International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 01,January-2015

Electro-Hydraulic PID Force Control for Nonlinear Vehicle Suspension System

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***Abstract*— This work proposed a design of a two loop (inner and outer) PID control of generated force and suspension parameters technique respectively for a four degree of freedom, nonlinear, half vehicle active suspension system model. The two loop arrangement is made up of an inner hydraulic actuator PID force control loop and an outer suspension parameters PID control loop. Simulation using the same model parameters 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 showed a better performance in the active system when compared to the passive system at the expenses of cost and power consumption.**

Keywords — Active suspension; half vehicle model; hydraulic actuator; Matlab/Simulink; PID control.

I. INTRODUCTION

It is demonstrated clearly that there is constantly uprising concern in providing a satisfactory suspension system performances in automobiles, particularly through the last one to two decades [1, 2]. Automotive suspension systems has some numerous life-sustaining functions: for example, it helps in supporting weight of vehicles, isolating vehicle body effectively from any road irregularities, maintaining tyre close interaction (contact) with ground, at the same time keeping vehicles wheel in appropriate positioning on the road surface [2]. Vehicle suspension system serves a significant function in ensuring that the stability and improvement in the suspension performances are attained. It deserves nothing that the problem for control system design of vehicle active suspension systems should be given much attention. Furthermore, vehicle suspension system rendered vehicular ride comfort for passengers at the same time guarantees other suspension performances through sufficing basic purposes of keeping apart the passengers from any road-hastened shocks and vibration [3]. Therefore, the appropriate active suspension system control design problem is considered as an important research topic for accomplishing the demanded vehicle suspension performances.

Due to increase in vehicle capabilities, the performance in vehicle suspension systems has constituted a great increased. In order to achieve a good design in suspension systems, indefinite number of performance characteristics need to be put into consideration [4]. These characteristics take into consideration of regulating the body motion, regulating the suspension motion and also the force distribution. Ideally, suspension systems should be able to keep apart vehicle bodies from any road interruptions, also should be able to isolate inertial interruptions related to vehicle braking, maneuvering or acceleration of the vehicle. Suspension system also should be capable of reducing the vertical forces that are imparted to both driver and passengers for their comfort [1]. By way of reducing the vehicle vertical body acceleration, this objective can be attained. Unreasonable vehicle wheel deflection will result in unfavorable condition of the tyre proportional to road surface which induces poor handling of vehicle and adherence. In addition, to keep to a good road holding characteristic, an optimum tyre to road contact is highly recommended and must be kept intact throughout for the four wheels of the vehicle. For the conventional suspension systems, the required characteristics are contravening also do not fulfill all those conditions [5]. Many research works carried out in automotive industries and schools have examined the suspension system in a widespread way through different experiments and works. The primary goal of those works was to make sure that there is an improvement in the traditionally designed tradeoff that exist between the vehicle ride comforts and the road handling ability through a direct control of suspension forces to accommodate the performance characteristics for the system [5].

The unpleasant aspect of design in suspension system is the challenges in the compromise effect between the vehicle safety and ride comfort. In order to overcome this compromise conflict, the automotive industries now considered a new suspension system development which is contrary to the initial conventional system that is in existence for decades (i.e. the passive suspension). This new improvement in suspension system design and construction has made a great contribution to the aspect of passengers ride comfort and the vehicle driving safety [6]. This development is known as active suspension system and it is electronically controlled.

Active suspension systems are distinctly different compared to conventional passive system and semi active system because it poses a potentiality of generating energy into the system, also, it is capable of storing and dissipating the energy generated by applying hydraulic, magnetic or pneumatic actuators to produce the desired force [7]. These actuators in the active vehicle suspension system are situated in parallel to spring and damper. Due to conception that the actuator connects vehicle body to wheel and axle; it is capable of controlling the vehicle body motions, suspensions deflection and the wheel hop speed. Consequently, the active system now improves the performances of the suspension system parameters such as passengers ride comforts, suspensions deflection and ride handling at the same time.

Many researchers reckoned into various means of heightening and improving the suspension system through redesigning or by optimization of the already designed parameters, but most of which concentration goes to the passive and semi-active suspension system. In recent time, researcher's attention is repositioned to the active suspension system despite its complexity.

Rough road disturbances always affect the vehicle handling and stability negatively; it is therefore observed that using the passive suspension for vehicles is of paramount disadvantage since the design of passive suspension is basically for ride comfort and to support the vehicle body without given much consideration on the roll and pitch control [8]. In spite of the fact that versatile control techniques such as adaptive control [9], fuzzy control [10], optimal state-feedback control [7], robust sliding mode control [4] etc were suggested in controlling vehicle active suspension systems, most of this research works considered only the linear parameters of the suspension system; only few considered the nonlinearity of the system. More so, very few researchers acknowledge the real actuator nonlinearity. The techniques were successfully implemented in computer simulation based applications.

Most of the work that were carried out on suspension systems are limited to the simple quarter vehicle model or for those that considered the half and full vehicle models, we found that mostly the nonlinearity characteristics and system actuator uncertainties are not given much consideration which we intend to consider in this work due to the fact that, to have a better dynamics performance of the system this uncertainties cannot be ignored.

II. SYSTEM MATHEMATICAL MODEL

A. *Half Vehicle Model*

A half vehicle suspension system can simply be model as four degrees of freedom (DOF) systems, see Fig. 1. The vehicle cross-section representation is what determined the DOF of the vehicle model. Half vehicle models are produced to discover either vehicle pitching (i.e. when considered lengthwise) or rolling (i.e. when considered base wise) motion of the vehicle.

Let f and r represent the front and rear suspension components, z and x represent the vertical up and the longitudinal forward directions respectively. m_s And I_θ represent the body sprung mass and mass moment of inertia for pitch motion respectively. m_{uf} , m_{ur} is the front and rear unsprung masses, F_{ksf} , F_{ksr} , F_{bsf} and F_{bsr} represent the front and rear suspension forces by the springs and dampers respectively. F_{tf} , F_{tr} denotes the front and rear tire forces respectively. The vehicle body vertical displacement at the centre of gravity is z_c whereas θ is the pitch angular displacement, z_{uf} , z_{ur} are the unsprung masses vertical displacements respectively.

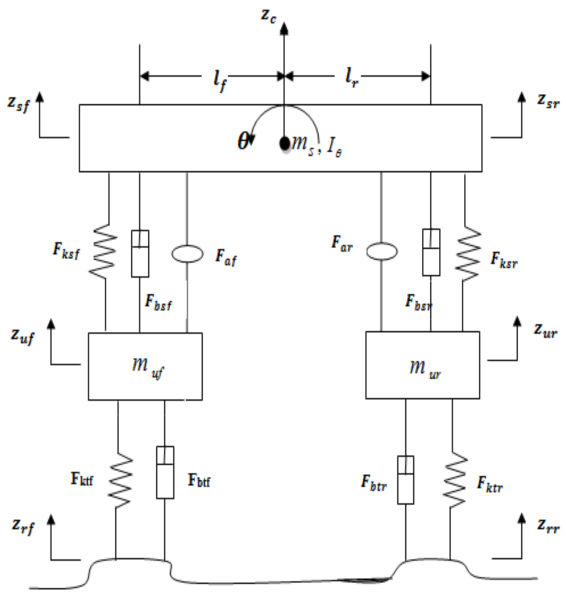

Fig. 1: Half Vehicle Model

z_{rf} , z_{rr} are the road inputs. l_f , l_r represent the suspension distances from the centre of the vehicle sprung mass, and F_{af} , F_{ar} are the control inputs actuator forces for front and rear suspensions respectively.

By using either Lagrange or Newton's second law of motion and assuming that the pitching angle is small, the following equations are obtained;

$$z_{sf} = z_c - l_f\theta \tag{1}$$

$$z_{sr} = z_c + l_r\theta \tag{2}$$

From the figure 1 above, the front and rear nonlinear suspension forces can be obtained as follows;

$$F_{ksf} = k_{sf}(z_{sf} - z_{uf}) + \zeta.k_{sf}(z_{sf} - z_{uf})^3 \tag{3}$$

$$F_{bsf} = b_{sf}(\dot{z}_{sf} - \dot{z}_{uf}) + \zeta.b_{sf}(\dot{z}_{sf} - \dot{z}_{uf})^2 \operatorname{sgn}(\dot{z}_{sf} - \dot{z}_{uf}) \tag{4}$$

$$F_{ksr} = k_{sr}(z_{sr} - z_{ur}) + \zeta.k_{sr}(z_{sr} - z_{ur})^3 \tag{5}$$

$$F_{bsr} = b_{sr}(\dot{z}_{sr} - \dot{z}_{ur}) + \zeta.b_{sr}(\dot{z}_{sr} - \dot{z}_{ur})^2 \operatorname{sgn}(\dot{z}_{sr} - \dot{z}_{ur}) \tag{6}$$

Where, ζ is a constant known as Empirical factor with a value of *0.1*.

And the tyre forces as;

$$F_{tf} = k_{tf}(z_{uf} - z_{rf}) + b_{tf}(\dot{z}_{uf} - \dot{z}_{rf}) \tag{7}$$

$$F_{tr} = k_{tr}(z_{ur} - z_{rr}) + b_{tr}(\dot{z}_{ur} - \dot{z}_{rr}) \tag{8}$$

B. *Road Input Disturbance Modeling*

Road surface is believed to be a natural changing condition as well as the major cause of input disturbance when dealing with vehicle suspension systems. For a better comfort during riding, it is necessary to have a perfect road surface model to design an active vehicle suspension control system. Road inputs analytically can be distinguished in many possible ways, which can be classified either as shock or vibration [11].

A discrete type of road input disturbance is used in this work which is commonly classified as a shock induced road input disturbance due to its ability to convey an impact forces to the vehicle within a short period of time. This class of road input disturbances include pronounced bumps, potholes, steps etc [7]. One of the examples which is speed bumps are used to impel most vehicle drivers in order to reduce the vehicle speed to levels off within the speed limits of a specific place such as residential area, schools, markets etc.

The input disturbances for front wheel and rear wheel, z_{rf} and z_{rr} respectively, are expressly shown in (9) and (10) for which *a* denotes amplitude of the bump, *t* denotes simulation time given in second, *L* represent the disturbance wavelength, t_d denotes time delayed between front wheel and the rear wheel and finally *V* stands for the vehicle forward velocity.

$$z_{rf} = \begin{cases} \frac{a}{2}\left(1 - \cos\left(\frac{2\pi V}{L}\right) t_f\right), & 0 \le t_f \le \frac{L}{V} \\ 0, & otherwise \end{cases} \quad (9)$$

$$z_{rr} = \begin{cases} \frac{a}{2}\left(1 - \cos\left(\frac{2\pi V}{L}\right) t_f\right), & t_d \le t_r \le t_d + \frac{L}{V} \\ 0, & otherwise \end{cases} \quad (10)$$

Where,

$$t_d = \frac{(l_f + l_r)}{V} \quad (11)$$

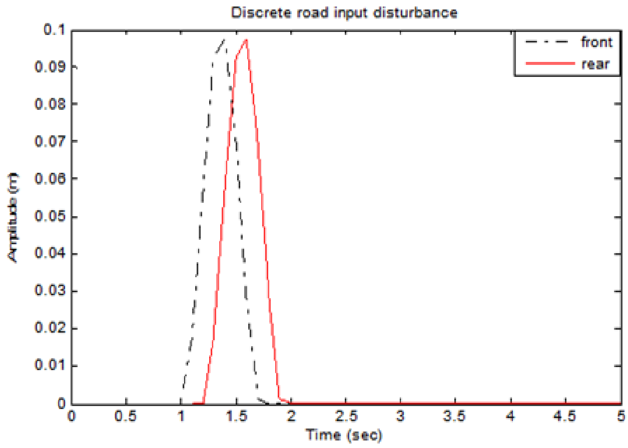

Fig. 2: Discrete road input disturbance

C. Hydraulic Actuator Dynamics

To understand the real performance of suspension systems and to build up a robust controller for the active suspension system, it is very important to generate a precise dynamic model for the hydraulic servo mechanism [12]. The hydraulic actuator serves as an appropriate force generator between the vehicles sprung and unsprung masses to enhance and improve the vehicle performance qualities.

Magnitude of the hydraulic actuation forces F_{ai} is controlled by a three land four-way spool servo-valves. These servo-valves are specified to operate within the range $u_i \le u_{\max}$ where $u_{\max}$ represent the allowable maximum control input voltage, and i denotes either the front or rear suspension parameter. 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 (12).

$$\dot{x}_{vi} = \frac{1}{\tau}(k_{vi}u_i - x_{vi}) \quad (12)$$

Where, τ is the hydraulic actuator time constant, x_{vi} represent 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;

$$Q_i = \alpha.C_d\omega.x_{vi}\sqrt{\frac{1}{\rho}(P_s - \mathrm{sgn}(x_{vi})P_{Li})} \quad (13)$$

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

$$\dot{P}_{Li} = \frac{4\beta_e}{V_t}[Q_i - C_{tp}P_{Li} - A_p(\dot{z}_{si} - \dot{z}_{ui})] \quad (14)$$

Let assume the following terms;

$$\alpha = \frac{4\beta_e}{V_t}, \quad \beta = \alpha.C_{tp}, \quad \gamma = \alpha.C_d\omega\sqrt{\frac{1}{\rho}}$$

Substituting the above assumptions, (15) is obtained

$$\dot{P}_{Li} = \gamma.Q_i - \beta.P_{Li} - \alpha.A_p(\dot{z}_{si} - \dot{z}_{ui}) \quad (15)$$

The hydraulic actuator force is;

$$F_{ai} = A_pP_{Li} \quad (16)$$

Where, i denotes either front or rear component.

Note that, the controlled input signal u_i and the output generated force F_{ai} from the actuator has a nonlinear dynamics relationship [13].

D. Suspension System Dynamics with Hydraulic Actuator

Dynamics equation of motion for half vehicle nonlinear systems model with hydraulic actuator force can be obtained as;

$$m_s\ddot{z}_c = -F_{ksf} - F_{ksr} - F_{bsf} - F_{bsr} + F_{af} + F_{ar} \quad (17)$$

$$I_\theta\ddot{\theta}_c = l_f(F_{ksf} + F_{bsf} - F_{af}) - l_r(F_{ksr} + F_{bsr} - F_{ar}) \quad (18)$$

$$m_{uf}\ddot{z}_{uf} = F_{ksf} + F_{bsf} - F_{tf} - F_{af} \quad (19)$$

$$m_{ur}\ddot{z}_{ur} = F_{ksr} + F_{bsr} - F_{tr} - F_{ar} \quad (20)$$

Where all the used symbols are defined in the sub-section A of section II above and the parameters values used for the simulation studies are given in Tables I and II below;

TABLE I. PARAMETER VALUES OF HALF VEHICLE SUSPENSION MODEL

Parameters	Description	Values	Units
m_s	Sprung mass	730	kg
I_θ	Pitch moment of inertia	2460	kgm^2
m_{uf}	Front unsprung mass	40	kg
m_{ur}	Rear unsprung mass	35.5	kg
k_{sf}	Front suspension stiffness	19,960	N/m
k_{sr}	Rear suspension stiffness	17,500	N/m
b_{sf}	Front suspension damping coefficient	1290	Ns/m
b_{sr}	Rear suspension damping coefficient	1620	Ns/m
k_{tf}	Front tire spring stiffness	175,500	N/m
k_{tr}	Rear tire spring stiffness	175,500	N/m
b_{tf}	Front tire spring damping coefficient	14.6	Ns/m
b_{tr}	Rear tire spring damping coefficient	14.6	Ns/m
l_f	Distance from m_s C.G to front axle	1.011	m
l_r	Distance from m_s C.G to rear axle	1.803	m
F_{af}	Front actuator force	-	-
F_{ar}	Rear actuator force	-	-

TABLE II. 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	10342500	Pa
τ	Time constant	0.003	s
k_{vi}	Servo valve gain	0.001	m/V

III. DESING OF CONTROLLER

The structural architecture of controller designed that was adopted for this research work is depicts in Fig. 3 below. Fundamentally, it's consisting of two controller loops namely; the inner controller loop corresponding to the hydraulic actuator control system and the outer controller loop corresponding to the vehicle suspensions control system. The inner loop controller must be capable of tracking the optimal targeted force for the actuator that was measured by the outer loop controller. The hydraulic actuator actual force obtained is introduced to the vehicle in order to resist the effects of road input disturbance [14]. The outer loop control system is employed for the elimination control of the road input disturbance in order to minimized undesirable vehicle motion. The reference signal desired actuator force is produced by the PID suspension output loop (which can be suspension travel or any other desired output by the designer).

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

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

$$e_i = r_i - y_i \quad (22)$$

Where, e_i is the control error and r_i is the reference signal. Considering suspension travel as one among the suspension output and according to suspension travel regulation, the suspension travel reference signal is always set to zero (i.e. $r_i = 0$). Therefore, it is hoped to designed a controller which obey the control law that states $e_i(t) \rightarrow 0$, as $t \rightarrow \infty$.

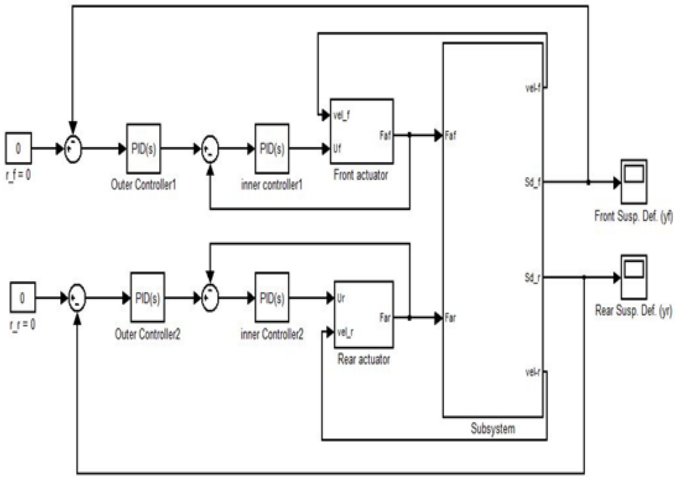

Fig. 3: Proposed Controller Architecture

TABLE III. INNER/OUTER PID CONTROL TURNING PARAMETERS

	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

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

A. Controller Performance Criteria

- Due to mechanical structure, suspension travel maximum allowable deflection is set to be;

$$sd = |z_{si} - z_{ui}| \le z_{i,\max} \tag{23}$$

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.

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

$$\left|k_{tf}(z_{uf} - z_{rf}) + b_{tf}(\dot{z}_{uf} - \dot{z}_{rf})\right| \le F_{sf} \tag{24}$$

$$\left|k_{tr}(z_{ur} - z_{rr}) + b_{tr}(\dot{z}_{ur} - \dot{z}_{rr})\right| \le F_{sr} \tag{25}$$

Where,

$$F_r = \frac{m_s g l_f + m_{ur} g (l_f + l_r)}{l_f + l_r} \tag{26}$$

$$F_f = (m_s + m_{uf} + m_{ur})g - F_r \tag{27}$$

F_{sf} And F_{sr} Denotes the front and rear static tire loads respectively.

- The maximum allowable actuator control force is given as $F_{ai,\max}$

Where,

$$F_{ai,\max} \le m_s g \tag{28}$$

Where, g is the acceleration due to gravity.

- The maximum control voltage allowed is;

$$u_{i,\max} \le 10V \tag{29}$$

Other important parameter is the Root Mean Square (RMS) values for vehicle suspension parameters which are:

- Vehicle sprung mass acceleration is given as;

$$\ddot{z}_{c,RMS} = \sqrt{\frac{1}{n}\sum_{0}^{n}(\ddot{z}_c)^2} \tag{30}$$

- Vehicle pitch angular acceleration is given as;

$$\ddot{\theta}_{c,RMS} = \sqrt{\frac{1}{n}\sum_{0}^{n}(\ddot{\theta}_c)^2} \tag{31}$$

- Vehicle suspension deflection is given as;

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

- Vehicle tyre travel is given as;

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

B. Simulation and Discussion of Result

To support that the design of a control system for a half vehicle model with hydraulic actuator is essential in fulfilling the control objectives or not, the time response of the proposed model was first investigated without controller (i.e. the open loop passive system).

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. This plots of the open loop system will show if the control objectives can be achieve without using control system for the proposed model or not. The proposed objectives of the research are; minimizing the vibration sensed by the passengers when travelling on sinusoidal discrete road profile and the avoidance of vehicle pitch motion when a critical maneuver occurs.

Responses of the passive systems as well as the active suspension systems were established in time domain analysis for a discrete road input profile. The input profile characterized a vehicle moving on a road that is having a sinusoidal of waves with amplitude of *10 cm*, wavelength of *9.1m* and forward velocity of *75km/hr (20.83m/s)*. Matlab/Simulink environment was used to build the half vehicle suspension system model.

Presented in Fig. 4 and Fig. 5 are vehicle suspension deflections time histories of front as well as rear suspensions for both active and passive suspension system respectively. It is clearly depicted that the uttermost travelled level by the suspension for wheels at utmost height of road disruption input and there values were less than the defined suspension travel limits of 0.1 *m*.

The RMS values obtained for both systems are given in table IV below. It can be seen from the percentage reduction that the vehicle rattle space was successfully minimized using the proposed control system and hence, passengers ride comfort was improved.

Vehicle handling time histories was shown in Fig. 6 and Fig. 7 respectively. The uttermost road holding capability value of 0.006 *m* and 0.017 *m* was attained for front active and passive wheels respectively. Whereas, 0.0043 *m* and 0.0085 *m* was attained for rear active and passive wheels respectively. The RMS values obtained show an improvement in the road handling capacity.

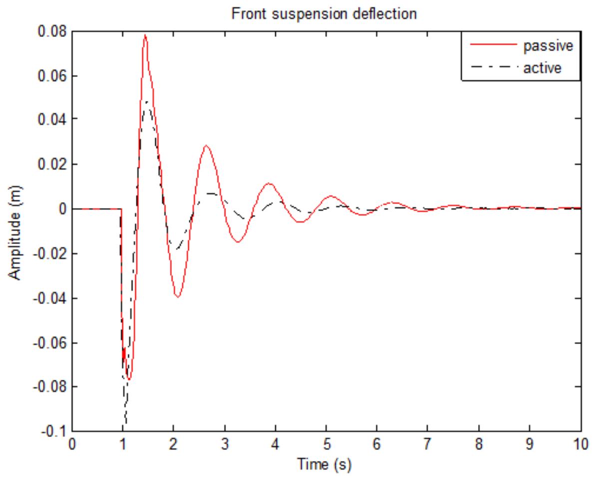

Fig. 4: Passive vs. Active front suspension deflection

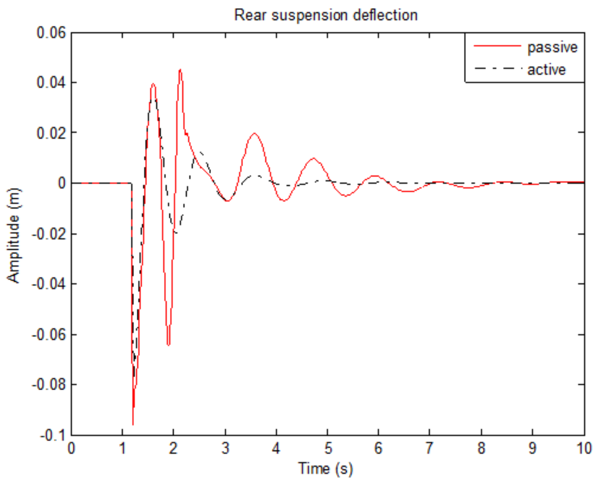

Fig. 5: Passive vs. Active rear suspension deflection

The active system was able to reduce the disturbance by suppressing it to about 2.3*s* in Fig. 8 when compared to about 5.0*s* of the passive system. The sprung mass acceleration deviate from -0.56m/s^2 and 1.16m/s^2 (-0.057*g* & 0.118*g*) for the active system, whereas, it is between -1.65m/s^2 and 4.35m/s^2 (-0.168*g* & 0.443*g*) for passive system which is within the ISO 2631-1 classified condition range of Not Uncomfortable and Little Uncomfortable for active and passive system respectively, for occupants in public transport (ISO 2631-1: 1997).

TABLE IV. RMS VALUES FOR ROAD INPUT DISTURBANCE

Parameters	**Passive System**	**Active System**	**% Reduction by Active System**
Front Suspension Deflection (m)	0.0268	0.0128	52.29
Rear Suspension Deflection (m)	0.0221	0.0094	57.47
Front Tyre Deflection (m)	0.0044	0.0024	45.46
Rear Tyre Deflection (m)	0.0023	0.0013	43.48
Sprung Mass Acceleration (m/s^2)	1.1867	0.1749	85.26
Pitch Angular Acceleration (rad/s^2)	0.0568	0.0268	52.26
Front Actuator Control Voltage (V)	-	0.4931	-
Rear Actuator Control Voltage (V)	-	0.4318	-
Front Spool-valve Displacement (m)	-	1.772e-4	-
Rear Spool-valve Displacement (m)	-	6.352e-5	-
Front Actuator Force (N)	-	540.329	-
Rear Actuator Force (N)	-	250.4649	

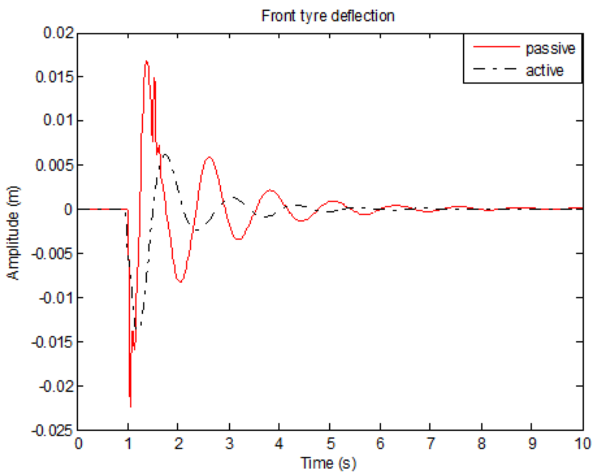

Fig. 6: Passive vs. Active front tyre deflection

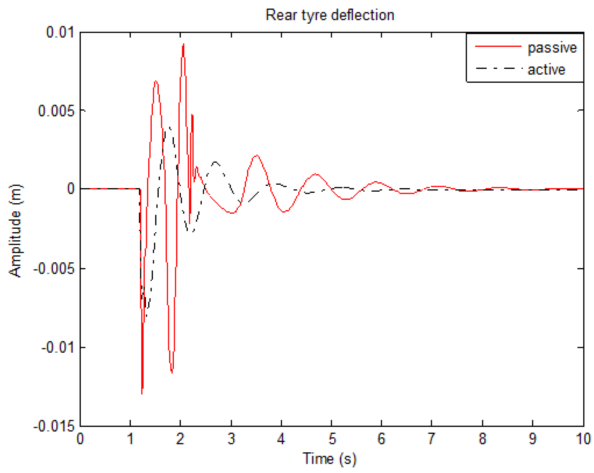

Fig. 7: Passive vs. Active rear tyre deflection

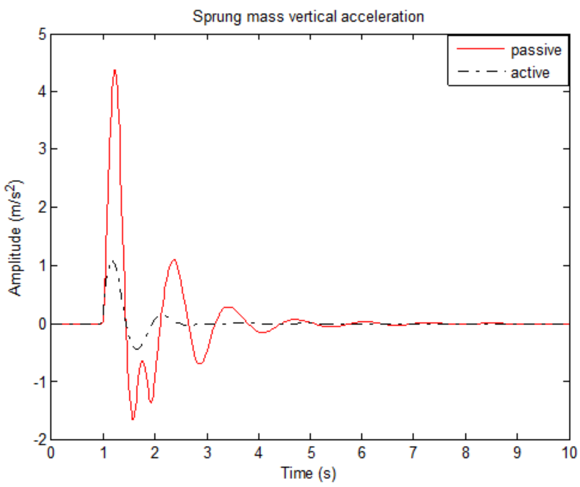

Figure 8: Passive vs. Active sprung mass vertical acceleration

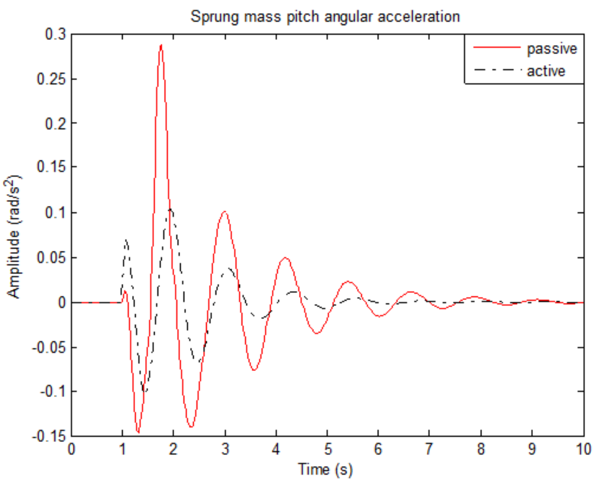

Fig. 9: Passive vs. Active sprung mass pitch angular acceleration

Fig. 9 described vehicle pitch angular acceleration with magnitude ranging between -0.1*rad/s*^{2} and 0.1 *rad/s*^{2} for active suspension, whereas, it is between -0.15 *rad/s*^{2} and 0.29 *rad/s*^{2} for passive suspension system respectively. The obtained RMS value gave about *52.8%* improvement in the passengers comfort compared to the passive system.

The actuator control input voltages was presented in Fig. 10. The control voltage ranges between *-3.1V* and *5.5V* for front actuator with RMS of *0.4931V*, and it is between *-4.0V* and *3.8V* for the rear actuator with RMS of *0.4318V*, which are all found to be less than the maximum specified value in this work. The disturbances effects generated by the road input are oppressed completely at about *1.2s* for both actuators.

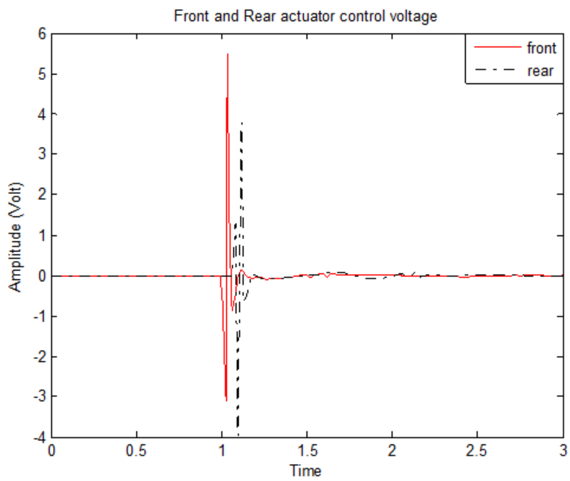

Fig. 10: Front vs. Rear actuator control voltage

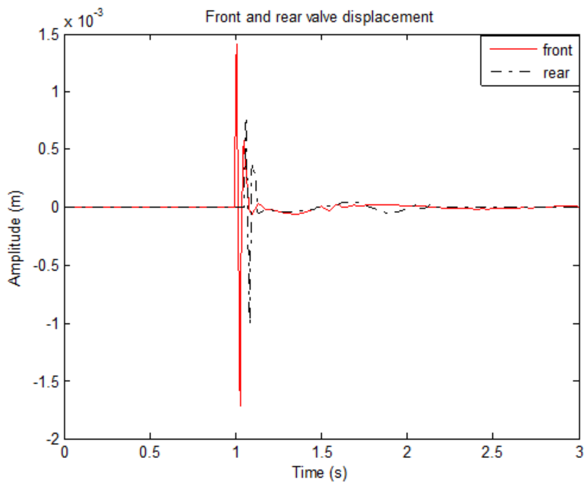

Figure 11: Front vs. Rear spool-valve displacement

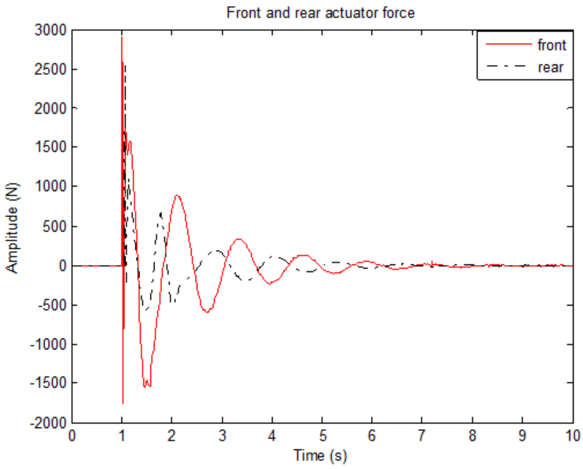

Figure 12: Front vs. Rear actuator force

The spool-valve displacement was depict in Fig. 11, the valve displacement was obtained between *-1.7mm* and *1.4mm* for front suspension whereas, is between *-0.93mm* and *0.75mm* for the rear suspension with RMS of *0.177mm* and *0.0635mm* respectively. Similarly, the actuator force ranges from *-1400N* to *2337N* for front actuator and from *-470N* to *1800N* for rear actuator (see Fig. 12).

IV. CONCLUSION

This work discussed in details the mathematical model of a nonlinear, half vehicle active suspension system with hydraulic actuator dynamics. This was carried out by firstly specifying forces that are generated from the nonlinear suspension elements; the spring forces are associated to the relative displacements between vehicle body and the wheels, whereas damper forces are associated to the relative velocities between vehicle body and the wheels.

A sinusoidal road input disturbances model was applied based on the literature. The road input disturbances exerted some forces on the vehicle through the wheels. Contrary to most of the models found in the literature, in this model the tyre damping was not ignored but the angular displacement was assumed to be very small. By the application of Newton's second law, the complete system model was produced. A nonlinear half vehicle passive suspension system was looked at for results comparison. Simulation of the developed model was implemented using Matlab/Simulink environment and the overall performance for the suspension system in terms of the root mean square (RMS) parameters reduction was found better in the active suspension system than that of the passive suspension system. Also, the suspensions travel, actuator input voltages and the actuator generated forces were found to be less than the specified limits in section III subsection A of this work. Hence, we concluded that regardless of the challenges of power consumption with relatively cost effects that is attached to active system; it poses better performance criteria compared to conventional passive system.

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