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Sliding Mode Controller for Different Road Profiles of Active Suspension System for Quarter-Car Model

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Abstract The main purpose of the vehicle suspension system is to induce more comfortable riding and well handling with the road inputs. The principal objective of this research paper is to reduce vibration and improve passenger comfort of car suspension through the sliding mode controller. The mathematical model of passive and active suspensions mechanism for a quarter car model that subjects to excitation from road profiles is obtained. The active suspension system is developed through sliding mode control for a quarter car model. The performance of the sliding mode control is evaluated through a simulation approach using MATLAB and SIMULINK toolbox. The simulated results plotted in the time domain and root mean square values. It is concluded that the active suspension using a sliding mode controller improves the ride comfort and reduces vibration.

Keywords: *quarter car model, active suspension system, sliding mode control, road profile*

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1. Introduction

A car suspension system is used to separate the car body from the wheels and improve the ride comfort, road handling and stability of vehicles. Good vibration isolation and ride comfort should be provided by suspension system through holding a small acceleration of the body mass and a small rattle space which is the maximum allowable relative displacement between various suspension components and the car body. A passive suspension system which consists of spring and damper has the ability to store energy through the spring and dissipate it through the damper in order to achieve a certain level of compromise between load carrying, road handling and ride comfort. Though, an active suspension is capable of storing, dissipating and inserting energy to the suspension system in addition to alter its parameters depending upon operating conditions [1,2].

In general, the passive suspension system is believed as an open loop control system which attains certain condition only because it's fixed and cannot be adapted. For passive suspension system if it designs heavily damped or too hard suspension it will shift much of road roughness to the passengers. And also, if it quietly damped or soft suspension it will reduce the stability of the car in turns or alter lane or it will swing the car [3,4,5].

As reported by many researchers, the active suspension system can gave better performance of suspension through using force actuator, which is a close loop control system. The actuator is a mechanical part that enhanced the passive suspension system and operates through a controller which will calculate either feed or dissipate energy to the system via the help of sensors. Sensors will import the input data of road profile to the controller. Therefore, the active suspension system collects different operation conditions and controls it to improve the operation of the suspension system [6,7,8,9]. Various control strategies such as adaptive control [10], fuzzy control [11] H8 Control and optimal control have been proposed in the past years to control the active suspension system [12,13].

In this study, the mathematical model for the passive and active suspensions systems for quarter car model which subject to excitation from a different road profiles is constructed. The active suspension system is obtained based on sliding mode control (SCM) for a quarter car model. Comparison between passive and active suspensions system are conducted for different road profiles. The performance of the SCM is shaped MATLAB and SIMULINK toolbox. Comparison between passive and active suspensions system are examined.

2. Mathematical Modelling of Suspension System

By adding hydraulic actuators with controller to the passive suspension system it will be convert to active syspension system, as shown in Figure 1. When the active control system fails, the passive components come into action. The second-degree differential equations of motion for the suspension system can be written as follows:

$$\begin{gathered} M_w \ddot{x}_w + k_s\left(x_w - x_b\right) \\ + C_s\left(\dot{x}_w - \dot{x}_b\right) + K_t\left(x_0 - x_w\right) = 0.0 \end{gathered} \tag{1}$$

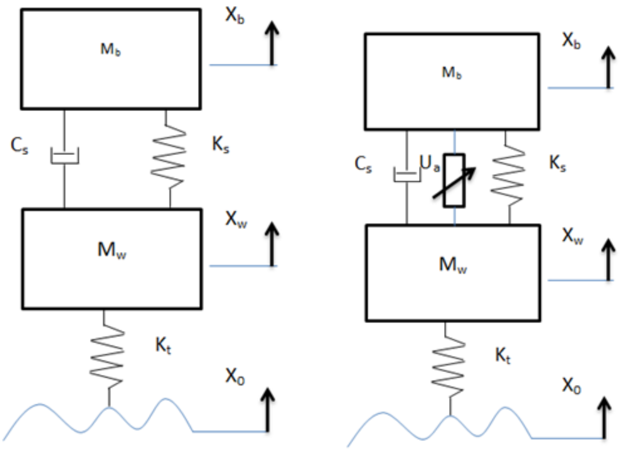
$$M_b \ddot{x}_b + k_s\left(z_w - z_b\right) + C_s\left(\dot{x}_w - \dot{x}_b\right) = 0.0 \tag{2}$$

Active suspension system:-

$$\begin{gathered} M_w \ddot{x}_w + k_s\left(x_w - x_b\right) \\ + C_s\left(\dot{x}_w - \dot{x}_b\right) + K_t\left(x_0 - x_w\right) + u_a = 0.0 \end{gathered} \tag{3}$$

$$M_b \ddot{x}_b + k_s\left(z_w - z_b\right) + C_s\left(\dot{x}_w - \dot{x}_b\right) - u_a = 0.0 \tag{4}$$

Where: u_a is the control force from the actuator. When the control force $u_a = 0$, then Equations (3, 4) become the equation of passive suspension system [1,3]. The control parameters are shown in Table 1.



(a) Passive suspension . (b) Active suspension.

Figure 1. Quarter car suspension model

Table 1. Parameter values of passive and active suspension

No	Symbol	Value
1	M_b(Sprung mass)	250 kg
2	M_{us}(Unsprung mass)	50 kg
3	K_a(stiffness coefficient)	16812 N/m
4	C_a(Damping coefficient)	1000 N.sec/m
5	K_t(Tyre stiffness)	190000 N/m

3. Road Profiles

In this paper, three types of road profiles are simulated namely; step input, random and bump road profiles. Step input where $x_0 = 0.1$, as shown in Figure 2 and Bump road profile, as shown in Figure 3. The third road is random road profile, as shown in Figure 4.

Numerous researchers report that when the car speed is constant, the roughness of the road is a stochastic character that is considered by Gauss distribution not by mathematical equations. The power spectral density of the vehicle is a constant, that couple with the statistical features of the white noise in order to it can be considered as road roughness time domain model. Corresponding to the ISO/TC108/SC2N67 international standard, it is urged that the power spectral density of road vertical elevation (PSD), $G_q(n_o)$, depicted as the following formula as a fitting expression [10]:

$$G_q(n) = G_q(n_0)\left(\frac{n}{n_0}\right)^{-w} \tag{5}$$

n: Spatial frequency, the reciprocal of the wavelength, unit m^{-1};
n_0: Reference Spatial frequency, $n_0 = 0.1\ m^{-1}$;
$G_q(n_0)$: Road roughness coefficient, unit m^2/m^{-1};
w: Frequency index, it determines the road surface frequency spectrum structure, usually take the frequency index $w = 2$. The road surface vertical displacement power spectral density (PSD), $G_q(n_0)$, defined as the following formula as a fitting expression:

$$\frac{q(s)}{w(s)} = \frac{2\pi n_0\sqrt{G_q(n_0)v}}{s + 2\pi f_0} \tag{6}$$

(PSD), values are showed in Table 2. Class C road profile is chosen in this paper to be the main road profile. Where $w(t)$ is Gaussian white noise filter, $V = 20\ m/s$, $Gq\ (no) = 256e-6\ m^3$, $no = 0.1\ m^{-1}$ and $f_o = 0$.

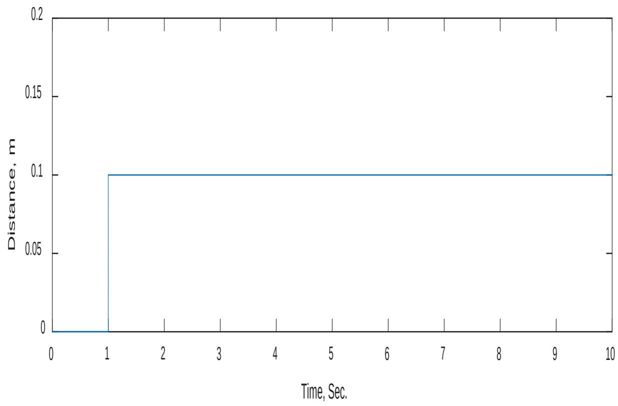


Figure 2. Step input signal

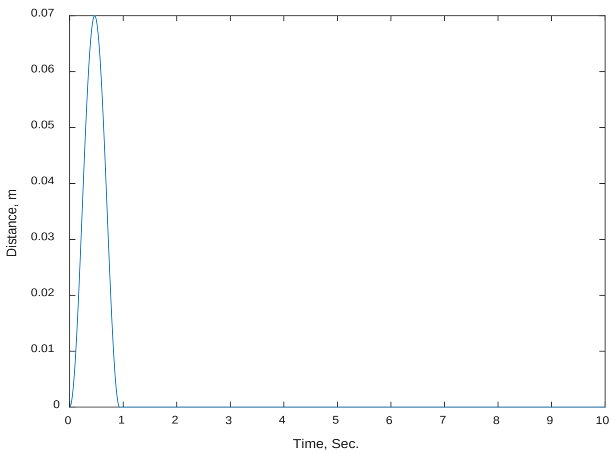


Figure 3. Bump road input

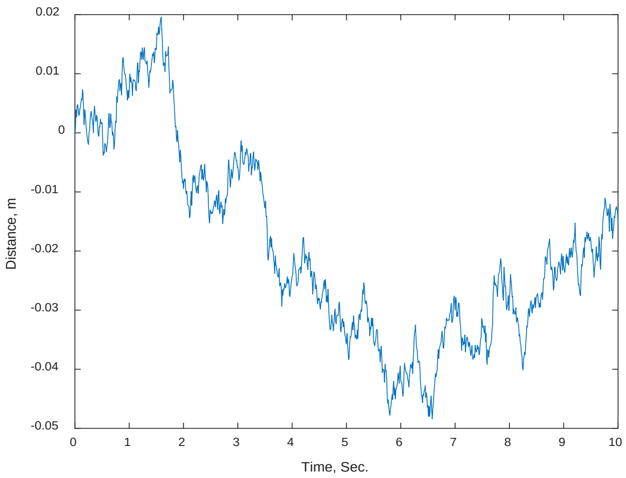


Figure 4. Class C Road roughness signal

Table 2. Eight degree of road roughness [10]

Road Level	$G_q(n_0)/(10^{-6}m^3)$ $n_0 = 0.1\,m^{-1}$	$\sigma_q(n_0)/(10^{-6}m^3)$ $0.11m^{-1} < n > 2.83\,m^{-1}$
	Geometric Average	Geometric Average
A	16	3.81
B	64	7.61
C	256	15.23
D	1024	30.45
E	4096	60.90
F	16384	121.80
G	65536	243.61
H	262144	487.22

4. Sliding Mode Control Design

The main aim of control design is to provide the desired dynamic behavior of passenger car under road variations. Where u_a is the control force from the hydraulic actuator. It can be noted that if the control force $u_a = 0$, then Equations (3, 4) become the equation of passive suspension system [1,3]. Considering ua as the control input, the state-space representation of Equations (3, 4) become,

$$\dot{z}_1 = z_2 \tag{7}$$

$$\dot{z}_2 = -\frac{1}{M_s}\left[K_s\left(z_1 - z_3\right) + C_a\left(z_2 - z_4\right)\right] \tag{8}$$

$$\dot{z}_3 = z_4 \tag{9}$$

$$\dot{z}_4 = \frac{1}{M_{us}}\left[K_s\left(z_1 - z_3\right) + C_a\left(z_2 - z_4\right) + K_t\left(z_3 - z_r\right)\right] \tag{10}$$

Where $z_1 = z_s$, $z_2 = \dot{z}_s$, $z_3 = z_{us}$ and $z_4 = \dot{z}_{us}$.

In order to develop an active suspension system, The hydraulic actuator installed between sprung mass and un-sprung mass is shown in Figure 5, including a valve and a cylinder, where U_h is the actuator force obtained through the hydraulic piston and ($x_{act} = x_1 - x_3$) is the actuator displacement. U_h (equal to U_a) is employed dynamically to improve ride comfort with varing input signals [8].

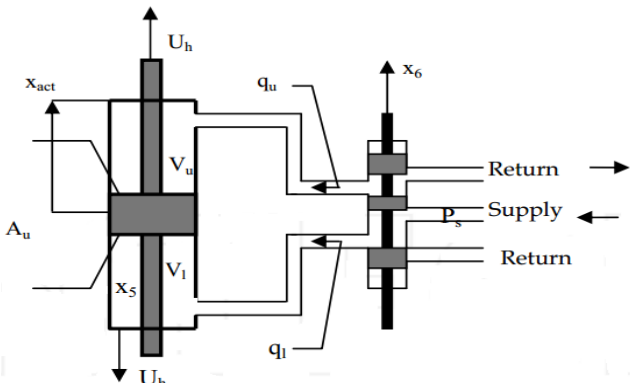


Figure 5. Hydraulic valve and cylinder

The predicted equations of the active suspension system, including the hydraulic actuator is rewritten in Equations (11) to (15).

$$\dot{x}_1 = x_2 \tag{11}$$

$$\dot{x}_2 = \frac{K_s}{M_s}x_1 + \frac{C_a}{M_s}x_2 + \frac{K_s}{M_s}x_3 + \frac{C_a}{M_s}x_4 + \frac{A_1}{M_s}x_5 \tag{12}$$

$$\dot{x}_3 = x_4 \tag{13}$$

$$\begin{aligned}\dot{x}_4 = &\frac{K_s}{M_{us}}x_1 + \frac{C_a}{M_{us}}x_2 + \frac{K_s + K_t}{M_{us}}x_3 \\ &+ \frac{C_a}{M_{us}}x_4 + \frac{A_1}{M_s}x_5\end{aligned} \tag{14}$$

$$\dot{x}_5 = \beta x_5 + A\left(x_2 - x_4\right) + x_6\omega_3 \tag{15}$$

$$\dot{x}_6 = \frac{x_6}{\tau} + U_c \tag{16}$$

Where, $\omega_3 = \mathrm{sgn}[P_s - \mathrm{sgn}(x_6)x_5]\sqrt{|P_s - \mathrm{sgn}(x_6)x_5|}$.

Thus equations become state feedback model of active suspension system including hydraulic dynamics. The sliding surface given by $\sigma = x_2 + z, z(0) = -x_2(0)$ is selected. The auxiliary variable z is defined as:-

$$\dot{z} = -\frac{1}{M_s}\left[-K_s\left(x_1 - x_3\right) - C_a\left(x_2 - x_3\right)\right]. \tag{17}$$

Differentiating σ and using (7) and (17) and with $e = -f_s - f_d$ is the lumped uncertainty.

$$\dot{\sigma} = \frac{1}{M_s}e + \frac{1}{M_s}u - \frac{1}{M_s}\begin{bmatrix}-K_s\left(x_1 - x_3\right) \\ -C_a\left(x_2 - x_3\right)\end{bmatrix} \tag{18}$$

Assumption 1. The lumped uncertainty e is such that $\left|\frac{de}{dt}\right| < \mu$, , where μ is a small number [9].

Now, the control u is designed, where u_n is the component that takes care of uncertainty in (16). Let:

$$u = \frac{M_s}{M_s}e\begin{bmatrix}-K_s\left(x_1 - x_3\right) \\ -C_a\left(x_2 - x_3\right)\end{bmatrix} - M_sK\sigma + u_n \tag{19}$$

Where K is a positive number. Using (17) in (16)

$$\dot{\sigma} = \frac{1}{M_s}e - K\sigma + \frac{1}{M_s}u_n. \tag{20}$$

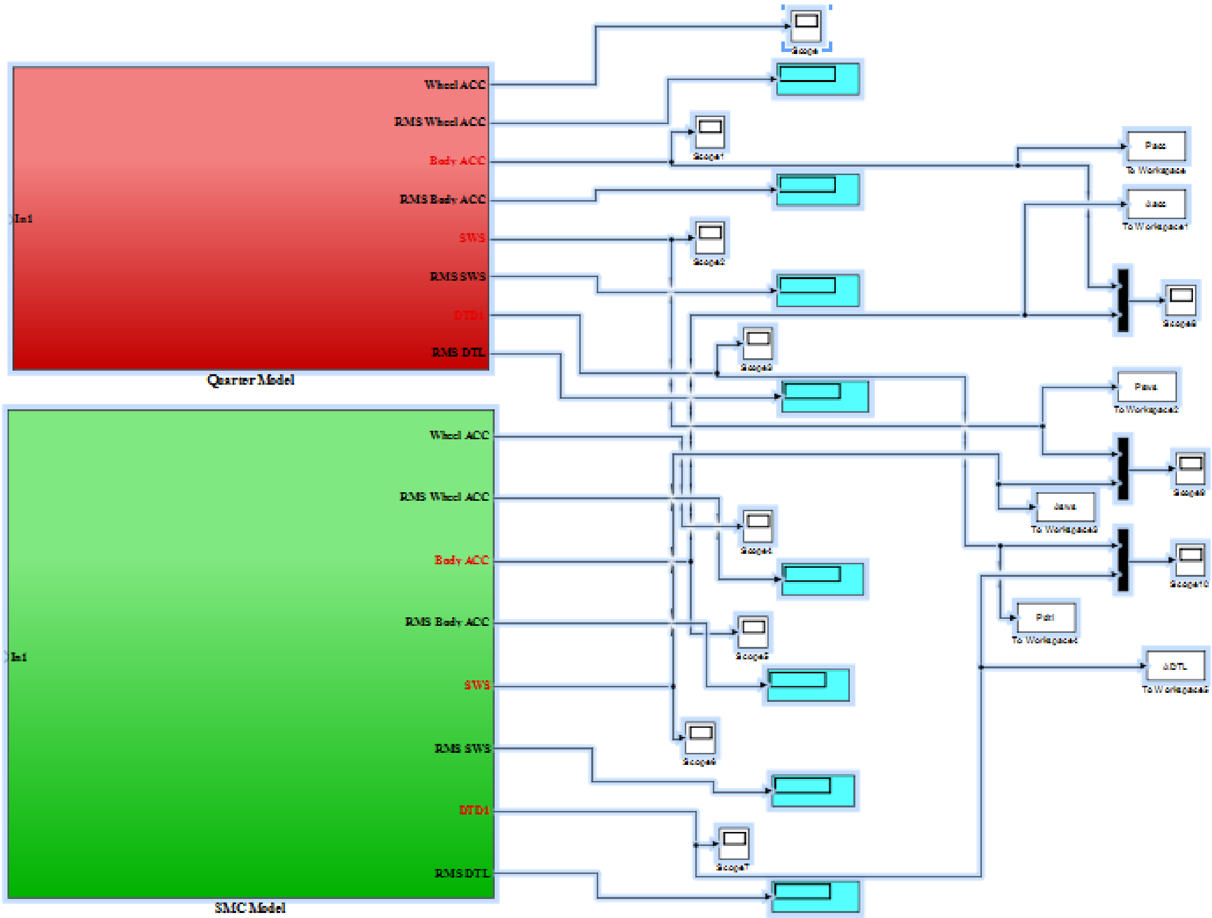


Figure 6. Simulink model of active suspension system using SCM

Let ê be the estimate of uncertainty e. Define the estimation error as:-

$$\tilde{e} = e - \hat{e}. \tag{21}$$

Now selecting

$$u_n = -\hat{e} \tag{22}$$

and substituting (22) in (20), we get

$$\dot{\sigma} = -K\sigma + \frac{1}{M_s}\tilde{e}. \tag{23}$$

If the estimate ê is such that ẽ goes to zero asymptotically, sliding condition will be satisfied and thereby model following can be assured. Simulink model of active suspension system using SCM is shown in Figure 6.

5. Results and Discussions

Simulation based on the mathematical model for quarter car using MATLAB/SIMULINK software is performed. Evaluation of the suspension system in term of ride comfort and car handling is detected, where road disturbance is accepted as the input for the dynamic system. Suspension travel, tyre load, and body acceleration for quarter car are obtained for the passive and active suspension with different road profiles, as shown in the following sections:

5.1. Step Road Input

For the step road input, the ride comfort parameters are showed in time domain, as shown Figure 7. The relations between body acceleration, suspension travel and wheel load with time domain is plotted. It is showed that active suspension system using SCM gives better ride performance than passive suspension system. The reduction percentage in r.m.s (root mean square) values of the various parameters for the step road input is showed in Figure 8. It is found that the active suspension using SCM capable to reduce body acceleration which increase ride comfort upto 71%.

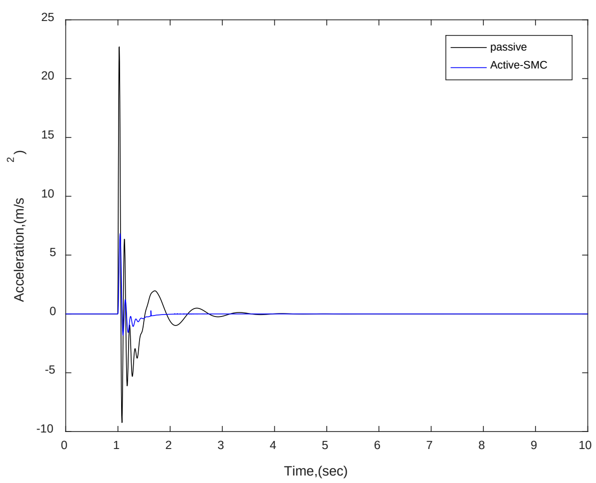


Figure 7a. Car body acceleration for step road input

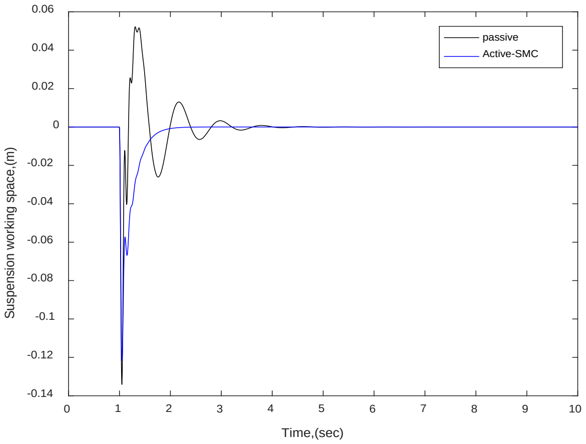


Figure 7b. Suspension travel for step road input

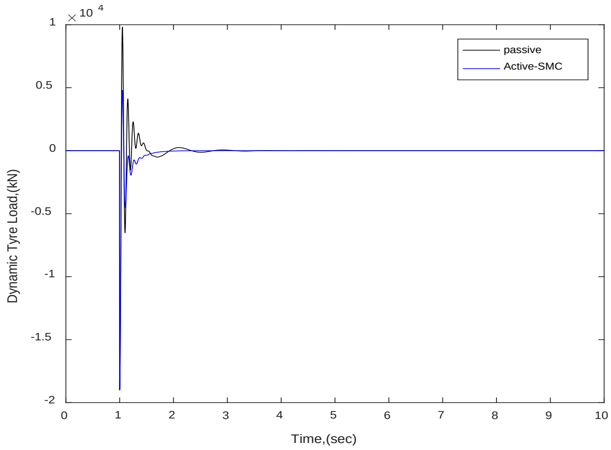


Figure 7c. Wheel load for step road input

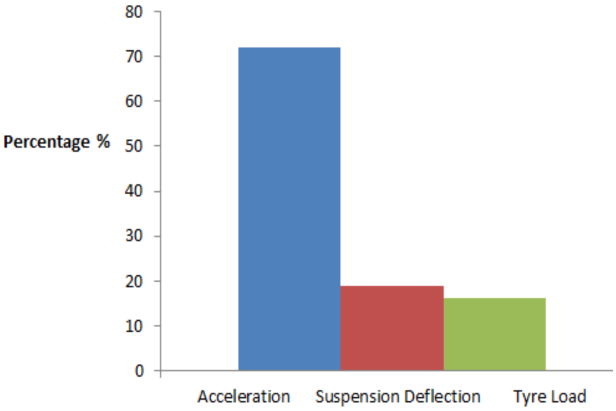


Figure 8. Reduction percentage of active suspension over passive suspension for step road input.

5.2. Bumpy Road Input

For the bumpy road input, Figure 9 illustrates the acceleration, suspension working space and dynamic tyre load in time domain. It is indicated that active suspension system using sliding mode control gives better ride performance than passive suspension system. Figure 10 shows the percentage reduction in r.m.s values of the various parameters for the sinusoidal input (Bump road). It is found that the active suspension using SCM reduces body acceleration up to 74% and improves ride comfort.

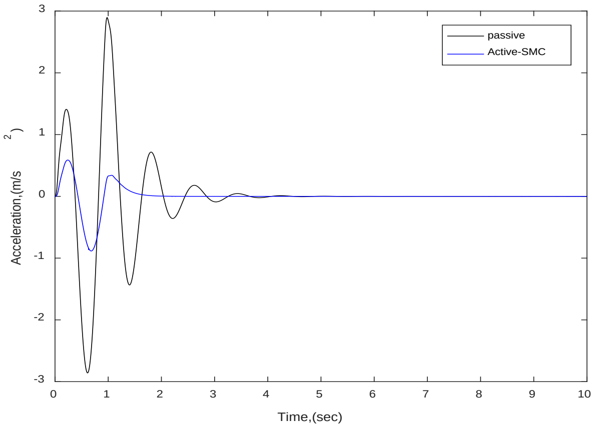


Figure 9a. Car body acceleration for bumpy road input

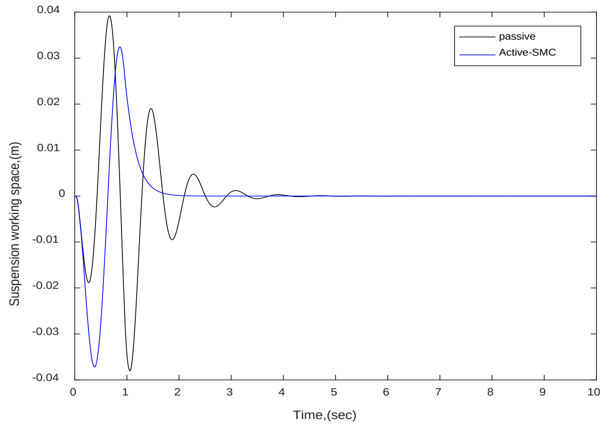


Figure 9b. Suspension travel for bumpy road input

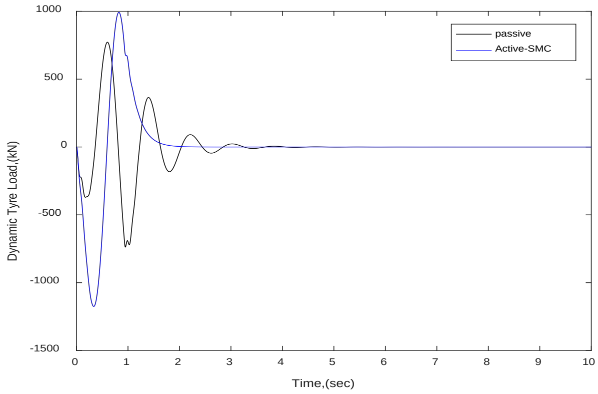


Figure 9c. Wheel load for bumpy road input

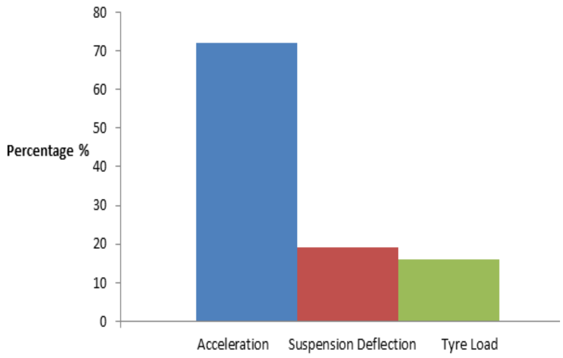


Figure 10. Reduction percentage of active suspension over passive suspension for bumpy road input

5.3. Random Road Input

Figure 11 shows ride comfort of sprung mass acceleration, suspension working space and tyre load in time domain for the random road input. It is showed that active suspension system using SCM gives better ride performance than passive system. Figure 12 show the percentage reduction in r.m.s values of the ride comfort parameters for the random road. It is observed that the active suspension using SCM reduces body acceleration upto 72% and improves ride comfort for the random road profile.

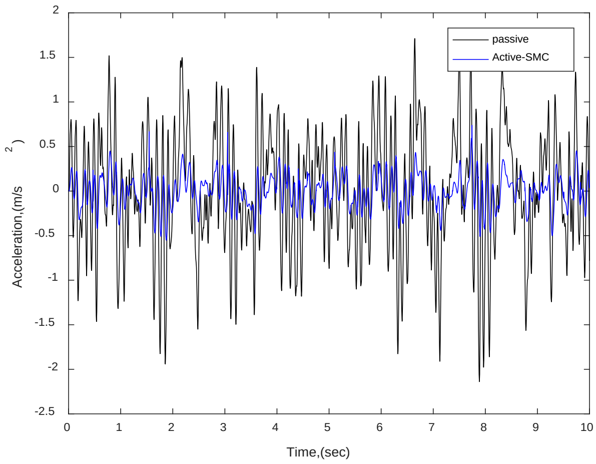


Figure 11a. Car body acceleration for random road input

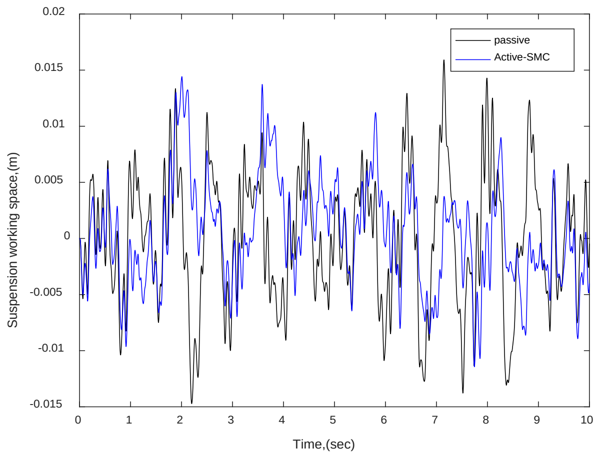


Figure 11b. Suspension travel for random road input

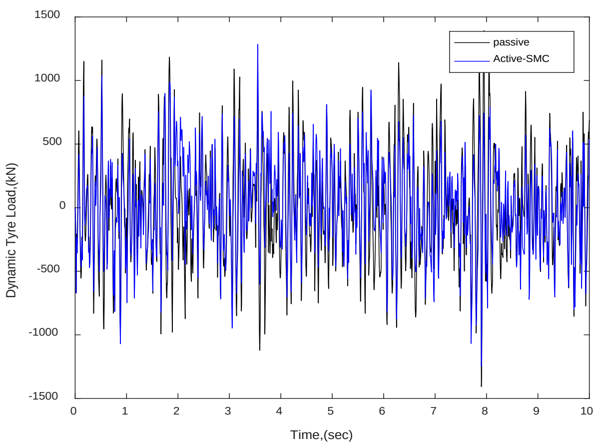


Figure 11c. Wheel load for random road input

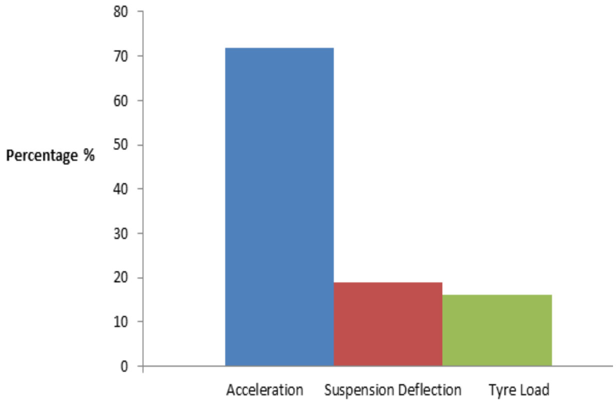


Figure 12. Reduction percentage of active suspension over passive suspension for random road input

6. Conclusion and Further Work

The paper presents a control option for an active suspension system. Mathematical modeling is performed using a quarter car models for passive and active suspension system considering only bounce motion to evaluate the parameters of ride comfort. SCM design approach is examined for the active suspension system for three types of road namely: step, bumpy and random road profiles. The results showed that active suspension system using SCM improves the body acceleration and ride comfort better than the passive suspension system. The future work is directly related with expanding the model to half and full passenger car model active suspension system using SCM.

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