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ACTIVE ELECTROMAGNETIC SUSPENSION SYSTEM FOR GROUND VEHICLES*

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Abstract- Application of a new control strategy on actuators used in electromagnetic vehicle suspension systems has been studied in this paper. The vehicle is modeled as a quarter car in which the unsprung mass is supported by two vertically arranged electromagnetic actuators having a nonlinear behavior. The nonlinear actuators, having forces proportional to the square of the electric current and inversely proportional to the square of their distances from the mass, are controlled using a simple and feasible control strategy named "Performance Based Time-Delay". It is shown that this control strategy can result in excellent system performance compared to the conventional passive suspension system or other control strategies implemented in active suspension systems. The vibrations due to road roughness are controlled by changing magnetic fluxes of the actuators. Less required computational steps, use of an on-line digital computer, overall system simplicity, and achievement of a better response to larger high frequency disturbances are among the advantages of the system controller.

Keywords – Active suspension, electromagnetic actuator, time-delay, nonlinear model, adaptive control

1. INTRODUCTION

An active system is one in which an actuator either replaces the passive components or acts in parallel with them. Energy, usually a significant amount, may be fed into or taken out of the system by the active element. The idea of using active suspension has been around for decades. Engineers have been seeking the best strategy for dealing with the ride quality and handling characteristics of the vehicle simultaneously, both from a theoretical as well as a practical point of view [1-5]. Although the basic idea of electromagnetic levitation goes back 150 years, many of the developments in the design and application of electromagnetic levitation systems have taken place in the last two decades. A number of studies have been reported on the problem of magnetic bearings, linear motor cars, and levitation systems [6-7]. The nonlinear characteristics of magnetic actuators make the analysis and design of related control systems rather complicated. Therefore in most applications to date, actuators have been considered to have a linear behavior. Nagaya and Arai [8], presented a particular type of actuator in which the essential supporting force is applied by means of two permanent magnets with opposite poles. Using a linear mathematical model and the optimal regulator theory, they showed that vibrations of the magnetic levitated body can be controlled.

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2. ANALYSIS

For simplicity of analysis, a simple quarter car model shown in Fig. 1 is considered. The model contains nonlinear active and linear passive suspension elements, a pair of electromagnetic actuators, a conventional spring and a shock absorber, all in parallel. In fact, the electromagnetic actuators are installed as auxiliary elements in order to improve the suspension system behavior. The vibrations due to disturbances are controlled by changing the magnetic flux of the actuators. Each actuator force is proportional to the square of electric feed-current and inversely proportional to the square of distance from the sprung mass.

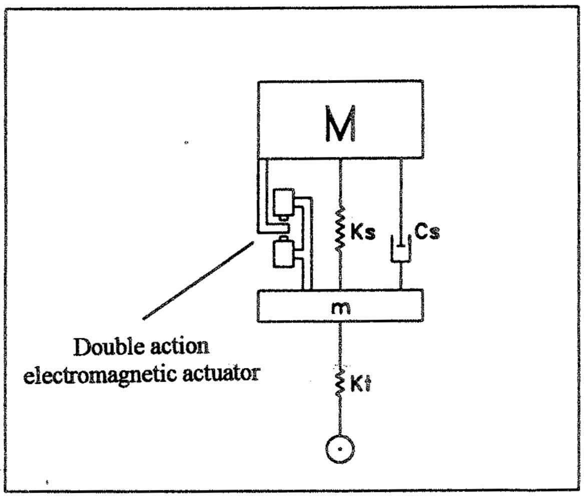

Fig. 1. Active suspension system

Using an on-line digital computer, the "Time-Delay" strategy is applied to the system. This will make the system adaptive to both input disturbances and the changes in system dynamic parameters. Variation of the value of the sprung mass in a vehicle is completely normal and possible. Having a different number of passengers in a car, the sprung mass may increase or decrease accordingly. The controller must be capable of achieving a good response for the system with such changes.

Vibrations due to external disturbances are controlled by changing magnetic flux densities of the actuators. Each actuator is considered to have uniform and unidirectional field along its longitudinal axis. The magnetic interference between the actuators has been ignored. The force exerted by each actuator, Fa, is considered as:

$$F_{a1} = \frac{d \cdot i_1^2}{(\delta - Y)^2} \tag{1}$$

in which δ is the initial clearance between unsprung mass and the actuator, assumed to be equal for both actuators, and Y is the relative position regarding the reference point. It must be noted that a change in the direction of the electric current does not change the direction of the force, i.e., the actuators always attract the mass. Therefore, if the mass is to be repelled by an actuator, that actuator will be deactivated and the opposite actuator will be activated to exert an attracting force equal to:

$$F_{a2} = -\frac{d}{(\delta+Y)^2}\left[i_1 \cdot \frac{\delta+Y}{\delta+Y}\right]^2 \tag{2}$$

The term in the brackets represents the required current that is to be supplied by a drive amplifier and applied to the opposite actuator as shown in Fig. 2.

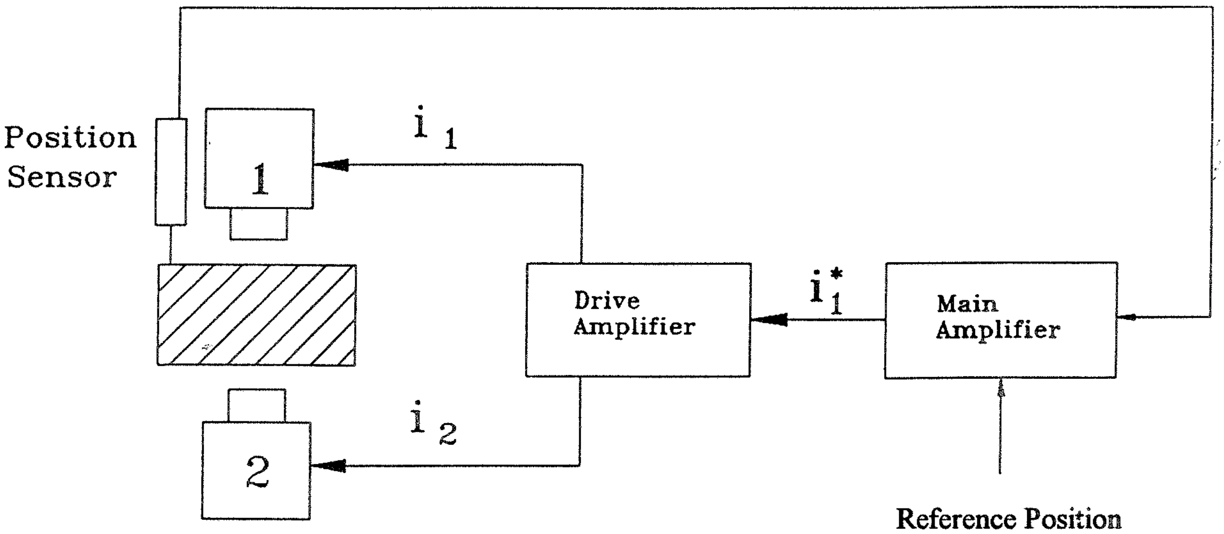

Fig. 2. The drive amplifier current control

Defining the state variables as:

$$X_1 = y_0 - y_1,\ X_2 = y_1 - y_2,\ X_3 = \dot{y}_1,\ X_4 = \dot{y}_2 \tag{3}$$

The system state equations will be:

$$\dot{X}_1 = \dot{y}_0 - X_3,\ \dot{X}_2 = X_3 - X_4$$

$$\dot{X}_3 = \frac{F_a}{m} - \frac{K_s}{m}X_2 - \frac{C_s}{m}(X_3 - X_4) + \frac{K_t}{m}X_1 \tag{4}$$

$$\dot{X}_4 = \frac{K_s}{M}X_2 + \frac{C_a}{M}(X_3 - X_4) - \frac{F_a}{M}$$

In which *Fa* is the electromagnetic actuator force, *Ks* is the secondary suspension spring rate, *Cs* is the shock absorber damping rate and *Kt* is the tire stiffness.

From ISO 2631, the ride comfort criterion is the vertical acceleration of sprung mass, i.e., the effective force acting on the driver and passengers. Also the stability and road-hold of the vehicle are directly related to the tire and road interaction. Hence, a quadratic performance index in the following form can be defined:

$$PI = \int_{t_0}^{t_f} (X^T.Q.X + \lambda_2.\dot{X}_4^2)dt \tag{5}$$

Where:

$$Q = \begin{bmatrix} \lambda_1 & 0 & 0 & 0 \\ 0 & \lambda_2 K_s^2 & 0 & 0 \\ 0 & 0 & \lambda_2 C_s^2 & -\lambda_2 C_s^2 \\ 0 & 0 & -\lambda_2 C_s^2 & \lambda_2 C_s^2 \end{bmatrix} \quad (6)$$

and λ_1 is the weighting coefficient of wheel to road contact, while λ_2 is that of the ride quality.

It must be noted that the terms involving λ_2 should exist in the matrix Q to consider the effects of spring K_S and shock absorber C_S. According to the comfort criterion, the objective here is to minimize the vertical acceleration of the sprung mass. This vertical acceleration is caused by both the actuator force and the suspension passive elements. The term λ_2 also defines a minimum value for the control force F_a. It is deduced that the actuators' forces also act on the sprung mass. This force in turn produces discomfort, and must be optimized accordingly. As a result, the energy consumption of the system is also be minimized.

For an application in which input disturbances experience rapid changes, the use of an on-line adaptive controller seems inevitable. To deal with the nonlinear nature of the problem, and to achieve a cost effective, competitive, and feasible solution, a control strategy named "Time-Delay Programming" was designed. The basic idea is to control a system in equal, sufficiently small time steps. In each small time step, the optimal control values are calculated and applied to the system as a constant control for the next time step. The steps can be chosen short enough to include only the final value of each state variable in performance index. Hence optimization of state variables during the time steps can be eliminated.

One advantage of this method is that the control in each time step will be constant and therefore the constrained optimization is simply replaced by putting a limit on the control value. Another advantage is that the system performance index becomes very simple. Using this control strategy, the performance index is defined as:

$$J_n = [X(t_f)]^T \cdot Q \cdot [X(t_f)] + \lambda_2 \cdot F_a |_n^2 \quad (7)$$

in which $[X(t_f)]_n$ is the state vector at the end of the nth time step. In each time step the following approximate relation can be written:

$$[\dot{X}]_n = [a(X, I, t)]_n \approx \frac{[X(t_f)]_n - [X(t_0)]_n}{\Delta t} \quad (8)$$

or:

$$[X(t_f)]_n \approx [X(t_0)]_n + \Delta t . [a(X, I, t)]_n \quad (9)$$

Hence the performance index can be restated as:

$$J_n = \{[X(t_0)]_n + \Delta t[a(X,I,t)]_n\}^T \cdot Q \cdot \{[X(t_0)]_n + \Delta t \cdot [a(X,I,t)]_n\} + \lambda_2 \cdot F_a|_n^2$$

$$= \lambda_1 \cdot \{X_1(t_0)|_n + \Delta t \cdot (\dot{y}_0|_n - X_3(t_0)|_n)\}^2 + \frac{\lambda_2 . K_s^2}{M^2} \cdot X_2(t_f)|_n^2 +$$

$$+ \lambda_2 \cdot \frac{C_s^2}{M^2}(X_3(t_f)|_n - X_4(t_f)|_n)^2 + \lambda_2 \cdot \frac{F_a|_n^2}{M}$$

$$+ 2\frac{\lambda_2}{M^2}(C_s \cdot (K_s \cdot X_2(t_f)|_n + F_a) \cdot (X_3(t_f)|_n - X_4(t_f)|_n) - F_a \cdot K_s \cdot X_2(t_f)|_n)$$

$$= \lambda_1 \cdot \left\{X_1(t_f)|n\text{-}1 + \Delta t \cdot \left(X_3(t_0)|n\text{-}1 + \Delta t\left(\frac{F_a|_{n-1}}{m} + \frac{K_s}{m}X_2(t_0)|n\text{-}1 + \frac{K_t}{m}X_1(t_0)|_{n-1}\right)\right)\right\}^2$$

$$+ \frac{\lambda_2 . K_s^2}{M^2} \cdot X_2(t_f)|_n^2 + \lambda_2 \cdot \frac{C_s^2}{M^2}(X_3(t_f)|_n - X_4(t_f)|_n)^2 + \lambda_2 \cdot \frac{F_a|_n^2}{M} \tag{10}$$

$$+ 2\frac{\lambda_2}{M^2}(C_s \cdot (K_s \cdot X_2(t_f)|_n + F_a) \cdot (X_3(t_f)|_n - X_4(t_f)|_n) - F_a . K_s \cdot X_2(t_f)|_n)$$

In order to have an optimum performance index, we should have:

$$\frac{\partial J_n^*}{\partial F_a} = 0 \tag{11}$$

In which J^*_n is the optimum value of the performance index. For the reason of having a minimum value for the performance index at t_f and considering the nonlinear behavior of the electromagnetic actuators, the optimum value for the control current will be:

$$i_1|_{n-1}^* = \left\{ \frac{(\delta - X_2(t_0)|_n)^2}{d\left(\frac{\lambda_2 . m}{\lambda_1 M^2 . \Delta t^2} + \frac{\Delta t^2}{m}\right)} \left[\begin{array}{l} X_1(t_0)|n + \Delta t \cdot \overset{*}{y}_0|_n - \Delta t \cdot X_3(t_0)|_{n-1} - \frac{\Delta t^2 \cdot K_t}{m} X_1(t_0)|_{n-1} \\ + \frac{\Delta t^2 \cdot C_s}{m}[X_3(t_0)|_{n-1} - X_4(t_0)|_{n-1}] + \frac{\Delta t^2 \cdot K_s}{m} X_2(t_0)|_{n-1} \\ + \frac{\lambda_2 \cdot m}{M^2 \cdot \Delta t^2}[C_s(X_3(t_0)|_n - X_4(t_0)|_n) + K_s \cdot X_2(t_0)|_n] \end{array}\right] \right\}^{1/2} \tag{12}$$

Equation (12) may also be attained by considering that the controller compares the performance index of the system with a reference function that traces the ideal trajectory. The ideal trajectory in this case is zero. By this, the controller tries to modify the system behavior based on the detected error between the actual or measured performance index of the system, and the optimal function. In this strategy, at every time step the input disturbances acting on the system give some displacements to the system. These displacements are measured and used as the initial conditions for the next step.

Actually, at each time step the control action is based on the system conditions of one step before, and will have a time delay equal to one time step. This control strategy is therefore called "Performance Based Time-Delay". This is how the (n-1)th time step is converted to the nth in the calculations of the above theory, with no loss of generality.

One important point to be noted here is that the optimum control current of Eq. (12) is calculated in a manner as if the actuator is able to attract or repel the sprung mass. In practice this current is fed into an intermediate drive amplifier that will feed the appropriate actuator to produce attraction force in the correct direction.

3. RESULTS

The dynamic behavior of an actual system is simulated using a fourth order Runge Kutta algorithm. Since the approximation of the derivatives of the above-mentioned theory was of the first order, the adoption of fourth order algorithm in the simulation of the system seems sufficient.

For a particular vehicle the sprung mass is 500 Kg, the unsprung mass is 45 Kg, the coefficient of electromagnetic effect of the actuator is equal to 1 $N.m^2/amp^2$, and the gap between the mass and the actuators is required to be 5 cm at the reference position. As a reasonable value, the duration of each time step for this problem is taken equal to 0.01 sec. The passive suspension spring rate is 5000 N/m and for the tires it is 30000 N/m. The system is equipped with a shock absorber having a damping coefficient equal to 2500 N.Sec/m. The comfort ratio of the active system, i.e., the ratio of λ_2/λ_1 in the performance index of Eq. (5), in this particular application is taken equal to 10000. Therefore the resulting system has great comfort but wheel to road contact behavior similar to the passive system. It is obvious that the system may be tuned inversely to achieve optimized vehicle stability characteristics.

Figure 3 illustrates the acceleration of the sprung mass for both optimal and passive systems, when the wheel reaches the road irregularities in the form of a sudden change similar to a step input. As it is seen, the active system is able to decrease the peak response of the vertical acceleration up to five times, hence producing a better ride quality. The figure also consists of curves for the actuator exerted force and the optimal electric control current. One important point to be noted here is that all the presented control currents are the main controller outputs fed to the drive amplifier for final processing to produce actuators requiring currents based on Eq. (2). Furthermore, for a different road condition the system responses due to a limited ramp input are shown in Fig. 4, with a similar result for a sinusoidal input shown in Fig. 5. Note that the active system behavior is very stable and comfortable in comparison with the passive system, especially when the system traces a sinusoidal path.

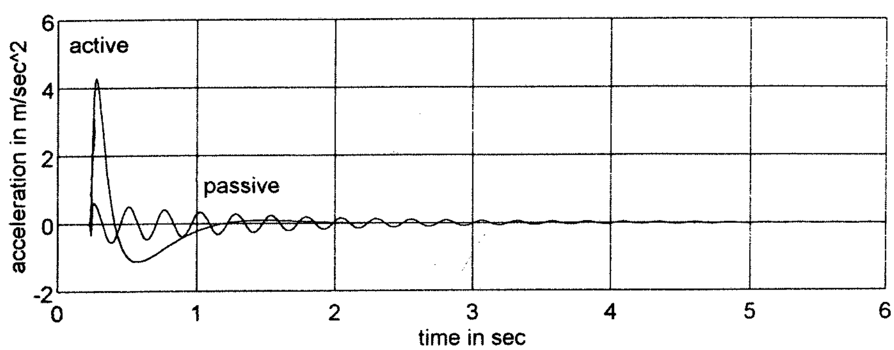

Fig. 3 a. Comparison of passive and active sprung mass accelerations, step input

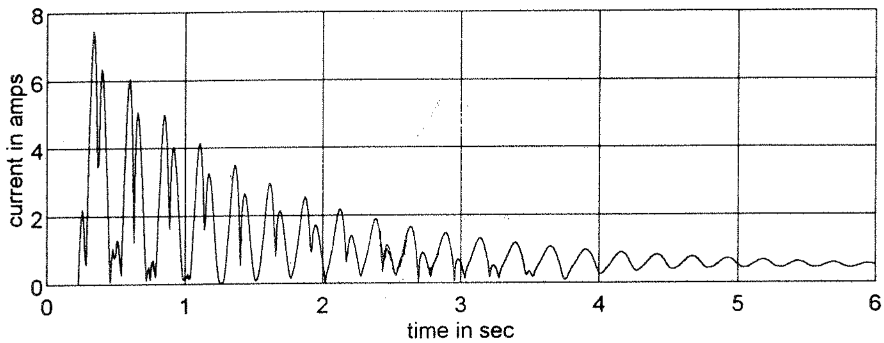

Fig. 3 b. Drive amplifier input control current, step input

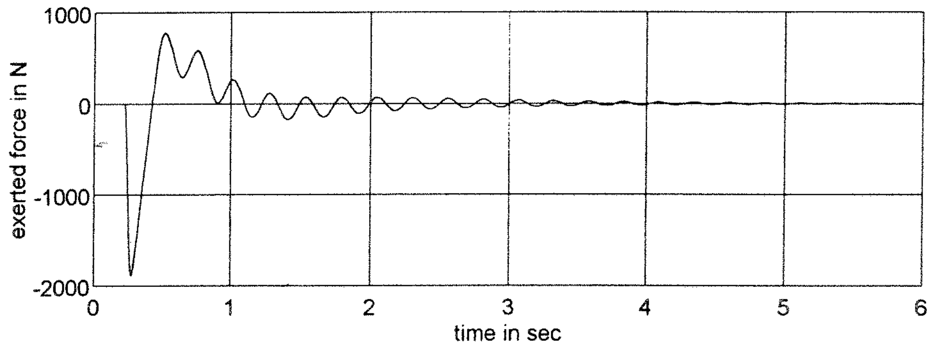

Fig. 3 c. Actuator exerted force, step input

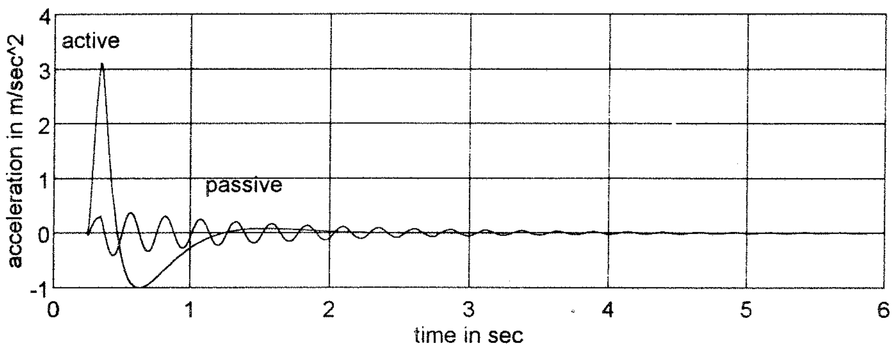

Fig. 4 a. Comparison of passive and active sprung mass accelerations, ramp input

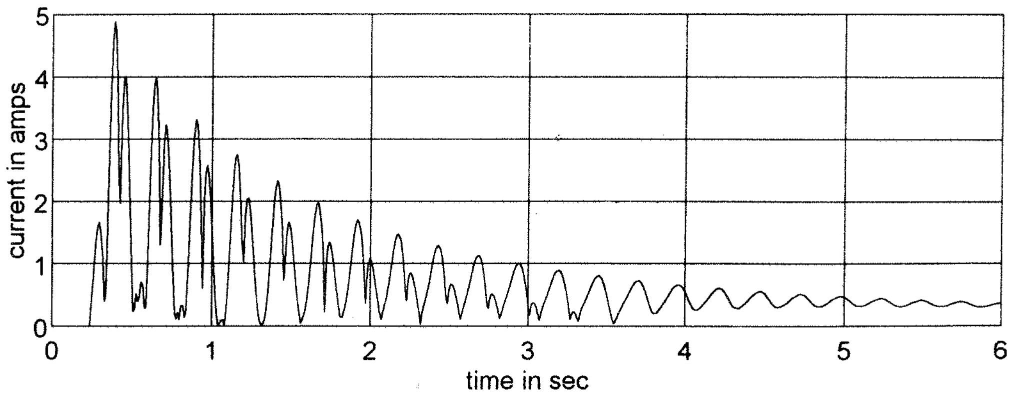

Fig. 4 b. Drive amplifier input control current, ramp input

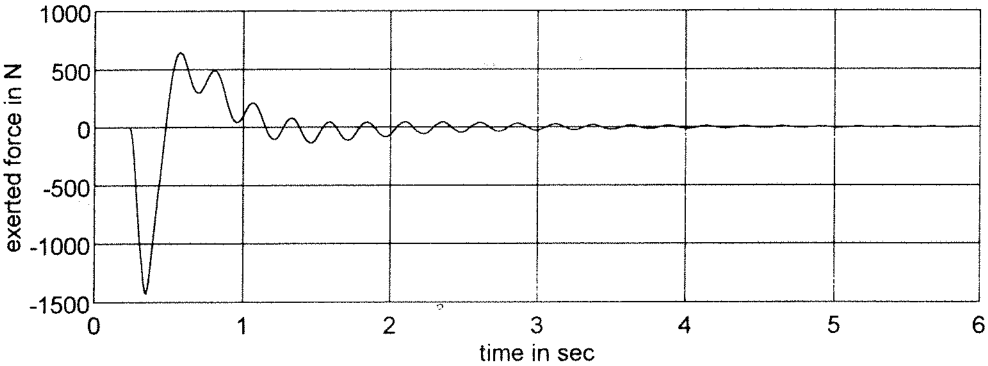

Fig. 4 c. Actuator exerted force, ramp input

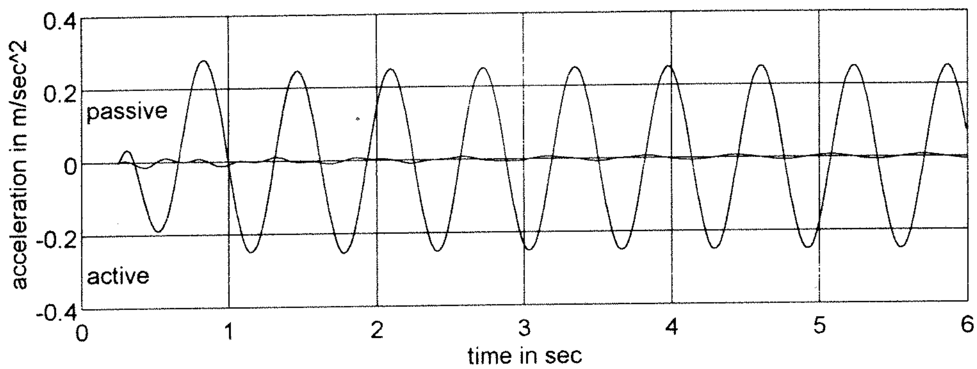

Fig. 5 a. Comparison of passive and active sprung mass accelerations, sinusoidal input

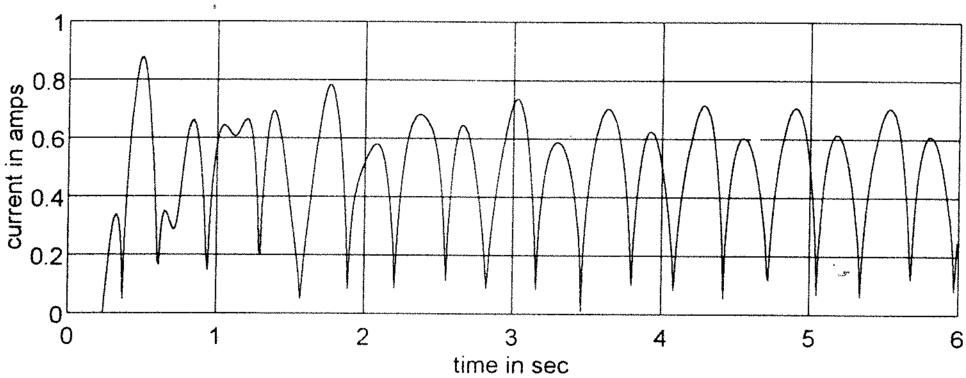

Fig. 5 b. Drive amplifier input control current, sinusoidal in amps

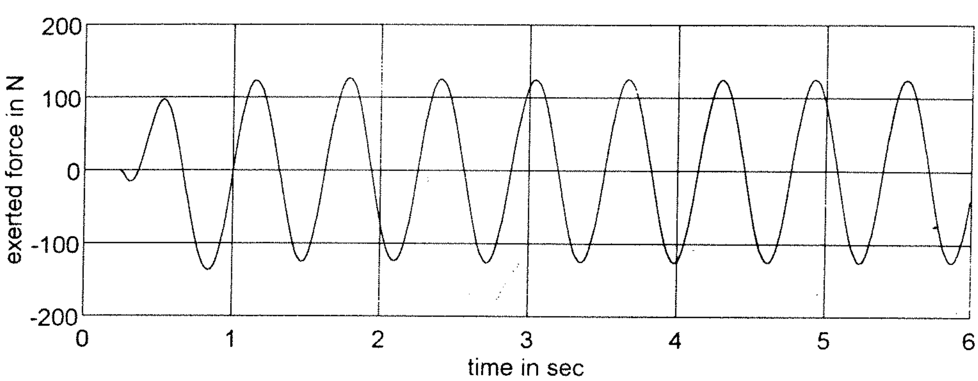

Fig. 5 c. Actuator exerted force, sinusoidal input

4. CONCLUSION

A special type of active suspension system is investigated in which a controlled nonlinear electromagnetic pair of actuators is chosen in parallel with a conventional passive spring and shock absorber system. A new type of controller is introduced. The time-delay control strategy can bring about acceptable results in controlling the sprung mass acceleration and wheel-to-road contact. Fewer computations are required and hence it is feasible to use a simpler and more cost-effective processor. The control strategy presented here also can be used in other applications. Simplicity and low computation volume of the controller are appreciated in applications with a large number of control channels or under cost pressure.

NOMENCLATURE

C_S	Viscose damping coefficient
F_a	Actuator force
J_n	Performance index of the nth step
Jn^*	Optimum value of performance index of the nth step
K_S	Suspension spring stiffness
K_t	Tire stiffness
M	Sprung mass
X	State vector
X_1 to X_4	State variables
a	Derivative of the state vector (vector of right hand sides of the state equations)
d	Coefficient of electromagnetic effect of the actuators
i_1	Actuator control current
i_1^*	Optimum value of actuator control current
m	Unsprung mass
n	Step number
t_0	Step initial time
t_f	Step final time
y_0	Road input disturbances
y_1	Unsprung mass displacement
y_2	Sprung mass displacement
δ	Initial clearance between mass and the actuator
λ_1	Coefficient of wheel to road contact
λ_2	Coefficient of ride quality
Δt	Time step duration

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