Designing PID Controller Based Semi-active Suspension System Using MATLAB Simulink

Mohsin Jamil[1,2], Salman Zafar[3], and Syed Omer Gilani[1(✉)]

[1] School of Mechanical and Manufacturing Engineering, National University of Sciences and Technology (NUST), Islamabad, Pakistan
{mohsin,omer}@smme.nust.edu.pk

[2] Department of Electrical Engineering, Faculty of Engineering, Islamic University Madinah, Medina, Saudi Arabia

[3] College of Electrical and Mechanical Engineering, National University of Sciences and Technology (NUST), Islamabad, Pakistan
salman.zaffar@gmail.com

Abstract. The suspension system of any vehicle is responsible for not only to support the weight of the vehicle, but also to improve ride comfort and vehicle handling by damping out the roughness of the road before transferring it to the passengers. When the vehicle experiences an uneven road profile, the suspension should not generate too large oscillations, and even if it does, then these oscillations must be removed as quickly as possible. In this paper, we have investigated the functioning of a semi-active suspension system of a vehicle by modelling it as a quarter car semi-active suspension system. The model is designed as a PID controller based semi-active suspension system. MATLAB Simulink has been used in the process. The system considered in the paper is a linear system, which can apprehend basic performance parameters of a suspension system like body and suspension travel and give results in terms of rise time, settling time and over-shoot. The performance of the system is taken as better ride quality given by body travel. The lesser body displacement (over-shoot) in earlier time (settling time) are used to depict these performance standards. These performance indicators are also compared with a passive suspension system of similar specifications. The results achieved through the simulation show that the semi - active suspension system, using the designed PID controller to adjust its damping parameters, demonstrates much better performance than the passive system, having fixed damping. The designed controller can be used to design more comfortable and stable suspension systems.

Keywords: Semi-active suspension system · PID controller
Quarter car model · MATLAB Simulink

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1 Introduction

The objectives of the vehicle suspension system is to ensure the comfort of the passengers, maximize the road grip of the tyres and to provide maximum stability for the steering. An efficient suspension has to maintain balance among all these factors. Vehicle suspensions can be categorized in three types, passive systems, semi-active systems and active suspensions systems (Fig. 1). The conventional suspension system [1], utilizing un-controlled shock-absorbing dampers and springs with constant parameters are the passive suspension [2,3]. They have fixed specifications which are designed for a specific range of operating conditions and hence cannot re-adjust these parameters with the variations in the conditions [4]. This causes the passive suspension system to generate fixed response for all types of road profiles. Passive system possesses an element to absorb energy; a damper, and an element to store energy; a spring. Since there is no source for any additional energy in the system, therefore the system is called a "passive suspension system". This system is subject to multiple trade-off when it experiences a large bandwidth of oscillating frequencies. To get the most optimized response for all these frequencies, passive systems are designed for such an operating condition, which the vehicle is experiencing the most, utilizing a constant stiffness spring and a fixed damping coefficient damper. This spring and damper cannot adjust their coefficients to suit the variations in the operating conditions [5]. An adjustable system is ideally required to cater for the variety of road disturbances that a vehicle can encounter while on road, which should have varying response for different conditions. A semi active suspension system does exactly this. It offers a remarkable upgrade in the suspension of the vehicle by utilizing fluid dampers, which can change their damping coefficient per the variations in the road disturbances. This system has the capacity to adapt either its damping coefficient, and/or the stiffness of the spring to cater for the continuously changing profiles of the road [6]. This auto-controlled adjusted system is especially beneficial for the suspensions due to its low energy requirements. Normally, in semi active systems, the spring already in use of the passive suspension is kept, while an added system is introduced to modulate the damping force of the damper to achieve a range of damping force for multiple operating conditions. "Electro Rheological (ER)" and "Magnetic Rheological fluid dampers" [7] are favoured for the fact that they can change their damping stiffness coefficient [5].

In an active suspension system, the conventional (passive) components are supplemented with the help of actuators which can provide extra force to pull or push the sprung mass of the vehicle to achieve the required level of comfort [8]. An active shock absorber may be used as an active control which can generate force instantaneously, to support the body weight and to provide stability and comfort in varying road disturbances [9]. The biggest drawback of this type of system is the increase in the cost caused by the added apparatus (external source) to provide the required actuation energy to the system [10]. Although there is a variety of options for actuators, like electromagnetic and hydraulic, the electromagnetic actuators are generally favoured due to their speed of actuation and rapid response. In short, the active suspension system can give better performance over a wide-ranging road disturbances. However, this active system is handicapped by being more complex, heavy in weight and requiring high external energy [5].

In current paper the concept of a semi active suspension system is studied due to its advantage of providing better driving experience and safety without adding any additional burden on the power requirements or overall vehicle weight [11]. A PID controller is designed for controlling the parameters of a semi active suspension system to demonstrate its advantages over the conventional passive approach. The paper tries to analyse the proposed system in the application of a quarter car model. The response of the proposed system and a passive system, caused by multiple road disturbances is simulated with fixed system parameters and their performances are compared to establish the better solution.

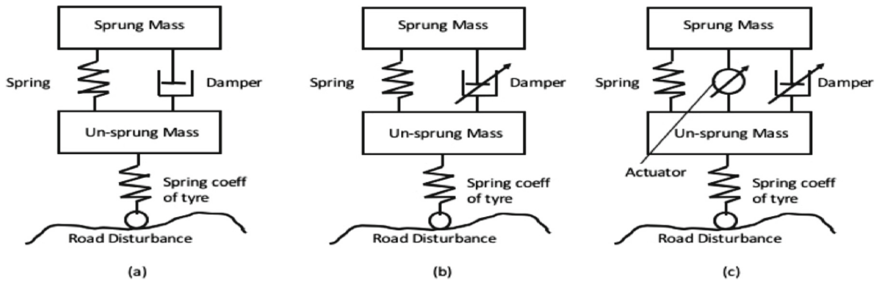


Fig. 1. (a) Passive system, (b) Semi-active system (c) Active suspension system [5]

2 Problem Statement

The semi-active suspension system has been considered for the paper due to its clear edge over the conventional passive suspension. The semi-active system demonstrates added adjustment in the stiffness of the damping force against multiple kinds of road disturbances, which enhances the drive comfort and increases the stability of the vehicle. The semi-active system is at an advantageous position as compared to active suspension system as well, which although possess much superior adjustment capabilities against the road disturbances, but also adds extra weight on the vehicle and at the same time requires extra external power to activate the actuators. This is where the semi-active suspension system gets its edge over active system. It is for the same reason such system is advised to be implemented on normal commercial cars. In the paper the suggested model is tested for a normal passenger car. A PID controller has been designed to control the damping force or stiffness of the adjustable damper of a semi-active suspension. For the ease of the simplicity, a quarter car model is taken to model and simulate the system. The quarter car model considers the working of a single tire and associated suspension components of a normal road car. The effects of various forces acting on one tire are considered and studied. The same results can be expected to be effecting all the tyres separately, considering the uniformity of the car.

3 Mathematical Modelling

The mathematical model for a quarter car passive and semi-active suspension system Fig. 2 has been derived by using the basic Newton's laws of motion and the free body diagram approach. The modelling has been done considering certain assumptions to keep the model simple yet effective [12].

- Two degree of freedom system has been considered for the suspension system modelled here. Moreover, the overall vehicle design is assumed to be a linear or uniform to support the quarter car model.
- For the ease of the design, certain minor factors, like backlash and movement in various gear systems, linkages and joints and the vehicle chassis flex have been disregarded to reduce the complexity. As the effect of these forces is negligible, therefore, these have been neglected in the model.
- The tyre is considered to act as having both damping and spring properties.

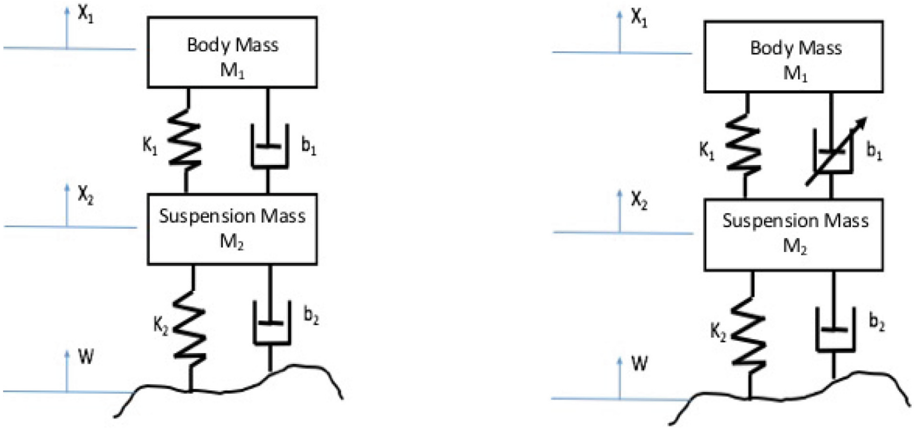


Fig. 2. (a) Quarter car passive suspension system model, (b) Semi-active suspension model

3.1 Passive Suspension System Model

The equation for quarter car passive suspension system are

$$M_1\ddot{x}_1 + b_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = 0 \quad (1)$$

$$M_2\ddot{x}_2 + b_1(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) + b_2\dot{x}_2 + k_2x_2 = b_2\dot{w} + k_2w \quad (2)$$

Taking Laplace transform of Eqs. (1) and (2), we get

$$M_1S^2X_1(s) + b_1S(X_1(s) - X_2) + K_1(X_1(s) - X_2(s)) = 0 \quad (3)$$

$$\begin{aligned} M_2S^2X_2(s) + b_1S(X_2(s) - X_1(s)) + k_2(X_2(s) - X_1(S)) \\ + b_2SX_2(s) + k_2X_2(s) = b_2SW(s) + k_2W(s) \end{aligned} \quad (4)$$

by separating variable in (3)

$$X_1(s)\{M_1S^2 + b_1S + k_1\} = X_2(s)\{b_1S + k_1\} \quad (5)$$

Substituting in (4) to get transfer function

$$G(s) = \frac{X_1(s)}{W(s)} = \frac{b_1b_2S^2 + (b_1k_2 + b_2k_1)S + k_1k_2}{m_1m_2s^4 + \{m_1(b_1 + b_2) + b_1m_2\}S^3 + \{m_1(k_1 + k_2) + b_1b_2 + k_1m_2\}S^2} + \frac{1}{(b_1k_2 + b_2k_1)S + k_2^2} \quad (6)$$

3.2 Semi-active Suspension System Model

The semi-active suspension utilizes a damper with variable coefficient instead of a linear one. The equations for quarter car semi-active suspension system derived from Newton's laws of motion are:

$$M_1\ddot{x}_1 + \bar{b_1}(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = 0 \quad (7)$$

$$M_2\ddot{x}_2 + \bar{b_1}(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) + b_2\dot{x}_2 + k_2x_2 = b_2\dot{w} + k_2w \quad (8)$$

By taking Laplace transform of Eqs. (7) and (8), we get,

$$M_1S^2X_1(s) + \bar{b_1}S(X_1(s) - X_2) + K_1(X_1(s) - X_2(s)) = 0 \quad (9)$$

$$\begin{aligned} M_2S^2X_2(s) + \bar{b_1}S(X_2(s) - X_1(s)) + k_2(X_2(s) - X_1(S)) \\ + b_2SX_2(s) + k_2X_2(s) = b_2SW(s) + k_2W(s) \end{aligned} \quad (10)$$

Separating the variables in (9)

$$X_1(s)\{M_1S^2 + \bar{b_1}S + k_1\} = X_2(s)\{\bar{b_1}S + k_1\} \quad (11)$$

Substituting in (10) to get transfer function

$$G(s) = \frac{X_1(s)}{W(s)} = \frac{\bar{b_1}b_2S^2 + (\bar{b_1}k_2 + b_2k_1)S + k_1k_2}{m_1m_2s^4 + \{m_1(\bar{b_1} + b_2) + \bar{b_1}m_2\}S^3 + \{m_1(k_1 + k_2) + \bar{b_1}b_2 + k_1m_2\}S^2} + \frac{1}{(\bar{b_1}k_2 + b_2k_1)S + k_2^2} \quad (12)$$

where :

$$\begin{aligned}
M_1 &= \text{Bodymass/Sprung mass (kg)} \\
M_2 &= \text{Suspension mass/Un} - \text{Spring mass (kg)} \\
x_1 &= \text{Body mass displacement (m)} \\
x_2 &= \text{Suspension mass displacement (m)} \\
b_1 &= \text{Suspension damping coefficient (N.s/m)} \\
b_2 &= \text{Tyre damping coefficient (N.s/m)} \\
k_1 &= \text{Suspension spring coefficient (N/m)} \\
k_2 &= \text{Tyre spring coefficient (N/m)} \\
w &= \text{Road profile} \\
\bar{b_1} &= \text{Variable damper stiffness coefficient for semi} - \text{active system (N.s/m)}
\end{aligned}$$

The semi-active suspension system utilizes a varying damping stiffness coefficient damper, and is operated by an external power source and an embedded controller with a set of sensors. The level of damping required as per the road profile is selected by the controller, which then adjusts the damper to achieve the optimized damping. A Proportional Integral Derivative (PID) controller was designed and tested for various types of road disturbances and the results were compared with a similar passive suspension system. The values of proportionality constants i.e. kp, ki and kd, for the designed PID controller, are determined using the trial and error method.

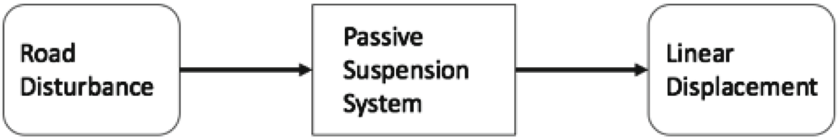


Fig. 3. Block diagram of passive suspension system

4 Simulation Model

The derived mathematical model, given in Eqs. (1), (2), (7) and (8), were simulated using MATLAB Simulink software. Separate models were developed for both, the passive suspension system (Fig. 3) and the semi-active suspension system (Fig. 4). The block diagram of the passive suspension system shows a road disturbance, the suspension system and the vehicle body mass displacement (output).

The semi-active suspension system has a feedback mechanism to control the damping coefficient of the damper. This feedback is fed to a PID controller which generates its response and as a result adjusts the damping stiffness of the damper

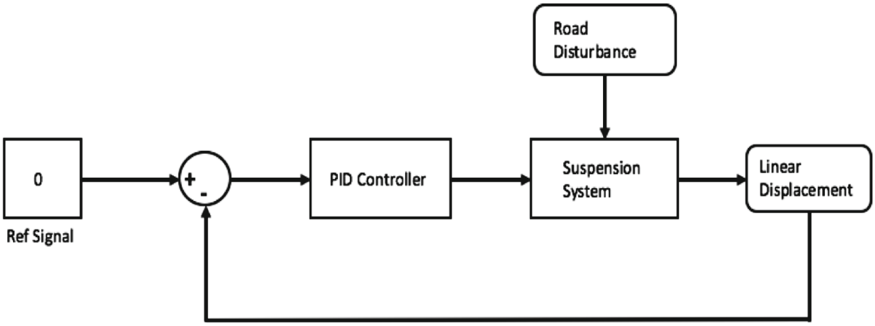


Fig. 4. Block diagram of semi-active suspension system

used. All other components of the system remain the same. Figure 4 shows the block diagram of a semi-active suspension system.

Basing on the block diagrams of the passive and the semi-active suspension system and the Eqs. (1), (2), (7) and (8), the Simulink models designed and used for the simulation are shown in Figs. 5 and 6.

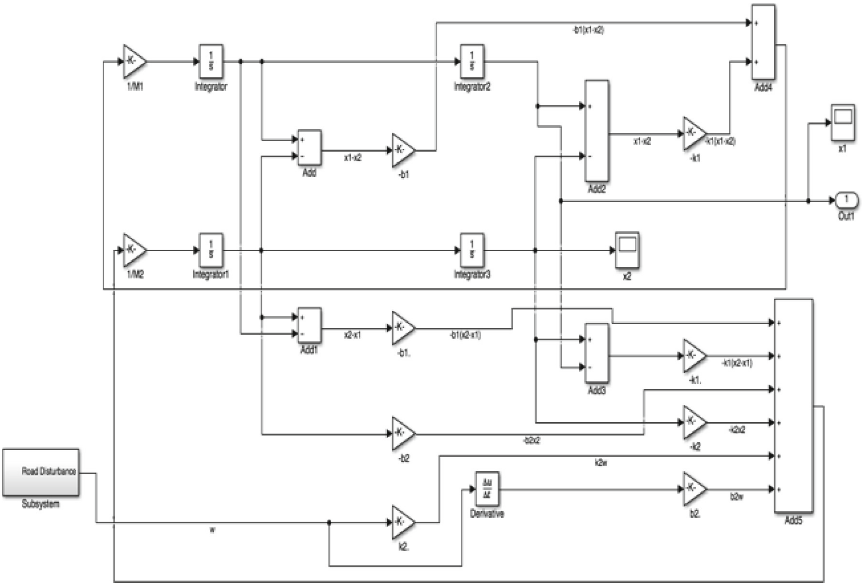


Fig. 5. Simulink model for passive suspension system

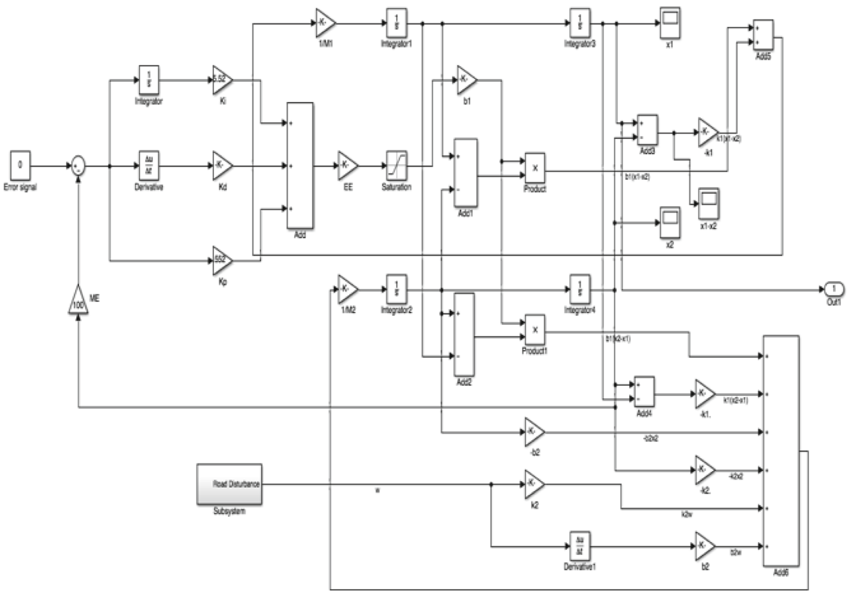


Fig. 6. Simulink model of semi-active suspension system having a PID controller

5 System Parameters and Conditions

Following parameters and conditions were set for dynamical modelling.

5.1 Parameters

Parameters of a normal passenger car considered for the simulated analysis are given in Table 1 [13].

Table 1. System parameters

Parameters	Value
M_1 (Sprung mass)	350 kg
M_2 (Un – Sprung mass)	40 kg
K_1 (Suspension spring coefficient)	18000 N/m
K_2 (Tyre spring coefficient)	1950000 N/m
b_1 (Suspension damper coefficient)	600 N.s/m
b_2 (Tyre damping coefficient)	800 N.s/m

The tuning of the PID controller was carried out manually to reduce overshoot and minimize settling time. The optimal gain values for various road profiles achieved were;

Profile-1: $K_p = 10$, $K_i = 70$, $K_d = 0.2$
Profile-2: $K_p = 3$, $K_i = 10$, $K_d = 0.2$
Profile-3: $K_p = 3$, $K_i = 23$, $K_d = 0.1$

5.2 Road Disturbances

Three kinds of road disturbances were considered for the purpose to cater for maximum types of real life conditions. The road profile-1 is taken as a sinusoidal bump of 10 cm (0.1 m) spread over a 5 s interval. This profile (Fig. 7) is given by following expression [14,20].

$$w = 1 \begin{cases} a[u(t-5)-(t-10)]sin(0.2\pi t) & 5\,\mathrm{s} \leq t \leq 10\,\mathrm{s} \\ 0 & \text{otherwise} \end{cases} \tag{13}$$

where: a = 0.1 m (bump height).

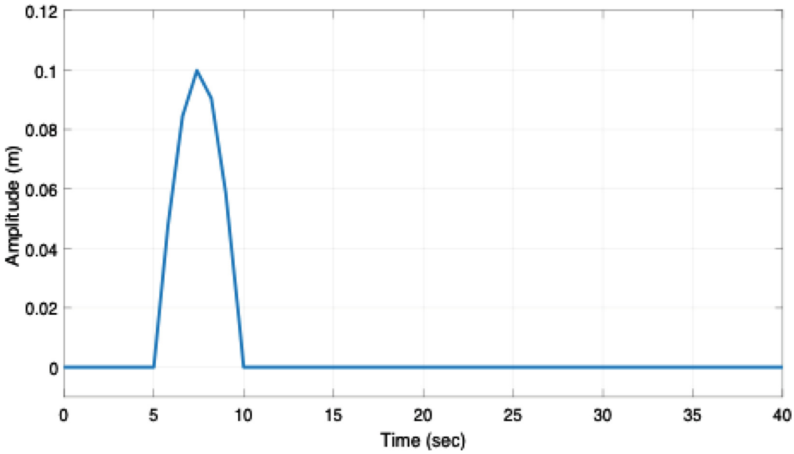


Fig. 7. Road disturbance profile-1

The profile-2 is taken as a single step of 10 cm. It is considered as a sudden step on the road surface [15]. The response of the suspension on this sudden step was also evaluated. This profile (Fig. 8) is expressed by the following expression

$$w = \begin{cases} 0 & 5\,\mathrm{s} \leq t \leq 10\,\mathrm{s} \\ 0.1 & t \geq 5\,\mathrm{s} \end{cases} \tag{14}$$

The road disturbance profile-3 is taken as a series of two consecutive jerks within a time period of 3 s. The bumps are sharp and depict the disturbance

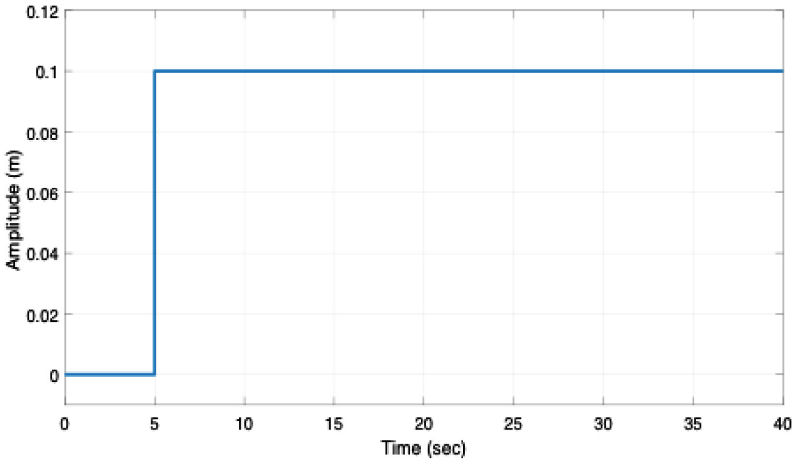


Fig. 8. Road disturbance profile-2

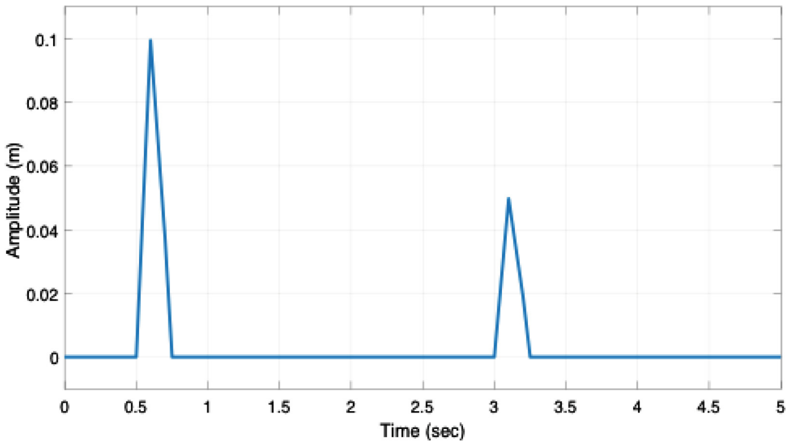


Fig. 9. Road disturbance profile-3

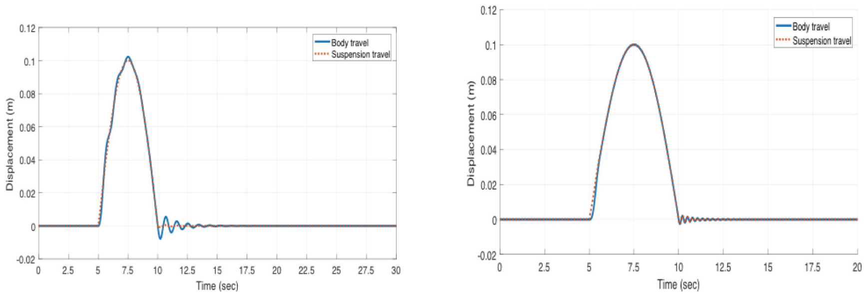
experienced by tyres at relatively faster speeds. The profile (Fig. 9) details are taken from [16,19] and the expression is given by:

$$w = a\{\frac{1 - cos(8\pi t)}{2}\} \tag{15}$$

$$a = \begin{cases} 0.11\,\text{m} & 0.5\,\text{s} \leq t \leq 0.75\,\text{s} \\ 0.55\,\text{m} & 3.0\,\text{s} \leq t \leq 3.25\,\text{s} \end{cases} \tag{16}$$

6 Results and Discussion

The simulation analysis, based on the Simulink models for quarter car passive and semi active suspension system, was carried out in MATLAB. The suspension travel and the displacement of the body mass (body travel) of the car in terms of linear displacement was taken as the performance parameter. The road disturbance was taken as input for both the systems. The performance criteria were taken in terms of rise time (Tr), settling time (Ts), percentage over-shoot (%OS) and the steady state error (eSS) [15]. The system which demonstrates smaller amplitude of displacement and lesser settling time for the body travel is considered better for the drive comfort and vehicle stability [16,17]. The settling time was taken as time taken to reach steady state with an error of 2% of the referenced amplitude/displacement [18]. The response of passive and semi-active suspension system for road disturbance profile-1 is as depicted in Fig. 10.

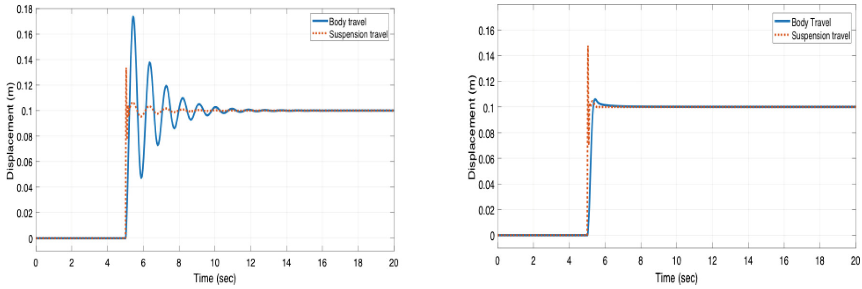


(a) Passive suspension displacement (b) Semi-active suspension displacement

Fig. 10. Displacements with road profile-1

Profile-1 represents a smooth and relatively long duration (5 s) bump. It is observed from Fig. 10a that even the passive suspension (body travel) displays smooth response and experiences a percentage over-shoot of 2.3% and has a settling time of 7.05 s. The body travel for the designed PID controller based semi active suspension system (Fig. 10b), however, shows 0.1% over-shoot and a settling time of 5.22 s.

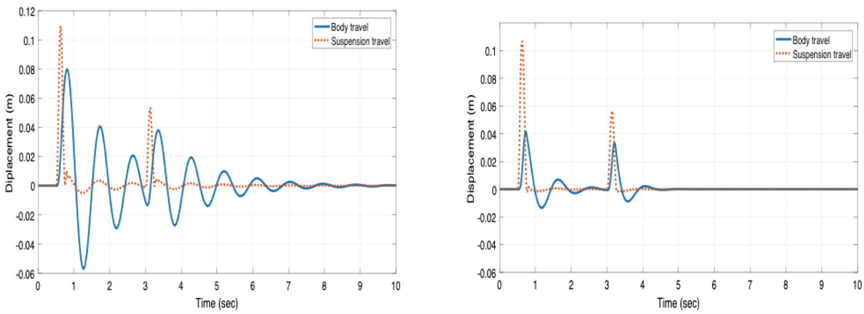
Figure 11 shows the response of passive and semi-active suspension system, for the road disturbance profile–2. Here the sharp step of 0.1 m in this disturbance, indicates the effectiveness of PID controller based semi-active system. Figure 11a shows the percentage over-shoot and settling time of body travel of passive system which is equal to 73.8% and 5.49 s. On the contrary, the body travel of semi active system (Fig. 11b) has over-shoot of just 6.1% and a settling time of 0.82 s.



(a) Passive suspension displacement (b) Semi-active suspension displacement

Fig. 11. Displacements with road profile-2

The comparison of passive and semi-active system for road profile-3 is given in Fig. 12 respectively. The percentage over-shoot of body travel of passive system (Fig. 12a) is 27.45% with settling time of 6.99 s. The body travel of semi-active system (Fig. 12b) shows much improved response with over-shoot of 11.98% and settling time of 3.07 s.



(a) Passive suspension displacement (b) Semi-active suspension displacement

Fig. 12. Displacements with road profile-3

The complete comparison of both passive and semi active suspension simulated models is carried out in Table 2. The analysis clearly demonstrates the efficiency of PID controller based semi active suspension system to be much higher in terms of the performance parameters considered in this paper.

Table 2. Comparison of body travel for passive and semi-active suspension system

Road profile	System	Tr (sec)	Ts (sec)	%OS	eSS (m)
Profile-1	Passive	2.49	7.05	2.3	0
	Semi-active	2.50	5.22	0.1	0
%Improvement/Reduction		−0.4%	25.96%	%95.65	0
Profile-2	Passive	0.42	5.49	73.8	0
	Semi-active	0.44	0.82	6.1	0
%Improvement/Reduction		−4.76%	85.06%	%91.73	0
Profile-3	Passive	0.31	6.99	27.45	0
	Semi-active	0.13	3.07	11.98	0
%Improvement/Reduction		58.06%	56.08%	%56.36	0

7 Conclusion

Passive and Semi-active suspension systems have been reviewed in this paper and a comprehensive simulated comparison has been done among them. A PID controller for a controller based semi active suspension system was designed and tested in parallel with a conventional passive suspension system. Their analysis was compared using common performance parameters like rise time, settling time, percentage over-shoot and steady state error. The results reveal that the PID controller designed here, for the semi active system is highly efficient and shows a lot of improvement in terms of body/suspension oscillations and settling time. Hence we conclude that by adding an active damping element, we can greatly enhance the ride comfort and vehicle stability in all kinds of poor road conditions. For any future study on the subject, emphasis must be given to designing a self-tuning PID controller or exploring other controlling methods. Secondly the results achieved in this paper with the help of simulation may be verified using experiments.

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