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Modeling, Simulation and Control of Semi Active Suspension System for Automobiles under MATLAB Simulink using PID Controller

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Abstract: This paper aims to investigate the performance of a quarter car semi-active suspension system using PID controller under MATLAB Simulink Model. Dynamic system used in this study is a linear system. A linear system can capture basic performances of vehicle suspension such as body displacement, body acceleration, wheel displacement, wheel deflection, suspension travels. Performance of suspension system is determined by the ride comfort and vehicle handling. It can be measured by car body displacement and wheel deflection performance. Two types of road profiles are used as input for the system. Results show that the performance of body displacement and wheel displacement can be improved by using the proposed PID controller.

Keywords: Semi-Active suspension, Quarter car Model, MATLAB Simulink, step and random road disturbance

1. INTRODUCTION

Vehicle suspensions are mainly classified into three types i.e., passive, semi active and active suspensions, which depend on the operation mode to improve vehicle ride comfort, vehicle safety, road damage minimization and the overall performance.

Normally, passive suspension is an older conventional system having non-controlled springs and shock observing dampers with fixed parameters and no online feedback action is used- Appleyard et al. (1997) and Sun et al. (2009).Passive suspension design performance is used for specific operating conditions .On contrary, active suspensions can have a wide range of operation condition and can adapt to the system variations based on online changes of the actuating force. Therefore, active suspensions have been extensively studied since 1960s and various approaches have been proposed Harvat et al. (1997).However active suspensions normally require large power supply, which is the main drawback that prevents this technique from being used extensively in practice. From 1970s, semi-active suspensions have received much attention since they can achieve desirable performance than passive suspensions and consume much less power than active suspensions. Especially some controllable dampers are available in practice that is electrorhelogical (ER) dampers and magneto-rheological (MR) dampers. Semi-active suspensions are more practical than ever in engineering realization. Semi-active control with MR dampers for vehicles suspensions have been studied by many researchers- Yao et al. (2002) and Lai et al. (2002). Many control strategies such as skyhook, groundhook and hybrid control - Ahmadian et al. (2000), H -infinity control- Choi et al. (2002) and model following sliding mode control -Yokoyama et al. (2001) have been evaluated in terms of their applicability in practice.

In this paper, the PID controller is considered for providing the fine control to vehicles suspension system. The proposed approach has more advantages as compared with the conventional passive approach- Constantin et al. (2009), Hanafi et al. (2010) and Kumar et al. (2008).The modelling and simulation is carried in Simulink environment and further a sophisticated controller is implemented i.e. PID controller. Simulink is a versatile interface which can handle various types of controllers easily –Lai et al. (2002) and Herren et al. (2008).

The present work tries to analyse the performance of semi-active suspension control in the application of two degree freedom vehicle suspension. Dynamic response with road disturbances is simulated with fixed parameter of the system. Further implements PID controller and performance improvement analyses carried out for such system.

2. SYSTEM MODELLING

The mathematical modelling of a two degree of freedom quarter car body for a semi-active suspension system is being carried out by using basic laws of mechanics.

Modelling of suspension system has been taking into account the following observations.

- The suspension system modelled here is considered two degree of freedom system and assumed to be a linear or approximately linear system for a quarter cars.

- Some minor forces (including flex in the vehicle body, movement and backlash in various linkages, joints and gear system) are neglected for reducing the complexity of the system because effect of these forces is minimal due to low intensity. Hence these left out for the system model.

- Tyre material is considered as having damping property as well as stiffness.

Mathematical Modelling of Semi-active Suspension System

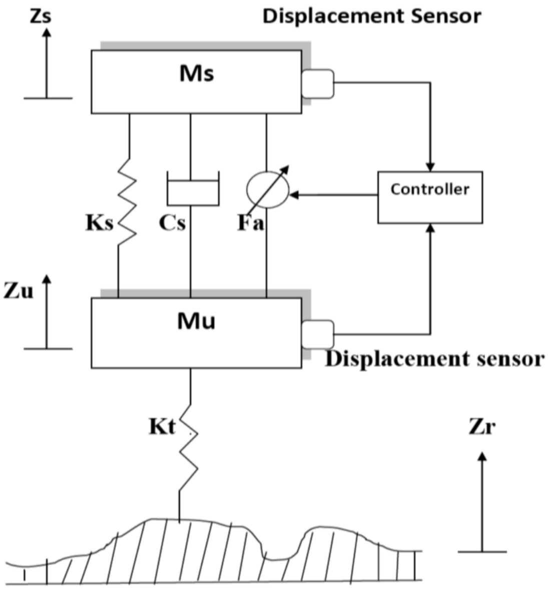

Fig 1. Quarter car semi active suspension model

From fig.1 we have the following equations,

$$M_s\ddot{Z}_s + K_S(Z_s - Z_u) + C_s(\dot{Z}_{s}-\dot{Z}_u) + U(t) = 0$$

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_{s}-\dot{Z}_u) = -U(t) \quad (1)$$

$$M_u\ddot{Z}_u + K_S(Z_u - Z_s) + C_S(\dot{Z}_{u}-\dot{Z}_s) + K_t(Z_u - Z_r) = U(t) \quad (2)$$

After choosing State variables as,

$$x_1(t) = Z_s(t) - Z_u(t)$$

$$x_2(t) = Z_u(t) - Z_r(t)$$

$$x_3(t) = \dot{Z}_s(t)$$

$$x_4(t) = \dot{Z}_u(t)$$

From equation (1), we have

$$M_s\dot{x}_3(t) + C_s[x_3(t) - x_4(t)] + K_s[x_1(t)] = -U(t) \quad (3)$$

From equation (2), we have

$$M_u\dot{x}_4(t) + C_s[x_4(t) - x_3(t)] - K_s[x_1(t)] + K_t[x_2(t)] = U(t) \quad (4)$$

Disturbance caused by road roughness,

$W(t) = \dot{Z}_r(t)$

Therefore,

$$\dot{x}_1(t) = x_3(t) - x_4(t)$$

$$\dot{x}_2(t) = x_4(t) - W(t)$$

$$\dot{x}_3(t) = -\frac{K_s * x_1(t)}{M_s} - C_s * \frac{x_3(t)}{M_s} + C_s * \frac{x_4(t)}{M_s} - \frac{U(t)}{M_s}$$

$$\dot{x}_4(t) = -\frac{K_s * x_1(t)}{M_u} - K_t * \frac{x_2(t)}{M_u} + C_s * \frac{x_3(t)}{M_u} - C_s * \frac{x_4(t)}{M_u} + \frac{U(t)}{M_u}$$

State space equation can be written as form,

$$\dot{x}(t) = Ax(t) + BU(t)$$

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\end{bmatrix} = \begin{bmatrix}0 & 0 & 1 & -1\\ 0 & 0 & 0 & 1\\ -K_s/M_s & 0 & -C_s/M_s & C_s/M_s\\ K_s/M_u & -K_t/M_s & C_s/M_s & -C_s/M_s\end{bmatrix}\begin{bmatrix}x_1\\ x_2\\ x_3\\ x_4\end{bmatrix} + \begin{bmatrix}0\\ 0\\ -1/M_s\\ 1/M_u\end{bmatrix}U + \begin{bmatrix}0\\ -1\\ 0\\ 0\end{bmatrix}W$$

Where,

$$A=\begin{bmatrix}0 & 0 & 1 & -1\\ 0 & 0 & 0 & 1\\ -K_s/M_s & 0 & -C_s/M_s & C_s/M_s\\ K_s/M_u & -K_t/M_s & C_s/M_s & -C_s/M_s\end{bmatrix} \quad B=\begin{bmatrix}0\\ 0\\ -1/M_s\\ 1/M_u\end{bmatrix} \quad Bw=\begin{bmatrix}0\\ -1\\ 0\\ 0\end{bmatrix}$$

$$C=\begin{bmatrix}1 & 0 & 0 & 0\\ 0 & 1 & 0 & 0\\ 0 & 0 & 1 & 0\\ 0 & 0 & 0 & 1\end{bmatrix}, \; D= [0; 0; 0; 0]$$

3. SIMULATION UNDER MATLAB SIMULINK

In this approach, system needs to be simulated for getting the dynamic response. The entire suspension is simulated with the application of MATLAB Simulink. In the last section mathematical modelling of proposed system is done.

The Simulink library and logic is developed according to the mathematical equations (1) and (2) and the entire system is simulated in the Simulink as shown in Fig.2

Table 1. Parameters used in system simulation

S.N.	Parameter	Symbol	Quatities
1	Mass of vehicle body	Ms	504.5kg
2	Mass of the tyre and suspention	Mu	62 kg
3	Coefficient of suspension spring	Ks	13100 N/m
4	Coefficient of tyre material	Kt	252000 N/m
5	Damping coefficient of the dampers	Cs	400 N-s/m

The parameter values are taken from-Haiping et al. (2005) and are listed in Table 1.

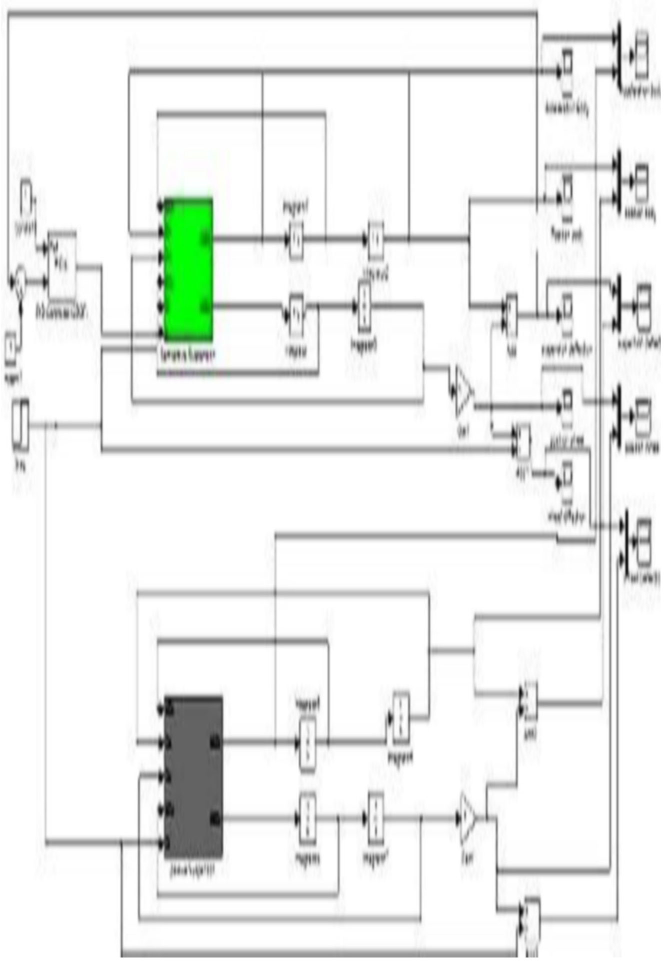

Fig 2. Simulation of basic suspension system with PID controller in MATLAB simulink

4. SIMULATION RESULTS

It is a Proportional Integral derivative (PID) as a feedback loop controller for the proposed system. In this closed loop an error signal is fed to adjust the input in order to reach the output to desired set of point. For tuning the controller in order to reduce the overshoot and settling time the following gain values are taken into consideration:

K_P=800, K_D=5000 and Ki =3500

The above selected values of gains are taken into account by adjusting it manually in Simulink where the minimum settling time and Peak overshoot is possible, while Simulink platform also have a provision by which it can auto tune the gains.

Now the performance of the designed suspension system under two types of road excitation i.e. Jerk (step input) and random input is evaluated through computer simulation

4.1 *System responses under step input(Jerk)*

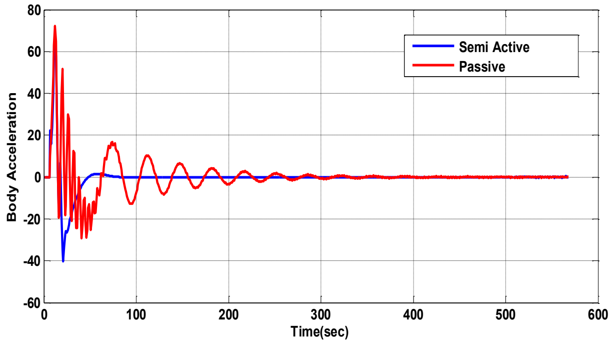

Fig 3. Time response of vehicle body acceleration

From fig 3, peak overshoot of vehicle body acceleration for Passive system is 70 m/s^2 whereas for semi-active suspension system it is 40m/s^2.

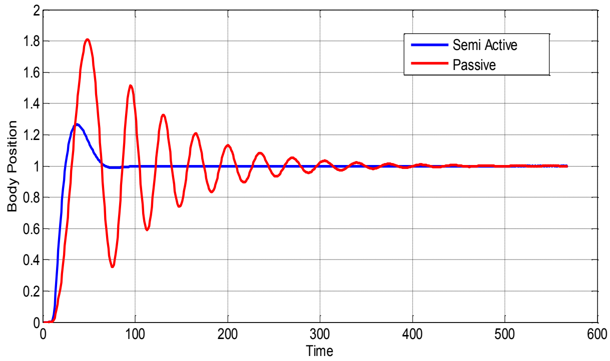

Fig 4. Time response of vehicle body position

From fig 4, peak overshoot of vehicle body position for Passive system is 1.8 m whereas for semi-active suspension system it is 1.25 m

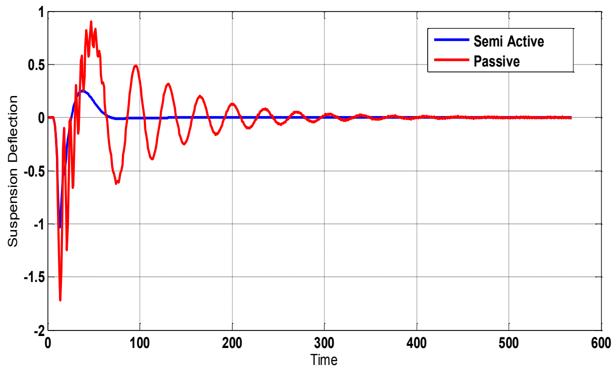

Fig 5. Time response of vehicle suspension deflection

From fig 5, peak overshoot of vehicle suspension deflection for passive system is 0.95m whereas for semi-active suspension system it is 0.2 m.

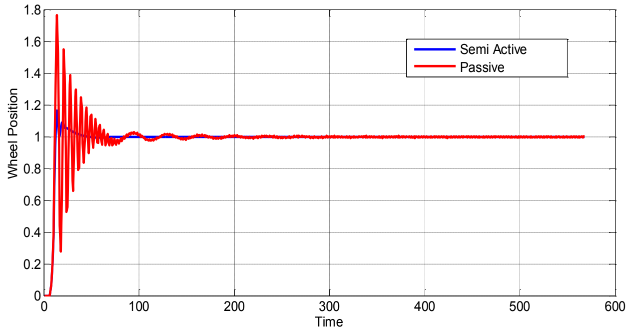

Fig 6 .Time response of vehicle wheel position

From fig 6, peak overshoot of vehicle body wheel position for Passive system is 1.78 m whereas for semi-active suspension system it is 1.19m

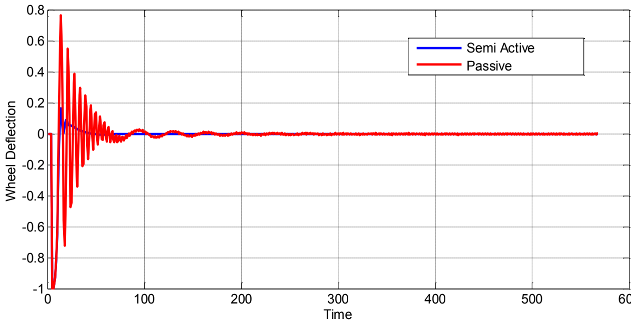

Fig 7. Time response of vehicle wheel deflection

From fig 7, peak overshoot of vehicle wheel deflection for Passive system is 0.79 m whereas for semi-active suspension system it is 0.19 m

Table 2. Performance Evaluation

	Settling Time(Sec)		
	Passive	**Semi-active**	**Improvement**
Body Acceleration	300	50	83 %
Wheel Deflection	250	60	76 %
Wheel Position	255	65	74.5 %
Suspension Deflection	300	85	71.66 %
Body Position	300	75	75%

From Table 2,settling time of vehicle body acceleration,wheel deflection,wheel position ,suspension deflection and body position for passive system is 300 sec,250sec,255sec,300sec,300sec respectively whereas for semi-active suspension system it is 50sec,60sec,65sec,85sec,75sec respectively. . It has been observed that semi-active system performances is improved in the form of settling time for body acceleration, wheel deflection, wheel position, suspension deflection and body position by 83 %,76 %,74.5 %,71.66 and 75% respectively.

4.2 System responses under random input

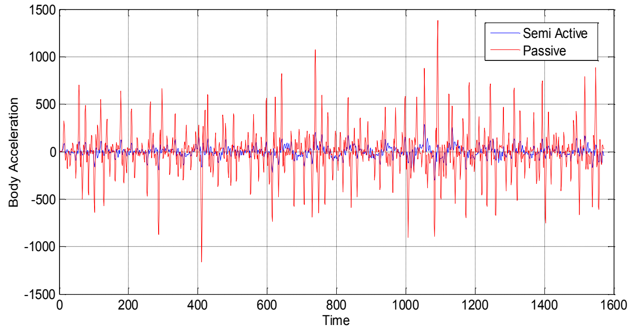

Fig 8. Time response of vehicle body acceleration

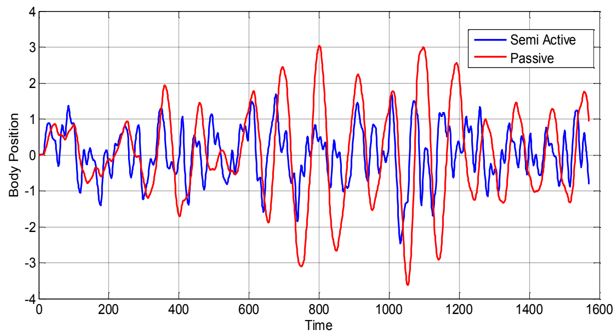

Fig 9. Time response of vehicle body position

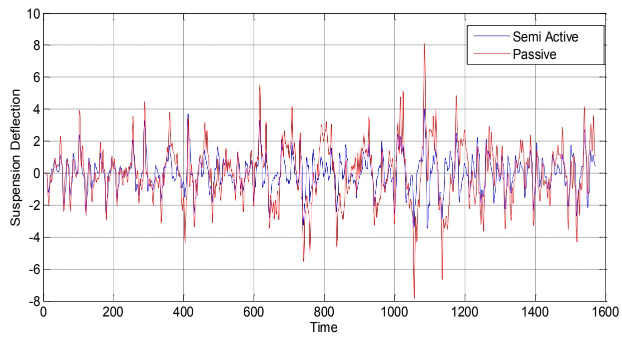

Fig 10. Time response of vehicle suspension deflection

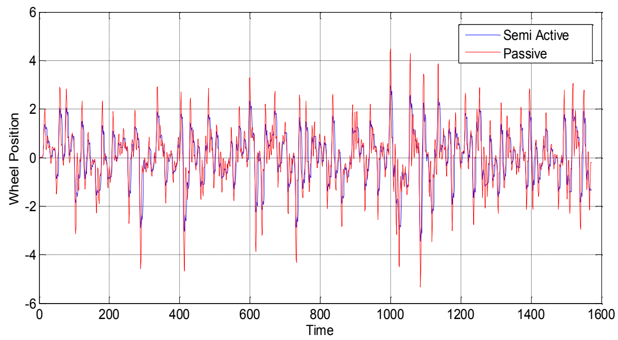

Fig 11 .Time response of vehicle wheel position

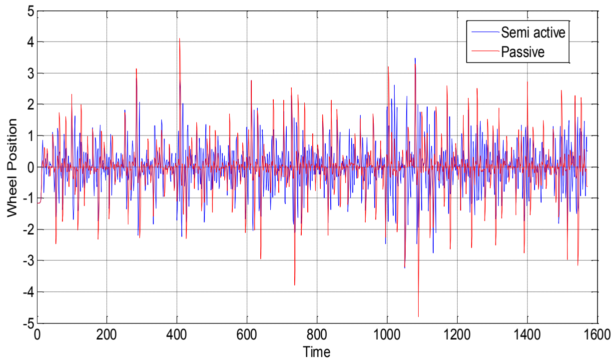

Fig 12. Time response of vehicle wheel deflection

From fig 8, peak overshoot of vehicle body acceleration for Passive system is 1400 m/s^2 whereas for semi-active suspension system it is 100 m/s^2.

From fig 9, peak overshoot of vehicle body position for Passive system is 3.7m whereas for semi-active suspension system it is 2.4 m.

From fig 10, peak overshoot of vehicle suspension Deflection for Passive system is 8m whereas for semi-active suspension system it is 3.8 m.

From fig 11, peak overshoot of vehicle body wheel position for Passive system is 5.7 m whereas for semi-active suspension system it is 2.5 m

From fig 12, maximum vehicle wheel deflection for Passive system is 0.79 m whereas for semi-active suspension system it is 0.19 m

5. Conclusions

In the research work carried out in this paper, a comparison between semi-active and passive suspension system is presented and their dynamic characteristics are also compared. It has been observed that performances is improved in reference with the performance criteria like settling time and Peak overshoot for body acceleration, wheel deflection, wheel position, suspension deflection and body position. This performance improvement in turn will increase the passenger comfort level and ensure the stability of vehicle.

It has also been demonstrated that for random excitation the body acceleration, wheel deflection, wheel position, suspension deflection and body position response of semi active suspension system is superior compared to passive system.

The use of semi active suspension system is therefore advocated in perspective of the ride comfort and vehicle handling.

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