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Design and Development of Vehicle Active Suspension Using PID Controller

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ABSTRACT--- **The main aim of suspension system is isolate vehicle body from disturbance caused by road surface variation. In most of the vehicles passive types of suspension are used. In passive suspension system, there is compromise between ride handling of vehicle and ride comfort of passenger. In this paper, an active suspension uses pneumatic actuator which controlled by PID has been generated.Actuator generates opposite force by sensing the various road inputs and thus providing good handling along with ride comfort. The mathematical model has been derived and is used to create Matlab/Simulink model. There has been 80%, 78.5%, 78.52% reduction in sprung mass displacement for step input, Sine input and White noise input respectively and vehicle stability has been improved as There has been 79.9%, 60.9%, 78.4% reduction in unsprung mass displacement for step input, Sine input and White noise input respectively.**

***Keywords:Active Suspension System, Matlab/Simulink, PID controller, Pneumatic actuator*.**

I. INTRODUCTION

As per the vehicle suspension System dynamic trademark, this venture utilizes present day control theory in the dynamic suspension system [1]. With a specific end goal to be generally utilized on the vehicle, numerous car organizations and research foundation have contributed both labor and material assets in order to get practical suspension. Utilization of new control innovation, innovative work to a control framework, is financially savvy and requires low vitality utilization is objective of connected research as well as imperative confirmation criteria for theory research [2].

Active suspension has been interesting issue of research and had examined in a wide range of controllers already. Yahaya Md. Sam had explored on LQR controller for Active suspension framework. They have performed investigation for latent and active suspension for the vehicle's performance and they have accomplished change in it [3].Based on Kruczek, Active Suspension's riding solace and taking care of can likewise be controlled by H-infinity controllers [4-5]. Wise controllers, for example, fluffy rationale have been executed into the dynamic framework [6]. Faraz Ahmed Ansari, Rajshree Taparia utilized Sliding control mode controller for Activesuspension .The execution of these controllers figured out how to give great Ride handling and ride comfort when contrasted with the latent system [7]. A nonlinear sliding control law is connected to an electro-water powered suspension framework [8]. Senthil Kumar, an introducer of PID control System could decrease the peak overshoot of the sprung mass relocation when contrasted with detached system [9].

II. MATHEMATICAL MODELING

Vibration control system outline ought to begin with the scientific model foundation, and after that decide the plan requirement, and formal portrayal of it.The parameters are taken from reference [7].

Table I

Suspension Parameters

Parameter	Value
Sprung mass	250 kg
Unsprung mass	50 kg
Spring stiffness	18,600 N/m
Damping coefficient	1,000 Ns/m
Tire stiffness	196,000 N/m

Notation used:

M_s: Sprung Mass, kg

M_{us}: Unsprung Mass, kg

K_1: Spring Stiffness, N/m

K_2: Tire Stiffness, N/m

C_a: Damping coefficient, Ns/m

X_s: Vertical Sprung mass displacement, m

X_{us}: Vertical Unsprung mass displacement, m

X_r: Vertical road profile displacement, m

$\dot{x}_s$: Sprung mass velocity, m/s

$\dot{x}_{us}$: Unsprung mass velocity, m/s

$\ddot{x}_s$: Sprung mass acceleration, m^2/s

$\ddot{x}_{us}$: Unsprung mass acceleration, m^2/s

F_a: Force generated by actuator, N

X_a: Actuator displacement,m

A. Passive Suspension

In the car universe of complex vibration System, simplifications are done in light of investigation of issue. There are several ways to show mathematical model. We have shown in figure mention below:

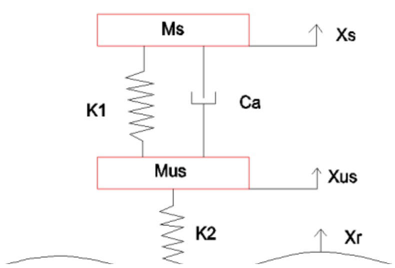

Fig 1. Passive Suspension System

The quarter model has been generated and two degree of freedom motion differential equations has been derived. These equations derived have been shown below.

For sprung mass:

$$M_s\ddot{x}_s + C_a(\dot{x}_s - \dot{x}_{us}) + K_1(X_s - X_{us}) = 0 \quad (1)$$

For unsprung mass:

$$M_{us}\ddot{x}_{us} + C_a(\dot{x}_{us} - \dot{x}_s) + K_1(X_{us} - X_s) + K_2(X_{us} - X_r) = 0 \quad (2)$$

B. Active Suspension

Active suspension uses pneumatic actuator to control the force. To measure suspension variables active suspension requires sensors.The ¼ vehicle dynamic vibration demonstration does not include whole vehicle parameters and it does not focus into vehicle pitching and roll motion but still system load and suspension information has been included.The active suspension system is uses actuator to add or dissipate the energy from the system. This helps in controlling the inertia of the vehicle and also beneficial in braking and cornering, thus improving vehicle stability.

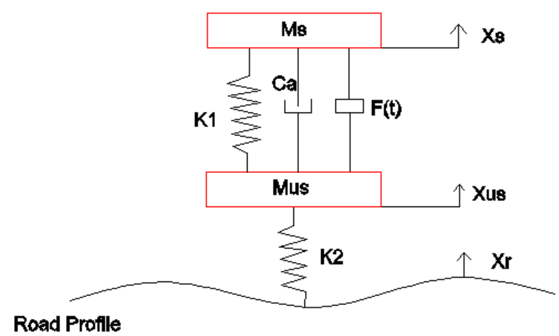

Fig 2. Active Suspension

Active suspension system works with hydraulic or pneumatic actuators, so it is clear that the desired external force is introduced into the system. Pneumatic actuator has been discussed in this paper.

The figure 2 shows the active suspension system and y the use of Newton's second law of motion the following equations are derived.

For sprung mass:

$M_s \ddot{x}_s + C_a (\dot{x}_s - \dot{x}_{us}) + K_1 (X_s - X_{us}) = F_a$(3)

For unsprung mass:

$M_{us} \ddot{x}_{us} + C_a(\dot{x}_{us} - \dot{x}_s) + K_1(X_{us} - X_s) + K_2(X_{us} - X_r) = -F_a$(4)

Here, $F_a = A_1P_1 - A_2P_2$

Where, P_1 and P_2 are the absolute pressures in the actuator chambers and A_1 and A_2 are the piston effective areas.

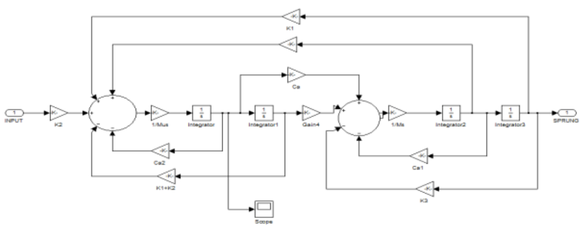

Fig 3. Active system Simulink Model

C. *Pneumatic Actuator System*

Pneumatic actuators are affordable power source, simple, high power to weight ratio and have a prompt access.The pneumatic cylinder consists of three subsystems: the valve, the cylinder containing the piston and chambers and the load and scientific model of a pneumatic actuator has been all around described in this section. [13]

$\ddot{X}_a = \frac{\mathrm{Fa}}{\mathrm{Ms-Mus}} = \frac{A_1P_1 - A_2P_2}{\mathrm{Ms-Mus}}$(5)

Pneumatic actuator with proportional valves has been considered as proportional valves offer fine-grained control of the port size and are also less noisy. [14]

From the reference [14] the relation between mass flow rate and the effective pressures are obtained.

$\dot{M}_1 = a_c \times v \times P_1$ (6)

$\dot{M}_2 = a_c \times v \times P_2$(7)

Where $\dot{M}_1$ and $\dot{M}_2$ are mass flow rates of chamber and a_c,v are orifice area of valve and applied voltage to actuator valve respectively.According to cylinder chamber model and cylinder chamber dynamics, the time derivative of pressure is given by, [15]

$$\dot{P}_i = \frac{K}{A_i l X_a}(RT_iM_i \pm A_iP_i\dot{X}_a) \quad (8)$$

Where, i= 1, 2 are cylinder chamber index, K is adiabatic index, R is ideal gas constant and Tis respective temperatures for respective chambers.With the help of equation 5, 6, 7, and 8 following simulink model is obtained.

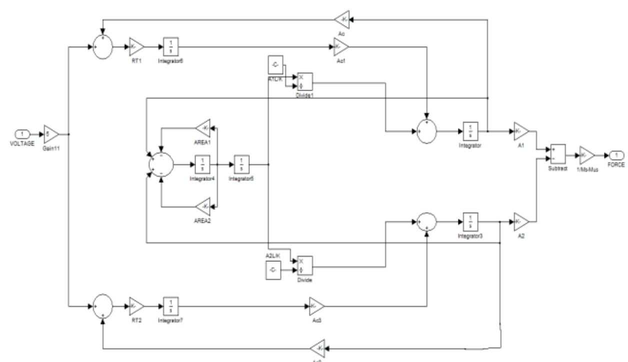

Fig 4. Actuator Simulink Model

III. PID CONTROL SYSTEM

The sprung mass displacement and the impact of disturbancehave been seen as the controlled variable reaction of the System. Additionally, PID control for the most part performs superbly for a system with little unsettling impact and works at a low speed. PID control system is one of the establish control system so, active force control part gives the additional strength featurethrough its Disturbance dismissal capacity.

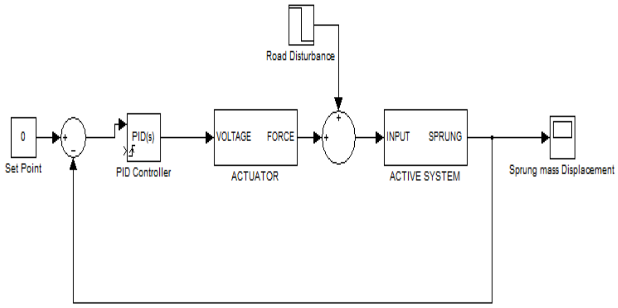

Fig 5. PID Controller design

Ziegler- Nicholas method is being used to find the values of K_p , K_i and K_d which are proportional, integral and derivative gains respectively. After tuning, K_p=61, K_i=9.1 and K_d = 0.7 were chosen for this system.

IV. EXPERIMENTAL SETUP

Fig 6. Experimental Setup

Excitation source validation could have been done using all three types of input but Sinusoidal input is fairly sufficient and that it is very easy and economically done. A Spur gear (B) was used and a small disturbance has been created on the surface of gear. The base (A) was formed which holds the spur gear shaft. The tire shaft is attached to slider and pneumatic actuator(D) is mounted on the shaft carrying tire.

V. RESULTS AND DISCUSSION

The road input signals taken into consideration are step input, sine input and White noise input signal.

A. *Step input signal*

The very first thing in any suspension system is it should sustain an impact occurred by jounce or rebound provided by road surface. The step input signal is basic need of any suspension system. We have considered rebound condition. The result has been taken with step of 1 m.

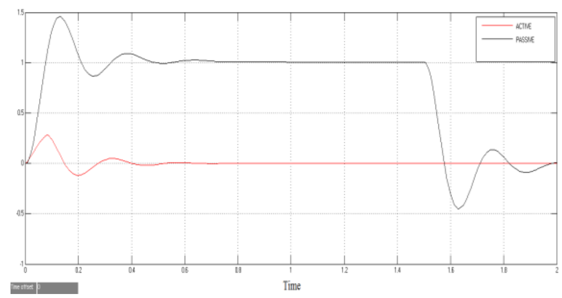

Fig 6.Sprung mass displacement Vs Time

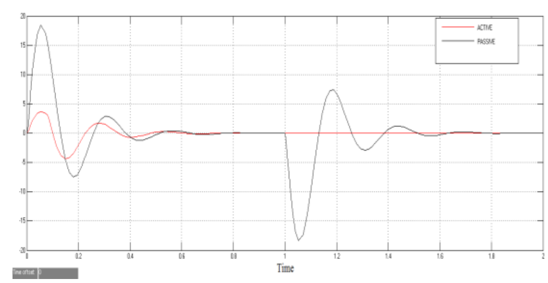

Fig 7. Sprung mass Acceleration Vs Time

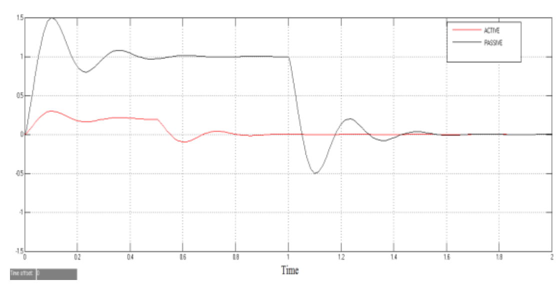

Fig 8.Unsprung mass Displacement Vs Time

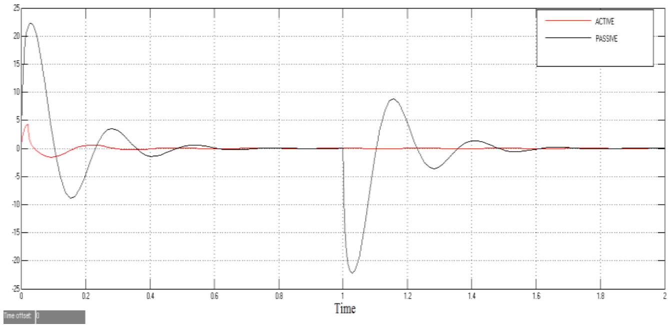

Fig 9.Unsprung mass Acceleration Vs Time

Table II
Step input Results

Parameters	Overshoot Values		%Reduction
	Passive	*Active*	
Sprung mass Displacement(m)	1.4593	0.2914	80.035
Sprung mass Acceleration(m/s^2)	18.4306	3.6950	79.95
Unsprung mass Displacement(m)	1.5018	0.3005	79.99
Unsprung mass Acceleration(m/s^2)	22.2859	3.4500	84.51

The vehicle will be more stable with active as there has been 80 % reduction in unsprung mass displacement and 84.5% reduction in unsprung mass acceleration. Unsprung mass provides the lesser roll of vehicle, thus vehicle will be much more stable with active suspension system.Also, the settling time for sprung and unsprung mass displacement has been reduced by 30% and 18% respectively while unsprung mass acceleration by 45%.

B. Sine input signal

Every automotive industries uses sine wave pavement input signal before any vehicle drives on road [10].

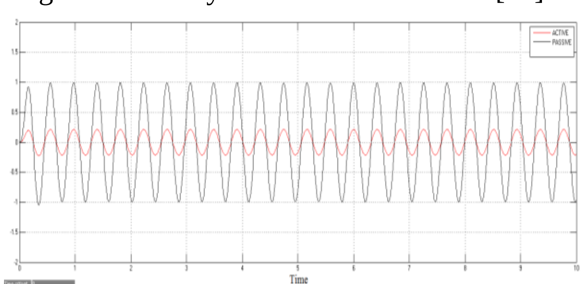

Fig 10. Sprung mass displacement Vs Time

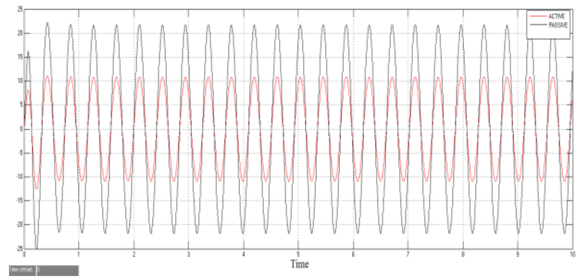

Fig 11.Unsprung mass Acceleration Vs Time

Table III
Sine input Results

Parameters	Peak Values		%Reduction
	Passive	**Active**	
Sprung mass Displacement(m)	1.0781	0.2316	78.51
Sprung mass Acceleration(m/s^2)	23.650	11.84	49.93
Unsprung mass Displacement(m)	1.0407	0.4066	60.93
Unsprung mass Acceleration(m/s^2)	25.144	13.50	52.58

The peak value obtained for sprung mass displacement for passive type of suspension is 1.0787 m, while for active type of suspension is 0.2316 m. The Sprung mass acceleration for passive suspension is 23.650m/s^2 and for active suspension system is 11.84m/s^2. Unsprung mass displacement for passive suspension is 1.0407 m, whereas for active suspension is 0.4066 m. The Sprung mass acceleration for passive suspension is 25.144m/s^2 and for active suspension system is 13.50m/s^2.

C. White noise input signal

When vehicle is moving on road with constant velocity the road input is random. According to ISO/TC108/SC2N67 international standard, the usual way to represent the random variable is in terms of its power spectral density (PSD). Filtering white noise generation method has been used as it has clear physical meaning and easy in computing for steady state stochastic Gaussian process [11]. So we have following equation,

$$\dot{q}(t) = -2\pi f_0 q(t) + 2\pi n_0 \sqrt{G_q(n_0)v}\, w(t) \qquad (9)$$

Where,

q(t)	-	Random road input signal
f_0	-	Filter lower-cut-off frequency
$G_q(n_0)$	-	Road roughness coefficient, unit;m^2/m^{-1}
w(t)	-	Gaussian white noise

According to international standard, Depending on road roughness coefficient values the road is classified into different class from level A to level H. level A is best road surface while level H is worst road surface [12]

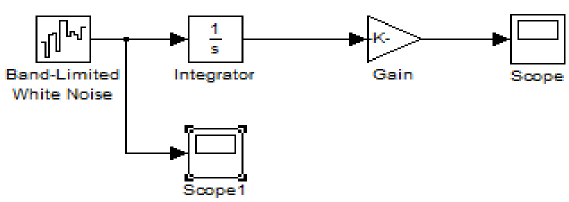

Fig 12: White noise input road Simulink model

The results obtained with the help of simulink model for white noise input signal are shown below.

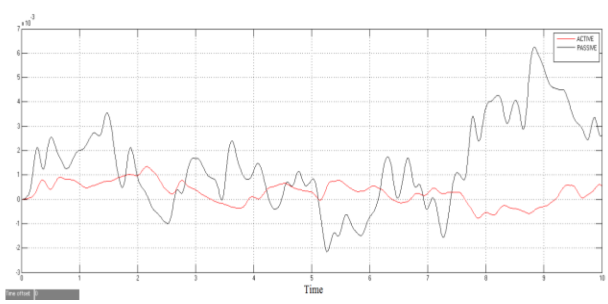

Fig 13: Sprung mass displacement Vs Time

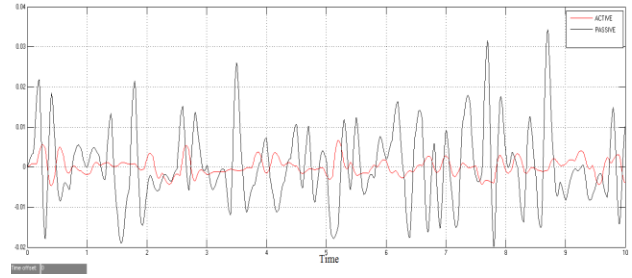

Fig 14: Unsprung mass Acceleration Vs Time

Table IV
White Noise input Results

Parameters	**Peak Values**		**%Reduction**
	Passive	***Active***	
Sprung mass Displacement(m)	0.00624	0.00134	78.52
Sprung mass Acceleration(m/s^2)	0.0312	0.00632	79.74
Unsprung mass Displacement(m)	0.00621	0.00134	78.42
Unsprung mass Acceleration(m/s^2)	0.0343	0.00634	81.51

VI. CONCLUSION

For step input, there is 80% reduction in peak overshoot value of passive suspension for sprung mass displacement. 79.95% reduction in peak overshoot value of passive suspension has been obtained for sprung mass acceleration. Stability has been improved as 84% reduction of peak overshoot value for unsprung mass acceleration. Peak overshoot value for unsprung mass displacement has been reduced by 79.9%.

For sine input, vehicle handling has been improved as there is 78.5% reduction of peak value of passive suspension. Sprung mass acceleration has been reduced by 49.9%. Stability and handling also has been improved as unsprungmass acceleration peak value has been reduced by 52.5%.Peak value for unsprung mass displacement has been reduced by 60.9%.

As white noise input is stochastic process, the peak value reduction obtained for sprung mass displacement is 78.5%. Sprung mass acceleration has been reduced by 79.7%. Stability has been improved as unsprung mass acceleration peak value has been reduced by 81.5%.Peak value for unsprung mass displacement has been reduced by 78.4%. Future work may include improvement of the suspensionsystem considering the experience of real world.

V. REFERENCES

[1] Lin, J.S and Kanellakopoulos, I. "Nonlinear Design of Active Suspensions," *IEEE Control Systems Magazine*, Vol.17, No.3, pp.45-59, 1997.

[2] William F Milliken and Douglas L Milliken "Race Car Vehicle Dynamics*", Milliken Research Associates, Inc*.

[3] Y. M. Sam, M. R. Ab.Ghani, and N. Ahmad. "LQR Controller for Active Car Suspension*". Mechanical Journal of Control and instrumentation.* pp.I-441-I-444, 2000.

[4] Thomas D Gillespie; Inc. "Fundamentals of Vehicle Dynamics", *Society of Automotive Engineers*, 1992.

[5] A .Kruczek and J. Honcu. "Active Suspension-Case study on Robust Control". World Academy of Science, Engineering and Technology, *International Journal of Computer, Electrical, Automation, Control and Information Engineering* Vol: 5, No: 6, 2011

[6] Ahmed Elmarakbi, Chandrasekaran Rengaraj, Alan Wheatley & Mustafa Elkady. "The Influence of Electronic Stability Control, Active Suspension, Driveline and Front Steering Integrated System on the Vehicle Ride and Handling".*Global Journal of Researches in automotive engineering*, 2013

[7] Faraz Ahmed Ansari, Rajshree Taparia, "Modeling, Analysis and Control of Active SuspensionSystem using Sliding Mode Control and Disturbance Observer*"5thInternational Colloquium on Signal Processing & Its Applications (CSPA),* 2009.

[8] Andrew Alleyne and J. Karl Hedrick "Nonlinear Adaptive Control of Active Suspensions"*IEEE transactions on control systems technology*, vol.1**,** no.1.march1995

[9] M. Senthil, "Development of Active Suspension System for Automobiles using PID Controller," vol. II, 2008.

[10] A. A. Aldair and W. J. Wang, "Design an Intelligent Controller For Full Vehicle Nonlinear Active Suspension Systems" *International Journal On Smart Sensing And Intelligent Systems* Vol. 4, No. 2, June 2011

[11] Qi Zhou, "Research and Simulation on New Active Suspension Control System",*Lehigh University*,2013

[12] SemihaTurkay and HuseyinAkcay "Spectral Modeling of Longitudinal Road Profiles",available:www.ccece2015.org

[13] G. Priyandoko, M. Mailah , H. Jamaluddin, "Vehicle active suspension system using sky hook adaptive neuro active force control", *Mechanical system and signal processing,Mechanical Systems and Signal Processing*Vol.23 pp-855–868,2009

[14] Tassa, Yuval, et al. "Modeling and identification of pneumatic actuators." *IEEE International Conference on Mechatronics and Automation. IEEE*, 2013.