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MODELLING AND CONTROL OF A TWO DEGREE OF FREEDOM DYNAMIC ABSORBER USING SHAPE MEMORY ACTUATOR

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Active vibration control using shape memory actuator is utilised to make a two degree of freedom dynamic absorber and demonstrate its effectiveness in vibration suppression. A two degree of freedom spring-mass system interconnected by shape memory wires is designed and constructed. This two degree of freedom system is modelled using state variables in the discrete domain and the model is identified using autoregressive moving average technique. The system is controlled using a decoupling PID control method. The experimental results indicate effectiveness of shape memory wire for vibration suppression at lower frequencies and satisfactory performance of the control strategy.

1. INTRODUCTION:

In the field of passive vibration suppression, dynamic absorber is the simplest and cheapest of all designs. In a dynamic absorber, a primary single degree freedom system is made into a two degree of freedom system by the addition of a secondary mass - spring system. First of such dynamic absorber was developed by Frahm in 1909. By simple attachment of a spring - mass system which has the natural frequency equal to the operating frequency, the high amplitude of a vibrating body at resonance was made zero. Den Hartog (1956) added damping to this two degree of freedom dynamic absorber. This helped in widening the frequency bandwidth of operation , even though the amplitude of the primary system increased from zero.

The functioning of the dynamic absorber mainly depend upon three variables namely mass, damper and the springs and most of the subsequent design variations of the dynamic absorber can be classified under one of these variables. For example, Tanaka and Kikushima (1992) modulated the initial conditions of the damper mass with an initial displacement. Van Berkel and Semercigil (1991) added one more spring mass system to reduce the high amplitude of vibration of the secondary mass. Ying and Semercigil (1991) isolated the third mass by suitably designed clearance and allowed this mass to act as an impact damper. Roberson (1952), Pipes (1953) and . Bert and Egle (1990) improved the frequency bandwidth of the dynamic absorber by using a combination of linear and cubic or square springs for primary mass and secondary mass respectively.

Apart from all these attempts by modification of passive elements, use of active and semi active vibration control techniques added a possibility of eliminating frequency bandwidth limitation. Lee and Sinha (1986) proved optimal active vibration control for a single degree of freedom system. Walsh and Lamancusa (1992) applied active vibration control to make a variable stiffness vibration absorber. Huang and Lian (1994) used DC servo motors with active vibration control in the two degree of freedom dynamic absorber.

This paper describes active vibration control of a two degree of freedom dynamic absorber using shape memory actuator. Active vibration control is implemented based on the principle that the shape memory actuator can undergo contraction and elongation on command. This results in control of displacement and hence addition of variable stiffness into the secondary system. The two degree of freedom dynamic absorber with variable stiffness is modelled using state - space approach. The system identification is done by ARMA technique. The performance of the shape memory actuator is verified for a harmonic input using a PID controller. The experimental and theoretical results are compared. The results indicate shape memory wires are more useful for vibration suppression at lower frequencies.

1.1 SHAPE MEMORY ALLOY ACTUATOR:

When a shape memory alloy material is plastically deformed at one temperature, it remembers the previous shape and completely recovers it when heated to a higher temperature. If the recovery is restrained, a proportional force or displacement will be available for doing work. This principle is widely used for actuator applications. Shape memory alloy actuators are versatile and compact due to the simplicity of operation. The actuation is caused by simple heating and cooling of the shape memory alloy. Since there are no moving parts, there is no friction and related wear and tear.

The shape memory alloy can be used in the form of wire, spring, sheet etc. For shape memory wire, the shape changes are in the form of contraction and expansion causing a linear displacement. The shape memory wire used in the experiment of this paper is of 0.2 mm diameter. The specifications of the shape memory wire are given in Table - 1. The wire is stretched or allowed to elongate by way of natural cooling in the presence of reset force. In the present paper, linear compressive springs are used for applying this reset force and restraining contraction. Heating is done by passing current through the wire and shape memory wire contracts to its initial position.

ACTUATOR MATERIAL	Shape memory alloy-'NiTinol'
SHAPE & SIZE	Wire - 0.2 mm diameter.
TYPE OF ACTUATION	By contraction.
SIGNAL INPUT	610 mA max.
TRANSITION TEMP.	+ 90 C
MAXIMUM STROKE	7%
MAX. FORCE AT 4%STROKE	590 grams
MAX. CONTRACTION TIME	1 sec
MAX COOLING TIME	2.2 sec

TABLE - 1 PROPERTIES OF SHAPE MEMORY ACTUATOR

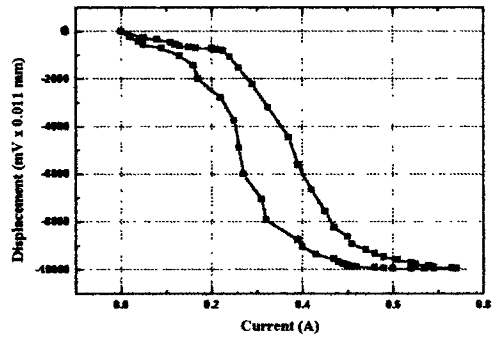

Fig. 1 HYSTERESIS CURVE FOR HEATING AND COOLING CYCLES.

The contraction and elongation of the wire is in the form of a hysteresis curve. The characteristic curve of the shape memory wire is evaluated by experiment and the characteristics are stored in the PC. The characteristic curve of the shape memory wire with respect to current is given in Fig.(1). The slope of the heating cycle is used as actuator gain 'K_s' to generate control responses in the experiment. For various inputs of current the shape memory wire undergo various levels of contraction thereby causing controlled actuation. For all actuations only heating is used due to better response time of shape memory wire in heating. Hence for reverting to lower actuation lengths, the current is always reset to zero and heating restarted. Since the response time of shape memory wire is of the order of 1 Hz only and the control signal is originated from the PC at a frequency of 1000 Hz, it is always possible to implement control in this method. However in the system level, poor response time of shape memory wire makes it suitable for low frequency applications only.

2. THEORETICAL MODELLING:

In order to describe the dynamic behavior of the dynamic absorber, it is considered as a linear continuos system and a lumped parameter state - space model generated. The schematic diagram of the set up is given in Fig. (2).

The primary and secondary masses are separated by springs and two separate shape memory wires are connecting them. The shape memory wire can generate harmonic oscillations for harmonic input signals. Both shape memory wires can be excited simultaneously or one at a time. Hence there can be two inputs and two outputs which represent a fourth order system with two inputs and two outputs. The equations representing such a MIMO continuos dynamic system require to be descretised for finding the system transfer functions. In order to simplify the complexity of calculations in the system identification, the system is first modelled for single input either to primary mass or secondary mass and the transfer functions are found out.

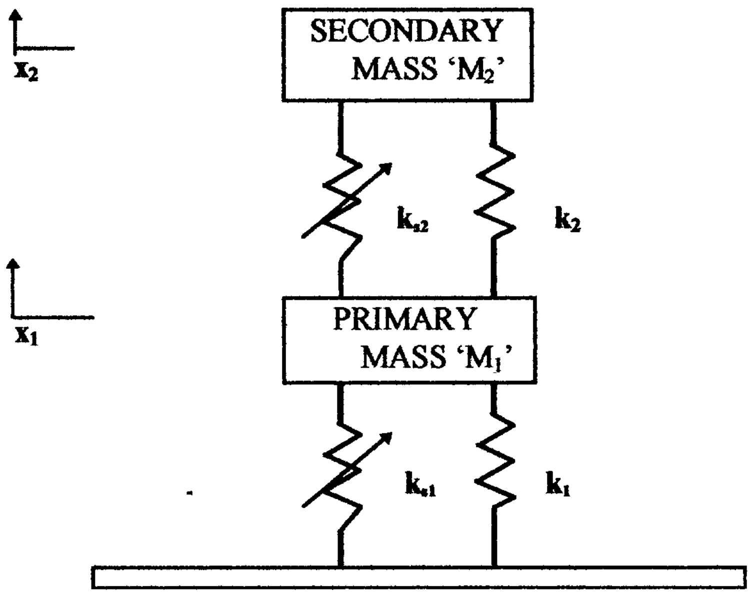

Fig. 2 Schematic diagram of dynamic absorber with shape memory actuator 'K_{s1}' and K_{s2}'

For primary input alone, the equations of motion can be written as,

$$M_1\ddot{x}_1 + k_1x_1 + (k_2 + k_{s2})(x_1 - x_2) = k_{s1}X_1 \quad (2.1)$$
$$M_2\ddot{x}_2 + k_2(x_2 - x_1) = 0 \quad (2.2)$$

Rewriting in terms of x_1 & x_2 and equating to the inputs,

$$M_1\ddot{x}_1 + (k_1 + k_2 + k_{s2})x_1 - (k_2 + k_{s2})x_2 = k_{s1}X_1 \quad (2.3)$$
$$M_2\ddot{x}_2 - k_2x_1 + k_2x_2 = 0 \quad (2.4)$$

Considering the system as time invariant, these equations can also be represented in the standard state - space form as,

$$\dot{x}(t) = A\,x(t) + B\,u(t) \quad (2.5)$$
$$y(t) = C\,x(t) + D\,u(t) \quad (2.6)$$

where, A = System matrix
B = Input matrix
C = Output matrix
D = Disturbance matrix
x = State Space Vector
u = Control input Vector

Assuming there is no disturbance in the system, $D\,u(t) = 0$ Eq. (2.6) will become,

$$y(t) = C\,x(t) \quad (2.6)$$

The state variables of this mechanical system can be defined as,

$$x_1 = x_1 \,;\; x_2 = \dot{x}_1 \,;\; x_3 = x_2 \,;\; x_4 = \dot{x}_2 \tag{2.7}$$

where x_1 = Primary Mass displacement
x_2 = Primary Mass velocity
x_3 = Secondary Mass displacement
x_4 = Secondary Mass Velocity

Substituting Eq.(2.7) in Eq. (2.3) and Eq. (2.4), and writing for the state variables $\dot{x}_2$ & $\dot{x}_4$ in terms of input variables X_1 and X_2, we obtain

$$\dot{x}_2 = -\frac{(k_1 + k_2 + k_{s2})}{M_1} x_1 + \frac{(k_2 + k_{s2})x_3}{M_1} + \frac{k_{s1}}{M_1} X_1 \tag{2.8}$$

$$\dot{x}_4 = \frac{k_2}{M_2} x_1 - \frac{k_2}{M_2} x_3 \tag{2.9}$$

Then the coefficients of state and output equations can be given in a matrix form,

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -(k_1 + k_2 + K_{s2})/M_1 & 0 & (k_2 + K_{s2})/M_1 & 0 \\ 0 & 0 & 0 & 1 \\ k_2/M_2 & 0 & k_2/M_2 & 0 \end{bmatrix} \tag{2.10}$$

$$B = \begin{bmatrix} 0 \\ K_{s1}/M_1 \\ 0 \\ 0 \end{bmatrix} \tag{2.11}$$

$$C = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \tag{2.12}$$

$$\dot{x} = [\dot{x}_1 \;\; \dot{x}_2 \;\; \dot{x}_3 \;\; \dot{x}_4]^T \qquad y = [x_1 \;\; x_2]^T \tag{2.13}$$

$$x = [x_1 \;\; x_2 \;\; x_3 \;\; x_4]^T \qquad u = [X_1 \;\; X_2]^T \tag{2.14}$$

For secondary input alone, the equations of motion can be written as,

$$M_1\ddot{x}_1 + k_1 x_1 + (k_2 + k_{s2})(x_1 - x_2) = 0 \tag{2.15}$$

$$M_2\ddot{x}_2 + k_2(x_2 - x_1) = k_{s2}X_2 \tag{2.16}$$

The state space equations 'for input to secondary mass condition' are as follows:

$$\dot{x}_2 = -\frac{(k_1 + k_2 + k_{s2})}{M_1} x_1 + \frac{(k_2 + k_{s2})}{M_1} x_3 \quad (2.17)$$

$$\dot{x}_4 = \frac{k_2}{M_2} x_1 - \frac{k_2}{M_2} x_3 + \frac{k_{s2}}{M_2} X_2 \quad (2.18)$$

For this condition, The coefficients A and C will remain same and the coefficient 'B' alone will change as,

$$B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ K_{s2} / M_2 \end{bmatrix} \quad (2.19)$$

From the coefficients A,B and C two direct transfer functions and two cross coupling transfer functions denoted by G_{ij} (where i = 1,2 represent input and j = 1,2 represent output) are found out. These transfer functions are used for simulating and response plotting of single input conditions.

3. EXPERIMENTAL ANALYSIS:

In order to evaluate the theoretical analysis, a two degree of freedom dynamic absorber is designed and constructed. The schematic diagram is given in Fig. (2). It consists of two platforms supported on compression springs. The platforms act as primary mass and secondary mass. Both are mounted on linear bearings sliding on two guide rods. The input to the system is generated by shape memory wire fitted between the rigid base and the primary mass. When current is passed through the shape memory wire in the form of a harmonic signal, the shape memory wire undergoes contraction and elongation, thereby exciting the primary mass. Similarly, another shape memory wire is fixed between the primary mass and secondary mass. Unlike the primary mass excitation wire (Exciter wire), control signal is passed through the secondary mass control wire (control wire) to modulate the displacement of the springs of secondary mass.

Experiments are conducted for the secondary and primary system independently. That is, single input is given to either of the masses in the form of shape memory wire actuation and outputs in the form of displacements of both primary and secondary masses are noted. We get two independent responses and two cross responses for two independent inputs. The displacements of the primary mass and secondary mass are measured by displacement transducers and data acquisition was done using PCL 812 card by a PC. These values represent the dynamic properties of this MIMO derived from single input responses. These data are analysed by ARMA model using MATLAB® to derive two independent transfer functions and two cross product transfer functions. The experimental results and theoretical simulation response plots obtained by state space technique are identical. The single input response plots are given in fig.(3).

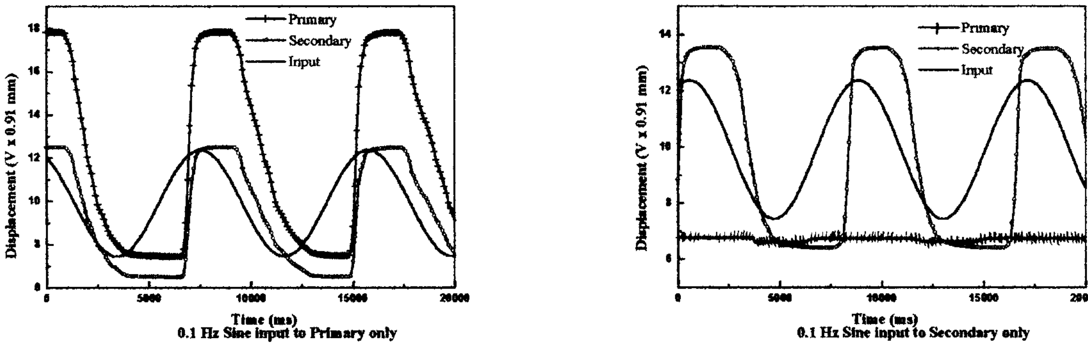

Fig.(3) Single input response plots of dynamic absorber with shape memory actuator

4. IMPLEMENTATION OF PID CONTROLLER:

Since it is intended to find the effectiveness of the shape memory actuator for active vibration control, a PID control is implemented for the secondary system. From the transfer functions derived by experimental and theoretical methods, the required PID controller at secondary mass is designed. The excitation is given in the form of harmonic signals of 0.1 Hz at the exciter wire resulting in harmonic displacement of primary mass. The corrective input is generated by PC using a software following PID algorithm and the control wire is actuated accordingly. The displacement data of both primary and secondary masses are acquired by PC and plotted. The response of the secondary mass with and without PID control are compared and given in the Fig.(4).

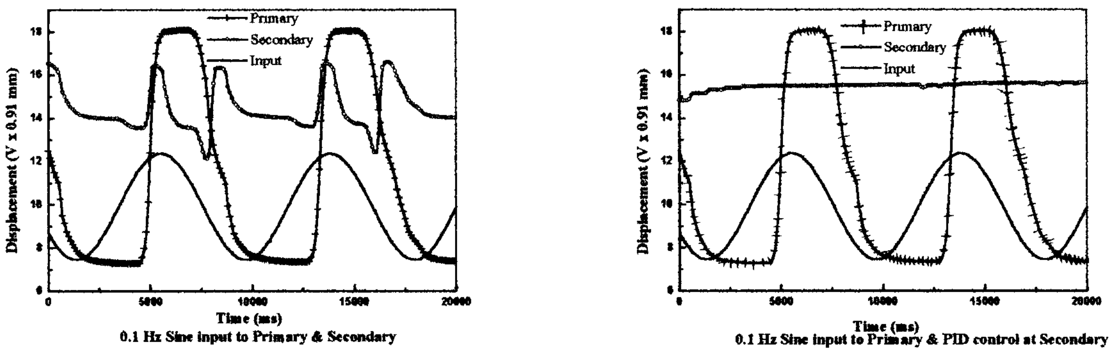

Fig.(4) Shape memory actuator responses for excitation of both primary and secondary masses

5. CONCLUSION:

Method of implementation of active vibration control is analysed by theoretical modelling and experimental testing for a dynamic vibration absorber system using shape memory actuator. The dynamic absorber is modelled using state - space variables. The system parameters are also identified experimentally and the decoupled transfer functions found by ARMA technique. Using the contraction and elongation of shape memory wire, actuation of primary mass and control of amplitude of the secondary mass system is accomplished. Due to the implementation of PID control, the amplitude of the secondary mass reduced and the results are plotted.

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