VOL. 10, NO. 23, DECEMBER 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

ACTIVE SUSPENSION FORCE CONTROL WITH ELECTRO-HYDRAULIC ACTUATOR DYNAMICS

Musa Mohammed Bello[1, 3], Adamu Y. Babawuro[2, 3] and Sado Fatai[3]
[1]Department of Mechatronics Engineering, Faculty of Engineering, Bayero University Kano, Kano. Nigeria
[2]Department of Mechatronics and Systems Engineering, Faculty of Engineering, Abubakar Tafawa Balewa University, Bauchi. Nigeria
[3]Department of Mechatronics Engineering, Kulliyyah of Engineering, International Islamic University Malaysia, Kuala Lumpur, Malaysia
E-Mail: mbmusa84@gmail.com

ABSTRACT

The purpose of vehicle suspension system is to keep-off the vehicle main body from any road geometrical irregularities thereby improving the comfort as well as maintaining a good handling stability. This work aim at designing a double loop PID control of generated force and vehicle suspension parameters using a four degree of freedom, nonlinear, half vehicle active suspension system model with hydraulic actuator. The loops arrangement is made up of an inner hydraulic actuator PID force control loop and an outer suspension parameters PID control loop. Simulation study was carried out; comparisons were made between the nonlinear active PID base suspension systems with a nonlinear passive system. Results obtained show a better performance improvement in the active system when compared to the passive system at the expense of cost and power consuming.

Keywords: force feedback, half car model, hydraulic actuator, PID control.

INTRODUCTION

Active vehicle suspension systems have been an area of vast research work for more than two decades due to their very much promising features. These systems pose the capability of responding to vertical changes due to road inputs 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 disperse energy from the system and this actuator can be control through different type of controllers which can be determined by the designer. The right control techniques gives rise to a better compromise to occur between vehicle ride comforts to road handling stableness, thus active vehicle suspension systems bids for a better suspension design.

To enable the control of vehicle active 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 [1]. These control strategies can be grouped based on different control techniques.

Linear control strategy based on optimal control concept is one of the most popular techniques that has being widely applied by most researchers in area of design of active vehicle suspension system [2]. Among the general idea applied for optimal control is the Linear Quadratic Regulator (LQR) method, 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 that are measures are a function of states as well as the inputs to the system [3].

State feedback control method was employed using active vehicle suspension in [4]. Other methods include Linear Parameter Varying (LPV) by [5], H-control strategy by [6]. Application of intelligent based control method such as Neural Network (NN), Fuzzy Logic Controls (FLC), Genetic Algorithms (GA) was also employed in the design of active suspension system [7].

Other control techniques in active suspension designs include nonlinearity nature of the system. Back stepping control techniques has been considered by [8]. [9] proposed a work on fuzzy control design technique using full vehicle model for nonlinear active suspension system with hydraulic actuator. [10] proposed a designed controller using sliding mode control method; also adaptive sliding mode control was looked into by [11]. All the results found in the literatures bring about some improvement into the system.

Active suspension systems in reality possessed a high nonlinearity and time-varying behavior. This behavior is contributed due to the existing nonlinear characteristics of passive suspension elements which consist of the springs, dampers and suspension bushes, active suspension actuators and the entire suspension geometry. The difficulty involved in modeling such behavior results in linear suspension models by most researches. Conventional and nonlinear modeling techniques require the exact models for the system accuracy.

It is believed that, most research's employed the used of intelligent control system when dealing with nonlinear systems, neglecting the fact that Feedback linearization (FBL) is introduced in a control system to linearized a nonlinear system, thereby, employing the use of a conventional control system like PID. Intelligent control system is capable of dealing with nonlinear uncertain systems without the need of an exact system

VOL. 10, NO. 23, DECEMBER 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

model. But in this work, we intend to deal with the real system model perhaps could help in reducing the possibility of poor performance on implementation due to nonlinearity of system and at the same time developed a control system which is less complex and also less expensive to utilize.

MATHEMATICAL MODEL

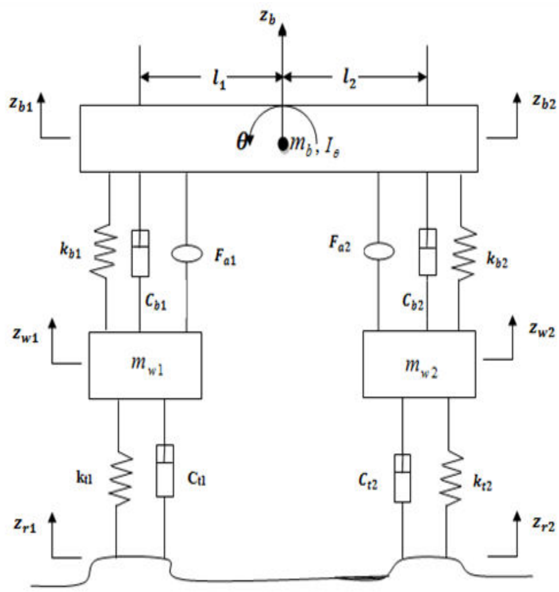

Figure-1. Half vehicle model.

Assuming a small pitching angle, and by application of Newton's second law of motion,

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

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

From the above Figure, the front and rear nonlinear suspension forces can be obtained as;

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

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

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

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

And the tyre forces as:

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

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

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

The dynamic equations of motion for the half vehicle nonlinear system model with hydraulic actuator forces can be obtained as;

$$\ddot{z}_b = -\frac{1}{m_b}[F_{kb1} + F_{kb2} + F_{cb1} + F_{cb2} - F_{a1} - F_{a2}] \tag{9}$$

$$\ddot{\theta}_b = \frac{1}{I_\theta}[l_1(F_{kb1} + F_{cb1} - F_{a1}) - l_2(F_{kb2} + F_{cb2} - F_{a2})] \tag{10}$$

$$\ddot{z}_{w1} = \frac{1}{m_{w1}}[F_{kb1} + F_{cb1} - F_{t1} - F_{a1}] \tag{11}$$

$$\ddot{z}_{w2} = \frac{1}{m_{w2}}[F_{kb2} + F_{cb2} - F_{t2} - F_{a2}] \tag{12}$$

Hydraulic Actuator Dynamics

The hydraulic actuator force is given as:

$$F_{ai} = A_{hyd}P_{Li} \tag{13}$$

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 Eqn. (14).

$$\dot{x}_{vi} = \frac{1}{\tau}(k_{vi}v_i - x_{vi}) \tag{14}$$

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;

$$Q_i = C_d\omega.x_{vi}\sqrt{\frac{1}{\rho}(P_s - \operatorname{sgn}(x_{vi})P_{Li})} \tag{15}$$

The rate of change of the pressure difference P_{Li} is given as;

$$\dot{P}_{Li} = \frac{4\beta_e}{V_t}[Q_i - C_{tp}P_{Li} - A_{hyd}(\dot{z}_{si} - \dot{z}_{ui})] \tag{16}$$

Where i denote either front or rear suspension. Let assumed the following terms;

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

Substituting the above assumptions, Eqn. (17) is obtained;

VOL. 10, NO. 23, DECEMBER 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

$$\dot{P}_{Li} = \gamma . x_{vi}\sqrt{(P_s - \mathrm{sgn}(x_{vi})P_{Li})} - \beta . P_{Li} - \alpha . A_{hyd}(\dot{z}_{bi} - \dot{z}_{wi}) \qquad (17)$$

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 [12]. For this work, 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 with a very poor (Class E) roughness coefficient

$(G(n_0) = 1024 * 10^{-6}\ m^2/(m/cycle))$.

The road surface input equation is given as;

$$\dot{z}_{ri} = 2\pi n_0 z_{ri} + 2\pi\sqrt{G(n_0) v w_0} \qquad (18)$$

Where, i is either front/rear suspension, 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 (see ISO 1982 values established for PSD).

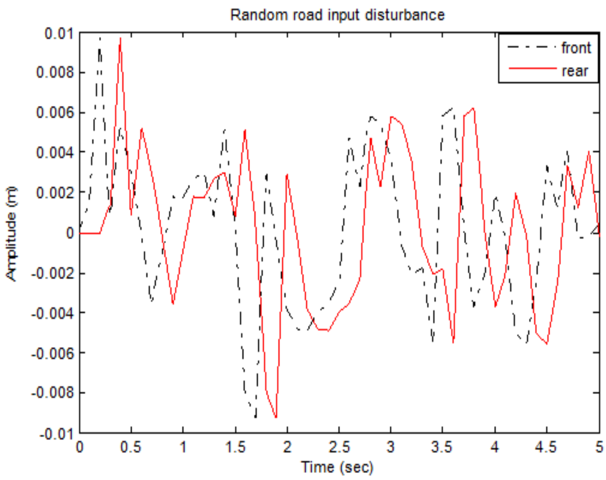

Figure-2. Random road input disturbance.

DESIGN OF CONTROLLER

The control architecture employed in this work comprises of two feedback control loops arrangement via. the inner controller loop checking the hydraulic actuator force control and the outer controller loop checking the vehicle suspensions control which are fed back through a PID control system respectively.

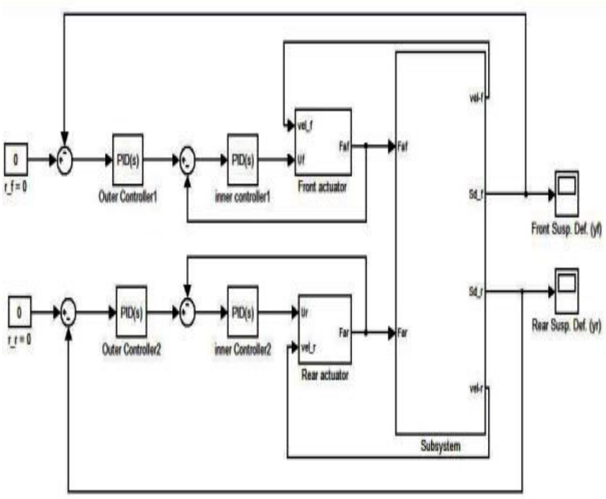

Figure-3. Proposed controller architecture.

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

$$u_i / F_{ai,ref} = K_{Pi} e_i(t) + K_{Ii}\int e_i dt + K_D \frac{de_i}{dt} \qquad (19)$$

$$e_i = r_i - y_i \qquad (20)$$

Table-1. Inner/Outer PID controller 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

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. $r_i = 0$) [13]. Therefore, it is hoped to designed a controller which obey the control law that states $e_i(t) \to 0$, as $t \to \infty$.

SIMULATION RESULTS

Transient responses of both passive and active suspension system were determined in time domain analysis for a random road input profile and the results were compared for both systems as depicted in the figures below. Matlab/Simulink environment was used to simulate the half vehicle nonlinear active suspension model with hydraulic actuators.

ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

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	kgm²
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 tire spring stiffness	175,500	N/m
k_{t2}	Rear tire spring stiffness	175,500	N/m
c_{t1}	Front tire spring damping coefficient	14.6	Ns/m
c_{t2}	Rear tire spring damping coefficient	14.6	Ns/m
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	$1.545*10^{9}$	$N / m^{5/2} /$
A_{hyd}	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

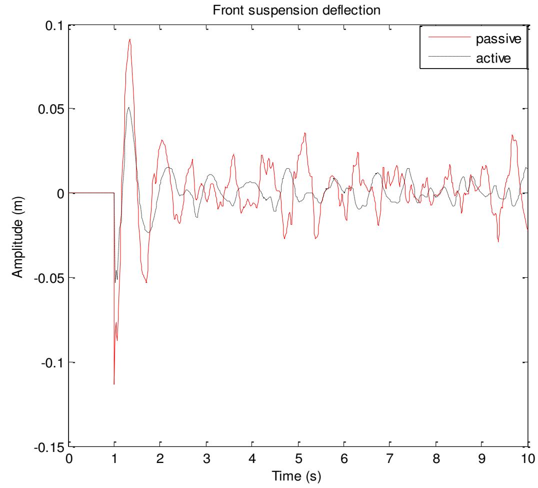

Figure-4. Passive vs. Active front suspension travel.

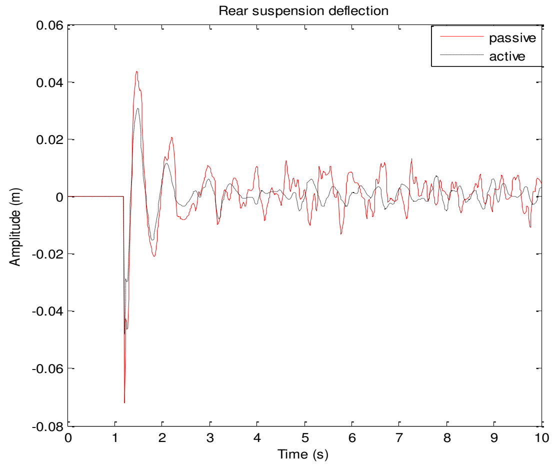

Figure-5. Passive vs. active rear suspension travel.

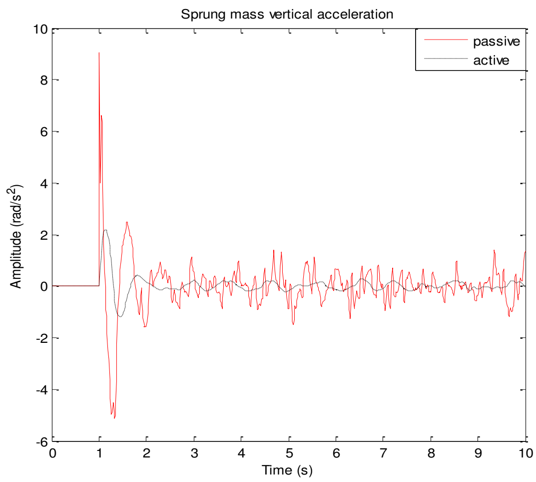

Figure-6. Passive vs. active sprung mass vertical acceleration.

ARPN Journal of Engineering and Applied Sciences

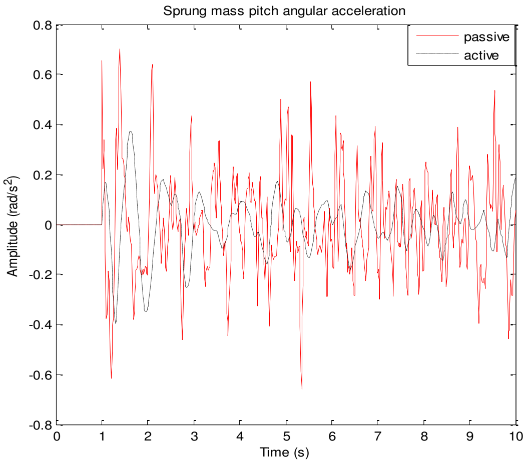

Figure-7. Passive vs. active sprung mass pitch angular acceleration.

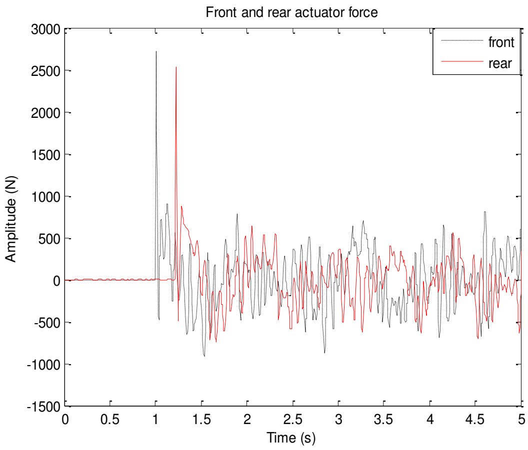

Figure-8. Front vs. rear actuator force.

CONCLUSIONS

This work discussed in detail the mathematical model of a nonlinear, half vehicle active suspension system with hydraulic actuator dynamics. Simulation studies was done and the overall system performances for active suspension system parameters with road input uncertainties was found better when compared to the passive suspension systems for a random road input disturbances, providing a better passengers ride comfort and also minimizing the suspension travel rattle space.

REFERENCES

[1] Y. M. Sam, J. H. S. Osman and M. R. A. Ghani. 2004. "A Class of Proportional-Integral Sliding Mode Control with Application to Active Suspension System," System & Control Letters, Vol. 51, pp. 217-223.

[2] Y. M. Sam. 2006. "Robust Control of Active Suspension System for a Quarter Car Model," Universiti Teknologi Malaysia.

[3] Y. M. Sam and J. H. S. Osman. 2005. "Modelling and Control of the Active Suspension System Using Proportional-Integral Sliding Mode Approach," Asian Journal of Control, Vol. 7, pp. 91-98.

[4] H. Du, N. Zhang and J. Lam. 2008. "Parameter-dependent input-delayed control of uncertain vehicle suspensions," Journal of Sound and Vibration, Vol. 317, pp. 537-556.

[5] C. Onat, I. B. Kucukdemiral, S. Sivrioglu and I. Yuksek. 2007. "LPV Model Based Gain-scheduling Controller for a Full Vehicle Active Suspension System," Journal of Vibration and Control, Vol. 13, pp. 1626-1666.

[6] H. Du and N. Zhang. 2007. "H-infinity control of active vehicle suspensions with actuator time delay," Journal of Sound and Vibration, Vol. 301, pp. 236-252, 2007.

[7] L. Yue, C. Tang and H. Li. 2008. "Research on Vehicle Suspension System based on Fuzzy Logic Control," International Conference on Automation and Logistics, Qingdao, China.

[8] N. Yagiz, Y. Hacioglu and I. Yuksek. 2008. "Backstepping control of a vehicle with active suspensions," Control Engineering Practice, Vol. 16, pp. 1457-1467.

[9] H. Du and N. Zhang. 2009. "Fuzzy control for nonlinear uncertain electro-hydraulic active suspensions with input constraint," in IEEE Transactions on Fuzzy Systems, pp. 343-356.

[10] N. Yagiz and I. Yuksek. 2001. "Sliding mode control of active suspensions for a full vehicle model," International Journal of Vehicle Design, Vol. 26, pp. 264–276.

[11] J. Lin, R. Lian, C. Huang and W. Sie. 2009. "Enhanced fuzzy sliding mode controller for active suspension systems," Mechatronics, Vol. 32, pp. 1178–1190.

[12] J. Cao, H. Liu, P. Li and D. Brown. 2008. "State of the art in vehicle active suspension adaptive control systems based on intelligent methodologies," IEEE Transactions on Intelligent Transportation Systems, Vol. 9, pp. 392-405.

[13] H. Gao, Lam J. and Wang C. 2006. "Multi-objective control of vehicle active suspension systems via load-dependent controllers," Journal of Sound and Vibration, Vol. 290, pp. 654-675.