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Vibration Isolation by Variable Stiffness and Damping

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***Abstract*—**Vibration isolation is a most widely used vibration protection method. The stiffness of vibration isolators in existing conventional type of vibration isolation system is usually of fixed value. This limits the system in exhibiting its vibration isolation effect is that, it has poor results for lower frequency vibration, especially for resonance frequency. Magneto-rheological elastomer is a new branch of Magneto-rheological materials. It's an intelligent materials in that it's shear modulus can be controlled by a magnetic field. It has wide application prospects in the vibration control area. This paper proposes using adjustable stiffness of magneto-rheological elastomer vibration isolation in vibration isolation system. By changing the current of vibration isolators coil to control the shear modulus of magneto-rheological elastomer, it can adjust the stiffness of the isolation system, making the system obtain wider vibration isolation frequency range. By exploying SimuLink software to analyze the vibration isolation system, it is found that such a design is effective and applicable.

Keyword- *Vibration Isolator, Stiffness,Damper,MR Fluid,Voigt Element*

I. INTRODUCTION

Vibration is the motion of a particle or a body or system of connected bodies displaced from a position of equilibrium. Most vibrations are undesirable in machines and structures because they produce increased stresses, energy losses, cause added wear, increase bearing loads, induce fatigue, create passenger discomfort in vehicles, and absorb energy from the system. Rotating machine parts need careful balancing in order to prevent damage from vibrations. Vibration occurs when a system is displaced from a position of stable equilibrium. The system tends to return to this equilibrium position under the action of restoring forces (such as the elastic forces, as for a mass attached to a spring, or gravitational forces, as for a simple pendulum). The system keeps moving back and forth across its position of equilibrium.

A. The Main Reasons of Vibration Are As Follows

- Unbalanced centrifugal force in the system. This is caused because of non-uniform material distribution in a rotating machine element
- Elastic nature of the system
- External nature of the system

Winds may cause vibrations of certain system such as electricity lines, telephone lines

B. Elementary Parts of Vibrating Systems

A vibratory system, in general, includes a means for storing potential energy (spring or elasticity), a means for storing kinetic energy (mass or inertia), and a means by which energy is gradually lost (damper). The vibration of a system involves the transfer of its potential energy to kinetic energy and of kinetic energy to potential energy, alternately. If the system is damped, some energy is dissipated in each cycle of vibration and must be replaced by an external source if a state of steady vibration is to be maintained. As an example, consider the vibration of the simple pendulum shown in Fig.1.

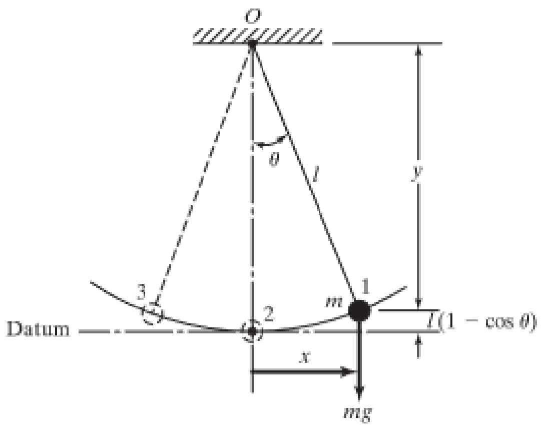

Figure 1: A Simple Pendulum

Let the bob of mass m be released after being given an angular displacement At position 1 the velocity of the bob and hence its kinetic energy is zero. But it has a potential energy of magnitude with respect to the datum position 2. Since the gravitational force mg induces a torque about the point O, the bob starts swinging to the left from position 1. This gives the bob certain angular acceleration in the clockwise direction, and by the time it reaches position 2, all of its potential energy will be converted into kinetic energy. Hence the bob will not stop in position 2 but will continue to swing to position 3. However, as it passes the mean position 2, a counter-clockwise torque due to gravity starts acting on the bob and causes the bob to decelerate. The velocity of the bob reduces to zero at the left extreme position. By this time, all the kinetic energy of the bob will be converted to potential energy. Again due to the gravity torque, the bob continues to attain a counter-clockwise velocity. Hence the bob starts swinging back with progressively increasing velocity and passes the mean position again. This process keeps repeating, and the pendulum will have oscillatory motion. However, in practice, the magnitude of oscil- lation gradually decreases and the pendulum ultimately stops due to the resistance (damping) offered by the surrounding medium (air). This means that some energy is dissipated in each cycle of vibration due to damping by the air.

II. VARIABLE STIFFNESS CONCEPT

The variable stiffness mechanism concept is shown in Fig 11. The Lever arm OA, of length L, is pinned at a fixed point O and free to rotate about O. The spring AB is pinned to the lever arm at A and is free to rotate about A. The other end E of the spring is free to translate horizontally as shown by the double headed arrow. Without loss of generality, the external force F is assumed to act vertically downwards at a point A. H is the height of the fixed point O from the ground. d is the horizontal distance of E from O. The idea is to vary the overall stiffness of the system by adjusting d accordingly. This is achieved through the control force u which will be designed in a subsequent section of this paper. Let k and l0 be the spring constant and the free length of the spring AE respectively, and Δ the vertical displacement of the point A. The overall free length Δ0 of the mechanism is defined as the value of Δ when no external force is acting on the mechanism.

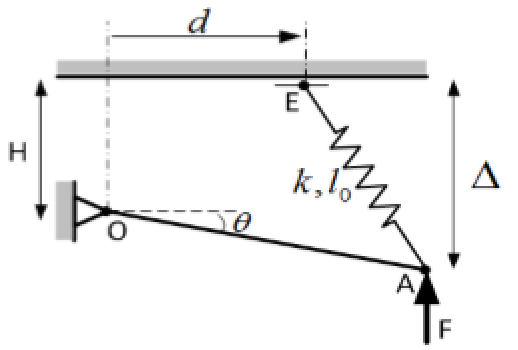

Figure 2: Variable Stiffness Mechanism

III. INTRODUCTION TO VARIABLE STIFFNESS AND DAMPING SYSTEM

In recent years, vibration isolation systems have been studied broadly and in great depth. The vibration control systems can be categorized as: passive, active and semi-active. Semi-active control systems fill the gap between passive and active control system and they represent a compromise between performance improvement and simplicity of implementation. They only expend a small amount of energy to change system parameters, such as damping and stiffness. The basic idea of variable damping systems have been proposed by many researchers to provide effective vibration control However, there is still room for further improvement because variable spring stiffness systems have not been thoroughly investigated in terms of their practical implementation, despite the fact that vibration systems with variable stiffness control were proposed by a few researchers. Kobori proposed a variable stiffness system to suppress buildings' response to earthquakes The aim of Kobori's work was to achieve a non-stationary and non-resonant state during earthquakes. Youn and Hac used an air spring in a suspension system to vary the stiffness among three discrete values The stiffness was changed only when the required control force could not be generated by variable damping alone. A vehicle system with variable stiffness demonstrated a good performance compared to a semi-active system with variable damping and fixed stiffness. However, conventional implementation of variable stiffness device is complicated. On the other hand, the variable damping can be easily produced by a controllable damper, such as a fluid damper with variable orifices or a magnetorheological (MR) damper The authors of this paper have proposed a structure using two Voigt elements (each one composed of a controllable damper and a constant spring) in series to realize variable stiffness and damping. In the system, the stiffness could be changed easily by damper. The proposed structure was experimentally implemented using two MR fluid dampers. The sinusoidal and random responses of one degree-of-freedom (DOF) and 2-DOF systems showed that the proposed damping and stiffness on-off control system using MR fluid dampers exhibited good vibration isolation performance However, because two controllable dampers were installed in series in the previous system, the damping and stiffness could not be changed independently. In this paper, a new variable stiffness and variable damping system in which the stiffness and damping can be independently and easily controlled is proposed. The responses of the proposed systems to the sinusoidal and random excitations are studied in numerical simulations and experiments.

A. Mechanical Structure

A new model of one-degree-of-freedom (1-DOF) vibration isolation system with two controllable dampers (damper 1 and damper 2 corresponding damping coefficients of c1 and c2) and two springs (spring 1 and spring 2 corresponding stiffnesses of k1 and k2) shown in Fig. 4 (a) is proposed. Damper 2 and spring 2 comprise a Voigt element. The Voigt element and spring 1 are in series. The stiffness values of the two springs are constant, however, the effective stiffness of the net system can be varied by the controllable damper 2. If the damping coefficient of damper 2 is small enough, the total system stiffness approaches the series stiffnesses of spring 1 and 2. However, if the damping coefficient of damper 2 is large enough, the total stiffness approaches the stiffness of spring 1. The damper 1 provides variable damping for the system.

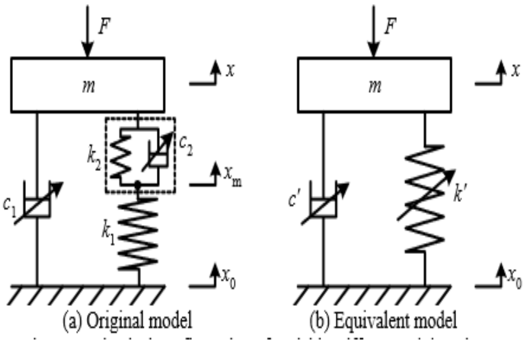

Figure 3: Mechanical Configuration Of Variable Stiffness And Damping

B. Equations of Motion

In Figure 3(a), F is an excitation force, x0, x and xm are displacements of base, mass m and the point between the Voigt element and spring 1, respectively. In the case of a vehicle suspension, x0 corresponds to the road bumpiness and F is produced by engine vibration. Figure. 3(b) shows the equivalent model of the system. Here k0 and c0 are equivalent stiffness and damping coefficient, respectively.The equations of motion for the system

$$m\dot{x} = -k_2(x - x_m) - c_2(\dot{x} - \dot{x}m) - c_1(\dot{x} - \dot{x}o) - f,$$

$$k_1(x_m - x_0) = k_2(x - x_m) + c_2(\dot{x} - \dot{x}m)$$

C. On– off Control Algorithms

The on–off control algorithm of damper 1 uses the sign of the absolute velocity and the relative velocity [1].

The force fd1 generated by damper 1 is

$$f_{d_1} = \begin{cases} -c_1\ on\ (\dot{x} - \dot{x}m)\ if\ \dot{x}(\dot{x} - \dot{x}o) \geq 0 \\ -c_1\ off\ (\dot{x} - \dot{x}m)\ if\ (\dot{x} - \dot{x}o) < 0 \end{cases}$$

where the damping coefficient c1 is equal to c1on in the on-state and c1off in the off-state. The control algorithm

for damper 2 uses the sign of _ xðx _ x0Þ [7]. The force fd2 exerted by damper 2 is

$$f_{d_2} = \begin{cases} -c_2\ on\ (\dot{x} - \dot{x}m)\ if\ \dot{x}(\dot{x} - \dot{x}o) \geq 0 \\ -c_2\ off\ (\dot{x} - \dot{x}m)\ if\ (\dot{x} - \dot{x}o) < 0 \end{cases}$$

CONCLUSION

A new variable stiffness and damping system configuration using two controllable dampers was proposed. Since the stiffness is controlled by changing the damping coefficient, this system is very simple and easy to apply in practical systems.

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