier and integrating preamplifier is used to monitor the vibracional displacement of the beam tip. Full strain-gage bridges are also used to measure the elastic deformation of the flexible structure. Random noise excitation, of the joint motion, is used to obtain the parametric model identification in the frequency domain. A computerized data acquisition and control system (DACS) based on a PC microcomputer with analog and digital interfaces, is used to monitor the sensor signals and control the system in real-time. A standard DC torque motor is used to actuate the slew motion of the micro manipulator.

Figure 5 shows the experimental FRF between the rigid link motion, measured with a potentiometer, and the motor input voltage. Also shown in this figure is the identified FRF based on the model given by equation (14). The system shows a clear mode resonance at the frequency of 26 Hz.

The same mode structure is shown in figure 6, which exhibits the experimental FRF and the corresponding identified model, for the acceleration at the flexible beam tip and the actuator input.

The strain gage experimental FRF shown in figure 7, has a more distinct mode structure. Besides the mode resonance at 26 Hz, it also shows a lower mode structure, around the frequency of 2 Hz. This frequency coincides with the clamped free mode resonance of the flexible beam. On the other hand, the mode structure described by equation (13) does not fit very well the high frequency behavior of the FSMS. Nonetheless, we consider that the proposed theoretical model indeed represents the main dynamic characteristics of the FSMS system.

These model validation studies and identified mode structure were applied to develop a nonlinear control strategy to actively damp the mechanical vibrations induced on the flexible structure mounted micro manipulator system. Instead of a computed torque model which is more appropriate for a current feedback system, we calculate from equation (10), the joint angle acceleration which results in a damped closed-loop system for the flexible coordinate, q_f. A simple algebra gives the following equivalent velocity control law for the established velocity servo system :

$$V_r = \int M_{fr}^{+}(G_f \dot{q}_f - \dot{M}_{fr}\dot{q}_r)dt \qquad (20)$$

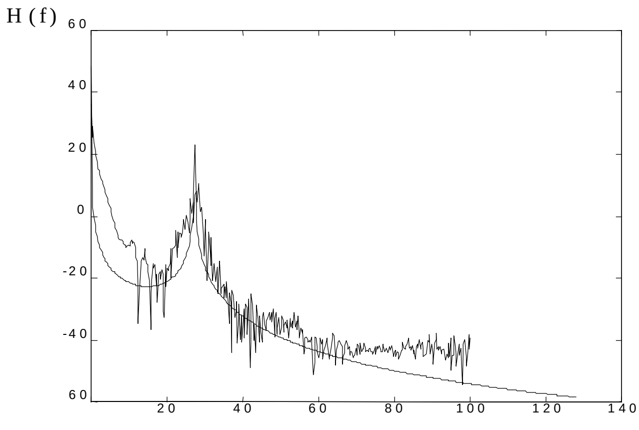

Figure 5 - Experimental and identified FRF for the micro manipulator joint angle.

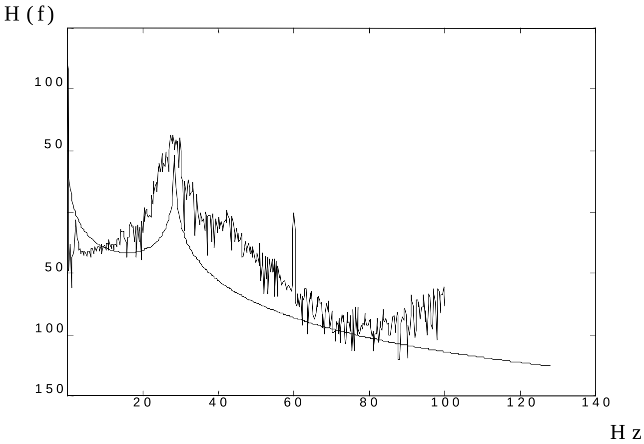

Figure 6 - Experimental and identified FRF for the accelerometer.

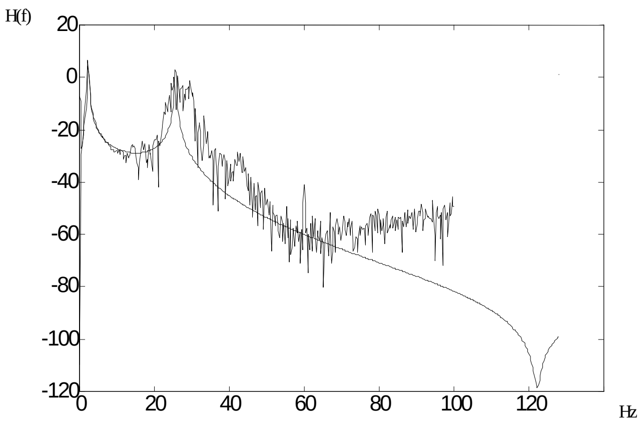

Figure 7 - Experimental and identified FRF for the strain gage sensor.

where M_{fr}^{+}, is the inverse of the inertia coupling parameter, and G_f is the control gain necessary to actively damp the vibrations of the FSMS. This latter gain was determined by extensive simulation studies of the FSMS closed loop dynamics.
The nonlinear control law describe by equation (20) was implemented on a PC based microcomputer with a standard AD/DA interfaces board. The flexible coordinate was estimated with the strain gage sensor as described by equation (12). The potentiometer signal was digitally derived to give an estimate of the joint angle velocity. The sensors and signal conditioning gains were experimentally determined from the fitted frequency response functions.

In figure 8, we show an estimate of the flexible modal coordinate for the open loop system. An impulsive disturbance was applied to the flexible structure and the time decay of the oscillations were recorded, giving a damping coefficient of approximately, ξ=0.018.

In figure 9, we show the active control of the FSMS, as implemented by the control law given by equation (17). The closed loop behavior has an equivalent damping coefficient approaching, ξ=0.038.

A residual mode oscillation is observed due to mismatches between the experimentally estimated parameters and the parameters of the real plant dynamics.

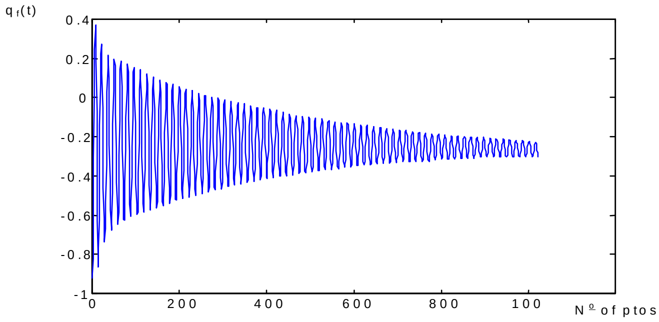

Figure 8 - Disturbance Response of the FSMS without the active control, time32.29 sec.

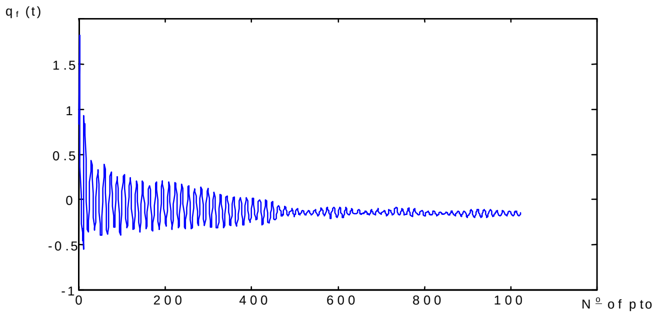

Figure 9 - Disturbance Response with the active control, time32.29 sec.

6. CONCLUSIONS

We have presented preliminary results of the implementation of an experimental micro/macro-manipulator system, consisting of a micro manipulator mounted on a flexible structure (Flexible Structure Mounted System/FSMS). We discussed a simplified model of the FSMS together with some simple control strategies for the active damping of the vibrations of the flexible base system. Symbolic manipulation, with the software MATHEMATICA, was used to derive the equations of motion for a generic FSMS. The procedures were validated with a simple case study both with simulation and experimental results. Several dynamic models for the sensors and actuator transfer functions were derived and compared with the experimental Frequency Response Functions (FRF), obtained by non-parametric modal identification. The FRF's were used to obtain the structural model of the FSMS. The identification results showed a dominant mode structure around the frequency of 2.01 Hz. These studies were important to determine the dynamic coupling between the macro manipulator servo actuation system and the FSMS system. The nonlinear control law implemented in this work showed to be effective to actively damp the vibrations of the FSMS. These results were observed both in the simulations and in the real time digital control.

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