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Linear and Nonlinear Controller Contrast for the Vibration Isolation System utilizing "Smart Spring"

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Abstract
The analysis of smart spring vibration isolation system is analyzed through MATLAB simulation. The results show suppressing dominant vibration mode of platform. In the paper controllers developed for the smart spring are linear and nonlinear. This paper deals with MIMO system analysis of low frequency dominant modes, which are critical, examined first by linear controller techniques and then with nonlinear controller. In this paper, instead of introducing the nonlinear model of system, nonlinear controller is developed in a trivial way, to get the improvement in vibration isolation, catering the disturbances in the system caused by environment, causing the controller to stop working as desired; a nonlinear controller is developed step by step by introducing more and more nonlinearity to the controller. In the paper results show frequency response of system without controller, with linear controller and with non linear controller and finally comparison between the linear and non linear controller. The article shows the achievement of requirement at the expense of introduction of nonlinear controller and shows that nonlinear controller gives a substantial reduction in the magnitude of vibration at the required frequency.

I. Introduction

Vibration is an integral component of the system and mostly the vibrations transmitted to the foundation from any system that could be an engine, machinery or a building under earthquake which are unavoidable. The vibration produced in structures of any kinds, may become catastrophic. High amplitude vibrations produced are usually due to the resonance at the natural frequencies of the system. To isolate the system from vibration, the most important thing is to isolate the cause of the vibration from structure. Typically the problem is solved by mounting the system on a set of resilient elastomeric mounts. The passive isolation techniques could be used but the passive isolation can perform better at high frequencies, where as at low frequencies it is difficult to isolate the vibration. Thus it is always a compromise between the isolation performance and machinery alignment [1-3].
Due to limitations and problems raised in passive vibration system design, active control of vibration has wide scope in research activity for vibration isolation. The research is based on either active or semi-active schemes of vibrations. For example active suspension of electro pneumatic type in vehicle, active repetitive control using "Smart Spring", Instantaneous harmonic control for multivariable systems. Other examples include isolation in air craft**,** automobiles, wafer production, Naval Systems and seismic response reduction[4-8].
Due to degraded performance in the vibration reduction using passive controllers, an alternative active approach is required, which uses the "Smart Spring". Electromagnetic of machinery raft using the fully active control, was expensive and was discarded because of large number of actuators.

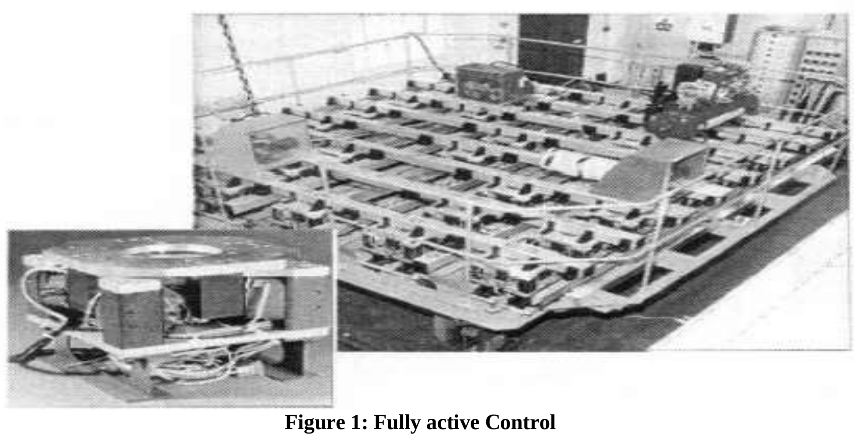

Figure 1: Fully active Control

In contrast to the fully active control BAE systems had developed a technology known as "Smart Spring" mounting system, which is hybrid active/passive solution and is also cost effective the fully active control [9].

II. Linear Controllers

The Linear controllers used in this research work for comparison purpose are Instantaneous harmonic controller, and velocity feedback controller [10]. The system used is shown in the figure1.

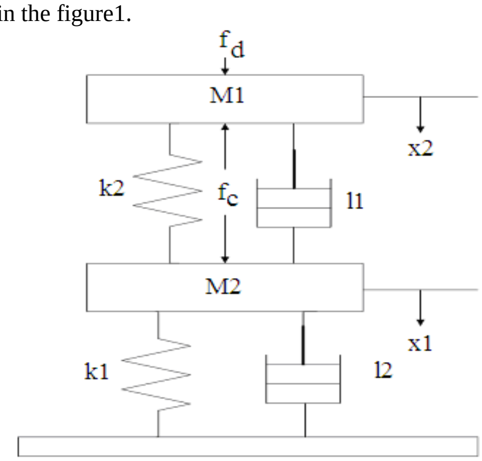

Figure 2: System Diagram

The main issue is to stop vibration in mass M2 due to f_d in mass M1, using controller force f_c as shown.

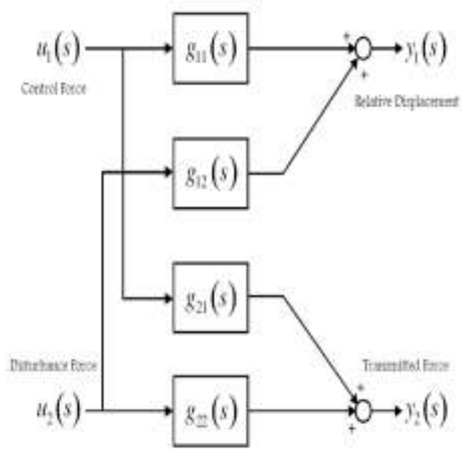

Figure 3: Block Diagram of System

The mass, stiffness and damping matrices are of the form:

$$M = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \quad \textbf{(1)}$$

$$L = \begin{bmatrix} l_1 + l_2 & -l_2 \\ -l_2 & l_2 \end{bmatrix} \quad \textbf{(2)}$$

$$K = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \quad \textbf{(3)}$$

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The system model is first converted to the state space form using the following state space representation:

$$\dot{\underline{x}} = A\underline{x} + B\underline{u} \qquad \textbf{(4)}$$

and

$$\underline{y} = C\underline{x} \qquad \textbf{(5)}$$

Since

$$M\ddot{\underline{q}} + C\dot{\underline{q}} + K\underline{q} = \underline{f} \qquad \textbf{(6)}$$

After rearranging equation (6)

$$\ddot{\underline{q}} = M^{-1}\underline{f} - M^{-1}C\dot{\underline{q}} - M^{-1}K\underline{q} \qquad \textbf{(7)}$$

And let $\dot{\underline{q}} = \dot{\underline{q}}$

So

$$\begin{bmatrix} \ddot{\underline{q}} \\ \dot{\underline{q}} \end{bmatrix} = \begin{bmatrix} -M^{-1}C & -M^{-1}K \\ I & 0 \end{bmatrix} \begin{bmatrix} \dot{\underline{q}} \\ \underline{q} \end{bmatrix} + \begin{bmatrix} M^{-1} \\ 0 \end{bmatrix} \underline{f} \qquad \textbf{(8)}$$

$$\underline{y} = \begin{bmatrix} 0 & I \end{bmatrix} \underline{x} \qquad \textbf{(9)}$$

Where $\underline{x} = \begin{bmatrix} \dot{\underline{q}} \\ \underline{q} \end{bmatrix}$ **(10)**

$$\underline{u} = \underline{f}$$

$$A = \begin{bmatrix} -M^{-1}C & -M^{-1}K \\ I & 0 \end{bmatrix} \qquad \textbf{(11)}$$

$$B = \begin{bmatrix} M^{-1} \\ 0 \end{bmatrix} \qquad \textbf{(12)}$$

$$C = \begin{bmatrix} 0 & I \end{bmatrix} \qquad \textbf{(13)}$$

First input is the disturbance force. Second input is the control force. First output is the relative displacement across the passive element; the second output is the transmitted force.

The response of Instantaneous Harmonic controller plotted below for the system is given by equation (14).

$$K(q) = \frac{2\beta\left(\mathrm{Re}\{C\} - 2\alpha\,\mathrm{Re}\{e^{-j\omega_\circ T}C\}q^{-1}\right)}{1 - 2\alpha\cos(\omega_\circ T)q^{-1} + \alpha^2 q^{-2}} \qquad \textbf{(14)}$$

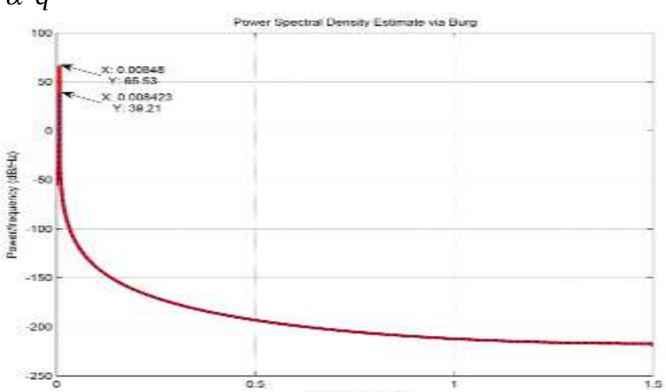

Figure 4: Comparison of uncontrolled and IHC.

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Feedback controller of simplest form is Velocity Feedback Controller of mathematical form.

$$u = k_1 \dot{x} \qquad \textbf{(15)}$$

Power spectral density of the system is shown in the figure below:

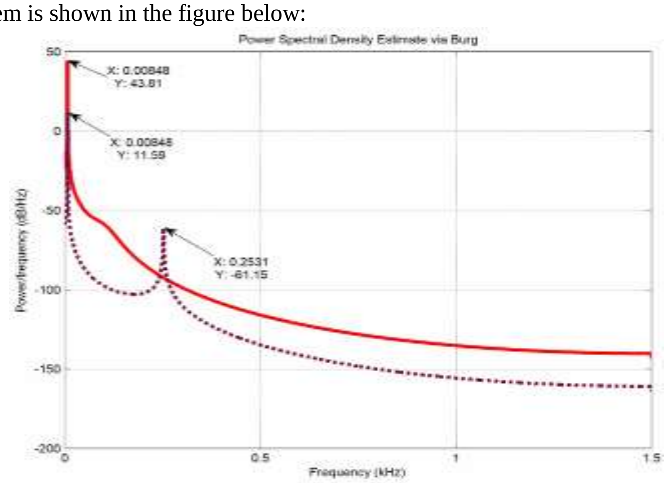

Figure 5: Velocity Feedback Controller

Designed feedback controller worked well at the resonance frequency of 8.4 Hz and at high frequencies, but showed perturbation at resonant frequency of the system i.e., 253.1 Hz.

III. Nonlinear Controller

The nonlinear controller can be used at any level of complexity. Nonlinear controller provides better response at the resonance frequency. The controller defined in literature is of mathematical form [11]:

$$u = k_1 \dot{x} + k_2 \dot{x}^2 + k_3 \dot{x}^3 \qquad \textbf{(16)}$$

To avoid complexities of the system behavior and analysis, controller used for this work is a velocity cube feedback controller of the form

$$u = k_1 \dot{x}^3 \qquad \textbf{(17)}$$

The output response of the system using velocity cube feedback controller is shown below:

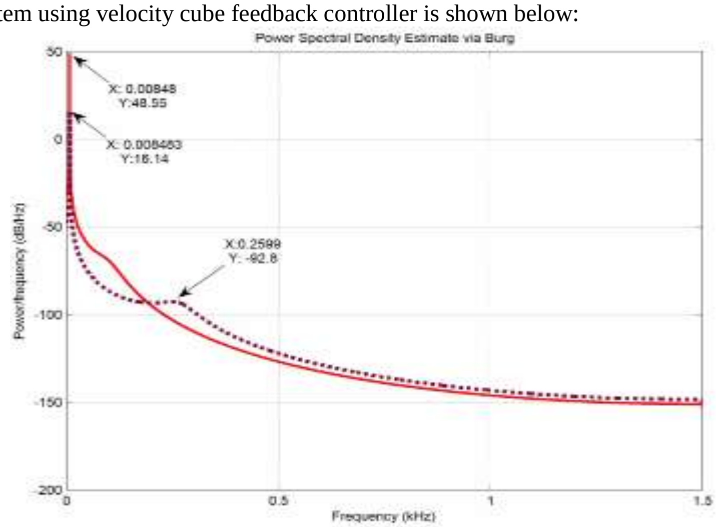

Figure 6: Velocity Cube Feedback

The solid line shows output of the system without implementation of the controller and dotted line shows power spectral density of the system after implementation of controller. At resonance frequency of 8.4Hz, the controller response shows reduction in magnitude from 48.55 dB to 15.14 dB and over high frequencies, response remains the same. The velocity cube feedback controller designed in state space has a gain of 0.2e16. The velocity cube feedback controller is a basic nonlinear controller and it provides a better insight to achieve desired results.

Nonlinear system identification using NARMAX (Nonlinear Auto Regressive Moving Average eXogenous) model [14,15] is used to identify nonlinear systems. Nonlinear terms are first fed into NARMAX model and then compared with the output from the system. All those terms which happen to appear in the data are kept while others are discarded. The designing of nonlinear controller is a reverse process of identification. Choose the basic terms first and implement the controller. The process continues by adding more terms unless a desired response is achieved. The velocity cube feedback controller works well in frequency domain for single degree of freedom and can be extended to multi degree of freedom.

The comparison of all controllers is shown below:

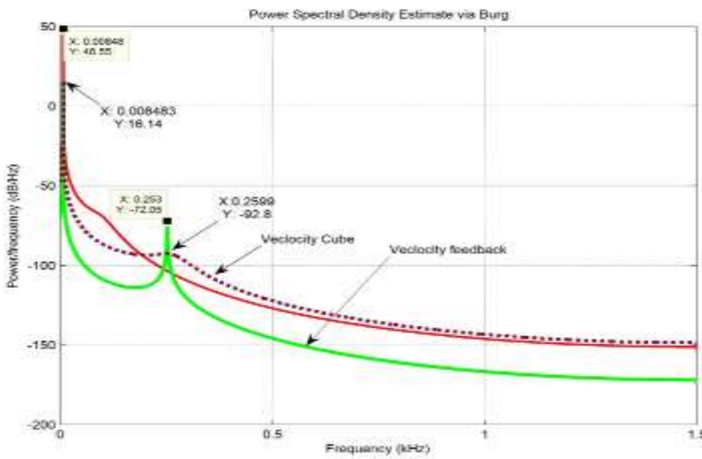

Figure 7: Comparison of controllers

Conclusion

The simulation results reveals that the nonlinear controller with the velocity feedback term provides the desired results at the low frequency and also the system does not get exited at the high frequencies. This work covers vibration isolation by comparing different types of linear controllers with nonlinear controller. The controllers can be designed to work with the random disturbances optimally. The instantaneous harmonic controller can be improved to work for not only the single tone disturbance but also for the random disturbances.

The nonlinear controllers can be explored in depth by adding more terms to attain the desired results. Although the complex nonlinear controllers can incur high computational cost; but for precision instruments, ships or submarines it can provide a good solution for vibration isolation.

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