International Journal of Engineering Research-Online
A Peer Reviewed International Journal
Articles available online http://www.ijoer.in

Vol.3., Issue.5., 2015
(Sept.-Oct.)

RESEARCH ARTICLE

ISSN: 2321-7758

OPTIMAL PIEZO-ELECTRIC PATCHES PLACEMENT FOR ACTIVE VIBRATION CONTROL OF SMART PORTAL FRAME STRUCTURE

PRABHAT KUMAR SINHA[1], HENRY SHEPHERED[2], ASHOK TRIPATHI[3], R.K.SRIVASTAVA[4]

[1]Research Scholar, Sam Higginbottom Institute of Agriculture, Technology and Sciences, Allahabad, U.P. INDIA

[2]Professor, Sam Higginbottom Institute of Agriculture, Technology and Sciences, Allahabad, U.P. INDIA

[3]Professor, Sam Higginbottom Institute of Agriculture, Technology and Sciences, Allahabad, U.P. INDIA

[4]Professor, Motillal Nehru National Institute of Technology, Allahabad, U.P. INDIA

PRABHAT KUMAR SINHA

ABSTRACT

The vibration control in portal frame is examined for the optimization of structure. In this paper, analysis and active vibration control of smart portal frame is done in which piezo patches are bonded in different location on the frame as sensor/actuator pair for control. The piezo patches are considerably using Proportional Integral Derivative (PID) controller for active vibration control. An analytical formulation is derived for modelling the behaviour of portal frame with integrated piezoelectric sensor and actuator. The voltage generated by the sensor layer and response of the frame to the actuator voltage can be computed independently. The obtained results are presented to demonstrate the ability of closed-loop system and actively control the vibration of portal frame. The frame is divided in six equal element and natural frequency of the frame is obtained by using Guyan's Reduction. Results are verified and compared by ANSYS. Analysis of active vibration control is done through Simulink (MATLAB) model. The effect of piezo patch on each element of portal frame are investigated for all the six elements. The optimum location is found for actively controlled structure.

Keywords: Active Vibration Control, PID Controller, ANSYS, Simulink, MATLAB, Finite Element Analysis, Guyan's Reduction.

INTRODUCTION

With the advancement of mechanical systems, it is develop smart structure which are capable of self-monitoring and control capabilities. A smart structure is an engineering structure containing sensors and actuators that, when active, modify the response of the structure to its environment and control the structure from failure. Research into smart structural control is growing due to new challenges in extreme conditions in the area such as space, undersea, polar, nuclear, chemical and biological applications. In the present research proposal a feasible application of active control of smart structures/mechanical equipment's would be studied. Advancement in sensor, actuator and microprocessors technologies enabled us to control these equipment's actively and precisely. The use of active control plays vital role in the improvement of machine-tool performances, vehicle dynamic characteristics, off-shore installations, aircraft structures and space shuttles etc. The piezoelectric sensor/actuator pairs that are collocated equidistant from the neutral axis for the active vibration control of smart structure using constant-gain feedback control. A response function of the system measured with optical, mechanical, electrical or chemical sensors create signals that are sent to control actuators. Advances in theory and practice of active structural control technology have modified the general perception of structures. Through mimicking living organisms, incorporation of intelligent control methodologies into structural engineering has potential to enhance the concept of structures. Self-diagnosis, self-repair and learning is examples of behavioral bio mimicry. Computing challenges that are important to the creation of the next generation of active structures are then identified. Active structural control is closely related to structural health monitoring (SHM). Inman (2001) has investigated that active bars to control a slewing frame test bend, the smart structure solution was concluded to be superior to a conventional solution and the closed-loop system of piezo ceramic actuators and sensors significantly improved controllability. Fest (2003) has experimentally studied shape control of a five module large-scale active tensegrity structure. The structure was controlled through active struts to ensure serviceability of the structure when it was subjected to additional loads. Campbell (2004) has presented that active strut control on six-axis hexapods used for space-based vibration isolation. Ziegler (2005)has presented an objective of shape control is elimination of the structural deformations caused by external disturbances. Other objectives include control of reception and transmission orientation as well as aerodynamic vibration control. Shape control is usually carried out through active cables and struts. Nudehi (2006) has investigated that experimentally the use of end forces for vibration control on a beam structure fitted with a cable mechanism and motor for applying the end force, and a piezoelectric patch for taking vibration measurements. Result shows that the modes of the beam were effectively controlled and the natural frequencies are less than the bandwidth of the motor. Adam and Smith (2007) have presented intelligence is one of the desirable qualities of biological systems. Through mimicking living organisms, incorporation of intelligent control methodologies into structural engineering has potential to enhance the concept of structures. Self-diagnosis, self-repair and learning is an example of behavioural bio mimicry. Annaswamy (2008) has presented active control of acoustic resonances produced by supersonic impinging jets. They experimentally demonstrated that acoustic resonances could be reduced using a closed-loop algorithm. Guo et al. (2008) have presented an algorithm to control the vibration of space structures with active tendons. Preumont (2008) has presented the classical problem of active and passive damping of a piezoelectric truss. They investigated that the Voltage control and charge (current) control implementations. They presented that the performance was controlled by the modal fraction of strain energy in the active strut and the electromechanical coupling factor. Malekzadeh (2009) has presented a three dimensional elasticity approach based on a layer wise theory. They investigated the dynamic response of cross-ply laminated thick plates subjected to the moving loads. Lin C.C (2010) have analysed soil–structure interaction effect on vibration control effectiveness of active tendon systems for an irregular building which modelled as a torsionally coupled structure subjected to base excitations such as those induced by earthquakes. A direct output feedback control algorithm was successfully applied to a two-way eccentric building with active tendon systems in outer frames.

FORMULATION

Large finite element models with thousand degree of freedom are used for stress and deformation analysis of ships, aircraft, automobiles, nuclear reactors, which is impractical to perform dynamic analysis. Guyan reduction is one of the popular method for dynamic analysis. The equation of motion for reduced stiffness and mass matrices are

$$M\ddot{Q} + MQ = F \qquad (1)$$

If we group the inertial force together with the applied forces, then

$$KQ = F \qquad (2)$$

Then the partition Q is

$$Q = \begin{Bmatrix} Q_r \\ Q_o \end{Bmatrix} \qquad (3)$$

The equations of motion can now be written in partitioned form as

$$\begin{bmatrix} K_{rr} & K_{ro} \\ K_{ro}^T & K_{oo} \end{bmatrix} \begin{Bmatrix} Q_r \\ Q_o \end{Bmatrix} = \begin{Bmatrix} F_r \\ F_o \end{Bmatrix} \qquad (4)$$

Now, retain those DOF in the r-set with large concentrated masses, which are loaded and which are required to describe mode shape. Setting $F_o = 0$, then Q_o is given by

$$Q_o = -K_{oo}^{-1} K_{ro}^T Q_r \qquad (5)$$

The strain energy in the structure is given by

$$U = \frac{1}{2}[Q_r^T \quad Q_o^T]\begin{bmatrix} K_{rr} & K_{ro} \\ K_{ro}^T & K_{oo} \end{bmatrix}\begin{bmatrix} Q_r \\ Q_o \end{bmatrix} \qquad (6)$$

Substituting Eq.4 into Eq.6, we get

$$U = \frac{1}{2} Q_r^T K_r Q_r$$

Where, $K_r = K_{rr} - K_{ro} K_{oo}^{-1} K_{ro}^T$ is reduced to stiffness matrix.

From Eq.5, Kinetic energy is given by

$$V = \frac{1}{2} Q_r^T M_r Q_r$$

Where, $M_r = M_{rr} - M_{ro} K_{oo}^{-1} K_{ro}^T - K_{ro} K_{oo}^{-1} M_{ro}^T + K_{ro} K_{oo}^{-1} M_{oo} K_{oo}^{-1} K_{ro}^T$ is reduced mass matrix.

Now,

$$K_r U_r = \lambda M_r U_r$$

Then recover

$$U_o = -K_{oo}^{-1} K_{ro}^T U_r$$

MODELLING OF SMART PORTAL FRAME

Consider a steel alloy portal frame with collocated Piezoelectric Sensor/Actuator pair. The properties of material and frame dimensions are given in table-1. An external disturbing force F_{dist} is acting at the mid of the horizontal portion of the frame. F_{actu} is the force generated by the Actuator. Fig – 1, 2, 3, 4, 5, 6 shows six positions of the PZT Sensor / Actuator pair. The frame is divided in 6 finite element. The more finite elements gives more accurate result. Primarily the modelling of frame and its element is done and then the modelling of smart frame element which contains PZT Sensor/Actuator pair.

Table-1

Sr.No.	Material properties	Dimensions
1.	Modulus of Elasticity = 2×10^5 N/mm^2	Length = 300 mm
2.	Density = 7803×10^{-9} kg/mm^3	Width = 25 mm
3.	Poisson's Ratio = 0.33	Thickness = 25mm

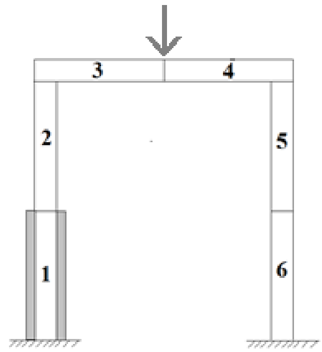


Fig-1

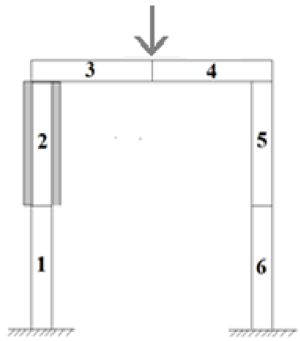


Fig-2

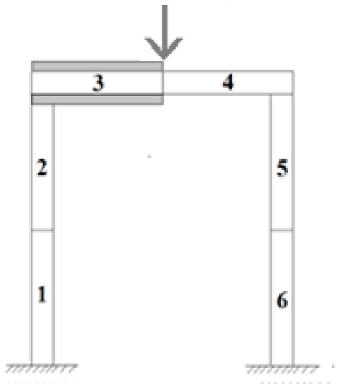


Fig-3

Sensor/Actuator Pair Element 1 Sensor/Actuator Pair Element 2 Sensor/Actuator Pair Element 3

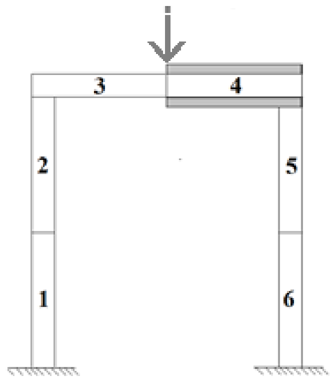


Fig-4

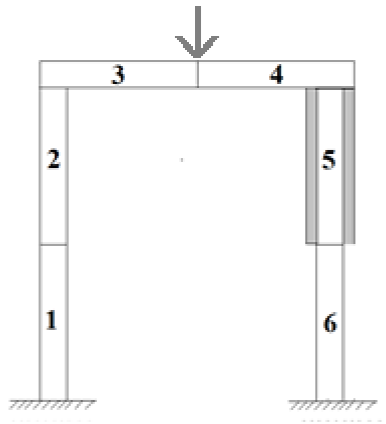


Fig-5

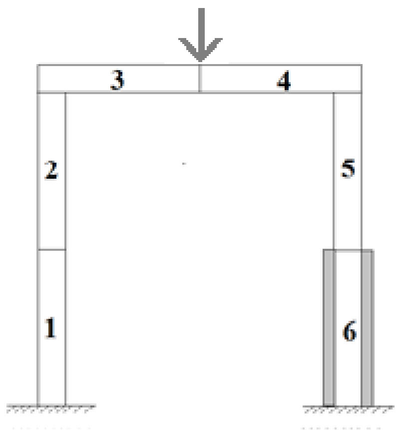
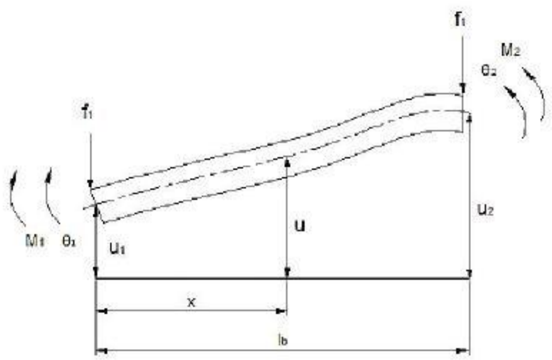


Fig-6

Sensor/Actuator Pair Element 4 Sensor/Actuator Pair Element 5 Sensor/Actuator Pair Element 6

FE formulation of regular frame element

A two nodes finite element of a rectangular beam element is shown in figure



The node undergoes both translational and rotational displacement and they are u_1, θ_1, u_2 and θ_2. The linear forces are F_1 and F_2 corresponding to displacements u_1 and u_2 and rotational forces (Bending moment) are M_1 and M_2 corresponding to the rotational displacements θ_1 and θ_2. The transverse displacement within the element is assumed to be a cubic polynomial as

$$u(x,t) = a_1 + a_2 x + a_3 x^2 + a_4 x^3 \qquad (1)$$

Substituting the boundary conditions the shape function of frame elements can be obtained as

$$\{N(x)\}^T = [N_1(x) N_2(x) N_3(x) N_4(x)] = \left[1 - \frac{3x^2}{l_b^2} + \frac{2x^3}{l_b^3} \; x - \frac{2x^2}{l_b} + \frac{x^3}{l_b^2} \; \frac{3x^2}{l_b^2} - \frac{2x^3}{l_b^3} \; \frac{-x^2}{l_b} + \frac{x^3}{l_b^2}\right]$$

(2)

The nodal displacement function can be written as

$$\{q\}^T = [u_1 \theta_1 u_2 \theta_2] \qquad (3)$$

The Lagrange's Equation gives the Kinetic energy and Potential energy of the system respectively as:

$$T = \frac{1}{2}\{\dot{q}\}^T [m]\{\dot{q}\}$$

$$U = \frac{1}{2}\{q\}^T [k]\{q\} \qquad (4)$$

Using Lagrange's Equation the element stiffness matrix and mass matrix of a element are computed given by[14][15]

$$[k_b] = \frac{E_b I_b}{l_b^3}\begin{bmatrix} 12 & 6l_b & -12 & 6l_b \\ 6l_b & 4l_b^2 & -6l_b & 2l_b^2 \\ -12 & -6l_b & 12 & -6l_b \\ 6l_b & 2l_b^2 & -6l_b & 4l_b^2 \end{bmatrix} \qquad (5)$$

$$[m_b] = \frac{\rho_b A_b l_b}{420}\begin{bmatrix} 156 & 22l_b & 54 & -13l_b \\ 22l_b & 4l_b^2 & 13l_b & -3l_b^2 \\ 54 & 13l_b & 156 & -22l_b \\ -13l_b & -3l_b^2 & -22l_b & 4l_b^2 \end{bmatrix} \qquad (6)$$

For deriving the sensor equation results, first and second spatial derivatives of the shape function are used.

$$\{n_1(x)\} = \left\{\frac{dN(x)}{dx}\right\} = \{N'(x)\} and \{n_2(x)\} = \left\{\frac{d^2N(x)}{dx^2}\right\} = \{N'(x)\} \qquad (7)$$

FE formulation of smart frame element

When PZT patches are assumed as Euler-Bernoulli frame elements the elemental mass and stiffness matrices of PZT frame element can be computed as:

$$[k_p] = \frac{E_p I_p}{l_p^3}\begin{bmatrix} 12 & 6l_p & -12 & 6l_p \\ 6l_p & 4l_p^2 & -6l_p & 2l_p^2 \\ -12 & -6l_p & 12 & -6l_p \\ 6l_p & 2l_p^2 & -6l_p & 4l_p^2 \end{bmatrix} \qquad (8)$$

$$[m_p] = \frac{\rho_p A_p l_p}{420}\begin{bmatrix} 156 & 22l_p & 54 & -13l_p \\ 22l_p & 4l_p^2 & 13l_p & -3l_p^2 \\ 54 & 13l_p & 156 & -22l_p \\ -13l_p & -3l_p^2 & -22l_p & 4l_p^2 \end{bmatrix} \qquad (9)$$

The smart frame element is obtained by sandwiching the regular beam element in between the two PZT patches. In which $EI = E_b I_b + 2E_p I_p$ is the flexural rigidity and $\rho A = b_b(\rho_b t_b + 2\rho_p t_p)$ is the mass per unit length of smart frame element, t_p is the thickness of PZT patches i.e. thickness of Actuator and Sensor. So the elemental mass and stiffness matrices of smart frame element are

$$[k] = \frac{EI}{l_p^3}\begin{bmatrix} 12 & 6l_p & -12 & 6l_p \\ 6l_p & 4l_p^2 & -6l_p & 2l_p^2 \\ -12 & -6l_p & 12 & -6l_p \\ 6l_p & 2l_p^2 & -6l_p & 4l_p^2 \end{bmatrix} \quad (10)$$

$$[m] = \frac{\rho A l_p}{420}\begin{bmatrix} 156 & 22l_p & 54 & -13l_p \\ 22l_p & 4l_p^2 & 13l_p & -3l_p^2 \\ 54 & 13l_p & 156 & -22l_p \\ -13l_p & -3l_p^2 & -22l_p & 4l_p^2 \end{bmatrix} \quad (11)$$

SENSOR EQUATION

Following linear PZT Constitutive equations [6], [13] will be used for driving the Sensor and Actuator equations.

$$\varepsilon_x = S_{11}^E \sigma_x + d_{31} E_z$$

$$D_z = d_{31}\sigma_x + \xi_{33}^\sigma E_z \quad (12)$$

Where ε is strain, σ is stress, S^E is compliance when electric field is constant, d_{31} is PZT constant, E is electric field, D is electric displacement, ξ^σ is dielectric constant under constant stress. The direct PZT effect is used to calculate the output charge on the sensor layer created by the strains in the beam. Since no electric field is applied to the sensor layer, we get

$$D_z = C_{11} d_{31} \varepsilon_x \quad (13)$$

Where C_{11} is the young's modulus of elasticity.

The charge measured through the electrode of the sensor is given by

$$q(t) = \int_S D_z ds \quad (14)$$

The current on the surface of the sensor is given by

$$i(t) = \frac{dq(t)}{dt} \quad (15)$$

We know that strain at a point in a beam is given as $\varepsilon_x = zd^2u/dx^2$, where z is a coordinate on the beam w.r.t. neutral axis. Width $b_b=b_s=b_a$. As such current generated can be written as [10]

$$i(t) = zC_{11} d_{31} b_b \int_0^{l_p} \{n_2(x)\}^T \{\dot{q}\} dx \quad (16)$$

Where $z=t_b/2+t_s$ for maximum strain.

Voltage generated by the sensor is

$$V^s(t) = G_s i(t) \quad (17)$$

Where G_s is the gain of the signal conditioning device

$$V^s(t) = G_s C_{11} d_{31} z b_b [0 \ \ -1 \ \ 0 \ \ 1][\dot{u}_1 \dot{\theta}_1 \dot{u}_2 \dot{\theta}_2]^T$$

(18)

This can be written as

$$V^s(t) = C_s[0 \ \ -1 \ \ 0 \ \ 1]\{\dot{q}\} \quad (19)$$

Where $C_s = G_s C_{11} d_{31} z b_b$ is sensor constant. The above equation can be written as

$$V^s(t) = \{g\}^T\{\dot{q}\} \quad (20)$$

Where {g} is a Constant Vector of size (4x1)

ACTUATOR EQUATION

From equation [12] the stress developed in the actuator is

$$\sigma_x = C_{11} d_{31} E_z \quad (21)$$

Where E_z is the Electric Field.

The resultant bending moment produced by the actuator is given by [10]

$$M_a = C_{11} d_{31}\left(\frac{t_a + t_b}{2}\right) V^a(t) \quad (22)$$

Where $V^a(t)$ is the voltage applied on the actuator which is given by

$$V^a(t) = Controller gain x V^s(t) \quad (23)$$

The force produced by the actuator is given by

$$\{F_{Actu}\} = C_{11} d_{31} b_b \left(\frac{t_a + t_b}{2}\right) \int_{l_a} V^a(t)\{n_1(x)\} dx$$

(24)

This can also be expressed as

$$\{F_{Actu}\} = \{H\} V^a(t) \quad (25)$$

Where {H} is a constant vector of size (4X1) and is given as

$$\{H\}^T = C_{11} d_{31} b_b \left(\frac{t_a + t_b}{2}\right)[-1 \ \ 0 \ \ 1 \ \ 0]$$

Or

$$\{H\}^T = C_a[-1 \ \ 0 \ \ 1 \ \ 0] \quad (26)$$

Where C_a is the Actuator Constant and is given by

$$C_a = C_{11} d_{31} b_b \left(\frac{t_a + t_b}{2}\right) \quad (27)$$

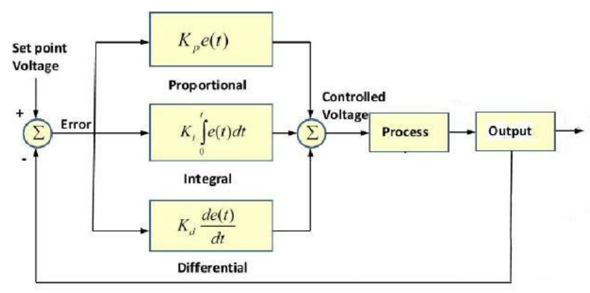
CONTROL LAW USING PID CONTROLLER

Now, calculating an appropriate controlled voltage that can be fed to the Actuator, for that a PID controller is used in this research. A typical PID control law that can be used for Active Vibration Control is:

$$y(t) = K_p + K_i \int e(t)dt + K_d \dot{e}(t) \quad (28)$$

Where y(t) is control signal, K_p, K_i and K_d are proportional, integral and derivative respectively. These three gains can be tuned in order to provide fine control for the application.

A typical PID controller shown below



DYNAMIC EQUATION OF SMART STRUCTURE

Now formulating and solving the equation of motion of entire structure that is give by

$$[M]\{\ddot{q}\} + [K]\{\dot{q}\} = \{F_{Dist}\} + \{F_{Actu}\} \qquad (29)$$

Consider a generalized coordinate using a transformation $\{q\} = [\psi]\{x\}$ for the first two dominant vibratory modes then the equation of motion becomes

$$[M^r]\{\ddot{x}\} + [K^r]\{x\} = \{F^r_{Dist}\} + \{F^r_{Actu}\} \qquad (30)$$

If damping of the structure is also considered then assuming proportional damping as

$$[C] = \alpha[M] + \beta[K] \qquad (31)$$

The generalized dynamic equation of motion is given as

$$[M^r]\{\ddot{x}\} + [C^r]\{\dot{x}\} + [K^r]\{x\} = \{F^r_{Dist}\} + \{F^r_{Actu}\}$$
(32)

STATE SPACE FORMULATION FOR THE FIRST TWO DOMINANT VIBRATION MODES

Let the {x}={y} as

$$\{x\} = \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \{y\} = \begin{bmatrix} y_1 \\ y_2 \end{bmatrix} \qquad (33)$$

And

$$\{\dot{x}\} = \{\dot{y}\} = \begin{bmatrix} \dot{y}_1 \\ \dot{y}_2 \end{bmatrix} = \begin{bmatrix} y_3 \\ y_4 \end{bmatrix} and \{\ddot{x}\} = \{\ddot{y}\} = \begin{bmatrix} \dot{y}_3 \\ \dot{y}_4 \end{bmatrix} \qquad (34)$$

Equation of motion now can be written as

$$[M^r]\begin{bmatrix} \dot{y}_3 \\ \dot{y}_4 \end{bmatrix} + [C^r]\begin{bmatrix} y_3 \\ y_4 \end{bmatrix} + [K^r]\begin{bmatrix} y_1 \\ y_2 \end{bmatrix} = \{F^r_{Dist}\} + \{F^r_{Actu}\} \qquad (35)$$

This can be simplified as

$$\begin{bmatrix} \dot{y}_3 \\ \dot{y}_4 \end{bmatrix} = -[M^r]^{-1}[K^r]\begin{bmatrix} y_1 \\ y_2 \end{bmatrix} - [M^r]^{-1}[C^r]\begin{bmatrix} y_3 \\ y_4 \end{bmatrix} + [M^r]^{-1}\{F^r_{Dist}\} + [M^r]^{-1}\{F^r_{Actu}\} \qquad (36)$$

The above equation can be written in state form as

$$\begin{bmatrix} \dot{y}_1 \\ \dot{y}_2 \\ \dot{y}_3 \\ \dot{y}_4 \end{bmatrix} = \begin{bmatrix} [0] & [I] \\ -[M^r]^{-1}[K^r] & -[M^r]^{-1}[C^r] \end{bmatrix}\begin{bmatrix} y_1 \\ y_2 \\ y_3 \\ y_4 \end{bmatrix} + \left[\frac{\{0\}}{[M^r]^{-1}\psi^T\{H\}}\right]V^a(t) + \left[\frac{\{0\}}{[M^r]^{-1}\psi^T\{f\}}\right]u(t) \qquad (37)$$

Now, the sensor voltage is taken as output of the structure which can be written as

$$V^s(t) = [\{0\}\{g\}^T[\psi]][y_1 y_2 \ y_3 \ y_4]^T \qquad (38)$$

So the state space model of smart structure for the first two dominant vibratory modes is given by

$$\{\dot{y}\} = [A]\{y(t)\} + [B]V^a(t) + [D]u(t)$$

And

$$V^s(t) = [E]^T\{y(t)\} + [F]V^a(t) \qquad (39)$$

Where

$$[A] = \begin{bmatrix} [0] & [I] \\ -[M^r]^{-1}[K^r] & -[M^r]^{-1}[C^r] \end{bmatrix}, [B] = \left[\frac{\{0\}}{[M^r]^{-1}\psi^T\{H\}}\right], [D] = \left[\frac{\{0\}}{[M^r]^{-1}\psi^T\{f\}}\right], [E] = \left[\frac{\{0\}}{\{g\}^T[\psi]}\right], and [F] = NullMatrix$$

THEORETICAL ANALYSIS

The mass and stiffness matrix is obtained, after applying boundary conditions and Guyan reduction given by,

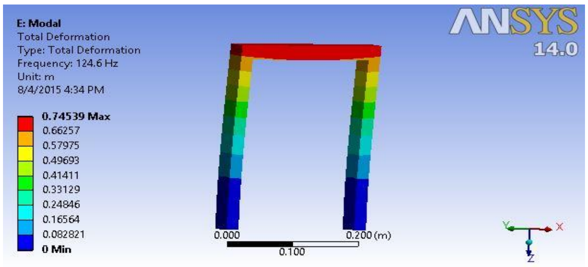
$$[M] = \begin{bmatrix} 1.0064 & 0 & 0.0224 & 0.2380 & 0 & 0 \\ 0 & 1.0064 & 0.0224 & 0 & 0.1836 & -0.0132 \\ 0.0224 & 0.0224 & 0.0024 & 0 & 0.0132 & -0.0009 \\ 0.2380 & 0 & 0 & 1.0064 & 0 & 0.0224 \\ 0 & 0.1836 & 0.0132 & 0 & 1.0064 & -0.0224 \\ 0 & -0.0312 & -0.0009 & 0.0224 & -0.0224 & 0.0224 \end{bmatrix}$$

$$[K] = \begin{bmatrix} 856.4 & 0 & 1.736 & -833.3 & 0 & 0 \\ 0 & 856.4 & 1.736 & 0 & -23.14 & 1.736 \\ 1.736 & 1.736 & 0.347 & 0 & -1.736 & 0.086 \\ -833.3 & 0 & 0 & 856.4 & 0 & 1.736 \\ 0 & -23.14 & -1.736 & 0 & 856.4 & -1.736 \\ 0 & 1.736 & 0.0868 & 1.736 & -1.736 & 0.347 \end{bmatrix}$$

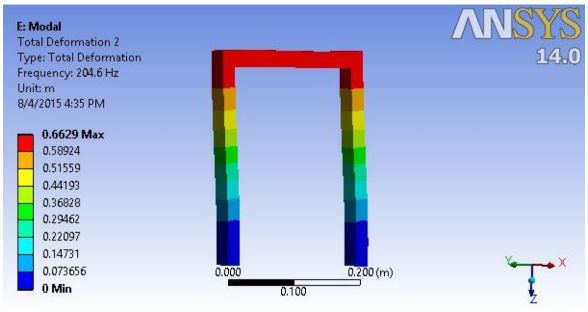
And solving $[\boldsymbol{M}]^{-1}[\boldsymbol{K}] = \boldsymbol{\omega}^2$for the natural frequencies, we get $\boldsymbol{\omega_1}, \boldsymbol{\omega_2}, \boldsymbol{\omega_3}, \boldsymbol{\omega_4}, \boldsymbol{\omega_5}, \boldsymbol{\omega_6}$.

MODELLING ANALYSIS

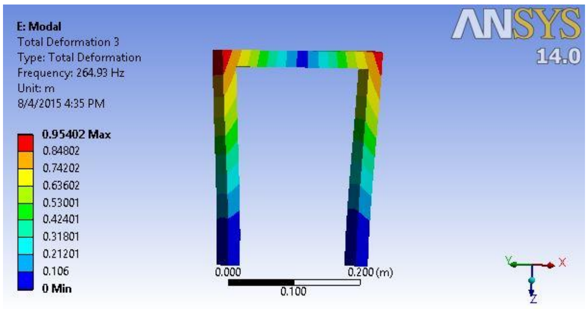
Modelling and analysis has been done on ANSYS Workbench 14 and following results are obtained.The result obtained by theoretical calculation is verified by modelling analysis result. After the verification, the basic six mode shapes are considered.



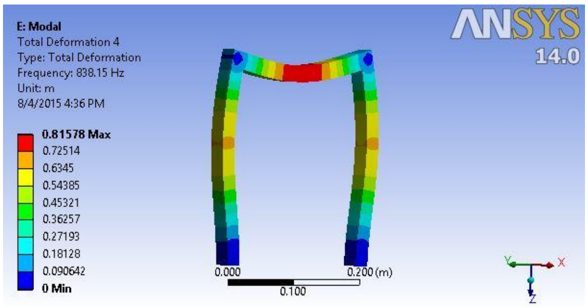
Mode shape-1



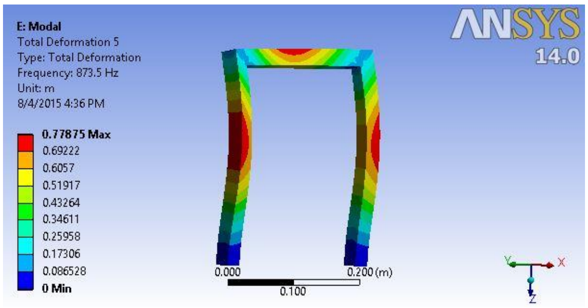
Mode shape-2



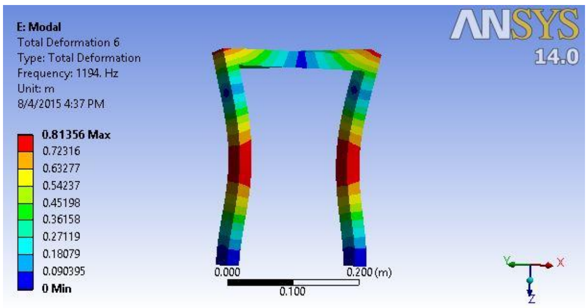
Mode shape-3



Mode shape-4



Mode shape-5



Mode shape-6

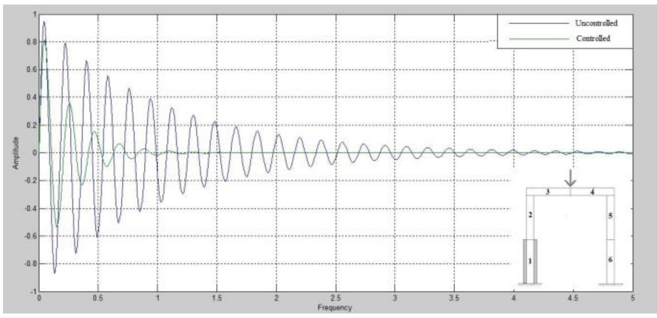
RESULT

Verification and comparison of result obtained by finite element method and ANSYS

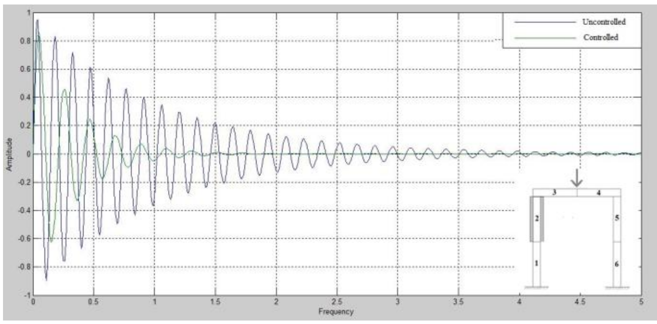
S.No.	Finite Element Analysis(Khz)	ANSYS(Khz)
1.	9.537	9.632
2.	5.193	5.568
3.	5.923	5.907
4.	2.969	2.952
5.	0.818	0.838
6.	1.380	1.259

Simulation Result of MATLAB(SIMULINK)

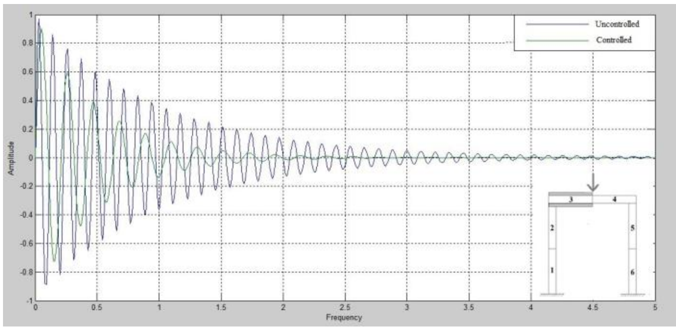
The portal frame is divided into six finite elements. Time response of the structure is studied after bonding the sensor actuator pair at different locations on the portal frame.A mid-point disturbance F_{dist} is applied at the junction of element 3 and element 4. The result is shown below



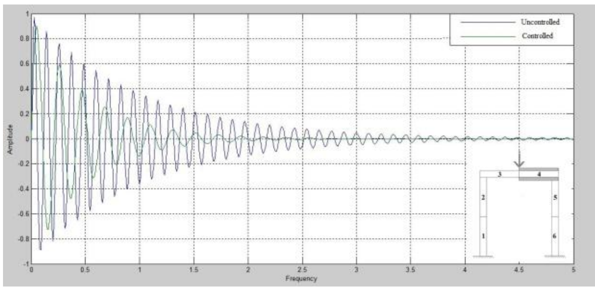
Sensor/Actuator pair is placed on element – 1



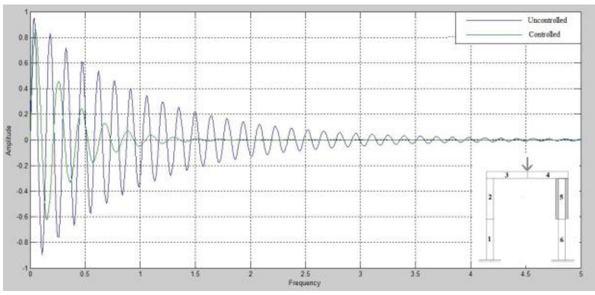
Sensor/Actuator pair is placed on element – 2



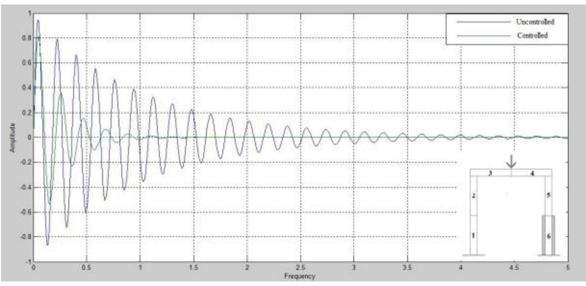
Sensor/Actuator pair is placed on element – 3



Sensor/Actuator pair is placed on element – 4



Sensor/Actuator pair is placed on element – 5



Sensor/Actuator pair is placed on element – 6

CONCLUSION

Present work deals with the mathematical formulation and the computational model for the active vibration control of a portal frame with piezoelectric smart structure. The responses are obtained for each of the models.. Here, the comparison and discussion of the simulation results of the vibration control for the smallest magnitude of the control effort required to control the vibrations of the smart portal frame is presented. Result show that, when the Sensor/Actuator pair is placed at the fixed end at element number 1 and element number 6 maximum control is obtained.

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