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The Utilisation of an Electric Field Sensitive Effect for Vibration Control of a Viscous Damped Isolation System

[1]E.C. Obinabo and [2]O. Ighodalo
[1]Department of Electrical and Electronic Engineering,
[2]Department of Mechanical Engineering, Ambrose Alli University,
P.M.B. 14, Ekpoma, Edo State, Nigeria

Abstract: This study describes a computer simulation approach to utilization of an Electric Field Sensitive (EFS) effect for vibration control of a viscous damped isolation system. The EFS effect was derived from a liquid whose properties were similar to those of a Bingham plastic with intensities of the applied electric field determining the fluidity constant and the yield stress. Vibration attenuation was due essentially to modulation of resistance to motion imparted to the liquid by subjecting it to a transverse electric field. Our results indicate that a significant degree of control is achievable for a wide range of the sprung mass.

Key words: Damped isolation system, digital computer simulation, electric field sensitive fluid, vibration attenuation

INTRODUCTION

The classical approach to simulation of viscous damped isolation systems (Golob and Tovornik, 2003) has been extended to demonstrate the resulting benefits of reduced vibration levels over a wide frequency range. It has been shown (Cao *et al*., 2000; Beards and Neroutsopoulos, 1980; Kim and Kim, 1994; Earles and Williams, 1972) that vibration may be available from other sources such as interface friction and it occurs in all machines and structures where parts are joined in a manner that permit small relative displacements between components such as riveted or bolted joints and press fits.

Dampers, absorbers and isolators of various types have been used to control and reduce vibrations in machine suspensions. Several machine suspensions make use of viscous fluid dampers (Cao *et al*., 2000; Cavailo *et al*., 1999) in which a flywheel is mounted within a hermetically sealed case and the case filled with a high viscosity fluid, usually silicone. Damping is achieved by the drag of the fluid film separating each of the two parts. This type of damper will equally attenuate all orders of vibration, and is very effective where several critical speeds lie within the operating range.

Controllable dampers and vibration isolators for a variety of applications such as electromagnetic dampers (Mizuno and Higuchi, 1984; Jayawant, 1976; Nikolajsen *et al*., 1979) have also been reported for active control of rotor-bearing systems and transmission shafts. In such an application, it is well known that when a rolling element in the bearing encounters a race defect, normally a small pit or crack, there will be an impulsive change of load on the system locally. This will cause a transient vibration to propagate. If the defect is on the inner race of the bearing, the pulse will initiate vibration of the rolling element and so be transmitted to the outer ring and to the bearing housing. Exactly what happens at impact is not known but if the defect is small the impulse is short and a transient will exist for several cycles.

Associated with these dampers are electromagnetic bearings (Golob and Tovornik, 2003; Schweitzer and Lange, 1976) which can support high rotor weights and exert specific bearing forces. Depending on the operating control loop, these electromagnetic forces have been used (Ellis and Mote, 1979) as elastic or as damping forces or as an arbitrary combination of both. Although the electromagnetic actuator was considered fascinating for vibration control and offers useful applications in various areas, it is argued (Schweitzer and Lange, 1976; Mizuno and Higuchi, 1984) that for industrial applications it has to compete with more conventional solutions. The electromagnetic damper needs space (Kim and Kim, 1994). If a rotor is supported on ball bearings, it would be possible to choose also such a type for the additional bearing which, in the case of journal bearings will reduce imbalance in vibration and increase the instability onset speed especially if it is mounted in such a way that its static load is sufficiently large. To this end, it has been shown (Obinabo, 1981; Bullough and Stringer, 1973) that a squeeze film damper utilizing EFS fluid as an operating medium could be more effective because of the great extent to which the EFS fluid can

Corresponding Author: E.C. Obinabo, Department of Electrical and Electronic Engineering, Ambrose Alli University, P.M.B. 14, Ekpoma, Edo State, Nigeria

be energized through the application of an electric field to cope with a wide range of vibrational forces occurring at different frequencies without special premium in space requirement for installation.

The purpose of the study was to explore the usefulness, from computer simulation point of view, of the controllable, variable effect, nonlinear characteristics of an EFS shock absorber in a vibration isolation system. By considering these features, the investigation was channeled towards achieving a steady state response of the system dynamics to varying forcing functions, in the shortest possible length of time.

MATERIALS AND METHODS

Vibration isolation systems capable of maintaining the design characteristics over a wide range of loading conditions have been reported (Obinabo, 1981; Bullough and Stringer, 1973) to incorporate essentially electric field sensitive damping. This concept as applied to fluid power engineering results from an instantaneous reversible change in apparent viscosity when the EFS fluid is subjected to an externally applied electric field. Fluids whose viscosity can be varied by the application of external voltage have received considerable attention in the past several decades (Shoureshi *et al*., 1999; Obinabo, 1996; Honda *et al*., 1978) because they exhibit such functionalities as shock or impact absorption and drive transmissibility. This effect (Tokhi and Veres, 2002; Honda *et al*., 1979; Yang *et al*., 2001) is a manifestation of an increase in the tangential force existing between the EFS liquid and the energized electrodes.

Various EFS fluids have been studied (Sievers and Flotow, 1988; Honda and Sasada, 1977) and classified by the dispersions of porous inorganic particles such as silica, alumina and talc in an electrically insulating fluid. Obinabo (1981) considered a mixture of equal parts of paraffin and silica gel powder to which sufficient water was added to give the mixture the desired resistivity. When the constituted EFS fluid (Obinabo, 1981) was introduced into a shock absorber housing, the fibers formed on application of an electric field enabled the fluid to sustain sufficient tension to increase the drag on the motion of the piston through it. The choice of silica is justified in this research by the fact that it is easily obtained on an industrial scale here in Nigeria, and is highly susceptible to improvement and manipulation.

Mount models incorporating EFS damping are categorized into two in which the EFS fluids are subjected to stresses either by pure shearing or by displacement. A combination of both the shear and flow models have also be used to provide the damping force. The flow mode performance (Klass and Martinek, 1967a) is generally known to be analogous to the shear mode; essentially the fluid is Newtonian with its fluidity unaffected by the applied field (Klass and Martinek, 1967b) except for an induced yield point, in pressure terms, which the forcing pressure must exceed in order that flow will commence. In motor vehicle suspensions where weight and space are at special premium, it is proposed (Obinabo, 1981) that the EFS effect be utilized in a flow mode by using a valve device where the EFS liquid is made to flow between two stationary electrified plates insulated from one another. If all losses such as leakage, pressure differential outside the valve and the inertia/compressibility ratio associated with the liquid flow are small compared with the valve pressure drop, then, the differential pressure across the piston will be the same as that over the controlling valve. The piston thus acts as a force magnifier for the valve. This principle with its potential for phased operation of modules has been shown (Bullough and Stringer, 1973) to lend itself to flexible and economical vibration systems and is explained in this paper by means of a simple simulated model in which the effects of input excitations were demonstrated for various sprung masses.

The research model: The simulated model is a novel approach to the problem of vibration control of a nonlinear viscous damped isolation system. This is an on-going research interest here at Ekpoma. Nigeria, in the use of a modulation effect applied to an electric-field sensitive damper for vibration control system design. The model was built up by a test mount which was traded with a compact mass, the down-side end of the mount being excited by a shaker which was controlled to generate the input excitation. The model was installed on a support or foundation of stiffness K_1 and consists of a damper of mass M_2 connected to the machine of mass M_1 by a linear spring of stiffness K_2 , linear viscous damper of damping coefficient C_1 and a nonlinear electric field sensitive damper all of which were in parallel with one another. The damping system utilized an electric field to energize the viscous fluid so that the performance was grossly enhanced.

Prior to application of the electric field, the damping rate was associated with the normal viscosity of fluid. However, by applying a control voltage at the instant the shock occurred, the slurrified EFS fluid was suddenly stiffened. When the machine suspension M_1 was displaced by the amount x_1 due to the applied external force P(t) , the damper of mass M_2 moved through a displacement x_2. Now, by allowing the masses of the system M_1 and M_2 to be accelerated by the input force P(t) and by applying Newton's second law of motion (Thompson, 1970; Yang *et al*., 2001) the accelerating force on the machine M_1 was $M_1 \ddot{x}_1$ and that on the damper of mass M_2 was $M_2 \ddot{x}_2$. Subsequently, the equations of motion for the system was derived as:

$$\begin{aligned} M_1\ddot{x}_1 &= -K_2x_1 - K_1x_1 - C_2\dot{x}_1 + K_2x_2 + C_2\dot{x}_2 + F \\ &= C_2\left(\dot{x}_2 - \dot{x}_1\right) + K_2\left(x_2 - x_1\right) - K_1x_1 + F \end{aligned} \quad (1)$$

and

$$M_2\ddot{x}_2 = -C_2\left(\dot{x}_2 - \dot{x}_1\right) - K_2\left(x_2 - x_1\right) \quad (2)$$

where, Kx is the restoring force due to the spring stiffness tending to accelerate the mass. K has the unit Nm^{-1}. The function $C\dot{x}$ is the viscous frictional force tending to retard the mass. C is the viscous frictional constant. The force F represents the algebraic sum of the forces exerting on the system and was expressed as:

$$F = P(t) + f_{EFS} \quad (3)$$

where f_{EFS} is the force induced in the system by the electric field sensitive damper. The equations of motion (1) and (2) of the viscously damped system was written in matrix notation as:

$$M\underline{\ddot{x}} + C\underline{\dot{x}} + K\underline{x} = \underline{F} \quad (4)$$

where M is the mass matrix, C is the damping matrix, K is the stiffness matrix, is the $\underline{x}$ displacement vector (column vector) and $\underline{F}$ is the force matrix (column vector). Equation (4) is identical to the response of a damped second order differential system represented generally by:

$$\ddot{\theta}_o + 2\zeta\omega\dot{\theta}_o = \omega^2\theta_o = \omega^2\theta_i \quad (5)$$

where, θ_i is the input function and θ_o the output function. From (4) we obtain:

$$\ddot{x} + \frac{C}{M}\dot{x} + \frac{K}{M}x = \frac{F}{M} \quad (6)$$

and comparing with (5) gave:

$$\frac{K}{M} = \omega^2$$

or $\left(K - \omega^2 M\right) = 0$ (7)

and $\dfrac{C}{M} = 2\zeta\omega$

or

$$\left(C - 2\zeta\omega M\right) = 0 \quad (8)$$

The vector solution to the problem (1) and (2) was given by:

$$\left(K - \omega_i^{\,2} M\right)\underline{R}_i = 0 \quad (9)$$

and $\left(C - 2\varsigma\omega_i M\right)\underline{R}_i$ (10)

where, the scalar quantities $\omega_i^{\,2}$ are the eigenvalues - natural frequencies of oscillation of the system, and the column vector $\underline{R}_i$ are the eigenvectors - normal modes associated with the eigenvalues. Pre-multiplying (9) and (10) by $\underline{R}_i^{\,T}$ yields:

$$\underline{R}_i^{\,T} K \underline{R}_i = \underline{\underline{R}_i^{\,2}}\omega_i^{\,2} M \underline{R}_i \quad (11)$$

and $\underline{R}_i^{\,T} C \underline{R}_i = \underline{R}_i^{\,T} 2\zeta\omega_i M \underline{R}_i$ (12)

Now if we let the vibration displacement of the machine suspension due to an input excitation to be defined as:

$$x = \sum \underline{R}_i r_i(t) \quad (13)$$

where, $r_i(t)$ is the function representing the displacement and $\underline{R}_i$ the position vector of the displacement relative to the origin, it was shown (Obinabo, 1981) that the relative motion between the damper and the machine upon excitation is:

$$x_{\mathrm{Re}l} = x_2 - x_1 \quad (14)$$

Substituting from (13) gave:

$$x_{\mathrm{Re}l} = \left[\sum_1^2 \underline{R}_i r_i(t)\right]_2 - x_1 \quad (15)$$

The measured value of vibration displacement $x_{\mathrm{Re}l}$ on the machine represents the relative displacement between the machine of mass M_1 and the damper of mass M_2. Using this value, it was possible to compute the actual values of x_1 and x_2 for a given input excitation function so that the effect of the electric field sensitive damping on the vibrating machine could be observed.

We now characterize the EFS effect by establishing a relationship between the effect of the induced EFS damper in the system and that due to an input excitation.

From (13) we substitute for into the matrix Eq. (4) to obtain:

$$\sum_{i=1}^{2} M\underline{R}_i\ddot{r}_i(t) + \sum_{i=1}^{2} C\underline{R}_i\dot{r}_i(t) + \sum_{i=1}^{2} K\underline{R}_i r_i(t) = \underline{F} \quad (16)$$

We define:

$$\underline{R}_i^T M \underline{R}_j = \delta_{ij} \quad (17)$$

where δ_{ij} is Kronecker delta defined as follows:

$$\delta_{ij} = 1 \text{, when } i = j$$
$$\delta_{ij} = 0 \text{, when } i \neq j \quad (18)$$

Then from (11) and (12) we write:

$$\begin{aligned} \underline{R}_i^T K \underline{R}_j &= \omega_i^2 \underline{R}_i^T M \underline{R}_j \\ &= \omega_i^2 \delta_{ij} \end{aligned} \quad (19)$$

and

$$\begin{aligned} \underline{R}_i^T C \underline{R}_j &= 2\omega_i \zeta_i \underline{R}_i^T M \underline{R}_j \\ &= 2\omega_i \zeta_i \delta_{ij} \end{aligned} \quad (20)$$

Pre-multiplying (16) by $\underline{R}_j^T$ gave:

$$\sum_{i=1}^{2} \underline{R}_j^T M \underline{R}_i \ddot{r}_i(t) + \sum_{i=1}^{2} \underline{R}_j^T M \underline{R}_i \dot{r}_i(t) + \sum_{i=1}^{2} \underline{R}_j K \underline{R}_i r_i(t) = \underline{R}_j^T F \quad (21)$$

Substituting from (17), (19) and (20) reduced (21) to:

$$\sum_{i=1}^{2} \left\{ \delta_{ij} r_j(t) + 2\zeta_i \omega_i \delta_{ij} \dot{r}_j(t) + \omega_j^2 \delta_{ij} r_j(t) \right\} = \underline{R}_j^T F \quad (22)$$

When $i = j$, that is, $\delta_{ij} = 1$ Eq. (22) becomes:

$$\begin{aligned} \sum_{j=1}^{2} \left\{ \ddot{r}_j(t) + 2\zeta_j \omega_j \dot{r}_j(t) + \omega_j^2 r_j(t) \right\} &= \underline{R}_j^T F \\ &= \underline{R}_j^T P(t) + \underline{R}_j^T f_{EFS} \end{aligned} \quad (23)$$

The developed model (23) is nonlinear due to the characteristics of the EFS damper. The nature of this nonlinearity is represented by a dead zone whereby the EFS fluid continues to offer resistance to motion inside it up to a point when the induced EFS force begins to bear proportional relationship with the motion. This effect has been demonstrated in a control loop (Obinabo, 1981) in which the vibration displacement signal x due to the input excitation force P(t) was differentiated and the resulting velocity signal $\dot{x}$ was fed to a power amplifier to generate a current I. This current then became converted by means of the EFS effect into a force which opposes the vibration velocity $\dot{x}$ of the machine suspension.

Table 1: Numerical values of the system parameters when the applied control voltage was zero

	M	K_1	K_2	C_1	C_2	f_{EFS}
I	1	25	75	6	4	0
II	2	25	75	6	4	0
III	3	25	75	6	4	0

Table 2: Numerical values of the system parameters when the applied control voltage was adjusted as the load was increased

	M	K_1	K_2	C_1	C_2	f_{EFS}
I	1	75	25	6	4	0
II	2	75	25	6	4	3
III	3	75	25	6	4	6

Table 3: Numerical values of the system parameters when the values of K_1 and K_2 were swapped and the input excitation halved (= 0.5 units), all other parameters remaining the same

	M	K_1	K_2	C_1	C_2	f_{EFS}
I	1	25	75	6	4	0
II	2	25	75	6	4	3
III	3	25	75	6	4	6

Table 4: The effect of controlled system with , and the condition shown in Table 1 retained unchanged

	M	K_1	K_2	C_1	C_2	f_{EFS}
I	1	25	75	0	10	0
II	2	25	75	0	10	3
III	3	25	75	0	10	6

RESULTS AND DISCUSSION

A digital computer program using real time computing and interactive graphics facilities was developed to compute the response of the EFS mount model to various input excitations. The program was structured to convert the time variable into the phase angle of the input function, and from knowledge of the displacement and vibration velocity of the mass, the forces acting on the body were determined. Integration of the equations of motion was performed by the Fourth Order Runge-Kutta method (Obinabo, 1981). In these calculations, steady state conditions of the vibrating system were of significant interest, and the response to the input functions gave the values of the vibration amplitudes at steady state. The major problem of concern was the response of the system to step and ramp input functions. The system responded reasonably well for the masses of 1.0, 2.0 and 3.0 units. An increase in the value of the damping ratio was considered significant in the

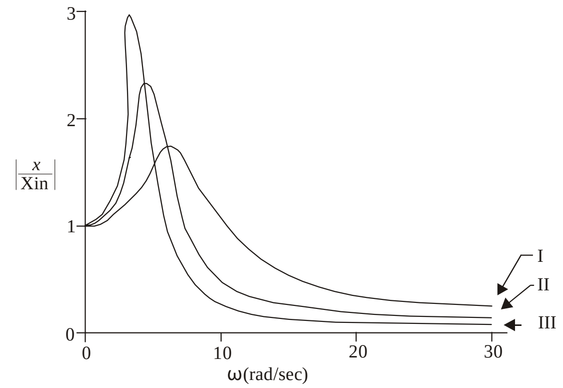

Fig. 1: Response of the system with zero applied control voltage, C_1/C_2 = 1.5; I, M = m; II, M = 2 m; III, M = 3 m

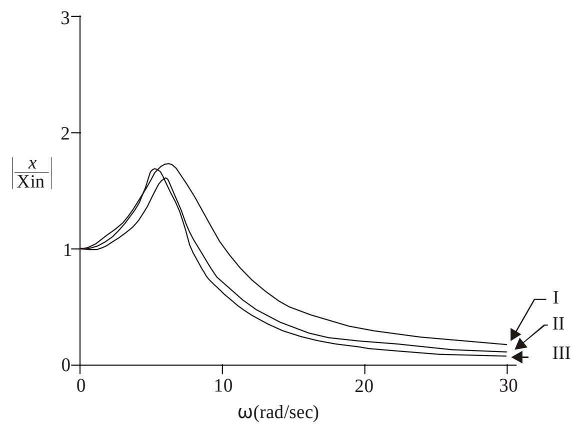

Fig. 2: Effect of varying the applied control voltage as the input level changes, C_1/C_2 = 1.5; I, M = m, F_{EFS} = 0; II, M = 2 m, F_{EFS} = 0.94 mg; III, M = 3 m, F_{EFS} = 0.2 mg

reduction of the peak and steady state response amplitudes.

In Fig. 1, all the parameters were kept constant. Only the mass was varied. The EFS damper was not activated (i.e., f_{EFS} = 0), and so it behaved like ordinary squeeze-film damper. The system parameters are shown in Table 1.

The increase in load gives a decrease in the natural frequency of oscillation and the damping ratio. Figure 2 shows the response of the system to a control voltage the value of which was adjusted as the load was increased. Here, a comparison is made between the peak amplitude ratio for low mass (M = 1 kg) with zero f_{EFS} and the higher masses (M = 2 kg, 3 kg). With introduced and varied for the same values of the input excitation (= 1.0 unit). The values of the system parameters are shown in Table 2.

Now with the values of K_1 and K_2 swapped and the input excitation halved (= 0.5 units), all other parameters remaining the same, Figure 2 reduces to that shown in

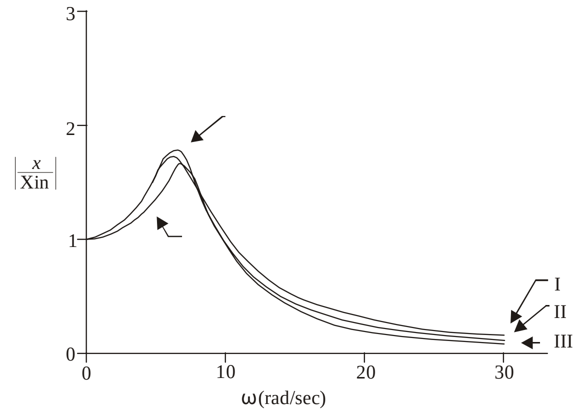

Fig. 3: Effect of varying, C_1/C_2: M = 3 m, F_{EFS} = 0.2 mg; I, C_1/C_2 = 1.5; II, C_1/C_2 = 0.66; III, C_1/C_2 = 0.25

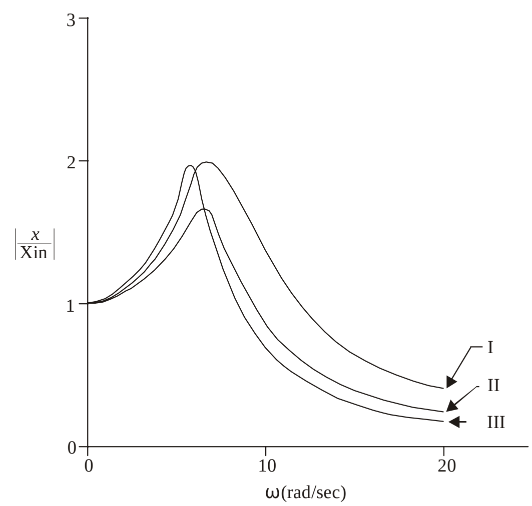

Fig. 4: Effect of Controlling the System with, C_1/C_2 = 0: I, M = m, F_{EFS} = 0; II, M = 2 m, F_{EFS} = 0.94 mg; III, M = 3 m, F_{EFS} = 0.2 mg

Fig. 3. The values of the system parameters are shown in Table 3.

With the condition shown in Table 1 retained, the effect of controlled system with C_1/C_2 = 0 was obtained and the parameters are as shown in Table 4 and Fig 4.

CONCLUSION

The results presented in this work have shown that a significant degree of control is achievable for the response of a viscous damped isolation system using the effects derived from electro-hydrodynamic agitation induced by the coulomb forces acting on the space charge layers of the damping fluid. The results are considerable improvement on the data reported in the existing literature. Our computation shows how the variation of the values of the mass and the damping ratio in combination with the EFS damping effect helped in the drastic reduction of the peak amplitude and hence in the achievement of the range of values of the parameters necessary to produce the desired performance characteristics of the system. The results show that with zero applied control voltage, an increase in the value of the damping ratio results in some reduction of the peak and steady state response amplitudes. With introduction of the control voltage, the responses of the system to each of the input excitations were computed for and for various sprung masses of the machine suspension. The results show considerable improvements in the response of the system. It was important to note that must always be finite in a practical application in order that resonance could be avoided when . However, the need for recourse to controller design was indicated in these results, the aim being to improve the response time and reduce the steady state errors due to the impulse and step input functions, and satisfy such requirements as stability and repeatability.

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