Simulation on Active Vibration Suppression Using Virtual Spring-Damper Combination

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***Abstract*-Active suspension is an emerging technology used to improve the contact, comfort and control of vehicles and in other vibration suppression applications. Different methods are used to control the actuator force exerted between the sprung mass and unsprung mass in order to achieve the active suspension. Most of the other research areas have introduced LQR controllers, Fuzzy logic controller methodologies. But this paper presents a control method based on a virtual spring-damper combination. A linear motor model is expected to be used as the actuator in realizing the values of the virtual spring damper. The simulation results show the sprung mass displacement with and without the active suspension when subjected to a large disturbance. The success and the limitations of the proposed virtual spring-damper combination are discussed with the simulation results.**

Keywords: Passive suspension;Active suspension;Virtual spring–damper; Vibration suppression

I. Introduction

The automobile suspension system physically separates the vehicle body from the wheel arrangement. The primary functions of a suspension system are to provide rider comfort, safe road handling and to minimize suspension deflections. Rider comfort is related to the vehicle body motion that is felt by the passengers. Passengers will feel seasick and dizzy when subjected continuously to the vibrations between 0.5 Hz to 1 Hz and 18Hz to 20 Hz[1]. Suspension system helps to improve the road handling capability by maintaining the tyre-road contact forces and controlling the rolling/pitching movements of the vehicle to be within the safe limits. Suspension deflection is the relative displacement between the automobile body and the wheel arrangement.

Conventional passive suspension systems are designed with physical spring and damper mechanisms and they are expected to suppress the vibrations in the region that they are designed. Passive suspension systems unable to achieve high quality suspension performance in changing conditions of the road profile as its parameters are fixed. In order to provide a high quality suspension performance, active suspension systems has being introduced.

Active suspension systems are mainly divided into two systems; semi-active suspension systems and pure active suspension systems. Semi-active suspension systems use active dampers with magneto rheological or electro rheological fluid which has the capability of supplying a real-time dissipation of energy. The drawback of these kinds of systems is that, the system can't provide an active force to reduce the response time. Pure active suspension systems use actuators to provide energy to the suspension system. So, the response time of the vibration suppression of the pure active suspension system is higher than the semi active suspension system.

The main component of the pure active suspension system is the actuator. Mainly two types of actuators are currently used; hydraulic actuators and electromagnetic actuators. Hydraulic actuators are slower due to high inertias present in the fluid, valves and pistons. Electromagnetic actuators are widely used in research areas as they have sufficient bandwidth having small system time constant. Electromagnetic actuators have the capability of storing energy when they are working in the generator mode [2].

Several controlling methodologies have been published by researchers with electromagnetic actuators in pure active suspension systems. Linear Quadratic Regulator (LQR) controller for quarter car model is proposed with the linear switched reluctance motor [3]. A mathematical model by using Linear Quadratic Regulator controller for quarter car model has been simulated and analyzed using MATLAB and SIMULINK toolbox [4]. Modified lead lag control, linear-quadratic servo control with a Kalman filter and a fuzzy controller with asymmetric membership functions and its performance were introduced for a vehicle quarter car model with a direct drive tubular linear brushless permanent magnet motor [5]. Fuzzy logic controllers have been introducing for actuator force control with the development on intelligent technologies. A fuzzy logic controller for an active suspension system in which the membership functions and control rules are optimized using a genetic algorithm has been discussed [6].

The main objective of this research is to introduce an active suspension system for large disturbances using virtual spring-damper controller. The actuator used to produce the force which results from virtual spring damper combination will be provided by an electromagnetic linear motor. This actuator force is controlled in order to stabilize the sprung mass of the system with minimal oscillation.

II. SYSTEM MODELING

Though virtual spring-damper combination can be applied generically for any vibration suppression application, the modeling has done for vehicle active suspension system as it is the widely using application. So a quarter car suspension system with two degrees of freedom is used for the system modeling.The system model consists of two masses M_S and M_U which are respectively known as sprung mass and unsprung mass. The system model is illustrated in Fig. 1.

M_S : Sprung mass (kg)
M_U : Unsprung mass (kg)
K_S : Suspension system spring coefficient (N/m)
K_T : Tyre spring coefficient (N/m)
C_S : Suspension system damping coefficient (N/m^2)
C_T : Tyre damping coefficient (N/m^2)
F : Active force (N)
X_R : Displacement of road surface (m)
X_U : Displacement of unsprung mass (m)
X_S : Displacement of sprung mass (m)

For modeling purposes it is assumed that all the above coefficients of the system and its masses do not vary with environmental factors. In addition, it is assumed that the suspension system acts as an ideal model that only has the capability to move vertically.

By Applying Newton's second law to M_S,

$$M_S\ddot{X}_S = -K_S(X_S - X_U) - C_S(\dot{X}_S - \dot{X}_U) + F \tag{1}$$

By Applying Newton's second law to M_U,

$$M_U\ddot{X}_U = K_S(X_S - X_U) + C_S(\dot{X}_S - \dot{X}_U) - K_T(X_U - X_R) - C_T(\dot{X}_U - \dot{X}_R) - F \tag{2}$$

A. Passive quarter car suspension system model

For passive suspension system,

$$F = 0$$

Using the Laplace domain Transfer function of (1) and (2), the passive suspension system transfer function can be written as,

$$X_S(s) = \frac{[C_T s + K_T][C_S s + K_S]}{\begin{bmatrix} M_S M_U s^4 + [M_S(C_S + C_T) + M_U C_S]s^3 + \\ [M_S(K_T + K_S) + M_U K_S + C_S C_T]s^2 \\ +[K_S C_T + K_T C_S]S + K_T K_S \end{bmatrix}} X_R(s) \tag{3}$$

Passive suspension system response to the road profile can be measured using the above equation (3).

B. Active suspension system model

The active force F provided by the control system is used to model the active suspension system. Following state variables were chosen to create a state space model of active suspension system.

x_1: Relative displacement between sprung mass and unsprung Mass
x_2: Velocity of sprung mass
x_3: Relative displacement between unsprung mass and road
x_4: Velocity of unsprung mass
Y : System output
F : Active force

$$x_1 = X_S - X_U \tag{4}$$
$$x_2 = \dot{X}_S \tag{5}$$
$$x_3 = X_U - X_R \tag{6}$$
$$\dot{x}_4 = \dot{X}_U \tag{7}$$
$$Y = [\ddot{X}_S] = \dot{x}_2 \tag{8}$$

Following equations are obtained by using the equations (1),(2),(4),(5),(6),(7) and (8).

$$\dot{x}_1 = \dot{X}_S - \dot{X}_U = x_2 - x_4 \tag{9}$$

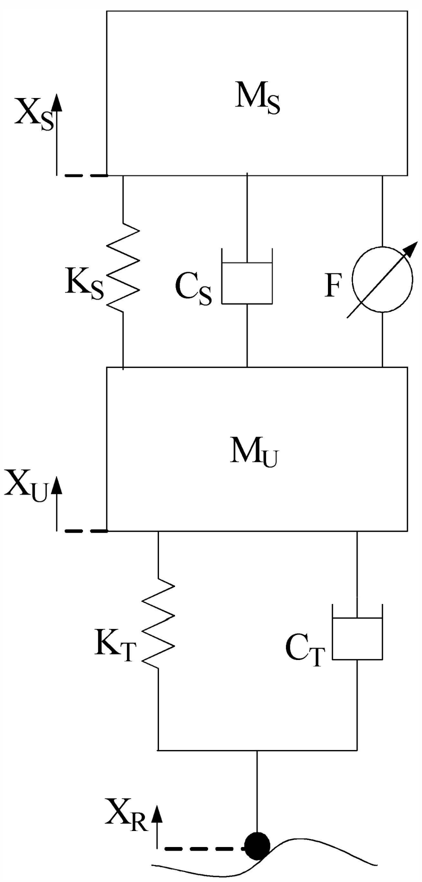

Figure 1. Active suspension system representation

$$\dot{x_2} = \frac{-K_S(x_1) - Cs(x_2 - x_4) + F}{M_S} \tag{10}$$

$$\dot{x_3} = \dot{X}_U - \dot{X}_R = x_4 - \dot{X}_R \tag{11}$$

$$\dot{x_4} = \frac{K_S(x_1) + C_S(x_2 - x_4) + K_T(x_3) + C_T\left(x_4 - \dot{X}_R\right) - F}{M_U} \tag{12}$$

The system state space can be written using equations (9), (10), (11) and (12)

$$\begin{bmatrix} \dot{x_1} \\ \dot{x_2} \\ \dot{x_3} \\ \dot{x_4} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & -1 \\ -\frac{K_S}{M_S} & \frac{-C_S}{M_S} & 0 & \frac{C_S}{M_S} \\ 0 & 0 & 0 & 1 \\ \frac{K_S}{M_U} & \frac{C_S}{M_U} & \frac{K_T}{M_U} & \frac{C_S - C_T}{M_U} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 1 \\ \frac{1}{M_S} \\ 0 \\ \frac{-1}{M_U} \end{bmatrix} F + \begin{bmatrix} 0 \\ 0 \\ -1 \\ \frac{C_T}{M_U} \end{bmatrix} \dot{X}_R \tag{13}$$

$$[Y] = \begin{bmatrix} \frac{-K_S}{M_S} & \frac{-C_S}{M_S} & 0 & \frac{C_S}{M_S} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} \frac{1}{M_S} \end{bmatrix} F \tag{14}$$

III. VIBRATION SUPPRESSION CONTROLLER

The input of the controller is the relative displacement between the sprung mass and the unsprung mass. The controller is a combination of two major sections. First one is the virtual spring and the second one is the virtual damper.

A. Virtual spring

The equation related with virtual spring force *Fs* is given in (15).

$$F_S = K_V(X_S - X_U) \tag{15}$$

Where Kv is the virtual spring constant.

The spring constant of the virtual spring is selected such that it is equal and opposite to the force due to the actual spring. Hence the resulting force acting on the sprung mass due to the virtual spring and actual spring become zero. Hence if the virtual damper is not present, the actual damping force becomes the resultant force acting on the sprung mass.

B. Virtual Damper

The second component of the controller is the virtual damper. The derivative of the relative displacement between the sprung mass and the unsprung mass is multiplied by the virtual damping constant in order to get the actuator force due to the virtual damper F_d as shown in (16).

$$F_d = -C_V(\dot{X}_S - \dot{X}_U) \tag{16}$$

Where C_V is the virtual damping constant.

The force exerted by the virtual damping component should be in the same direction of the force exerted by the actual damper of the system on the sprung mass. Hence the virtual damping constant is selected in order to get the virtual damper force in same direction with the actual damper force acting on the sprung mass.

The controller is used to calculate the actuator force which should be exerted on sprung mass and unsprung mass. The actuator force is used in order to achieve the active suspension of the sprung mass. The controller is designed to control the behavior of the actuator force as a virtual spring-damper combination to achieve the active suspension. The controller model is presented in Fig.2.

The total actuator force *F* is caused by the virtual spring-damper controller is given in (17).

$$F = F_S + F_d = K_V(X_S - X_U) - C_V\left(\dot{X}_S - \dot{X}_U\right) \tag{17}$$

Substituting the *F* in (1) provides the force acting on the sprung mass for active suspension in (18).

$$M_S\ddot{X}_S = -(C_S + C_V)(\dot{X}_S - \dot{X}_U) \tag{18}$$

Substituting *F* in (2) gives the force acting on the unsprang mass for active suspension in (19).

$$M_U\ddot{X}_U = (C_S + C_V)\left(\dot{X}_S - \dot{X}_U\right) - C_T\left(\dot{X}u - \dot{X}_R\right) - K_T(Xu - Xr) \tag{19}$$

By rearranging (18) and (19) and transforming into the Laplace domain the following linear equations were obtained.

$$\frac{X_S(s)}{s^2} = -\frac{(C_S + C_V)}{M_S} s[X_S(s) - X_U(s)] \tag{20}$$

$$\frac{X_U(s)}{s^2} = \frac{(C_S + C_V)}{M_U} s(X_S(s) - X_U(s)) - \frac{C_T}{M_U} s(X_U(s) - X_R(s)) - \frac{K_T}{M_U}(X_U(s) - X_R(s)) \tag{21}$$

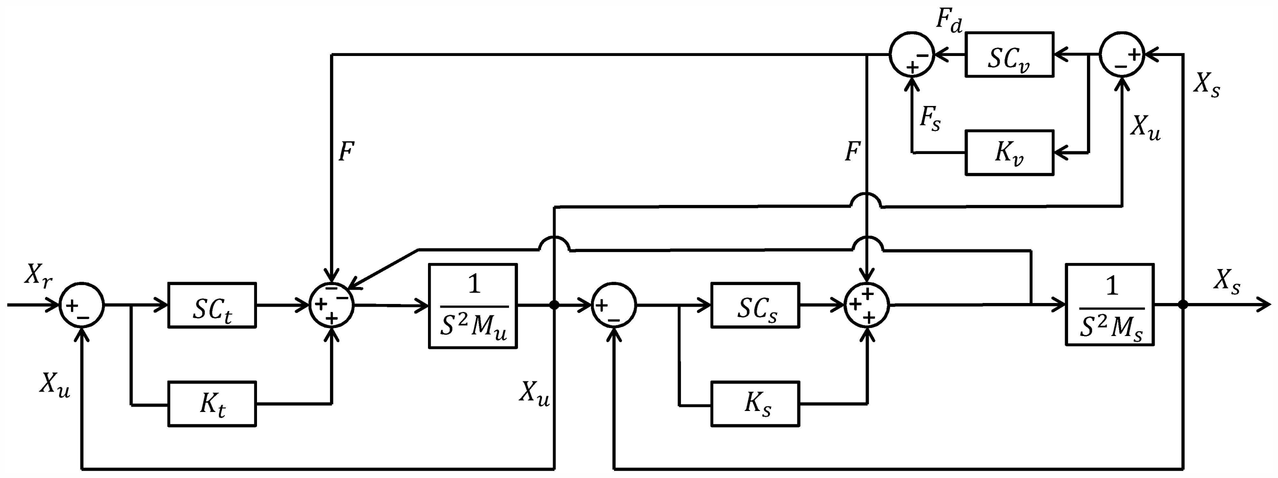

Figure 2.Controller block diagram

The transfer function for X_S over X_R can be obtained by solving (20) and (21).

By changing the magnitude of the virtual damping constant, the time taken to stabilize the sprung mass can be changed. But the following limitations should be taken in to account in order to achieve the value for the virtual damping constant.

1. The acceleration of the sprung mass ($\ddot{X_S}$)
2. The maximum force of the actuator (F)

The acceleration of the sprung mass should be kept on acceptable values in order to achieve the desired level of vibration suppression. The time taken to stabilize the sprung mass can be reduced by increasing the virtual damper coefficient which is needed to keep as minimum for disturbances with short time intervals. The maximum force capability of the actuator affects the virtual damper coefficient since the increase in virtual damper coefficient will also increase the force exerted by the actuator.

IV. Simulation and Results

The system was tested under both active suspension and passive suspension conditions to compare results. The parameter values used for the simulation are given in Table 1. X_r was given as a step input and the oscillation of the sprung mass under passive suspension was measured. A step input signal was provided after 0.5 seconds as shown in Fig.3.

There was a considerable oscillation of sprung mass under passive suspension system for the step input, and it was decayed with time. After introducing active control operation to the suspension system, the sprung mass oscillation had been removed, observed as shown in Fig.4.

Deflection of sprung mass had gone to a critical damping situation with the active control. The passengers of the vehicle will not be able to feel the oscillation due to the disturbance with the active suspension effect. The virtual spring force produced from the active controller part to achieve the critical damping state is shown in Fig.5.

TABLE 1
PARAMETER VALUES USED FOR SIMULATION

Parameter	Value
M_S	10 kg
M_U	2 kg
K_S	500 Nm
K_T	10000 Nm
K_V	500 Nm
C_S	5 Nsm^{-1}
C_T	15 Nsm^{-1}
C_V	90 Nsm^{-1}

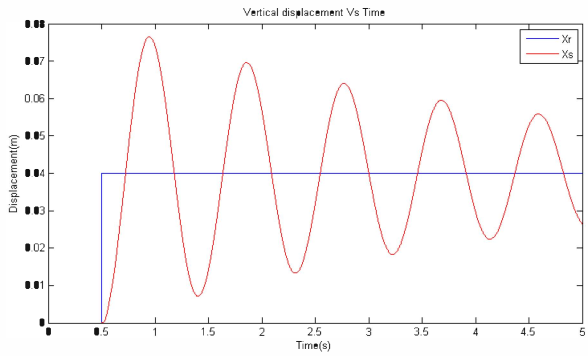

Figure 3. Sprung Mass deflection of passive suspension system for a step input

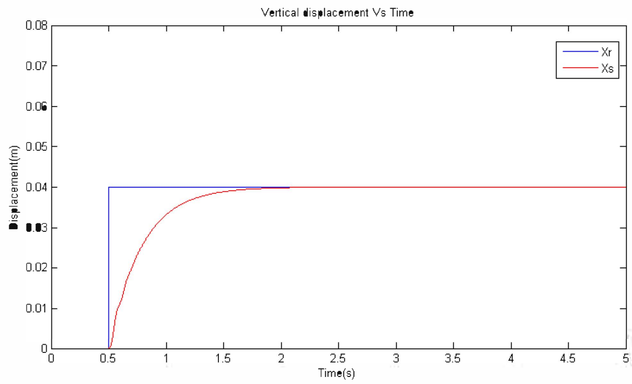

Figure 4. Sprung Mass deflection of active suspension system for a step input

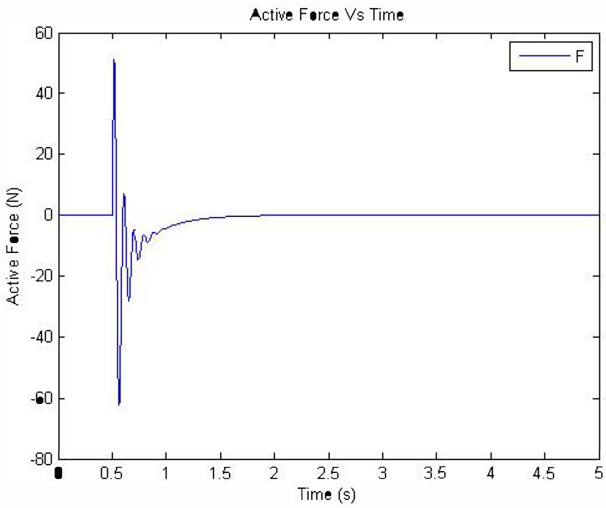

Figure 5. Virtual spring force produced from the active controller part

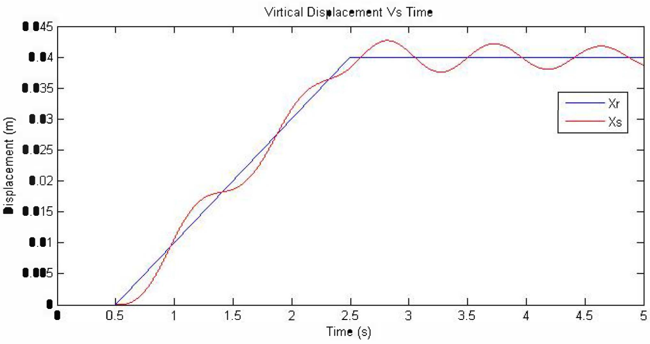

Fig.6: Sprung Mass deflection of passive suspension system for a ramp input

The system was tested by applying a ramp input for both passive and active operation separately. The ramp input was started at 0.5 seconds. Without active operation, the sprung mass of passive system shows an oscillation when the ramp input was given and it is decayed with time as shown in Fig.6.

The deflection of sprung mass was adapted to ramp input without such oscillations of the sprung mass in the active mode operation. Ride comfort had improved minimizing the oscillation when the road profile is acting as a ramp input. The active force for the ramp input is given in Fig.8.

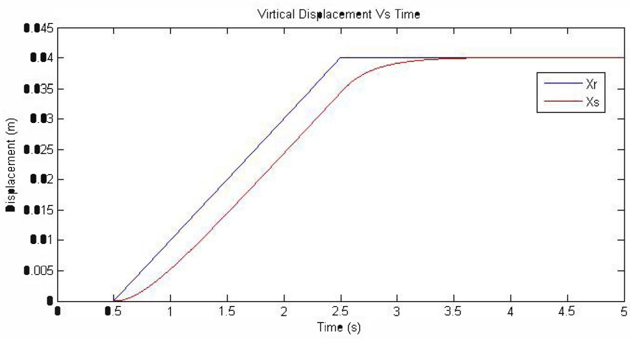

Figure 7. Sprung Mass deflection of active suspension system for a ramp input

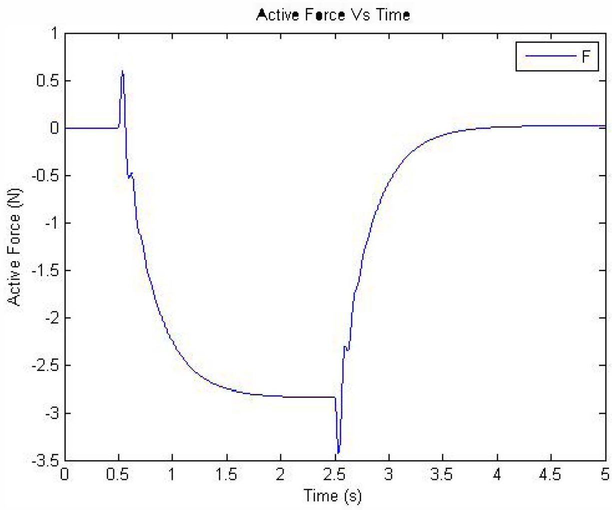

Figure 8. Active force producedfrom the active controller part for the ramp input

The virtual spring-damper combination was simulated for a ramp road profile having some random ripples along the ramp. The simulation results of the vertical displacement of the sprung mass and the unsprung mass against time are as shown in fig.9 and fig.10 for active and passive operation respectively. The active force of the actuator is shown in fig.11.

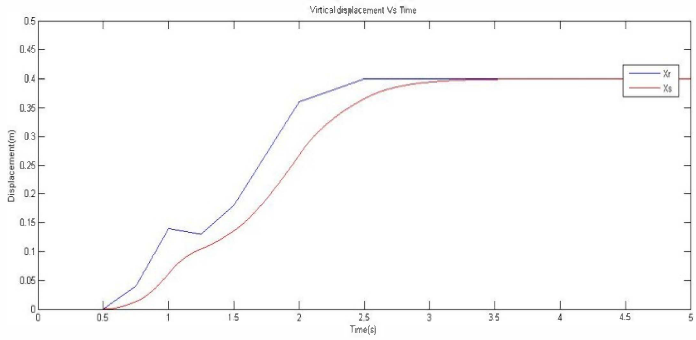

Figure 9. Sprung Mass deflection of active suspension system for a ramp input having random ripples

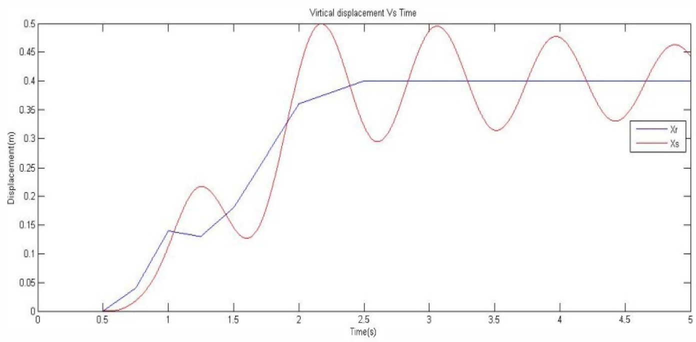

Figure 10. Sprung Mass deflection of active suspension system for a ramp input having random ripples

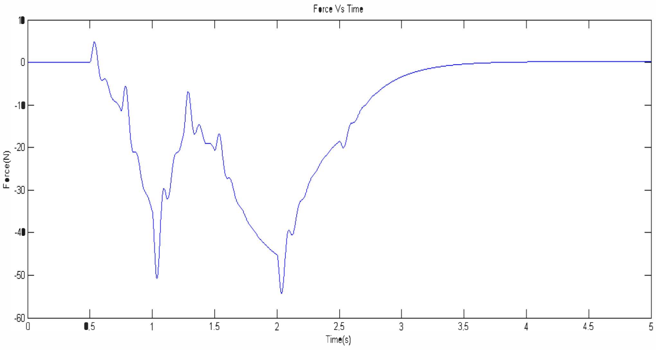

Figure 11. Active force produced from the active controller part for the ramp input having random ripples

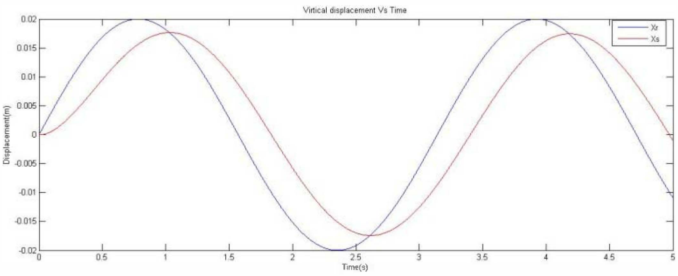

Figure 12. Sprung Mass deflection of active suspension system for a small magnitude sine wave

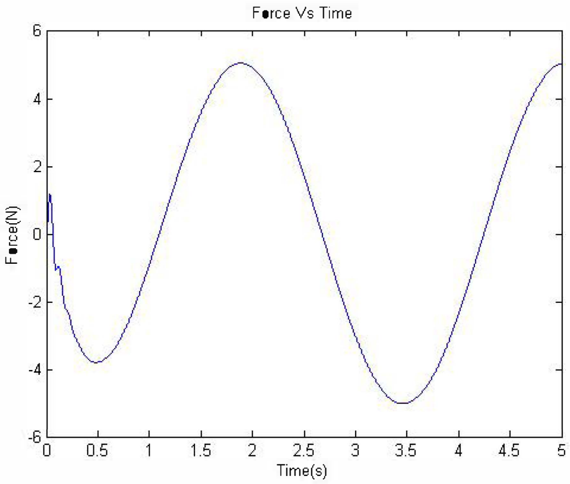

Figure 13. Active force produced from the active controller part for the small magnitude sine wave

The simulation results of the sprung mass vertical displacement where the input road profile having a small magnitude sine wave is shown in fig.12. Sprung mass is trying to follow the road profile as a result of the controlling methodology of virtual damping coefficient is being used is exerting a force as in the same direction as that of the actual damping force directed. But the vertical displacement of the sprung mass is expected to be zero while the unsprung mass's vertically movement is expected as same as the road profile. The active force of the actuator corresponding to this scenario is shown in the fig.13.

IV. Conclusion

A method of active vibration suppression based on virtual spring-damper combination was proposed in this paper. A linear motor is assumed to be used as the actuator of the active suspension system which produces active force to stabilize the sprung mass. A virtual spring-damper combination for the controller is going to be used in order to produce the active force of the linear motor.

The system was simulated for the large disturbances in the road profile with and without the active suspension. The effectiveness of the virtual spring-damper controller can be clearly observed in the results. The sprung mass was stabilized without oscillations in the active suspension system.

The sprung mass tends to oscillate slightly when there are ripples mix with large disturbances in the road profile. This limitation should be optimized further to achieve a better ride comfort.

The controlling methodology of the virtual damper should be changed when the road profile having small magnitude disturbances frequently. The force exerted by the virtual damper should be equal and opposite to the force generated by the actual damper in this scenario. A control mechanism of virtual damping coefficient for combining small magnitude disturbances and large magnitude disturbances should be developed.

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