

Vibration Control of Quarter Car Model Using Modified PID Controller

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Abstract

The purpose of this research is to control a quarter car suspension system and also to reduce the fluctuated movement caused by passing the vehicle over road bump using modified PID (Proportional Integral and Derivative) controller. The proposed controller deals with dual loop feedback signals instead of single feedback signal as in the conventional PID controller. The structure of the modified PID controller was created by moving the proportional and derivative actions in the feedback path while remaining the integral action in the forward path. Thus, high accuracy results were obtained. Firstly, modelling and simulation of linear passive suspension system for a quarter car system was performed using Matlab – Simulink software. Then the linear suspension system was activated and simulated by using an active hydraulic actuator to generate the necessary force which can be regulated and controlled by the proposed controller. The performance of whole system has been enhanced with a modified PID controller.

Keywords: Displacement, Modified PID controller, Quarter car, Vibration.

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1. Introduction

Vehicles can be exposed to different external sources of disturbances like bumps and potholes. These disturbances are transmitted to the passengers through a suspension mechanism which physically separates the car's body from the wheels. A good suspension system means good separation of the vehicle body from the wheel and on the same time an excellent isolation of the whole car from road disturbances. In addition, the passenger's comfort can be achieved by minimizing the transition of the vertical forces caused by road disturbances.

Therefore, controlling both the vertical displacement of vehicle and suspension system travel as well as regulating the distribution of the forces generated by exposure of the vehicle to any external loads can achieve the required performance.

Moreover, to ensure good curing and handling feature, the optimum connection between the tire and the road must be maintained on four wheels. In conventional suspension system, these features are inconsistent and do not meet all the required conditions. Poor adhesion and bad curing can result due to an excessive wheel movement which can lead to an imperfect tire position relative to the road. The suspension system can be categorized into passive, semi-active and active suspension system according to external power input to the system. Passive suspension system consists of a non-controlled spring and shock-absorbing damper. No power supply is needed to activate the suspension elements and the forces responsible for carrying both vehicle body and passengers with different external loads are generated by conventional spring and damper. The stability of the car and

wheel are achieved by limiting the suspension parameters such as acceleration, velocity and displacement depending on the typical specification and design feature of the system. In active suspension system a hydraulic actuator is added in parallel with the basic suspension elements (spring and damper) to give additional force. This force assists in support the vehicle's body and passengers on their seats and provide both ride comfort and ride handling if controlled. Hydraulic actuators have the ability to store energy and dissipate it.

They are activated by a power supply source and controlled by different manners. In general, hydraulic actuator is located between vehicle's body and wheel mass, and it connected to them perfectly, as a result the car performance is enhanced and improved [1].

Many automotive researchers have studied and developed the suspension systems and focused on different control methods extensively through both analysis and experiments.

Haris and Aboud [2] proposed control technique for active suspension systems by using weighted multiple model adaptive control scheme, the models were selected based on sprung mass values and the road profile input. An output feedback PID controllers were designed within the adaptive mixing control framework system stability.

Hassan and Dammed [3] used PID controller with semi-active suspension of a quarter car to provide the comfort of the ride and provide the long life of the vehicle. Yaseen and Abdulwahab [4] considered linear and nonlinear quarter car suspension models for the investigation of active vibration control using PID and Fuzzy logic controllers.



Rao [5] investigated the performance of a quarter car semi-active suspension system using PID controller. Ali and Hameed [6] proposed an air spring of the type of Nishimura and hydraulic damper instead of coil spring to control of linear quarter car active suspension system with the using of Fuzzy controller. Pekgökgö et al. [7] used Fuzzy logic to control the active suspension which optimized by using genetic algorithm operations. Anandan et al. [8] designed PID and LQR controllers and compared between them for different road profiles to a quarter car suspension model.

Shahid and Wei [9] investigated different damping control methods for active vibration control of a quarter-vehicle suspension system such as higher-order, first-order and integral sliding mode control in addition to PID and linear quadratic regulator controllers. Tandel et al. [10] presented modelling of quarter car double wishbone suspension prepared by using the toolbox Sim Mechanics and Simulink of MATLAB Software for physical and mathematical modelling of suspension respectively. PID controller is then implemented on both the models to minimize the vertical body acceleration. Ahmed et al. [11] constructed an active suspension control for a quarter car model subjected to excitation from a road profile using PID controller. Jamil et al. [12] proposed optimal control based intelligent controller for active suspension system to minimize the vehicle's body vertical displacement. Zeinali and Darus [13] used Magneto-rheological damper and a parallel structure of fuzzy PID controller to control the semi-active suspension system.

Akpakpavi [14] presented the application of P, PI, PID, GA-PID and Auto-tuned PID controllers to control the vibration of the 1/4 car suspension system. Other researchers focused on conventional PID, Fuzzy logic and Fuzzy – PID controllers for quarter car active suspension system as in [16], [17].

In this study, a linear passive and active quarter car suspension systems were modeled mathematically by using Newton's second law of motion and the model response to road disturbance was implemented using Matlab/Simulink software. System variables were analyzed and confirmed by applying suitable control method. Then, the system fluctuation and vibration were clearly reduced by regulating and controlling the action of the hydraulic actuator to enhance suspension system performance through the application of a modified PID controller.

2. Mathematical Modelling

2.1. Passive Suspension System

The car model investigated in this study consists of one-fourth of the body mass, suspension elements (spring and damper) and one wheel which has a stiffness and little damping as shown in Fig. 1.

It is assumed that the springs and dampers are linear and the tire is always in contact with the road surface and the effect of friction is neglected. The free body diagram for the quarter car model is shown in Fig. 2.

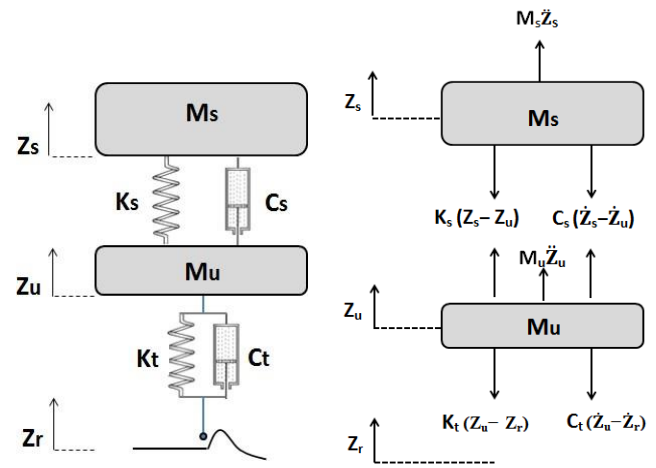


Fig. 1 Passive Quarter car model.

Fig. 2 Free body diagram.

Newton's second law of motion is used to derive the equations of motion of quarter car model. For the passive suspension system, the equation of motions related to the sprung and unsprung masses can be respectively written as bellow, while system parameters used for modelling and simulation are listed in Table 1.

$$M_s \ddot{Z}_s = -K_s(Z_s - Z_u) - C_s(\dot{Z}_s - \dot{Z}_u) \tag{1}$$

$$M_u \ddot{Z}_u = K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) - K_t(Z_u - Z_r) - C_t(\dot{Z}_u - \dot{Z}_r) \tag{2}$$

Table 1. System parameters for quarter car model [17].

Parameter	Description	Value	Unit
M_s	sprung mass	208	kg
M_u	unsprung mass	28	kg
K_s	spring stiffness of the suspension system	18709	N/m
C_s	damping constant of the suspension system	1300	N.s/m
K_t	spring stiffness of the tire	127200	N/m
C_t	damping constant of the tire	10	N.s/m
Z_s	sprung mass displacement	-----	m
Z_u	unsprung mass displacement	-----	m
Z_r	road profile	-----	m

2.2. Active Suspension System

For active suspension system, the hydraulic actuator is connected in parallel with the spring and damper as shown in Fig. 3. The force generated from actuator can be written in the form of (F_a) and can be calculated by measuring one of the system variables. Thus, the equations of motion related to the sprung and unsprung masses for the active suspension system are respectively given by [2] as follow:

$$M_s \ddot{Z}_s = -K_s(Z_s - Z_u) - C_s(\dot{Z}_s - \dot{Z}_u) + F_a \tag{3}$$

$$M_u \ddot{Z}_u = K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) - K_t(Z_u - Z_r) - C_t(\dot{Z}_u - \dot{Z}_r) - F_a \tag{4}$$

Where F_a is a hydraulic actuator force (N).

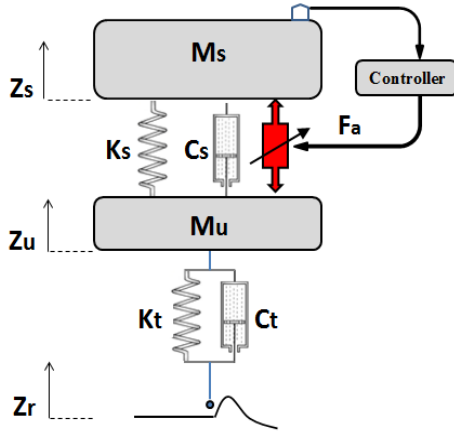


Fig. 3 Active quarter car model.

3. Control Method

3.1. Conventional PID Controller

The conventional PID controller consists of three main components which are proportional (P), integral (I), and derivative (D). The Proportional action reduces entire error and following the set point. The integral action accounts and improves the steady state error while the derivative action accounts the rate of change of error and reduces overshoot and oscillation. All these parameters are put in the forward path of the manipulated signal to do the control action as shown in Fig. 4.

The conventional PID controller is represented in Laplace transform as follow [15]:

$$G_C(s) = K_p + K_i/s + sK_d \tag{5}$$

Where:

$G_C(s)$ is the transfer function of the controller.

K_p , K_i and K_d are proportional, integral and derivative gains respectively.

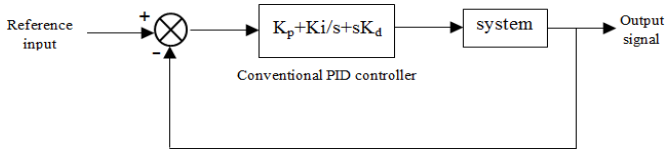


Fig. 4 Closed loop system with conventional PID [15].

3.2. Modified PID controller

The modified PID controller represents dual loop feedback signal with the structure of integral plus proportional and derivative (I – PD) as shown in Fig. 5. The integral control part has remained on forward path. There for the error signal is only affected this part. However, proportional and derivative actions have been moved on feedback path to affect the output signal. Thus, the noise resulted from error signal is significantly reduced. The modified PID controller is represented in Laplace transform as follow [15, 16]:

$$G_C(s) = K_i/s - K_p - sK_d \tag{6}$$

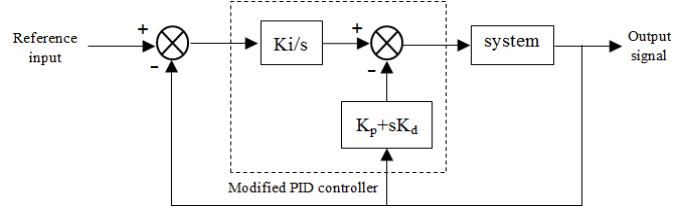


Fig. 5 Closed loop system with modified PID controller [16]

4. Simulation and Results

The quarter car system is analyzed with uncontrolled and modified PID controlled structures. According to the mathematical modelling represented by both equations (1) and (2) the Simulink block diagram is created by Simulink library as shown in Fig. 6. Firstly, the passive quarter car system is simulated to show system dynamic response like car body displacement, velocity, acceleration and suspension deflection. The control methodology is designed then using modified PID controller.

The error signal is fed to adjust the input in order to reach the output to the reference or set point. For controlling purposes, car body displacement is measured as a feedback signal to generate the desired hydraulic actuator force, the road bump is represented as a step function of 0.08 m and considered as an external disturbance source. The system response represented by the well-known properties such as rise time, overshoot, settling time and steady state error reflects the overall behavior to a step input reference signal.

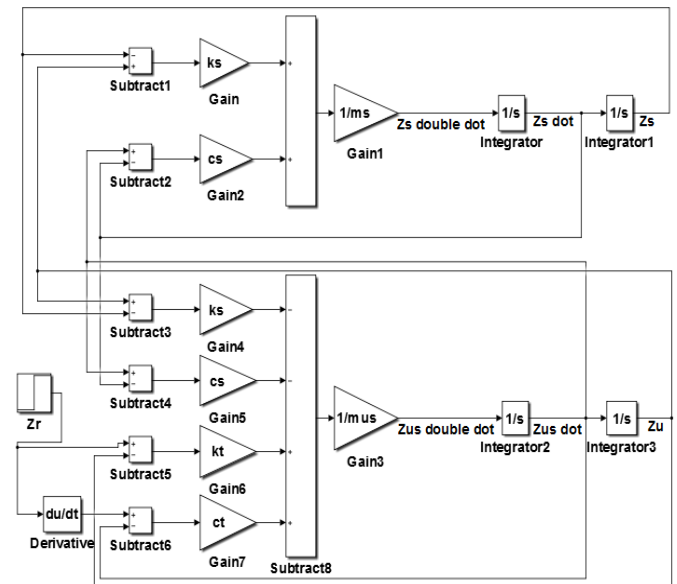


Fig. 6 Simulink diagram of passive quarter car.

The controller gains K_p , K_i and K_d is tuned manually by trial-and-error method in order to reach the desired values taking into a count closed loop response according to the Table 2.

Table 2. Response of proportional, integral and derivative controller gains [14]

Closed loop response	Rise time	Overshoot	Settling time	Steady state error
K_p	Decrease	Increase	No change	Decrease
K_i	Decrease	Increase	Increase	Eliminate
K_d	No change	Decrease	Decrease	No change

As mentioned earlier the structure of modified PID controller is created by moving the position of proportional and derivative gains on feedback path to affect only the output signal, while remaining the integral gain in the forward path. By using the equations 3, 4 and 6 the entire active system with a modified PID controller can be implemented in the Simulink environment as shown in Fig. 7.

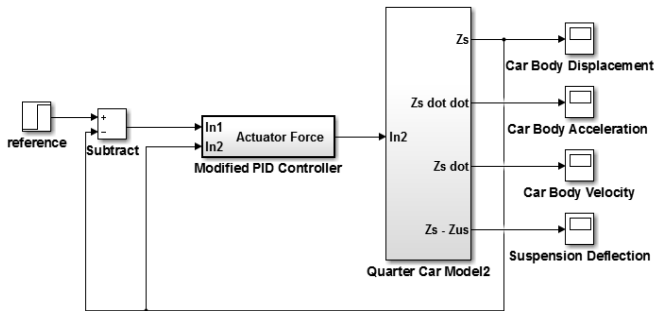


Fig. 7 Active quarter car model with modified PID controller.

The structure of the modified PID controller can be re-written in the form of I – PD controller to become a dual loop controller that is responsible for generating the desired actuator force as shown in Fig. 8. The PID controller gains are calculated as shown in Table 3.

Table 3. Parameters used in system simulation.

K_p	K_i	K_d
0.4	20	2250

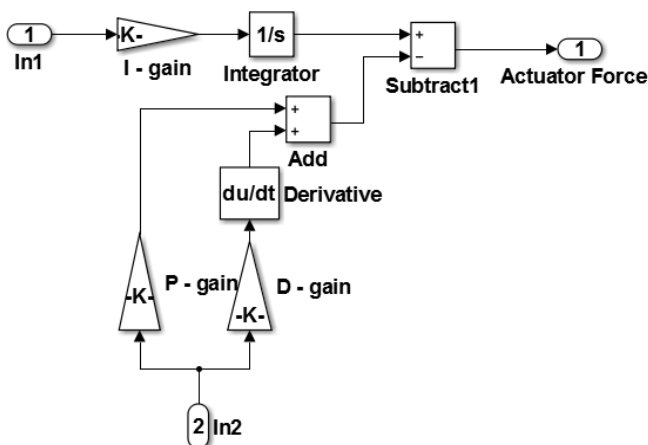
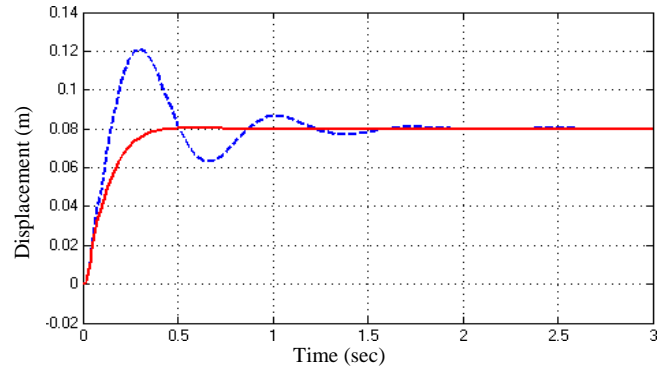


Fig. 8 Schematic of modified PID.

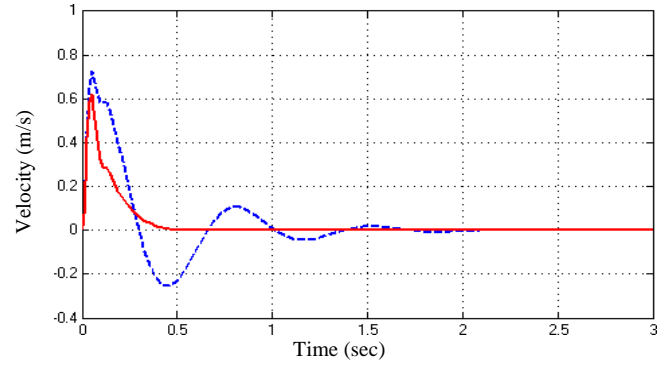
As a result, the dynamic response for both passive and active systems is shown in Fig. 9.

It can be deduced from the previous figures that the system was not stable due to vibrations resulted from the fluctuations in response parameters. Also, settling time was high. These fluctuations and settling time have been significantly reduced when using modified PID controller.

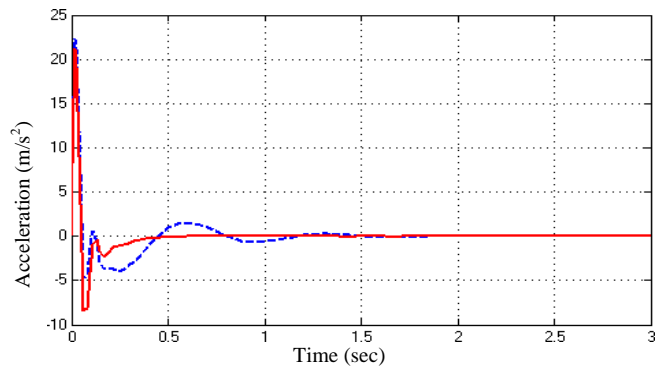
This improvement in system performance reflects both vehicle handling and ride comfort. Table 4 shows the improvement in system parameters, the reduction ratios in the overshoot have been calculated as in [11].



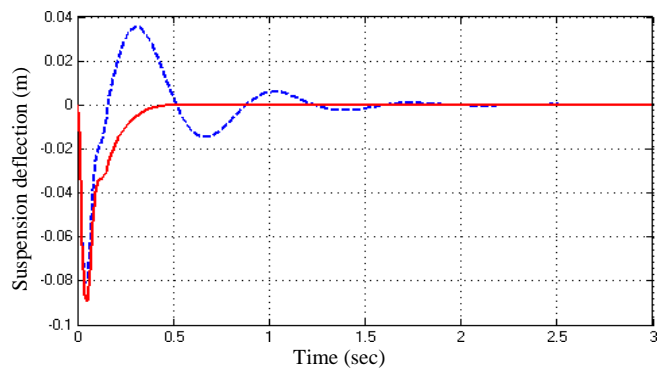
(a)



(b)



(c)



(d)

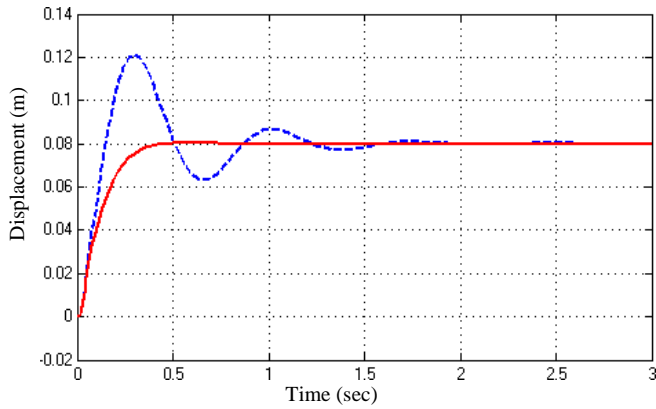
Fig. 9 System response with modified PID controller, (a) car body displacement, (b) car body velocity, (c) car body acceleration, (d) suspension deflection.

Table 4. Reduction in overshoot values for step road input.

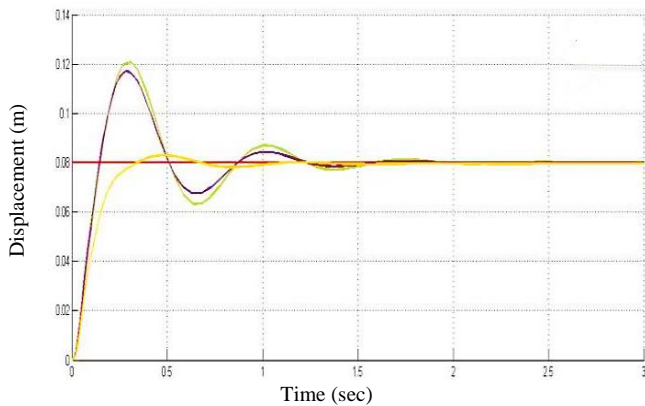
Parameters amplitudes (peak to peak)	Passive	Active	Reduction $\left(\frac{\text{Passive} - \text{Active}}{\text{Passive}}\right) \times 100\%$
Car body displacement(m)	0.12	0.08	33.3 %
Suspension Deflection (m)	0.1175	0.0885	24.7 %

5. Comparison with other studies

The results obtained from current study have been compared with the results in reference [17]. It is clear that the output oscillations in car body displacement in the current work has been obviously reduced when using modified PID controller especially when compared with the using of conventional PID controller, in addition the output oscillation from the proposed controller gave similar values when using Fuzzy-PID controller as shown in Fig. 10 and Table 5.



(a) Current study.



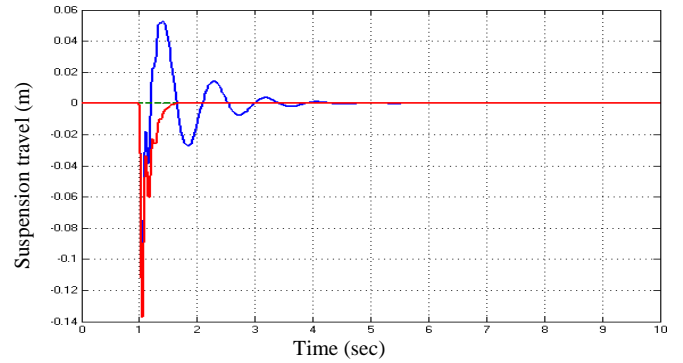
(b) Reference [17].

Fig. 10 Comparison of car body displacement with reference [17].

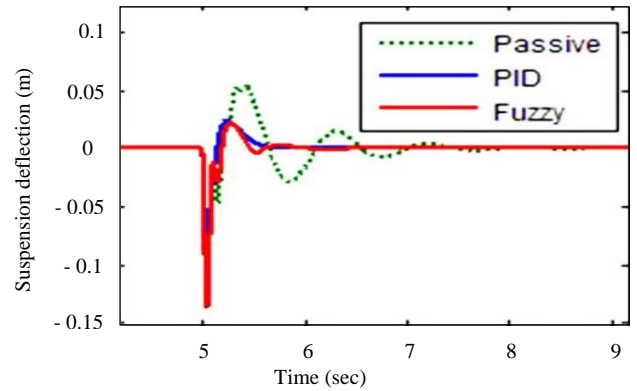
Table 5. Results comparison between current study and reference [17].

Parameters amplitudes (peak)	Passive	Current study	Reference [17]	
			Conventional PID	Fuzzy-PID
Car body displacement (m)	0.04	0.00	0.038	0.005
Settling time (sec)	1.6	0.5	1.5	0.75

Also, the results obtained from current study have been compared with the results in reference [18]. It is clear that the output oscillations in suspension travel in the current work has been obviously reduced when using modified PID controller compared with the using of conventional PID and Fuzzy controllers as shown in Fig. 11 and Table 6.



(a) Current study.



(b) Reference [18].

Fig. 11 Comparison of suspension travel with reference [18].

Table 6 Results comparison between current study and reference [18].

Parameters amplitudes (peak)	Passive	Current study (Modified PID)	Reference [18]	
			Conventional PID	Fuzzy
Suspension travel (m)	0.055	0.00	0.025	0.025
Settling time (sec)	3	0.5	0.5	0.5

6. Conclusions

In this study, in order to reduce the vibrations caused by road bumps, a modified PID controller was used to control the suspension system of a quarter car model. A comparison between passive and active quarter car models was done to show the dynamic characteristics for both systems. It is clear that the results of the current study demonstrate very good traceability. System dynamic response such as car body displacement has been improved in this work which reflects the stability and comfort of the passengers.

Other system dynamic responses such as acceleration, velocity and suspension travel have been improved when using modified PID controller.

According to that, the modified PID controller is better than the conventional PID controller. In addition, the simplicity of computing modified PID controller gains makes it easier to use for controlling purposes to prevent the long time needed to compute and tune both Fuzzy or Fuzzy-PID controllers.

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