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Fuzzy Sliding Mode Control for Active Vibration Control of Hydraulic Actuated Vehicle Suspension

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Abstract: ***Fuzzy Sliding model control (FSMC) has been suggested to control hydraulic actuated nonlinear active suspension using full car model. The development of efficient classical controller for active suspension has been difficult due to complex nature of full car mathematical model, nonlinear effect in suspension and unpredictable behaviour of hydraulic actuator. The sliding mode controller (SMC) has an ability to handle uncertain parameter, nonlinearity and complex mathematical model of dynamic system. Chattering phenomenon remains to be the only obstacle for nonlinear sliding mode controller which can be eliminated by integrating fuzzy logic with sliding mode control. In this work, fuzzy sliding mode controller proposed for hydraulic actuated nonlinear active suspension to handle nonlinearity, uncertainty in parameter variation and chattering effect. The simulation is carried out using Seven degree of freedom based full car model to measure body displacement, acceleration, roll and pitch angle of vehicle. The response of full car model confirms the feasibility of fuzzy sliding mode controller for hydraulic actuated active suspension.***

Keywords: ***Fuzzy sliding mode control, Vibration control, Active suspension, Actuator dynamics.***

I. INTRODUCTION

A suspension system consist of spring and oil damper between vehicle body and tire to control the vertical moment of vehicle body. The main purpose of suspension system is to reduce the road disturbance by minimizing the vehicle displacement and acceleration. The suspension system which imparts better ride comfort depends on soft suspension system whereas the suspension system which give better road handling capacity depends on hard suspension. The optimization techniques is used to design of such suspension where design parameters are selected by compromising on soft and hard suspension. There is no external energy source on passive suspension and its stiffness and damping values are almost constant. The active suspension systems are closed loop systems where the suspension travel measured to predict the actuator force needed for active suspension. Nowadays, many researcher focusing their research on active suspension systems due to its ability to operate wide range frequency [1-2]. Control theory [3-5] to design a controller for active suspension. The development of computer and micro-processor improved practical implementation of active suspension in automotive industries. A quarter car model has 1/4th of vehicle mass, spring and damper which is placed between wheel mass and vehicle mass. The quarter car model is a simplified model of car with two degree of freedom by using that the vertical motion of car body and wheel has been measured for active suspension. This model will be used to represent heave motion of 1/4th of vehicle mass. Initially, model based controller is designed with quarter car model for active suspension. But it cannot be used to measure the pitch motion of wheels. Alleyne and liu proposed a nonlinear control technique for hydraulic operated quarter car active suspension [6]. A nonlinear control law formulated to control dynamic nature of hydraulic actuator.

Half car model will be used to represent pitch and heave motion of vehicle body. Vehicle body coupled between front and rear wheel by center of gravity. Nurkan Yagiz et al proposed that the half car model based sliding mode control combined with Fuzzy logic [7]. The effect of chattering can be eliminated completely by adding fuzzy controller for sliding mode control. Both quarter car and half car model does not model the actual system for practical application. H infinity controller introduced for full car model to suppress the causes produced by road disturbances and parameter uncertainty in actuator dynamics [8]. A preview controller for full car model based active suspension introduced with two control approach [9].

The first controller optimize the displacement of actuator whereas the second controller control the roll, heave and pitch of vehicle body. The complexity in mathematical model for full car and non-linear behavior of actuator has increased the difficulties of applying conventional control schemes to active suspension system. Hence a model free controller based on intelligent control schemes like fuzzy logic, neural network are gaining more importance in recent times and they are applied successfully to control suspension system in real time.

The sliding mode control (SMC) introduced by utkin [10] is type of nonlinear control which is insensitive to external disturbance. The SMC control is commonly used in Robotics [11], mechatronics and Active suspension [12] .Kim and Ro proposed sliding mode control for nonlinear quarter car model [13]. Sam et al. introduced proportional-integral sliding mode controller for active suspension [14]. Yagiz et al developed sliding mode controller without chattering effect for automotive application [15].

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The Fuzzy logic introduced by professor lotfi zadeh in his seminar paper fuzzy algorithm in 1965 [17]. The Fuzzy is based on multi value logic where the true values lies in between 1 and 0.The performance of fuzzy depends on type and nature of linguistic variable whereas the values of variable graded using membership function. Fuzzy logic rules are written based on the behavior of system and then it will be converted in to equivalent mathematical model of a system. Fuzzy controller are simple and flexible hence it can handle imprecise data very well. Fuzzy logic can easily model the nonlinear function of any system.

Mamdani introduced fuzzy logic in control system for practical application. The control rules in fuzzy is written depends up on the knowledge of expert. Fuzzy has an ability to develop controller with-out any mathematical model of a system. The complex nonlinear behavior of actuator and its dynamics has been controlled effectively using fuzzy logic controller. In the fuzzy controlled active suspension, suspension deflection and its rate of change act as input whereas the control signal act as output to fuzzy controller. Hybrid intelligent algorithms was developed for active suspension using quarter car model where control signal is combination of two types of control.

Quarter car model based active suspension with Self-organizing fuzzy sliding mode controller (SFSC) created using sliding surface and its change as input to fuzzy controller. The stability of system improved by adaptive law of SFSC [19]. Takagi–Sugeno (T–S) fuzzy model is proposed to handle the nonlinear effect of active suspension system using half car model [20-23]. A fuzzy sliding-mode controller for half car model is introduced to remove chattering effect in active suspension where sliding mode controller combined with fuzzy logic controller to give better performance for suspension.

II. MODELING OF ACTIVE SUSPENSION

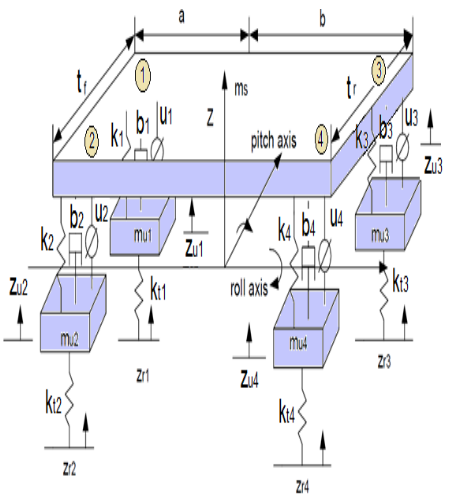

Figure 1. Schematic Representation of 7 DOF Full car model

A nonlinear full car model with 7 DoF as shown in Fig. 1 is used for modeling of active suspension [24]. The full car model consist of vehicle body mass m_s and four tires of mass m_{u1}, m_{u2}, m_{u3} and m_{u4} respectively. The four corners of vehicle body denoted by z_1, z_2 ,z_3 and z_4 respectively in order to measure the vertical displacement of vehicle body at each corner. A stiffness of k and damping coefficient of b along with actuator force u is placed in between vehicle body and tire on each corner. The vertical motion of tire is indicated by z_{u1}, z_{u2}, z_{u3} and z_{u4} respectively. The stiffness of tire at each corner is modeled by spring of stiffness k_t. The up and down motion of vehicle body along pitch axis is represented by symbol θ. Similarly α represent rolling motion of vehicle body along roll axis. The displacement of road for each tire on each corner is indicated by road profile z_{r1}, z_{r2}, z_{r3} and z_{r4}. The 7 Degree of freedom of full car model are represented as four vertical tire displacement namely z_{u1}, z_{u2}, z_{u3}, z_{u4} , heave z, Pitch θ, Roll α of an vehicle body.

The distance of CG from front and rear end of vehicle indicated by symbol a and b respectively. T_f and t_r indicates front and rear treat of vehicle body. The actuator arranged vertically between sprung and unsprung mass has been used to provide actuator force to active suspension. The full car model has ability to measure the pitch and roll motion of car boy which cannot be possible to measure in quarter car model.

A. Tire Model

The equation of motion for nonlinear full car model is derived from newton's second law, where the nonlinearity of suspension spring represented in cubic term . The vertical motion of tire on each corner for an external disturbance can be modelled as spring mass system by neglecting nonlinearity of damper as described in equation (1)-(4).

$$m_{u1}\ddot{z}_{u1} = b_1(\dot{z}_1 - \dot{z}_{u1}) + k_1(z_1 - z_{u1}) + k_{n1}(z_1 - z_{u1})^3 + k_{t1}(z_{r1} - z_{u1}) - u_1 \quad (1)$$

$$m_{u2}\ddot{z}_{u2} = b_2(\dot{z}_2 - \dot{z}_{u2}) + k_2(z_2 - z_{u2}) + k_{n2}(z_2 - z_{u2})^3 + k_{t2}(z_{r2} - z_{u2}) - u_2 \quad (2)$$

$$m_{u3}\ddot{z}_{u3} = b_3(\dot{z}_3 - \dot{z}_{u3}) + k_3(z_3 - z_{u3}) + k_{n3}(z_3 - z_{u3})^3 + k_{t3}(z_{r3} - z_{u3}) - u_3 \quad (3)$$

$$m_{u4}\ddot{z}_{u4} = b_4(\dot{z}_4 - \dot{z}_{u4}) + k_4(z_4 - z_{u4}) + k_{n4}(z_4 - z_{u4})^3 - k_{t4}(z_{r4} - z_{u4}) - u_4 \quad (4)$$

In the above equation, k_{ni} denotes nonlinear stiffness of suspension at each corner, where i =1…4. The vertical displacement of car body at corner 1,2,3,4 can be represented using heave z, pitch angle, roll angle of vehicle body as follow.

$$z_1 = z + t_f\phi_s + a\theta_s$$
$$z_2 = z - t_f\phi_s + a\theta_s$$
$$z_3 = z + t_r\phi_s - b\theta_s$$
$$z_4 = z - t_r\phi_s - b\theta_s \quad (5)$$

The road disturbance for each wheel of full car model indicated in equation (6)-(8). There will be small change in height of road Z_{r1} and Z_{r2} in order to measure the roll angle of full car model.

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$$Z_{r1} = \begin{cases} 0.15\left(1-\cos\left(2\pi t\right)\right) & 1 \le t \le 2 \\ 0 & otherwise \end{cases}$$

(6)

$$Z_{r2} = \begin{cases} 0.1\left(1-\cos\left(2\pi t\right)\right) & 1 \le t \le 2 \\ 0 & otherwise \end{cases}$$

(7)

$$Z_{r3,r4} = \begin{cases} 0.1\left(1-\cos\left(2\pi t\right)\right) & 6 \le t \le 7 \\ 0 & otherwise \end{cases}$$

(8)

B. Active Suspension Model

The equation of motion of active suspension including heave, pitch and roll angle of vehicle body described in equation (9), (10) and (11).

$$\begin{aligned} I_r\ddot{\varphi} = & -b_1t_f(\dot{z}_1-\dot{z}_{u1})+b_2t_f(\dot{z}_2-\dot{z}_{u2})-b_3t_r(\dot{z}_3-\dot{z}_{u3})+b_4t_r(\dot{z}_4-\dot{z}_{u4})-k_1t_f(z_1-z_{u1})-k_{n1}t_f(z_1-z_{u1})^3 \\ & +k_2t_f(z_2-z_{u2})+k_{n2}t_f(z_2-z_{u2})^3-k_3t_r(z_3-z_{u3})-k_{n3}t_r(z_3-z_{u3})^3+k_4t_r(z_4-z_{u4})+k_{n4}t_r(z_4-z_{u4})^3 \\ & +t_fu_1-t_fu_2+t_ru_3-t_ru_4 \end{aligned}$$

(9)

$$\begin{aligned} I_p\ddot{\theta} = & -b_1a(\dot{z}_1-\dot{z}_{u1})-b_2a(\dot{z}_2-\dot{z}_{u2})+b_3b\left(\dot{z}_3-\dot{z}_{u3}\right)+b_4b\left(\dot{z}_4-\dot{z}_{u4}\right)-k_1a\left(z_1-z_{u1}\right)-k_{n1}a\left(z_1-z_{u1}\right)^3 \\ & -k_2a\left(z_2-z_{u2}\right)-k_{n2}a\left(z_2-z_{u2}\right)^3+k_3b\left(z_3-z_{u3}\right)+k_{n3}b\left(z_3-z_{u3}\right)^3+k_4b\left(z_4-z_{u4}\right)+k_{n4}b\left(z_4-z_{u4}\right)^3 \\ & +au_1+au_2+au_3+au_4 \end{aligned}$$

(10)

$$\begin{aligned} m_s\ddot{z} = & -b_1\left(\dot{z}_1-\dot{z}_{u1}\right)-b_2\left(\dot{z}_2-\dot{z}_{u2}\right)-b_3\left(\dot{z}_3-\dot{z}_{u3}\right)-b_4(\dot{z}_4-\dot{z}_{u4})-k_1\left(z_1-z_{u1}\right)-k_{n1}\left(z_1-z_{u1}\right)^3-k_2\left(z_2-z_{u2}\right) \\ & -k_{n2}\left(z_2-z_{u2}\right)^3-k_3\left(z_3-z_{u3}\right)-k_{n3}\left(z_3-z_{u3}\right)^3-k_4\left(z_4-z_{u4}\right)-k_{n4}\left(z_4-z_{u4}\right)^3+u_1+u_2+u_3+u_4 \end{aligned}$$

(11)

C. Actuator Model

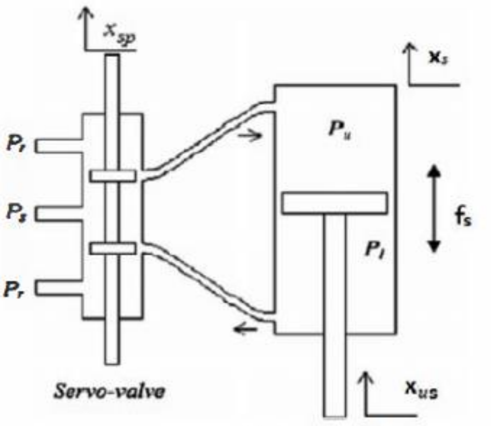

Figure 2. Representation of hydraulic actuator

Modelling of hydraulic actuator with four way piston - valve arrangement is as shown in Fig. 2. The force exerted by actuator is f_a=AaP_L where Aa and P_L represent area of piston and pressure drop inside cylinder respectively.

The rate of change of pressure drop expressed as in equation (12).

$$\frac{V_t}{4\beta_e}\dot{P}_L = C_{tp}P_L - A_a\left(\dot{x}_b-\dot{x}_w\right)+Q$$

(12)

Q is the hydraulic flow which is derived from Equation (13).

$$Q = C_d w x_v \sqrt{\frac{1}{\rho}\left[P_s - \operatorname{sgn}\left(x_v\right)P_L\right]}$$

(13)

The actuator force is described by the nonlinear equation (14).

$$\dot{u} = \dot{f}_s = -\beta f_s - \alpha A_a^2\left(\dot{x}_b-\dot{x}_w\right)+A_a x_v\sqrt{P_s - \frac{\operatorname{sgn}\left(x_v\right)f_s}{A_a}}$$

(14)

Where α = 4βe/Vt, β = αCtp, $\Gamma = \alpha\, cd\, w\sqrt{\frac{1}{\rho}}$ and $x_v = k_c U$. k_c is the gain of conversation, and u is the voltage of servo valve . The values for the full car model is represented in Table 1.

Symbol	Parameter	Value
m_s	mass of the car body	1136 kg
m_{u1}, m_{u2}	mass of the front wheel at corner 1&2	63 kg
m_{u3}, m_{u4}	mass of the front wheel at corner 3 & 4	60 kg
I_p	Pitch moment of inertia	2400 kgm^2
I_r	Roll moment of inertia	400 kgm^2
b_1, b_2	Damping coefficient of front suspension at corner 1&2	3924 Ns/m
b_3, b_4	Damping coefficient of rear suspension at corner 3	2943 Ns/m
k_1, k_2	Stiffness of front suspension at corner 1&2	36297 N/m
k_3, k_4	Stiffness of rear suspension at corner 3 &4	19620 N/m
k_{n1}, k_{n2}, k_{n3}, k_{n4}	Nonlinear Stiffness of front suspension at corner 1,2,3,4	850 N/m
k_{t1}, k_{t2}, k_{t3}, k_{t4}	Linear Stiffness of tire at corner 1,2,3,4	182470 N/m
a	Distance between front end to CG of Vehicle body	1.15 m
b	Distance between rear end to CG of Vehicle body	1.65 m
t_f	Front treat	0.505 m
t_r	Rear treat	0.557 m
P_s	Hydraulic pressure	10,342,500 Pa
A_a	Actuator ram area	3.35x10^{-4} m^2
k_c	Conversion gain	0.001 m/V
α	α	4.515x 10^{13} N/m^5
B	B	1 s^{-1}
Γ	Γ	1.545x 10^9 N/$m^{5/2}kg^{1/2}$

Table 1. Values of Parameter used in full car model

III. CONTROLLER DESIGN

A. Sliding Mode Control (SMC)

The sliding mode control (SMC) developed by Utkin . The application of SMC and variable structure problem introduced in reference . The theoretical work of SMC was developed by Slotine and Edwards and Spurgeon. Sliding mode control (SMC) received more attention by researcher in active suspension because it can handle the actuator dynamics very well. Design of SMC is based on reaching phase and sliding phase. During the reaching phase, the system states moves towards sliding surface using Lyapunov's stability theory. During the sliding phase, the system slides through sliding surface towards origin.

Sliding mode control produce control signal $u(t_i)$ in such way that the output of active suspension system track desired output of system $y_d(t_i)$ in order to minimize the error defined in (15-16).

$$e(t_i) = y_a(t_i) - y_d(t_i)$$

(15)

$$e(t_i) = [e(t_i), \dot{e}(t_i)......\overset{n-1}{e}(t_i)]^T \in R$$

(16)

The sliding surface for an *n*th-order system written as

$$\sigma(x,t_i) = \left(\frac{d}{dt} + \lambda\right)^{n-1} e(t_i)$$

(17)

n indicates order of the system. Lypunov function (18) and stability condition (19) is expressed as.

$$V = \frac{1}{2}\sigma^2$$

(18)

$$V(0) = 0, \quad \sigma > 0, \quad V(\sigma) > 0$$

(19)

$$\dot{V} = \frac{1}{2}\frac{d}{dt}\sigma^2 \le -\eta|\sigma|$$

(20)

The criteria for convergence (21) and sliding mode (22) is expressed as

$$\sigma\dot{\sigma} \le -\eta|\sigma|$$

$$\sigma\dot{\sigma} \le -\eta\,\mathrm{sgn}(\sigma)\sigma$$

(21)

$$\dot{\sigma}\,\mathrm{sgn}(\sigma) \le -\eta$$

(22)

If η > 0, the states of system reach sliding mode. The control signal $u(t_i)$ consists of two parts . The first part is equal to control signal whereas the second part indicates switching of signal using SMC as shown in equation (23)

$$u(t_i) = u_{eq}(t_i) + u_{sw}(t_i)$$

(23)

$$u_{sw}(t_i) = K\,\mathrm{sgn}(\sigma)$$

(24)

Gain K =2000 is positive number and sgn(.) is signum function.

$$\mathrm{sgn}(\sigma(t_i)) = \begin{cases} +1, & \sigma(t_i) > 0 \\ 0, & \sigma(t_i) = 0 \\ -1, & \sigma(t_i) < 0 \end{cases}$$

(25)

While $\sigma \neq 0$, the control signal switch the state of system to reach sliding surface. Once the system reaches sliding surface, the control signal is disabled by enabling equivalent control signal as described in (25).The switching gain of control signal of electro hydraulic valve introduce unwanted chattering effect into active suspension system by signum () during the implementation of sliding mode controller in real time. This undesirable chattering phenomena can be eliminated completely by using fuzzy logic controller instead of signum () in sliding mode control.

B. Fuzzy sliding mode Control

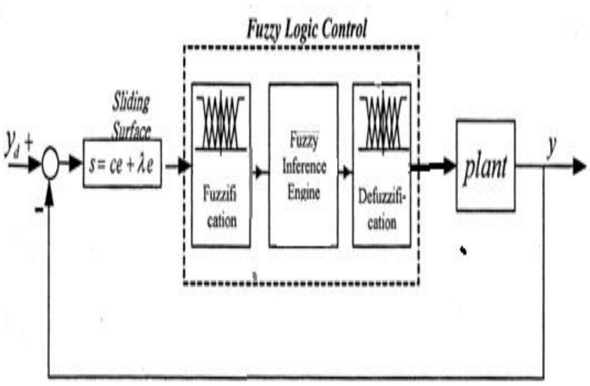

Figure 3. Overall Layout of Fuzzy sliding mode controller

Fuzzy sliding mode control has ability to handle nonlinearity, chattering and undesirable behaviour of actuator dynamics . The overall layout of FSMC i0.s shown in fig 3. The actual suspension travel of each wheel and its derivative in the form of sliding surface act as control input parameter for fuzzy controller whereas the control voltage of spool valve act as an output to fuzzy controller. The fuzzy rules base and defuzzification process carried out to obtain control voltage for spool valve in such way that will reduce the chattering effect of conventional sliding mode controller.

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The fuzzification stage converts the Sliding surface (S) and rate of change in sliding surface ($\dot{S}$) of suspension deflection into fuzzy values using membership function. The knowledge of expert about an active system converted into rules base for fuzzy controller.

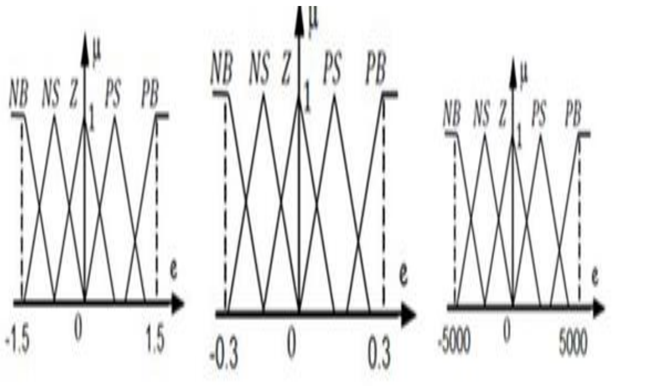

Figure 4. Membership function for Fuzzy controller

The membership function for sliding surface and its change and control voltage represented in Fig 4 with linguistic variable. The rules base is constructed based on elucidation of Mamdani. Simulation carried out to determine the range of input and output for fuzzy controller. The rules table is represented in Table 2. The rules are formulated based on minimization of vertical velocity and acceleration of car body.

S/$\dot{S}$	NEB	NES	ZER	PES	PEB
NEB	NEB	NEB	NES	NES	ZER
NES	NEB	NES	NES	ZER	PES
ZER	NES	NES	ZER	PES	PES
PES	NEM	ZER	PES	PES	PEB
PEB	ZER	PES	PES	PEB	PEB

Table 2. Fuzzy Rules for Controller

The control voltage which is used to drive the electro hydraulic valve obtained from fuzzy controller using defuzzification process .The defuzzification process carried out by centroid method which is easy and simple to construct. The expression for centroid defuzzification technique is shown

$$Z_{COG} = \frac{\int_Z \mu_A(Z) Z dz}{\int_Z \mu_A(Z) dz}$$

(26)

IV. SIMULATION

The simulation was carried out for active suspension system using full car model described in section 2. The performance of proposed fuzzy controller measured against passive and Proportional integral Derivative (PID) control of Active suspension system. The variables namely body response, suspension deflection, control force and body Acceleration of each wheel measured to find the performance of proposed fuzzy controller.

Table 3 lists RMS of displacement, suspension travel and acceleration of corner at 1,2,3,4 respectively. It also indicates displacement, pitch angle, roll angle of whole body for passive, PID and fuzzy sliding mode controlled suspension system. The values of parameter in the table indicates the fuzzy sliding mode controller has less suspension deflection and acceleration compare to passive and PID controller. Hence, fuzzy sliding mode controller will outperform both passive and PID controlled suspensi

Position	Parameter	Root Mean Squared Value		
		PASSIVE	PID	FSMC
Corner 1	Displacement (cm)	2.94	2.49	2.47
	Suspension Travel (cm)	0.87	0.26	0.20
	Acceleration (m/s^2)	1.01	0.75	0.62
Corner 2	Displacement (cm)	2.23	1.96	1.90
	Suspension Travel (cm)	.76	0.35	0.30
	Acceleration (m/s^2)	0.79	0.53	0.59
Corner 3	Displacement (cm)	2.01	1.69	1.64
	Suspension Travel (cm)	0.75	0.29	0.18
	Acceleration (m/s^2)	0.62	0.43	0.43
Corner 4	Displacement (cm)	2.04	1.69	1.65
	Suspension Travel (cm)	0.80	0.30	0.22
	Acceleration (m/s^2)	0.64	0.43	0.54
Body	Displacement (cm)	1.71	1.47	1.46
	Acceleration (m/s^2)	0.58	0.49	0.44
	Pitch angle (Deg)	5.8	3.9	3.4
	Roll Angle (Deg)	4.1	3.0	2.7

Table 3. RMS of Suspension Parameter

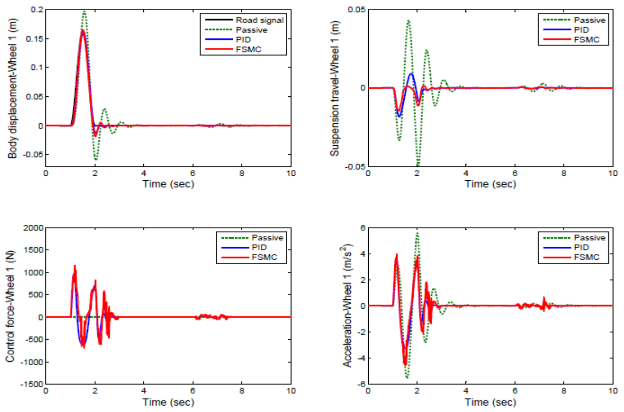

Figure 5. Response of Wheel 1 for Road profile

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The response of wheel 1 & 2 is represented in Fig. 5-6 for response of passive, PID and fuzzy sliding mode active suspension. Table 3 lists the RMS values of suspension parameters. The Passive suspension at corner 1 has displacement of 2.94 cm and high settling time compare to FSMC which has displacement of 2.47 cm and less settling time. The acceleration of fuzzy sliding mode controlled suspension is 0.62 m/s^2 which is far better than 1.01 m/s^2 of passive suspension.

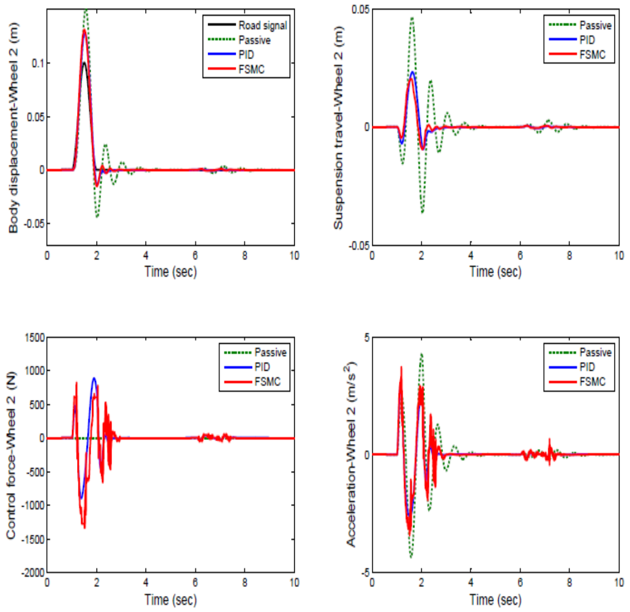

Figure 6. Response of Wheel 2 for Road profile

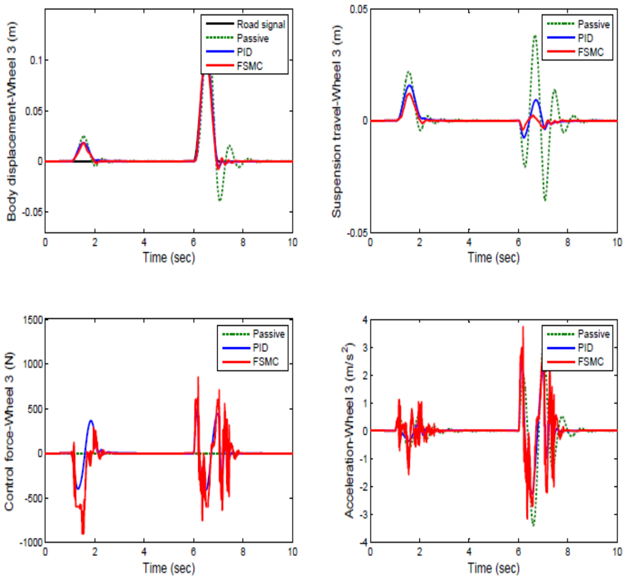

Figure 7. Response of Wheel 3 for Road profile

The response of front wheel 3 at corner 3 & 4 is represented in Fig 7-8. The suspension travel of passive suspension of corner 3 is 2.01 cm whereas suspension travel of fuzzy sliding mode controller is 1.64 cm. The acceleration of fuzzy sliding mode controlled suspension is 0.54 m/s^2 which is better than 0.64 m/s^2 of passive suspension. So, fuzzy sliding mode controller reduces suspension travel and acceleration better than passive suspension system.

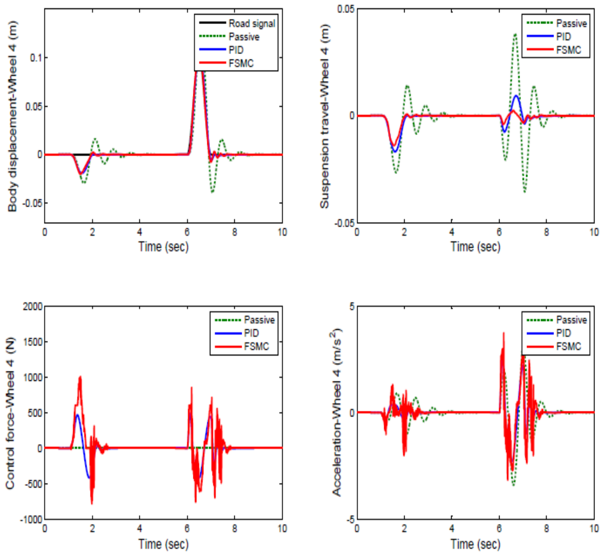

Figure 8. Response of Wheel 4 for Road profile

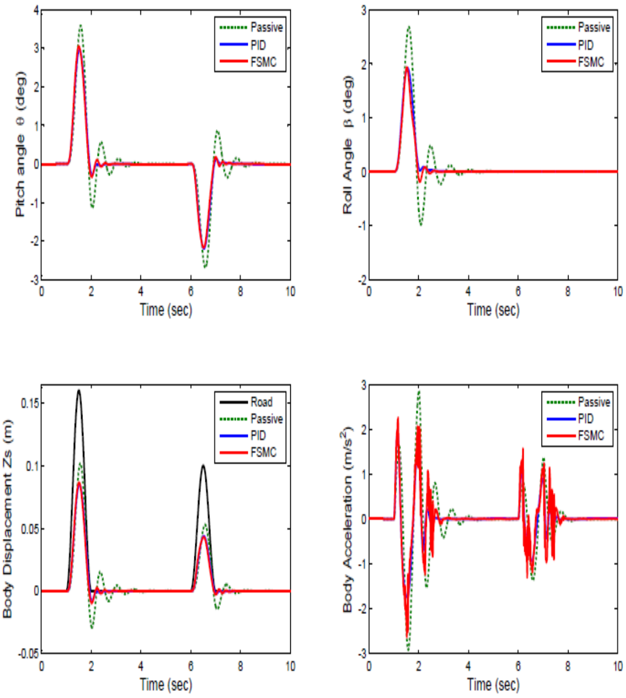

Figure 9. Response of car body for Road profile

The response of whole vehicle body about centre of gravity is represented in fig 9. The body displacement of fuzzy sliding mode controlled suspension is 1.46 cm which is less than 1.710 cm of passive suspension. The acceleration of vehicle body indicates that fuzzy sliding mode controlled suspension has acceleration of 0.44 m/s^2 which is less than passive suspension system. The pitch angle of fuzzy sliding mode controlled suspension is 3.4^0 whereas passive suspension has pitch angle of 5.8^0. The SMC controlled suspension provide roll angle of 4.1^0 whereas passive suspension has roll angle of 2.7^0.

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V. CONCLUSIONS

A full car model with 7 DoF was developed for hydraulic actuated active suspension. The development of mathematical model has been difficult due to nonlinear effect in suspension and unpredictable behavior of hydraulic actuator. The Fuzzy Sliding Mode Controller (FSMC) based on variable structure control method has ability to handle uncertain parameter, complex mathematical model and chattering effect in active suspension. Hence, fuzzy sliding mode controller was developed for hydraulic actuated active suspension to give better performance than analytical control scheme. The response of full car model confirms the feasibility of fuzzy sliding mode controller for hydraulic actuated active suspension. In the future work, self-organized fuzzy controller will be investigated to reduce the difficulty of designing fuzzy controller by using suitable membership function and fuzzy rules.

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