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Implementation of an Active Vibration Control Technique on a Full Scale Space Vehicle Structure

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Abstract

This paper addresses the implementation issues involved in an active vibration control (AVC) technique using piezoelectric stack actuators. It explains briefly the design and development of a broad band active vibration isolator (AVI) and subsequently presents the details of how these AVIs are employed in real time AVC application. The actuator electronics and the control scheme development have been highlighted. The AVC results are presented for four elastic modes in terms of Power Spectral Densities in open and closed loop configurations.

1. Introduction

Smart materials such as piezoelectric and shape memory alloys have found interesting applications in various engineering fields [Chopra,2002]. In particular, active control technologies have matured in recent years after the use of smart materials. Both feedback and feed-forward control concepts have been effectively applied in vibration control applications. Finite element procedures, involving electro-mechanical couplings are developed and implemented in beams, plates and shells etc [Mackerle, 2001; Suresh & Rao, 2005; Kapuria et al., 2009; Malgaca, 2010; de Godoy & Trindade, 2011]. Indeed numerical modeling capabilities have enhanced the understanding of smart structures, which have resulted in different applications, employing piezoelectric materials.

The brittle nature of monolithic piezoelectric materials has been overcome by developing piezoelectric composites, where PZT fibers are embedded in a polymer epoxy matrix [Bent, 1999; Wilkie, et al., 2000]. Numerical studies have been performed using piezoelectric composites to show the advantage of using actuators in a flexible form in vibration control applications [Gopinath et al., 2011; Kerur & Ghosh, 2011]. Though active control brings advantages such as weight reduction, improved control bandwidth, and better system stability, the hybrid approach, involving active-passive techniques, still attracts researchers and has been found very effective in vibration control [Araujo et al., 2010; Moita et al., 2011].

Sandwich construction is widely adopted to build light weight structures, for aerospace vehicles in particular. The use of smart materials in sandwich structures has received notable attention in recent years [Guo & Jiang, 2011; Rechdaoui & Azrar, 2010; Rechdaoui et al., 2009; Luo et al., 2008; Duigou et al., 2006; Baillargeon & Veel, 2005].

Piezoelectric materials have been extensively used in stack form (multilayered) and plate crystals (single/two layered/ multi-layered) in both vibration control and health monitoring applications. Currently the flexible PZT composites (Active Fiber Composite/Macro Fiber Composite) are commercially available in two actuation modes, namely d_{31} (Duract®, PI Ceramics) and d---$_{33}$ (MFC, Smart Materials®). These composite actuators not only possess strain and force capabilities but have also improved the actuators' damage tolerance abilities. CEDRAT® and APC® (American Piezoceramics) have already commercialized multilayer stack actuators with wider specifications (block force/free deflections/ geometry), for users to choose according to their structural control applications. The current work has employed the parallel pre-stressed multi-layer stack to develop a broad band active vibration isolator (AVI). Subsequently, these AVI's are used along with passive isolators to demonstrate their abilities as vibration attenuators [Raja *et al.*, 2010]. Active vibration control demonstrations on a typical space structure are accordingly considered. The necessary in-flight electronics are employed to conduct closed loop vibration control experiments, nearly simulating a real time environment, addressing the power requirements [Shankar et al., 2008]. The sandwich deck plate that is considered in the present work is made of aluminium faces and an aluminium core, on which critical mission mode electronics are mounted. During the flight of the space vehicle, it is necessary to protect this equipment in order to ensure mission safety and its accomplishment. Therefore, vibration control is preferred to expose these electronic subsystems to a lower vibration level. In this regard, an explorative research programme has been initiated by LPSC (ISRO, India) on the use of piezoelectric materials in space structural control applications.

2. Structural Details

The space vehicle structure is basically a sandwich construction, on which many mission critical electronic pieces of equipment are mounted. The structural assembly consists of five components, namely two doubler plates, one each (1 unit thick) on the top and bottom of the sandwich plate with face (1 unit thick)-core (40 units thick)-face (1 unit thick). Figure 1 shows the pictorial view

Figure 1: AVC Experimental Setup

of the structural system and the associated instrumentation.

2.1 Structural modelling and analysis

The sandwich structure (deck plate) is modelled and analysed using ABAQUS. The material data given in table 1 have been used to compute free vibration characteristics. The numerical model of the structure is shown in figure 2. The deck plate is a three layered configuration, which has been located in the strap-on booster of a space vehicle to carry important instruments on board. The structure is discretized by plate elements, considering it as an equivalent laminate. The deck plate is actually mounted on 16 passive isolators and they are characterized by an average elastic stiffness of 40 kg/mm in the analysis. The passive isolators are elastically represented by spring elements at their respective locations. A numerical model is then built with 4 PZT actuators (as the replacement for 4 passive isolators) and 12 passive isolators (represented by spring elements) to evaluate the effectiveness of the electro-mechanical coupling. For this purpose, the numerical models of the actuators are integrated with the FE model of a deck plate.

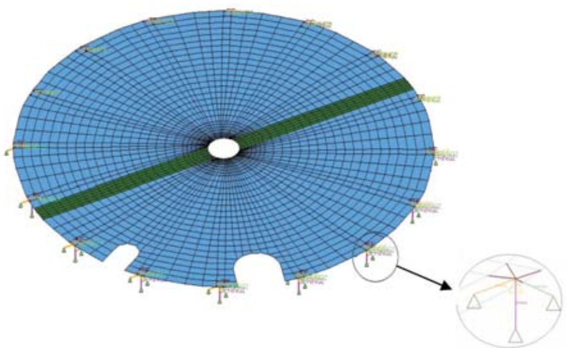

Figure 2: FE model of the Space Structure with passive isolators

In the analysis model, the boundary conditions are simulated such that they represent the stiffness characteristic of the passive isolators (16 members).

Table 1: Material data

Components	Doubler, Face Sheet	Core
Material	Aluminium	Al 5056 Alloy
Young's Modulus (GPa)	68.6	—
Poisson's ratio	0.30	0.0
Shear Modulus G12 (GPa)	—	—
Shear Modulus G13 (GPa)	—	0.4522
Shear Modulus G23 (GPa)	—	0.1972
Density (Kg/m^3)	2780	63

TABLE 2: Frequency Comparison

Mode No	FEM	Sine	Random		Mode Description
	Freq (Hz)	Freq (Hz)	Freq (Hz)	Damp (%)	
1	104.66	105.7	104.46	1.69	Transverse mode
2	194.21	184.4	184.53	0.94	Planar rotational mode
3	207.39	227.1	227.39	1.40	Planar rotational mode
4	287.72	256.9	241.45	1.37	Bending-Torsion mode

This configuration is treated as the reference one and the open loop system dynamics is accordingly established. In order to conduct the experiments on the deck plate, a proper fixture (supporting structure) is designed and fabricated. The fixture consists totally of 8 columns; over which four segmented steel plates are mounted. Solid elements are used to model the supporting structure (refer to figure 3). These columns are considered to be fixed at the bottom to simulate the experimental conditions in the numerical analysis.

Subsequently, ground vibration testing (GVT) is performed and the numerical simulation of the structure is validated. A single input and multi-output test are carried out to extract the frequencies and mode shapes. The mode shapes can be referred to in figure 3. Table 2 presents the comparison of the frequencies.

2.2 Active Vibration Isolator

The PZT stack actuator is a high power device but needs protection from excessive dynamic loading, particularly in the shear mode. The salient features of the AVI design are presented below.

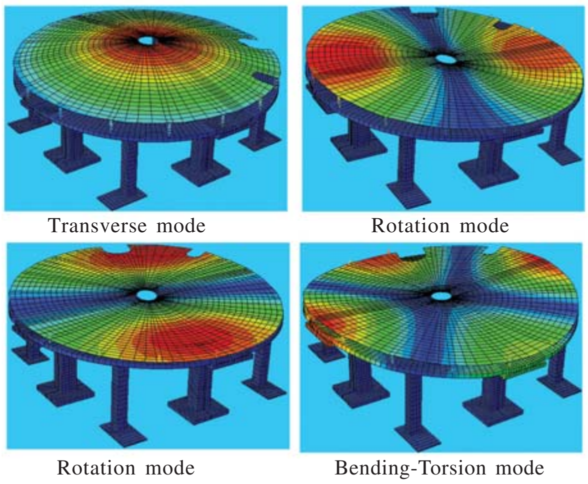

Figure 3: Mode shapes of the sandwich plate

- Due to a dimensional constraint for accommodating the proposed actuating system into the sandwich plate (available height < 40mm), a pre-stressed PZT stack actuator is selected.
- PPA20M (CEDRAT®) is chosen that has a dimension of 10x9x28 mm^3 and weighs around 12 gm.

The CAD model of the AVI and the fabricated components are shown in figure 4. The actuator is numerically characterized for its block force (800N) and free deflection (20μm). Indeed, the stiffness of the actuator is 50N/μm(manufacturer supplied data), which is quite sufficient to take the inertial loading of the sandwich plate. This actuator can be operated in the voltage range of -20V to +150V.

2.3 Stack actuator modeling and block force estimation

Piezoelectric materials in multilayered form (stack actuators) are widely used in a structural control applications. Parallel stack actuators consist of pre-stressed active materials that deform in an axial direction electro-mechanically with a limited amount of stroke length (d_{33} mode actuation). The deformation can be elastically constrained to generate induced force, which can be used in AVC applications. The blocked force (fully constrained state) or induced force (elastically constrained state) and free deflection (unconstrained state) are identified as design parameters to characterize the piezoelectric stack actuator. Stack actuators with a blocked force of 800 N and a free deflection of 40 mm are considered to develop active vibration isolators for the present application. The manufacture supplied data are used to model the stack actuator in ABAQUS®.

$$\Delta x = n.d_{33}.V \tag{1}$$

$$n = \frac{\Delta x}{d_{33}.V}$$

$$L = n \times t_1$$

where,

d_{33} = Piezoelectric constant (m/V), n = Number of layers , V= Applied voltage (V), $\Delta x =$ Free deflection (mm)

$$F_{block} = \frac{d_{33} \times A \times V}{S_{33} \times t_1} \tag{2}$$

where,

F--$_{block}$ = Block force (N), S_{33} = Compliance (m^2/N), t_1= Active layer thickness (m) , A=Area of cross section (m^2).

The stack actuator is modeled in ABAQUS® using C3D8E elements, where the voltages are imposed as enforced electrical boundary conditions. in table 3. Since individual layers in the stack are very thin (100 mm), a group of layers are made to form a single plate and are modeled as an equivalent solid element with a simulated electric field in each element to match the field strength of the individual layer. In the present analysis, a parallel type of electrical connection is idealized. The voltages are applied at the nodes of the top and bottom surfaces of each element with opposite potentials. This helps to introduce the opposite polarity across the elements to simulate a parallel connection as shown in figure 5. The free deflection of the stack is then estimated with clamped-free boundary conditions for the maximum operating voltage. It is observed that the present model predicts very closely the deflection with CEDRATÒ data (see tables 4,5). The blocked force is estimated by constraining both ends of the actuator and keeping the stack in actuation mode with maximum voltage. The finite element analysis results (free deflection, block force) are shown in figures 6 (a) and (b) as contour plots.

Table 3: Properties of FEA / Analytical models of stack actuator

Material Properties	FEA	Analytical
Elastic moduli (Gpa) 15.18317	E	15.183
Poissons ratio n	0.22	0.22
Piezoelectric constant (m/v) d_{33}	374×10^{-12}	374×10^{-12}
Dielectric constant (F/m) $Î_{33}$	1.531×10^{-8}	1.531×10^{-8}

Table 4: Characteristics of PPA40M actuator (Cedrato)

Description	Units	Data
Displacement	ìm	40
Blocked force	N	800
Stiffness	N/ìm	20
Voltage range	V	-20…150
Height (in actuation Direction)	mm	48.0
Width	mm	10.0
Depth	mm	9.0

Table 5: Comparison of analysis results

Analysis	Displace-ment (im)	Blocked force (N)
PPA40M CEDRAT Ò	40	800
Analytical	39.99	799.9
FEA-ABAQUSÒ	40	794.1

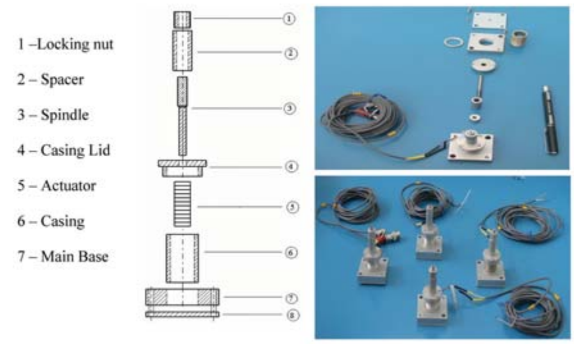

Figure 4: Active Vibration Isolators used in the AVC study

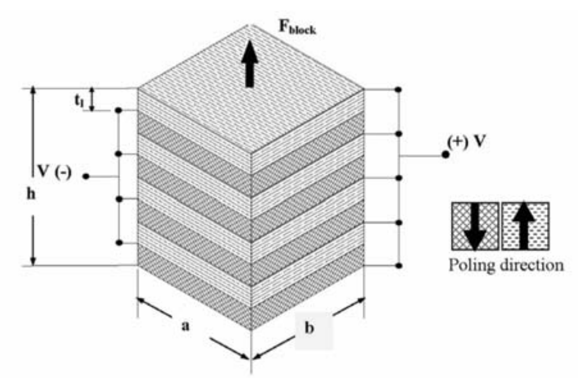

Figure 5: Stack actuator with parallel connection

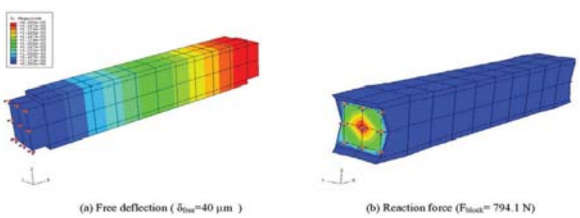

Figure 6: Free deflection and reaction force

3. Controller and system electronics details

The actuator electronics are designed in-house such that they can be operated through 28VDC. This closely simulates the actual power source that is available in the space vehicle. Since the PZT stack actuator needs -20 to 150 V, appropriate power electronics are designed and fabricated using DC-DC converters [Raja et al.,]. Figure 7 demonstrates the testing of these in-flight electronic modules in an open loop configuration.

Figure 7: In-flight electronics used to drive the stack actuators

A multi-input and multi-output control model is then developed using the LQG technique. It is a model based control design approach, where the numerical FE plant is represented in a state variable form. We measured the acceleration state and estimated the velocity through a Kalman filter in order to implement full state feedback control (See figure 8). A linear quadratic regulator is designed in the modal domain using the system matrices, obtained from finite element analysis. The stiffness, mass and actuator matrices of the deck plate are obtained in an uncoupled form in the modal domain by transforming them from physical coordinates (X,Y,Z) using modal orthogonality relations (mode shape vectors).

The dynamic equation of the deck plate is then written in modal form as,

$$\bar{M}_{uu}\ddot{\zeta}+\bar{D}_{uu}\dot{\zeta}+\bar{K}_{uu}\zeta-\bar{K}_{u\phi}=\bar{F}_d, \quad (3)$$

and

$$\bar{M}_{uu}=\psi^T M_{uu}\psi, \quad \text{(modal mass);}$$

$$\bar{D}_{uu}=\psi^T D_{uu}\psi, \quad \text{(modal damping);}$$

$$\bar{K}_{uu}=\psi^T K_{uu}\psi, \quad \text{(modal stiffness);}$$

$$\bar{K}_{u\phi}=\psi^T K_{u\phi}, \quad \text{(modal actuator stiffness);}$$

$$\bar{F}_d=\psi^T F_m, \quad \text{(modal force);}$$

where ψ is the modal vector.

Further, the closed loop system is built using the dynamic equation in state variable form as follows:

$$\dot{\chi}(t)=A\chi(t)+B\phi_a(t)+\beta(t),$$
$$\phi_s(t)=C\chi+\nu(t), \quad (4)$$

Equation (4) can be defined for the i^{th} mode using the independent modal space control (IMSC) approach in the state-space format as follows:

$$A_i = \begin{bmatrix} -2\zeta_i\omega_i & \omega_i \\ -\omega_i & 0 \end{bmatrix}, \quad B_i = \begin{bmatrix} \overline{F}_d(i) & \overline{K}_{u\phi}(i,a) \\ 0 & 0 \end{bmatrix}, \quad C_i = \begin{bmatrix} \overline{C}_d(s,i) & \overline{C}_v(s,i) \end{bmatrix},$$

$$\overline{C}_d = \frac{1}{C_{eq}}\overline{K}_{\phi u}, \ \overline{C}_v = R_{eq}\overline{K}_{\phi u}, \ \overline{K}_{\phi u} = K_{\phi u}\psi, \tag{5}$$

where R_{eq}= equivalent circuit resistance, and C_{eq}= equivalent circuit capacitance, 's' is the number of sensors, 'a' is the number of actuators, ω is the frequency of vibration, and ζ is the modal damping .

The control design has taken into account the process disturbance (   ) and the measurement noise (ν).

Therefore, the final closed loop system is built with a liner control law as,

$$\begin{aligned} \dot{\chi} &= (A - BG_c - G_eC)\chi + G_e\phi_s, \\ \phi_a &= -G_c\chi. \end{aligned} \tag{6}$$

Both the LQR gain (G_c) and the Kalman filter gain(G_e) are computed by minimizing the quadratic performance index, involving plant matrices and associated weighting vectors of Q_c, R_c, Q_e, R_e, respectively. The subscripts c and e denote controller
β and estimator, respectively.

3.1 Locations of AVIs on the sandwich structure

Actuator and sensor placement have been considered one of the important design parameters in smart structure based AVC solutions. It is, therefore, necessary to couple the active effects (actuation/sensing) with that of elastic strain fields (In-plane/bending/torsion) in order to compute the coupled responses (electro-mechanical) of large smart structural systems like the deck plate. Critical strains (modal) are obtained through modal response analysis in the interested frequency band (first four modes). Further, electro-mechanical responses are computed for arbitrary locations of the four AVIs, 12 being passive isolators. Finally, the locations are decided based on the ability of these four AVIs to simulate the critical modes (mode shapes/nodal lines) in the best possible manner.

4. Implementation of the AVC System

For the sake of clarity, the locations of actuators(AVIs) and accelerometers are shown in

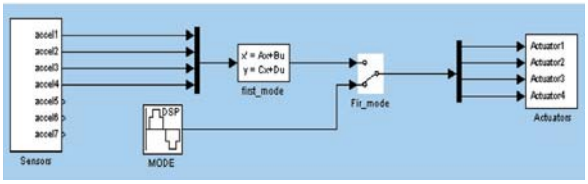

Figure 8: Multi-input multi-output control model in SIMULINK

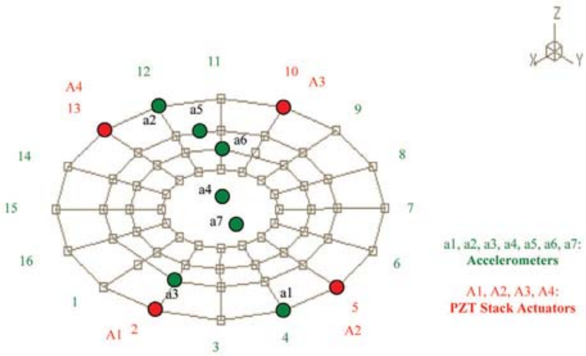

Figure 9: Accelerometer and Actuator locations on Space Structure

figure 9. As it is already mentioned, the sandwich plate with 16 passive isolators is considered an open loop system. Corresponding to this configuration, the closed loop performance is evaluated. In the closed loop configuration, four passive isolators are replaced with four AVIs. Experimentally the frequencies and mode shapes are established through a GVT process (Refer to table 2). The four elastic modes are excited with sine disturbance (simple resonance) in open loop conditions and they are independently controlled by applying the modal LQG controller.

The details of applied disturbances and the voltages of actuators are presented here.

Case (a): Ist mode control

Frequency: 115.2 Hz

Actuators/Sensors in loop: A1/a2 & A3/a1

Disturbance level: 1.57g

Applied voltage: 70V

Case (b): IInd mode control

Frequency: 215 Hz

Actuators/Sensors in loop: A1/a1 & A3/a2

Disturbance level: 0.461g

Applied voltage: 30V

Table 6: PSD and RMS energy at different locations (Mode I)

Sensors	Output PSD (g^2/Hz)			Energy band 20Hz (106-126 Hz) (g^2)		
	Open	Closed	Control (%)	Open	Closed	Control (%)
1	4.174e-2	6.125e-3	85.32	9.166e-2	3.665e-2	60.01
2	7.829e-2	1.076e-3	98.62	0.1255	1.543e-2	87.70
4	2.467	7.159e-3	99.70	0.7047	4.036e-2	94.27
7	2.448	6.262e-3	99.74	0.7020	3.784e-2	94.60

Table 7: PSD and RMS energy at different locations (Mode II)

Sensors	Output PSD (g^2/Hz)			Energy band 100Hz (160-260Hz) (g^2)		
	Open	Closed	Control (%)	Open	Closed	Control (%)
1	0.1375	2.011e-2	85.37	0.2775	0.1048	62.22
2	0.2130	3.306e-2	84.47	0.3455	0.1344	61.09

Table 8: PSD and RMS energy at different locations (Mode III)

Sensors	Output PSD (g^2/Hz)			Energy band 100Hz (225-325Hz) (g^2)		
	Open	Closed	Control (%)	Open	Closed	Control (%)
1	5.097e-2	3.053e-2	40.0	0.2176	0.1673	23.11
2	0.1224	6.125e-2	49.95	0.3372	0.2370	29.71
3	1.039e-2	2.789e-3	73.15	9.826e-2	5.056e-2	48.54
7	6.101e-3	2.425e-3	60.25	7.525e-2	4.715e-2	37.30

Table 9: PSD and RMS energy at different locations (Mode IV)

Sensors	Output PSD (g^2/Hz)			Energy band 100Hz (280-380Hz) (g^2)		
	Open	Closed	Control (%)	Open	Closed	Control (%)
1	5.438e-2	1.637e-2	69.89	0.1811	0.1021	43.62
3	1.402e-2	2.040e-3	85.44	9.195e-2	3.606e-2	60.78

Case (c): IIIrd mode control
Frequency: 265 Hz
Actuators/Sensors in loop: A1/a1 & A4/a2
Disturbance level: 0.35g
Applied voltage: 20V
Case (d): IVth mode control
Frequency: 328 Hz
Actuators/Sensors in loop: A2/a2 &A3/a3
Disturbance level: 0.23g
Applied voltage: 20V

5. Results and Discussion

The developed MIMO control scheme has been implemented in the dSPACE DS1104 board. The open and closed loop responses are presented in tables 6 to 9. The control performance of each mode is examined in terms of PSD at resonance frequency as well as the energy level in a frequency band for both open and closed loop conditions. More than 90% vibration reduction is seen for the fundamental mode both at resonant frequency (See figure 10) and in a 20 Hz energy band (Refer to table 6). It can also be seen that the second mode vibration is reduced significantly at resonant frequency; however, the overall reduction in the energy level is around 60% in a 100 Hz frequency band. This kind of assessment provides a realistic performance estimation for the implemented AVC system, in case the natural frequencies of the structure change due to flight conditions. A similar quantification for the third and fourth modes reveals that they are better controlled with a developed AVC system around 50 to 60%.

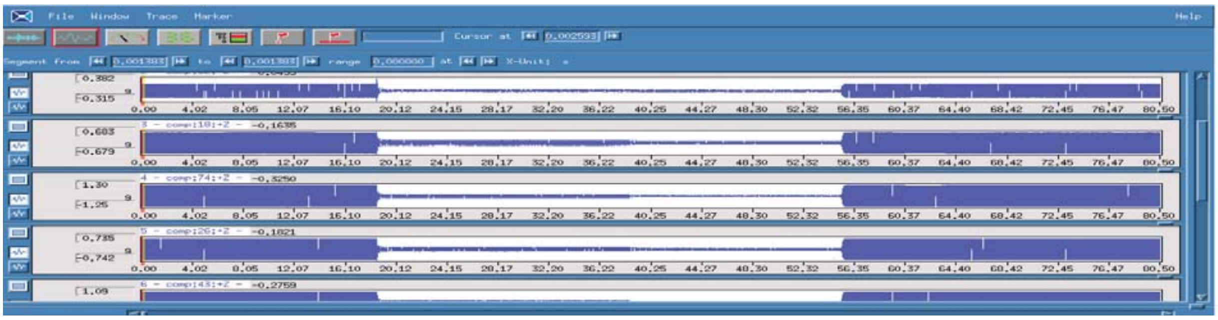

Figure 10: Time plot of response signals (open/closed)

It is worth noticing that for the higher modes' control, the actuators are operated at barely less than 20% of its full actuating potential (-20 to 150V).In the present study, the performance of the AVC system is examined through the PSD. Also the main goal is to develop an AVC system for amplitude control through active isolations concept. Therefore damping is not estimated in the current work.

6. Conclusions

The active vibration control technique using piezoelectric stack actuators has been developed and is implemented on a real time space structure. A four channel closed loop control system simulation and hardware in loop experiments are carried out to evaluate the AVC system performance. It has been found interesting that with four stacks, which are replacements for four passive isolators, the excessive vibrations (amplitude control) in four modes are controlled by more than 60%.

Acknowledgements

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