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Modelling and Simulation of Full Vehicle Model with Variable Damper Controlled Semi-Active Suspension System

P. Sathishkumar[a], S. Rajeshkumar[b,c], T.S. Rajalakshmi[b,d], J. Thiyagarajan[b,e] and J. Arivarasan[b,f]

[a]*Automotive Engineering Research Institute, Jiangsu University, Zhenjiang, China*
Corresponding Author, Email: sathishkumar8989@gmail.com
[b]*Dept. of Mechatronics Engg., SRM Institute of Science and Technology, Kattankulathur, India*
[c]*Email: rajeshkumar.s@ktr.srmuniv.ac.in*
[d]*Email: rajalakshmi.ts@ktr.srmuniv.ac.in*
[e]*Email: thiyagarajan.j@ktr.srmuniv.ac.in*
[f]*Email: arivarasan.j@ktr.srmuniv.ac.in*

ABSTRACT:

The main objective of the variable damper controlled vehicle suspension system is to reduce the discomfort identified by passengers which arises from road roughness and to increase the ride handling related with the rolling, pitching and heave movements. This imposes a very fast and accurate variable damper to meet as much control objectives, as possible. The method of the proposed damper is to reduce the vibrations on each corner of vehicle by providing control forces to suspension system while travelling on uneven road. Numerical simulations on a full vehicle suspension model are performed in the Matlab Simulink toolboxes to evaluate the effectiveness of the proposed approach. The obtained results show that the proposed system provides better results than the conventional suspension system.

KEYWORDS:

Full vehicle model; 7 degree of freedom; Semi-active suspension; Passive suspension; Simulation

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1. Introduction

Suspension system is one of the main components of the vehicle. Its purpose is to provide ride comfort to passengers, to provide road-holding and competent handling abilities, and to give support to the vehicle static weight [1]. Suspension is a compromise between the sprung mass acceleration, the rattle space or suspension deflection which is the maximal allowable relative displacement between the sprung mass and suspension components and the dynamic tyre load. When the vehicle travels over a bumpy road, the vehicle is exposed to different motions and dynamic loads. The vehicle body should be well isolated from the road unevenness with minimal suspension deflection yet provide good handling performance. The suspension system is a term that denotes a system of springs, dampers and linkages which connect the vehicle body and wheels. The springs are assumed to have almost linear characteristics while most of the dampers exhibit the non-linear relationship between the velocity and the force [2-6].

In the last few decades, many researchers have been carried out to improve vehicle suspensions. Among the suggested solutions, semi-active and active suspension is a potential way to improve suspension performance although the passive suspension system can effectively handle some control of suspension system [7]. Passive systems have been designed to attain performance of the vehicle but intrinsic limits prevent them from attaining the best performances for both objectives. Replacement of the passive suspensions system of the car by semi-active suspension systems has the potential of improving comfort and safety under nominal conditions because active system offers more design flexibility [8-11]. This paper deals the modelling and full car suspension system with variable damper and passive suspension system.

2. Mathematical modelling

Fig. 1 shows that seven degree of freedom suspension model. It consist of sprung mass (M) i.e. the above the suspension vehicle weight and unsprung mass (m_1, m_2, m_3 and m_4) for front left, rear left, rear right and front right respectively.

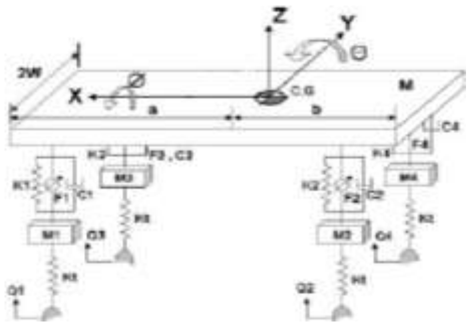

Fig. 1: Full car model

Using Newton's second law of motion and free-body diagram concept, the following equations of motion are derived [6, 12-13]. The Eqn. of motion for the sprung mass is given by,

$$M\ddot{Z} + (Z - \theta a + \varphi w - z_1)k_1 + (Z + \theta b + \varphi w - z_2)k_2 + (Z - \theta a - \varphi w - z_3)k_3 + (Z + \theta b - \varphi w - z_4)k_4 + (\dot{Z} - \dot{\theta} a + \dot{\varphi} w - \dot{z}_1)c_1 + (\dot{Z} + \dot{\theta} b + \dot{\varphi} w - \dot{z}_2)c_2 + (\dot{Z} - \dot{\theta} a - \dot{\varphi} w - \dot{z}_3)c_3 + (\dot{Z} + \dot{\theta} b - \dot{\varphi} w - \dot{z}_4)c_4 - F_1 + F_2 + F_3 - F_4 = 0 \quad (1)$$

The Eqn. of motion for the sprung mass's pitch and roll motions are given by,

$$I_{yy}\ddot{\theta} - (Z + \theta a - \varphi w - z_1)k_1 a - (Z + \theta b + \varphi w - z_2)k_2 b - (Z + \theta a + \varphi w - z_3)k_3 a - (Z + \theta b - \varphi w - z_4)k_4 b - (\dot{Z} + \dot{\theta} a - \dot{\varphi} w - \dot{z}_1)c_1 a - (\dot{Z} + \dot{\theta} b + \dot{\varphi} w - \dot{z}_2)c_2 b + (\dot{Z} + \dot{\theta} a + \dot{\varphi} w - \dot{z}_3)c_3 a - (\dot{Z} + \dot{\theta} b - \dot{\varphi} w - \dot{z}_4)c_4 b - F_1 a + F_2 b - F_3 a + F_4 b = 0 \quad (2)$$

$$I_{xx}\ddot{\varphi} - (Z - \theta a + \varphi w - z_1)k_1 w - (Z + \theta b - \varphi w - z_2)k_2 w - (Z + \theta a - \varphi w - z_3)k_3 w - (Z - \theta b + \varphi w - z_4)k_4 w - (\dot{Z} - \dot{\theta} a + \dot{\varphi} w - \dot{z}_1)c_1 w - (\dot{Z} + \dot{\theta} b - \dot{\varphi} w - \dot{z}_2)c_2 w + (\dot{Z} + \dot{\theta} a - \dot{\varphi} w - \dot{z}_3)c_3 w - (\dot{Z} - \dot{\theta} b + \dot{\varphi} w - \dot{z}_4)c_4 b + F_1 w + F_2 w - F_3 w - F_4 w = 0 \quad (3)$$

The Eqn. of motion for the unsprung mass for the front and rear left wheels are given by,

$$m_1\ddot{z}_1 - (Z + \theta a - \varphi w - z_1)k_1 - (\dot{Z} + \dot{\theta} a - \dot{\varphi} w - \dot{z}_1)c_1 + (z_1 - q_1)k_t + (\dot{z}_1 - \dot{q}_1)c_t - F_1 = 0 \quad (4)$$

$$m_2\ddot{z}_2 - (Z - \theta b - \varphi w - z_2)k_2 - (\dot{Z} - \dot{\theta} b - \dot{\varphi} w - \dot{z}_2)c_2 + (z_2 - q_2)k_t + (\dot{z}_2 - \dot{q}_2)c_t - F_2 = 0 \quad (5)$$

The Eqns. of motion for the unsprung mass for the rear and front right wheels are given by,

$$m_3\ddot{z}_3 - (Z + \theta a + \varphi w - z_3)k_3 - (\dot{Z} + \dot{\theta} a + \dot{\varphi} w - \dot{z}_3)c_3 + (z_3 - q_3)k_t + (\dot{z}_3 - \dot{q}_3)c_t - F_3 = 0 \quad (6)$$

$$m_4\ddot{z}_4 - (Z - \theta b + \varphi w - z_4)k_4 - (\dot{Z} - \dot{\theta} b + \dot{\varphi} w - \dot{z}_4)c_4 + (z_4 - q_4)k_t + (\dot{z}_4 - \dot{q}_4)c_t - F_4 = 0 \quad (7)$$

3. Magneto-rheological dampers

Semi-active suspension systems feature a damping ratio that can vary with bandwidth of 20-30Hz and relatively fast variation of damping can be attained by the electrohydraulic dampers that are hydraulic devices typically equipped with solenoid valve, Electro-rheological (ER) dampers and magneto-rheological (MR) dampers consisting of shock absorbers filled with rheological fluids, which may vary the viscosity of the fluid under the action of an electric or magnetic field respectively [14]. The description of a semi-active damper consists of two features. The first is signified by the force-velocity maps as a function of an electronic command. MR dampers exploit the physical properties of MR fluids. MR fluids vary their viscosity depending on the applied magnetic field. MR fluid comprises of a mixture of oil and micro-particles (iron particles) sensitive to the magnetic field. The MR fluid behaves as a liquid when no magnetic field is applied. In the case of a field applied to the fluid, the particles form chains and the fluid turns into very viscous [15]. The piston includes coils that are capable of delivering a magnetic field in the orifices. The piston may be worked as a MR valve, and the damping is the result of the friction between the orifice and the fluid. The MR characteristics can be affected by the phenomena such as variable friction and large hysteresis. [2 & 16-19]. The ideal and simplified characteristics of Magnet-rheological dampers are signified by the following relation [17],

$$\begin{cases} F_d(I, \dot{Z}) = C_0\dot{Z} + F_{MR}(I, \dot{Z}) \\ \quad 0 \le I \le I_{max} \end{cases} \quad (8)$$

Where C_0 is the damping given by the free flowing of the fluid through the piston orifices, Z is the suspension deflection, I is the current (electronic command) given to the coils, which must be between I = 0 and I = I_{max}. F_{MR} is the friction force between the piston orifices and the MR fluid.

4. ER dampers

ER dampers may change the damping abusing the physical property of the fluid flowing inside the shock absorber. ER fluids may consist of mixture of oil and micron sized particles which are sensitive to electric field. The fluid is almost free to flow in the damper orifices, when no field is applied. The particles react as dipoles and form chains, when the field is applied, so that the flowing of the fluid turns from free to visco-plastic. The ER damper can be viewed as a capacitor with the piston being a cathode and the external body an anode. In order to achieve the necessary forces, the surface of the piston is moderately large. The ideal and simplified characteristics of ER dampers are signified by the following relation [17],

$$\begin{cases} F_d(v, \dot{Z}) = C_0\dot{Z} + F_{ER}(v, \dot{Z}) \\ \quad 0 \le v \le v_{max} \end{cases} \quad (9)$$

Where C_0 is the damping given by the free flowing of the fluid through the piston Z is the suspension deflection. 'v' is the voltage (electronic command) given to the piston (cathode) and to the body of the damper (anode). FER is the friction force between the piston surface and the ER fluid and which is assumed as a non-linear function of suspension deflection speed $\dot{Z}$ and the command voltage (v).

5. Surface roughness characterization

Mainly four methods were used to define the basic properties of random data: root mean square values, auto correlation function, probability density function and power spectral density function. The PSD function signifies the rate of change of mean values with respect to frequency. It is assessed by computing the mean square value in a narrow frequency band at different frequencies, and dividing by the frequency band. The total area under the power spectral density curve over all frequencies will be the total mean value of the record. The partial area under the PSD curve from frequency f1 to f2 signifies the mean energy value of the record related to that frequency range. Fig. 2 shows the classification by ISO and the corresponding range of PSD values at a frequency of 1/2π cycles/m. Since the road surface roughness under consideration is a spatial disturbance, rather than a disturbance in time, it is necessary to define the PSD in terms of the spatial frequency. Ω, in cycles/m, rather than in terms of the conventional time frequency, f, in cycles/s. spatial frequency is additional depiction of frequency in the space domain. Then road surface roughness can be expressed in a concrete manner, disregarding the effect of various operating velocities of the vehicle. In terms of the spatial frequency argument, the power spectral density of the road surface is defined in the following manner [20-21],

$$S_y(\Omega) = \lim_{x\to\infty} \frac{2}{x} \left|\int_0^x y(x) e^{-2i\pi\Omega x}\, dx\right| 2 \quad (10)$$

Where S_y (Ω) is the one sided power spectral density of the road surface profile (m3/cycle), Ω is the spatial frequency (cycles/m), X is the length of course (m), x is the horizontal distance over surface (m), and y(x) is the surface elevation profile from a reference plane (m).

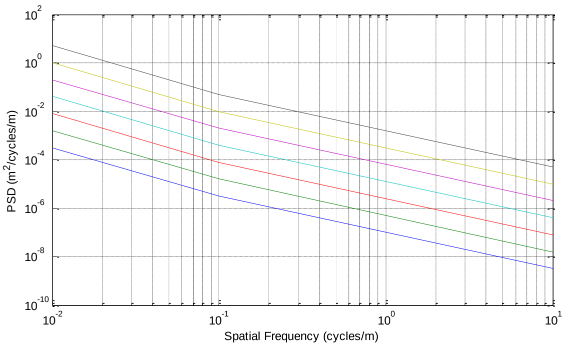

Fig. 2: Classification by ISO & corresponding range of PSD values

6. Results and discussion

To verify the variable damper controlled semi-active suspension systems, they are compared with passive suspension system. Passive system and semi active suspension system are all included in one analysis loop.

For the given input, the response of the system is observed on 100 seconds scale. Figs. 3-5 show the sprung mass vertical acceleration i.e. (Heave acceleration), lateral acceleration and longitudinal acceleration of passive suspension system and semi-active suspension system. The RMS of accelerations is given in Table 1. The peak acceleration of passive suspension is much higher than those of semi-active suspension systems. The heave acceleration is reduced by 53% than the passive suspension system. The roll acceleration of semi-active is reduced by 21% and pitch acceleration is reduced by 32% than the passive system.

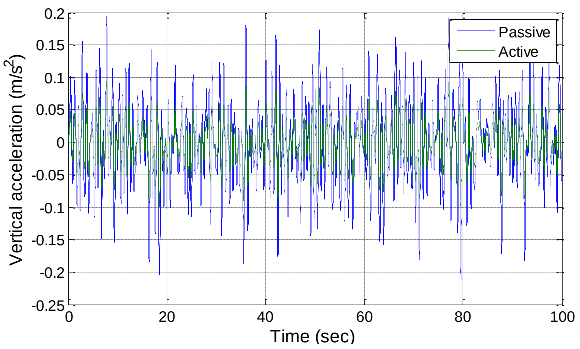

Fig. 3: Heave acceleration for passive & semi-active suspension

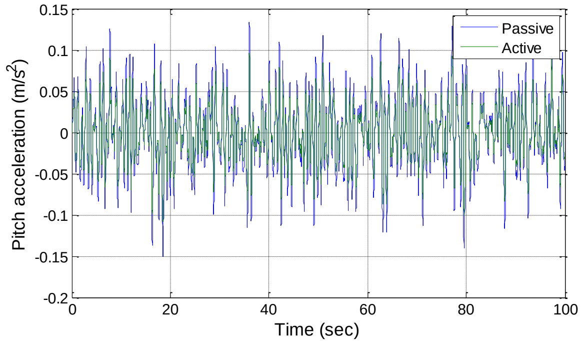

Fig. 4: Pitch acceleration for passive & semi-active suspension

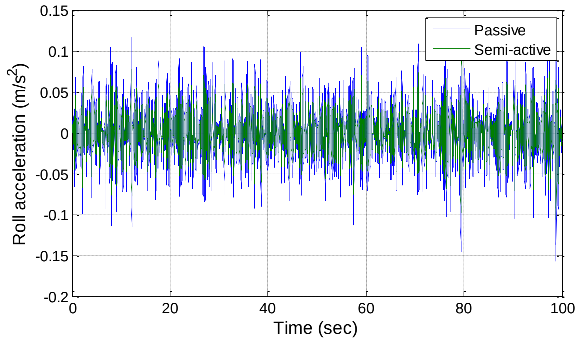

Fig. 5: Roll acceleration for passive & semi-active suspension

Table 1: RMS value comparisons for full car suspension system

Parameters	Passive	active	% reduction
Vertical acceleration (m/s^2)	0.0673	0.0316	53.04%
Roll acceleration (m/s^2)	0.0435	0.0340	21.83%
Pitch acceleration (m/s^2)	0.0360	0.0242	32.78

7. Conclusion

This study aimed at enhancing the vehicle ride performance for cars through simulation analysis by means of the control of the semi-active active suspension dampers. This method has enhanced the ride comfort of the vehicle suspension significantly for ISO random road profiles, where previous attention had only been on vertical acceleration. The use of variable dampers in some research literature has been with the damper simply modelled, without considering the dynamics of systems. But once it is properly represented, damper has been reported to contribute to a major degradation of the vehicle performance of lateral and longitudinal acceleration in contrast to the conventional. Therefore, there is motivation to concentrate on the enhancement of the vehicle comfort through the controls of the dampers across the suspension system.

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