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Active Vehicle Suspension Control using Electro Hydraulic Actuator on Rough Road Terrain

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Abstract – *Vehicle suspension system main purpose is to keep the main body of the vehicle apart from any geometrical road irregularities thereby improving passenger comfort and also maintain good handling stability. The present work proposes a design of a two loop PID controls of generated force (inner loop) and suspension parameters (outer loop) for a four degree of freedom, nonlinear, half vehicle active suspension system model. The two loop arrangements are made up of an inner hydraulic actuator PID force control loop and an outer suspension parameters PID control loop. Simulation using the same parameter model for both systems was carried out; a comparison was made between the nonlinear active PID based suspension systems with a nonlinear passive system. Obtained results show a better performance in the active system when compared to the passive system even though the earlier was more costly and power consuming.*

Keywords: Active suspension, Force feedback, Half car model, Hydraulic actuator, PID control

1.0 INTRODUCTION

Suspension system in automobiles is creditworthy for driving safety and comfort for the suspension takes the vehicle sprung mass (body) weight and conveys all the forces which act in between automobile body and road surface [1]. Vehicle suspension system consists of the springs, shocks absorbers usually called the dampers and the mechanical linkages known as the wishbones to transmit and also filter all the forces between vehicle body and the road. The springs accommodate the vehicle body mass and have cushion effect against road disturbances, thus contribute to driving safety. The shock absorbers absorb the vehicle body and wheel oscillations thus contribute to both driving safety and comfort. More so, a suspension system reduces the loss in traction that occurs between the tyre and road by sustaining road holding ability thus, improving vehicle road handling.

Due to the implicit contradictory nature of the system performance in the vehicle suspension systems, the problem becomes at large for improved solution to be recovered by researchers [2]. Three primary types of automobile suspension systems, that is to say, passive systems, semi and fully active systems were studied in accomplishing automobile required performances and to stay clear from the trade off [1, 3].

Passive vehicle suspension systems are the conventional mechanical arrangements consisting of the linear springs and viscous damper with constant stiffness and damping coefficient as can be pictured in Figure 1a. Most of today's commercial vehicles utilize passive suspensions in controlling dynamically vertical motions in vehicles as well as the vehicle roll and pitch

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motion [4]. A semi active suspension allows the smooth changing of passive damper with a semi-active damping coefficient as shown in Figure 1b. This type of system maintains the use of a fixed spring but employs the use of a continuous variable damper which can be varied in real time through closed loop feedback control design, which controls energy dissipated by the continuous variable damper and can rapidly be changed over some broad range, hence improving the suspension performance over the passive system [5]. In active vehicle suspension system, passive elements are increased with actuators that generate an additional external force to the system as pictured in Figure 1c. For the suggested vehicle suspension system, researchers generally acknowledge that the active suspension systems are the most efficient means of improving a suspension system performance credited to the possessed adaptability in dealing with the conflicting parameters [6].

Active vehicle suspension system has been an area of vast research work for more than two decades due to very much promising features. These systems pose the capability of responding to vertical changes due to road input irregularities. The springs as well as the dampers in the active suspension system are mediated by the actuator force. The task of this actuator is to bestow or disperse energy from the system and this actuator can be controlled through different type of controllers which can be determined by a designer. The right control techniques give rise to a better compromise to occur between vehicle ride comforts to road handling stableness, thus active vehicle suspension systems bid for a better suspension design.

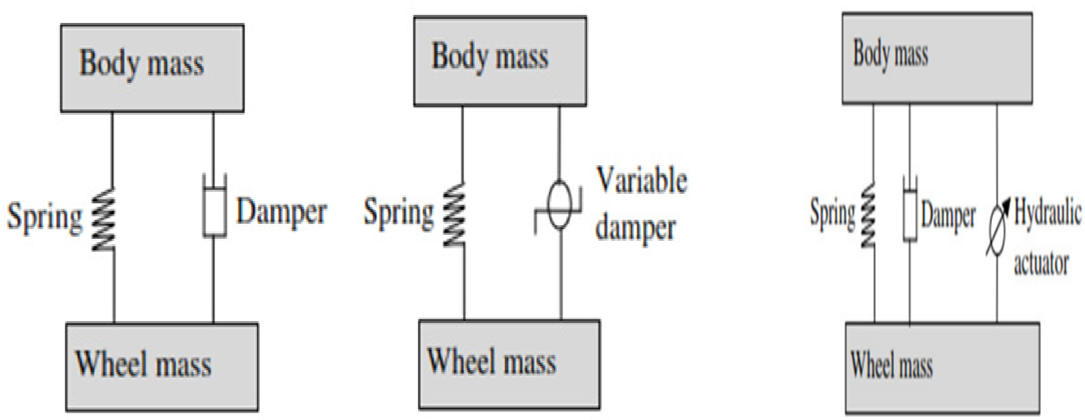

Figure 1a: Passive System **Figure 1b:** Semi Active System **Figure 1c:** Active System

To enable the control of active vehicle suspension system actuators in a pleasing manner, a suitable control algorithm is needed. Wide spectrums of research projects were carried out on active suspension systems and numerous control strategies were proposed by different researchers to bring about improvement in the conflict between vehicle ride comfort and road handling [7]. These control strategies can be grouped based on different control techniques. Results of literature survey based on different control methods utilized in active suspension system were briefly highlighted.

Linear control strategy based on optimal control concept is one of the most popular techniques that have been widely applied by most researchers in the area of designing active

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vehicle suspension system [8]. Amidst optimal control general ideas applied are the Loop Transfer Recovery (LTR) method, Linear Quadratic Regulator (LQR) method, the Linear Quadratic Gaussian (LQG) method etc. These general ideas were based on minimization of a cost function in a linear quadratic function where the parameter performances measured are the function of states as well as the inputs to the system [9].

LQR application method in active vehicle suspension system was proposed by [10-12]. State feedback control method was employed using active vehicle suspension in [13-15] etc. Other methods include Linear Parameter Varying (LPV) by [16], and H_∞ control strategy by [17, 18]. The application of intelligent based control method such as Neural Network (NN), Fuzzy Logic Controls (FLC), Genetic Algorithms (GA), etc. was employed in designing active vehicle suspension by [19-21] respectively.

Other control techniques in active suspension designs include nonlinearity nature of the system. Back stepping control techniques have been considered by [22, 23]. [24] proposed a work on fuzzy control design technique using full vehicle model for nonlinear active suspension system with hydraulic actuator. [25] proposed a designed controller using sliding mode control method; also adaptive sliding mode control was looked into by [26-29].

All the reviewed active suspension system control techniques are aimed at improving the vehicle suspension performance and to overcome the existing compromise in the predominately used passive suspension systems.

In the actual implementation of active suspension performance, actuator dynamics may quite be complicated, and it is a known fact that interaction between vehicle suspension and the actuator cannot be neglected. Also it is a hard task to generate actuator force that is very much closer to the targeted force without applying the inner loop control or force tracking control techniques. This is because hydraulic actuator shows an attribute of nonlinear behavior developing from the dynamics response of servo valve and the effects of undesirable back pressure as a result of reciprocal action (interacting) that occurs between the vehicle suspension and the hydraulic actuator system [9]. Hence, this research work will consider the building of an inner feedback control loop joined together with an outer feedback control loop of actuator force tracking and suspension parameters respectively considering a vehicle passing over a random road profile.

2.0 METHODOLOGY

Applying Newton's second law of motion and assuming that pitching angle is small, the following equations are obtained:

$$z_{b1} = z_b - l_1\theta \tag{1}$$

$$z_{b2} = z_b + l_2\theta \tag{2}$$

From Figure 2, the front and rear nonlinear suspension forces can be obtained as follows:

$$F_{kb1} = k_{b1}(z_{b1} - z_{w1}) + \zeta.k_{b1}(z_{b1} - z_{w1})^3 \tag{3}$$

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$$F_{cb1} = c_{b1}(\dot{z}_{b1} - \dot{z}_{w1}) + \zeta.c_{b1}(\dot{z}_{b1} - \dot{z}_{w1})^2 \operatorname{sgn}(\dot{z}_{b1} - \dot{z}_{w1}) \tag{4}$$

$$F_{kb2} = k_{b2}(z_{b2} - z_{w2}) + \zeta.k_{b2}(z_{b2} - z_{w2})^3 \tag{5}$$

$$F_{cb2} = c_{b2}(\dot{z}_{b2} - \dot{z}_{w2}) + \zeta.c_{b2}(\dot{z}_{b2} - \dot{z}_{w2})^2 \operatorname{sgn}(\dot{z}_{b2} - \dot{z}_{w2}) \tag{6}$$

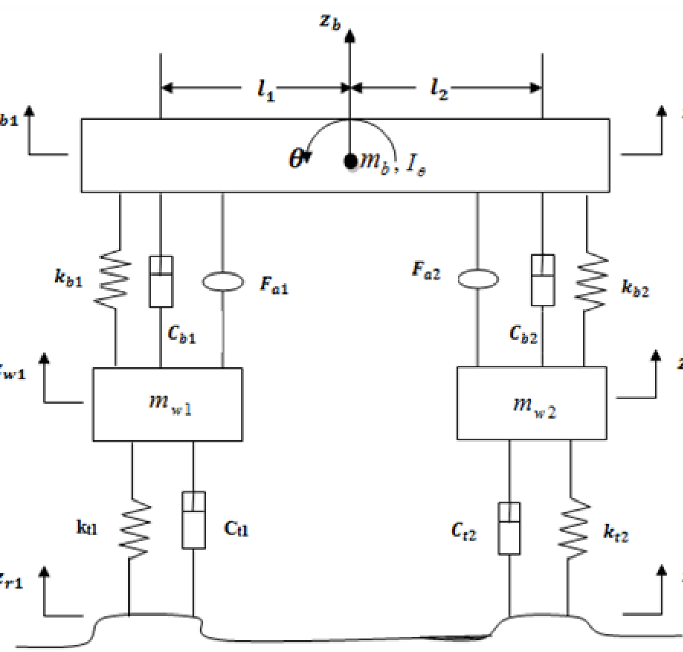

Figure 2: Half Vehicle Model.

where, ($\zeta = 0.1$) is a constant called the empirical factor,

and the tyre forces as;

$$F_{t1} = k_{t1}(z_{w1} - z_{r1}) + c_{t1}(\dot{z}_{w1} - \dot{z}_{r1}) \tag{7}$$

$$F_{t2} = k_{t2}(z_{w2} - z_{r2}) + c_{t2}(\dot{z}_{w2} - \dot{z}_{r2}) \tag{8}$$

2.1 Road Input Model

A random road input disturbance is characterized with road surface roughness of high frequency which is sometimes described as a Power Spectral Density (PSD) function that causes a maintained vehicular vibration [3]. Vibrations, from another point of view, are described by sustained and uniform excitations which are referred to as rough roads. Series of road roughness standard has been suggested for classification purposes by the International Standard Organization (ISO) using values established for PSD (ISO 1982), as established in Table 1. In space domain, the road displacement power spectral density (PSD) is expressed as Equation 9.

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$$G(n) = G(n_0)\left(\frac{n}{n_0}\right)^{-w} \tag{9}$$

where, n denotes frequency of space in m^{-1}, n_0 represents the reference space frequency, $G(n_0)$ stands for coefficient of the road roughness which is given in Table 1, w denotes coefficient of linear fitting with range of ($1.75 \leq w \leq 2.25$) but is mostly set to $w=2$. When the values of w are small, they indicate the presence of a greater amount of short wavelengths or longer wavelengths for larger values of w, thus, specifying road surface waves [30]. The random road profile was generated using the integral white noise method. Other research works that employed these techniques are work in [3, 30].
The road surface input equation is given as:

$$\dot{z}_{r1} = 2\pi n_0 z_{r1} + 2\pi\sqrt{G(n_0)Vw_0} \tag{10}$$

$$\dot{z}_{r2} = 2\pi n_0 z_{r2} + 2\pi\sqrt{G(n_0)Vw_0} \tag{11}$$

w_0 denotes a Gaussian white noise with a PSD of 1, V is the vehicle forward velocity and $G(n_0)$ represents the road roughness coefficient which is of different classification (Table 1).

Table 1: ISO classification of road roughness values ($256\text{x } 10^{-6}$).

Road Class	Range	Geometric Mean
A (very good)	< 8	4
B (good)	8 - 32	16
C (average)	32 - 128	64
D (poor)	128 – 512	256
E (very poor)	512 – 2048	1024

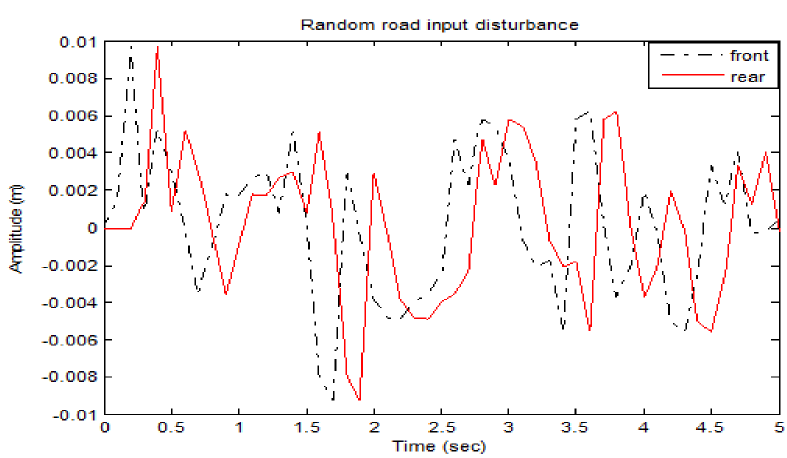

Figure 3: Random road input disturbance.

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2.2 Hydraulic Actuator Dynamics

To understand the concept of actuator dynamics in the system, there is a need for the description of the subsystem fluid dynamics, hydraulic cylinder and servo-valve, as well as load required. The electro-hydraulic system in this case is a piston that is controlled by an input voltage/current signal to the servo-valve. The cylinder is positioned in-between the vehicle body and wheel masses, and connected parallel to the passive system combining the spring and damper.

When control input u_i is supplied into the system, the spool-valve is moved by x_{vi} unit which then causes a high piston pressure difference. This piston high pressure difference multiplied by cross-sectional area of the piston is what produces hydraulic force F_{ai} for the suspension system. Hence, the spool valve motion needs to be controlled properly in order to track and generate the required force.

The hydraulic actuator force is given as:

$$F_{ai} = A_{hyd} P_{Li} \tag{12}$$

where, i denotes either front or rear component.

For a given voltage input u_i, the rate of change of servo-valve displacement $\dot{x}_{vi}$ can be approximated by a linear filter with time constant as Equation 13.

$$\dot{x}_{vi} = \frac{1}{\tau}(k_{vi} v_i - x_{vi}) \tag{13}$$

where, τ is the hydraulic actuator time constant, x_{vi} represents the servo-valve displacement and k_{vi} denotes the servo-valve gain, which is a conversion ratio from the control input voltage to the servo-valve displacement in meter.

The resulting hydraulic flow rate Q_i can be written as:

$$Q_i = C_d \omega . x_{vi} \sqrt{\frac{1}{\rho}(P_s - \operatorname{sgn}(x_{vi}) P_{Li})} \tag{14}$$

The rate at which P_{Li} changes with time, including hydraulic flow load Q_i is given as:

$$\dot{P}_{Li} = \frac{4\beta_e}{V_t}[Q_i - C_{tp} P_{Li} - A_{hyd}(\dot{z}_{bi} - \dot{z}_{wi})] \tag{15}$$

Let's assume the following terms:

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$$\alpha = {}^{4\beta_e}\!/_{V_t}, \quad \beta = \alpha . C_{tp},$$

Substituting the above assumptions, Equation 16 is obtained:

$$\dot{P}_{Li} = \alpha . Q_i - \beta . P_{Li} - \alpha . A_{hyd} (\dot{z}_{bi} - \dot{z}_{wi}) \tag{16}$$

1.3 Suspension System with Hydraulic Actuator Dynamics Model

The dynamics equation of motion for the half vehicle nonlinear system model with hydraulic actuator forces can be obtained as:

$$\ddot{z}_b = -\frac{1}{m_b}[F_{kb1} + F_{kb2} + F_{cb1} + F_{cb2} - F_{a1} - F_{a2}] \tag{17}$$

$$\ddot{\theta}_b = \frac{1}{I_\theta}[l_1(F_{kb1} + F_{cb1} - F_{a1}) - l_2(F_{kb2} + F_{cb2} - F_{a2})] \tag{18}$$

$$\ddot{z}_{w1} = \frac{1}{m_{w1}}[F_{kb1} + F_{cb1} - F_{t1} - F_{a1}] \tag{19}$$

$$\ddot{z}_{w2} = \frac{1}{m_{w2}}[F_{kb2} + F_{cb2} - F_{t2} - F_{a2}] \tag{20}$$

All the parameters involve in the Equations (1)-(20) are clearly defined in the given Tables 2 and 3 below.

Table 2: Parameter Values for half vehicle model.

Parameters	Description	Values	Units
m_b	Body mass	730	kg
I_θ	Body pitch moment of inertia	2460	kgm^2
m_{w1}	Front wheel mass	40	kg
m_{w2}	Rear wheel mass	35.5	kg
k_{b1}	Front suspension stiffness	19,960	N/m
k_{b2}	Rear suspension stiffness	17,500	N/m
c_{b1}	Front suspension damping coefficient	1290	Ns/m
c_{b2}	Rear suspension damping coefficient	1620	Ns/m
k_{t1}	Front tyre spring stiffness	175,500	N/m
k_{t2}	Rear tyre spring stiffness	175,500	N/m
c_{t1}	Front tyre spring damping coefficient	14.6	Ns/m
c_{t2}	Rear tyre spring damping coefficient	14.6	Ns/m

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l_1	Distance from m_s C.G to front axle	1.011	m
l_2	Distance from m_s C.G to rear axle	1.803	m
F_{a1}	Front actuator force	-	-
F_{a2}	Rear actuator force	-	-

Table 3: Parameter values of the hydraulic actuator

Parameters	Description	Values	Units
α	Actuator parameter	$4.515*10^{13}$	N/m^{-5}
β	Actuator parameter	1	s^{-1}
γ	Actuator parameter	$1.545*10^{9}$	$N/m^{5/2}/kg^{1/2}$
A_p	Piston cross-sectional area	$3.35*10^{-4}$	m^2
P_S	Supply pressure	1034250	Pa
τ	Time constant	0.003	s
k_{vi}	Servo valve gain	0.001	m/V

3.0 DESIGN OF CONTROLLER

The active suspension control architecture employed in this work consists of two feedback control loop arrangement that includes the inner controller loop corresponding to the hydraulic actuator control system and the outer controller loop corresponding to the vehicle suspensions control that are fed back through a PID control system respectively.

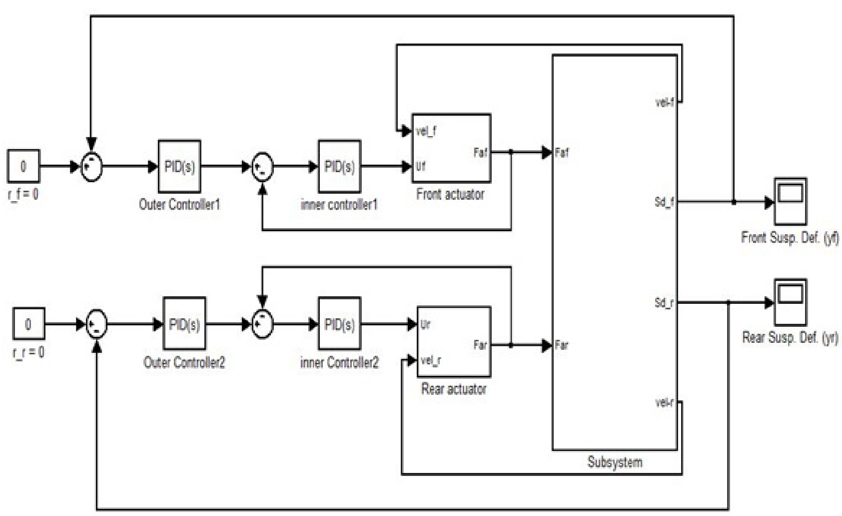

Figure 4: Proposed Controller Architecture

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The primary goal of this control system is to make sure that despite the deterministic disturbances generated from the road roughness, the output signals from the controller should be kept as closely as possible to the input reference signals.

The inner/outer loop PID control is defined as follows;

$$u_i / F_{ai,ref} = K_{Pi} e_i(t) + K_{Ii} \int e_i dt + K_D \frac{de_i}{dt} \tag{21}$$

$$e_i = r_i - y_i \tag{22}$$

where, e_i is the control error and r_i is the reference signal. Considering suspension travel as one among the suspension outputs and according to suspension travel regulation, the suspension travel reference signal is always set to zero (i.e. $r_i = 0$) [6]. Therefore, it is hoped that a controller which obeys the control law that states $e_i(t) \to 0$, as $t \to \infty$ could be designed·

Table 4 gives the inner/outer loop PID controller parameters which are determined through the use of Ziegler-Nichols turning rule with the desired goal of obtaining a better performance by reducing the RMS parameters of the active systems when compared with passive systems.

Table 4: Proposed Controller Architecture.

	Front Suspension		Rear Suspension	
PID Gains	Inner Loop	Outer Loop	Inner Loop	Outer Loop
K_P	0.000545	13600.016	0.000545	3155.021
K_I	0.000323	8267.840	0.000323	1232.820
K_D	0.0000156	318.220	0.0000156	306.251

3.1 Controller Specification

The control system proposed is expected to fulfill most of the following described requirements:

i. The controller ought to be capable of keeping off any divergence from the set reference point in order to comform with controller command;
ii. Demonstrate a good low frequency disturbance rejection;
iii. The control closed loops should be nominally stable;
iv. Due to mechanical structure, suspension travel maximum allowable deflection is set to be:

$$sd = \left| z_{bi} - z_{wi} \right| \leq z_{i,\max} \tag{23}$$

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where, $z_{i,\max}$ is the maximum suspension travelled and is set to be *0.1 m* for this work, sd is the suspension deflection and i is either the front or rear suspension.

v. Dynamic tyre load should not outmatch the static tyre load for both front and rear wheels in order to maintain a good road holding ability. This can be described as:

$$\left|k_{t1}(z_{w1} - z_{r1}) + b_{t1}(\dot{z}_{w1} - \dot{z}_{r1})\right| \leq F_{t1} \quad (24)$$

$$\left|k_{t2}(z_{w2} - z_{r2}) + b_{t2}(\dot{z}_{w2} - \dot{z}_{r2})\right| \leq F_{t2} \quad (25)$$

where,

$$F_{t2} = \frac{m_b g l_1 + m_{w2} g (l_1 + l_2)}{l_1 + l_2} \quad (26)$$

$$F_{t1} = (m_b + m_{w1} + m_{w2})g - F_{t2} \quad (27)$$

F_{t1} and F_{t2} denotes the front and rear static tyre loads respectively.

vi. The maximum allowable actuator control force is given as:

$$F_{ai,\max} \leq m_b g$$
(28)

where, g is the acceleration due to gravity.

vii. The maximum control voltage allowed is:

$$u_{i,\max} \leq 10V$$
(29)

Other important parameter is the Root Mean Square (RMS) values for vehicle suspension parameters which will equally be obtained in order to compare the active suspension system performance with that of passive suspension system, and this RMS values can be described.

viii. Vehicle sprung mass acceleration as:

$$\ddot{z}_{b,RMS} = \sqrt{\frac{1}{n}\sum_{0}^{n}(\ddot{z}_b)^2} \quad (30)$$

ix. Vehicle pitch angular acceleration as:

$$\ddot{\theta}_{b,RMS} = \sqrt{\frac{1}{n}\sum_{0}^{n}(\ddot{\theta}_b)^2} \quad (31)$$

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x. Vehicle suspension deflection as:

$$sd_{i,RMS} = (z_{bi} - z_{wi})_{RMS} = \sqrt{\frac{1}{n}\sum_{0}^{n}((z_{bi} - z_{wi}))^2} \tag{32}$$

xi. Vehicle tyre travel as:

$$Td_{i,RMS} = \sqrt{\frac{1}{n}\sum_{0}^{n}(z_{wi} - z_{ri})^2} \tag{33}$$

4.0 RESULTS AND DISCUSSION

The existence of active suspension control problems neccessitate for the betterment of the vehicle passenger ride comfort and road handling capability, at the same time, keeping the suspension rattle space limit within the suspension workspace. In real life application, active suspension system is expected to execute better performance in a dignified manner with actuator power supply and output force constraint [3, 10].

Matlab/Simulink environment was used to simulate the half vehicle nonlinear active suspension model with hydraulic actuators. Firstly, the open loop response of the half vehicle suspension systems when the proposed road inputs were applied as road excitation was investigated. The plots for the open loop system will show whether the control objectives can be achieved without using control system for the proposed model. The proposed objectives of this work are to minimize the vibration felt by the passengers on travelling through random road profile and avoidance of vehicle pitch motion when a critical maneuver occurs.

Passive suspension transient response as well as active suspension was determined in time domain analysis for input of random road profile. The input road profile characterizes a vehicle moving on a rough road terrain with a forward velocity of *45km/hr*. The rough road is characterized as a white noise disturbance of magnitude *0.1*.

Figures 5 and 6 are vehicle suspension deflections time history of both front and rear suspensions of active system as well as the passive system respectively, when the vehicle travels across a rough terrain. Clearly it is shown that the uttermost travelled level by the suspensions at utmost height of road disruption input and values were less than the defined suspension travel limits of 0.1*m*. The input disturbance was not completely suppressed but was minimized to a lower level of about *3.0sec* and *2.5sec* for both front and rear passive and active systems respectively. The RMS values obtained for both systems are given in Table 4 below. It shows that the percentage reduction in the vehicle rattle space was successfully minimized using the proposed control system and hence, passengers ride comfort was to some extent improved.

Vehicle road handling time histories were depicted in Figure 7 and 8 respectively. Uttermost road holding capability values of *0.01m* and *0.0175m* were attained for front active and passive wheels respectively. Whereas, *0.0065m* and *0.0137m* were attained for rear active and passive wheels respectively. The RMS values obtained show about *42%* improvements in the

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road handling capacity for both front and rear wheels. However, due to surface roughness, the settling time does not attain stability.

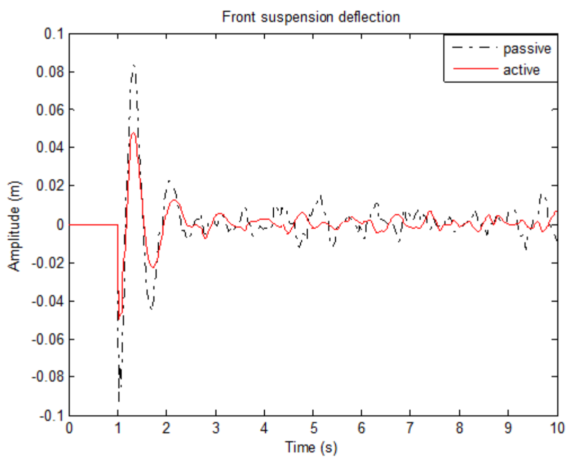

Figure 5: Front suspension travel.

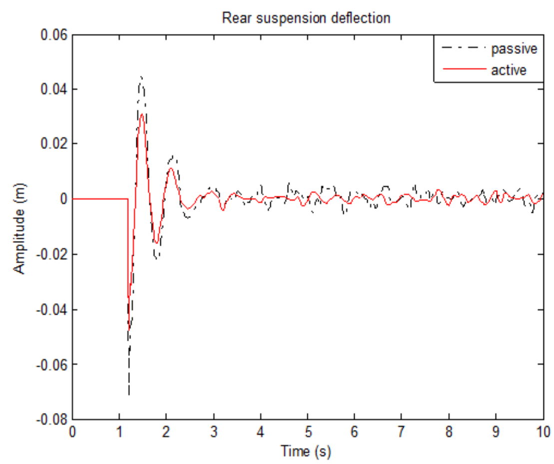

Figure 6: Rear suspension travel.

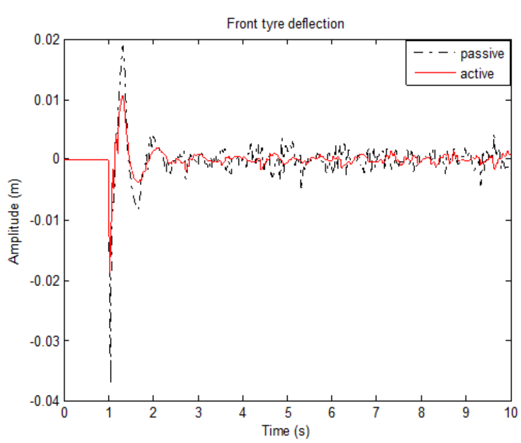

Figure 7: Front tyre deflection.

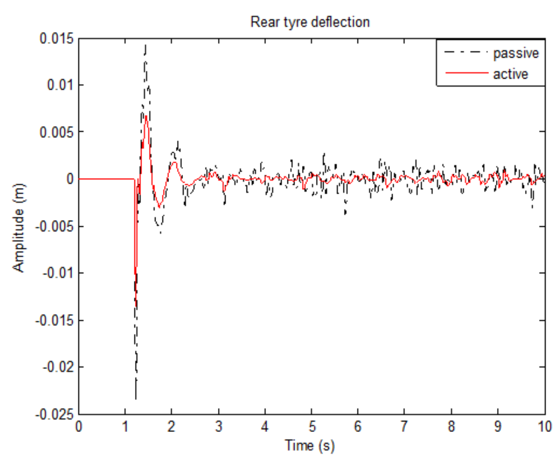

Figure 8: Rear tyre deflection.

Table 5: RMS values for random road input disturbance.

Parameters	Passive System	Active System	% Reduction by Active System
Front Suspension Deflection (m)	0.0135	0.0087	35.56
Rear Suspension Deflection (m)	0.0070	0.0051	27.14
Front Tyre Deflection (m)	0.0028	0.0016	42.86
Rear Tyre Deflection (m)	0.0019	0.0011	42.11
Body Mass Acceleration (m/s^2)	0.7078	0.2427	65.71

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Body Pitch Angular Acceleration (rad/s^2)	0.1157	0.1078	8.26
Front Actuator Control Voltage (V)	-	0.4831	-
Rear Actuator Control Voltage (V)	-	0.3933	-
Front Spool-valve Displacement (m)	-	7.644e-5	-
Rear Spool-valve Displacement (m)	-	4.722e-5	-
Front Actuator Force (N)	-	229.4483	-
Rear Actuator Force (N)	-	190.2236	-

Figure 9 shows the sprung mass acceleration time history with the ISO 2631-1 weighted root mean square (RMS) acceleration values given in Table 4. The Figure shows that the active suspension system was able to gain fear stability at about *3sec* compared to the passive suspension system which is unstable throughout the simulation time range. On the other hand, the weighted RMS acceleration for both passive and active systems was found not to remain at an uncomfortable level of discomfort throughout the simulation time with about *65%* performance improvement in active suspension when compared with conventional passive suspension.

Body pitch angular acceleration time history is shown in Figure 10. The RMS pitch angular acceleration was found to give about *8.3%* performance improvement for active system when compared with passive system but with instability in settling time throughout the simulation period which was due to the impact of the road roughness.

The front and rear actuator forces are depicted in Figure 11. The magnitudes of the RMS are given in Table 4 with a front force magnitude of *229.45N* and *190.22N* for the rear suspension. Carefully by examining the magnitudes of the RMS experience actuator forces for the random input disturbance, it shows that the required actuator force decreases with the amplitude of input disturbance.

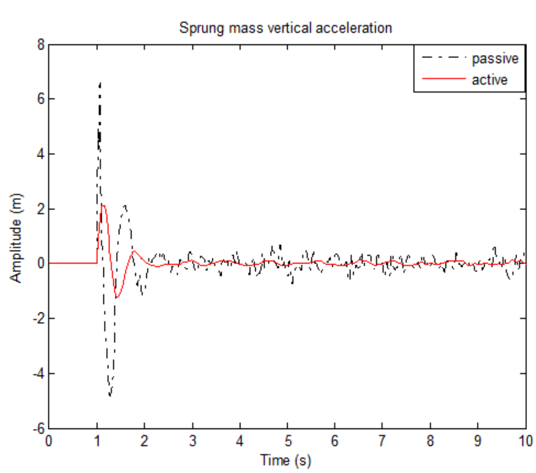

Figure 9: Body mass vertical acceleration.

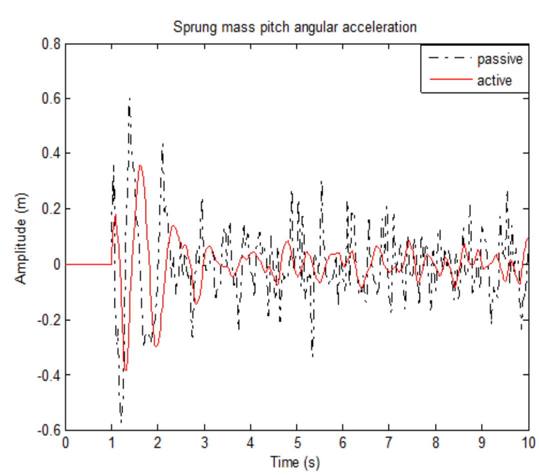

Figure 10: Pitch angular acceleration.

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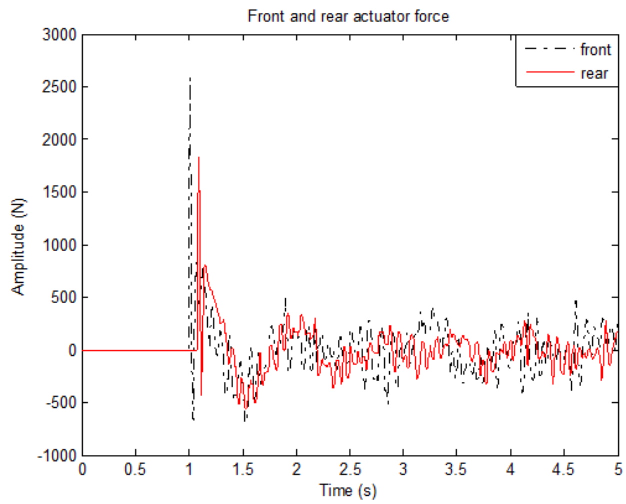

Figure 11: Actuator force.

4.0 CONCLUSSION

In this research paper, the designed and implementation of a PID based suspension system parameter controllers for active vehicle suspension through a way of two loop control system configuration were presented. The control system of these active vehicle suspensions was achieved by the application of a PID based control hydraulic actuator force feedback through simulation studies using a Matlab/Simulink simulation environment. The work was presented based on the circumstances that surround the current literature on active suspension control.

Nonlinear, four DOF (degree of freedom) half vehicle with hydraulic actuator dynamics active suspension model was used. In addition, the passive suspension system model used was a nonlinear, four DOF half vehicle with model parameters that are similar to those of the active suspension systems. It is a clear fact that hydraulic actuators are the most widely used of all actuators in active suspension system design; their drawbacks and benefits motivate us to use them in this research work. Furthermore, one of the motivating factors that cause us to include the nonlinearity in the vehicle suspension system models was to obtain the real effects and results of system models.

The overall performance of active suspension system parameters with road input uncertainties was found and proved better than that of passive suspension systems for random road input disturbances, providing a better passenger ride comfort, better road handling capacity and a minimum rattle space for suspension travel.

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