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Improving the Dynamic Response of Half-Car Model using Modified PID Controller

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Abstract
This paper focuses on the vibration suppression of a half-car model by using a modified PID controller. Mostly, car vibrations could result from some road disturbances, such as bumps or potholes transmitted to a car body. The proposed controller consists of three main components as in the case of the conventional PID controller which are (Proportional, Integral, and Derivative) but the difference is in the positions of these components in the control loop system. Initially, a linear half-car suspension system is modeled in two forms passive and active, the activation process occurred using a controlled hydraulic actuator. Thereafter, the two systems have been simulated using MATLAB/Simulink software in order to demonstrate the dynamic response. A comparison between conventional and modified PID controllers has been carried out. The resulting dynamic response of the half-car model obtained from the simulation process was improved when using a modified PID controller compared with the conventional PID controller. Moreover, the efficiency and performance of the half-car model suspension have been significantly enhanced by using the proposed controller. Thus, achieving high vehicle stability and ride comfort.

Keywords
Displacement, Dynamic Response, Half-Car Model, Modified PID Controller, Oscillation.

I. Introduction

Vehicle can be subjected to a shock as a result of the tire's interference with irregularities of the road surface like crossing over road bumps. These shocks can be transmitted to the vehicle body in the form of vibrations. The suspension system i.e., spring and damper, which is located between the car body and the wheel attempt to absorb encountered shocks. This process results in damping car body vibrations. Springs have the property of continuing to oscillate for period of time while the dampers dissipate the received energy of oscillations. Other basic tasks of the suspension system fall under its capability to isolate the car body from road disturbances in order to provide ride comfort, good handling and support the vehicle static weight [1]. Suspension system's absorbing and damping capabilities sometimes may be insufficient and may take a while to halt the oscillations. This may require adding other means to significantly dampen the vibrations and oscillations. A hydraulic actuator is one of the most suitable means that can be used in parallel with the suspension system to generate the needed force to damp the road disturbances. These actuators can be activated and controlled by using different methods in order to achieve high vehicle stability while driving. Many researchers have attempted to improve the dynamic response of the vehicles by controlling the developed force from the hydraulic actuator in order to achieve the principle of vehicle stability. A list of researches contributes in this field of study. P. S. Kumar et al. [2] proposed a hybrid intelligent control technique based on combination of neural network and fuzzy for hydraulic actuated active half car suspension system. Faried Hasbullah and Waleed F. Faris [3] proposed an active disturbance rejection control with input decoupling transformation for a half car model. Yan-Jun Liang et al. [4]

designed an optimal vibration controller for vehicle active suspension systems. A. Pati et al. [5] designed a controller for a half car suspension system based on sliding mode control using proportional-integral-derivative (PID) sliding surface. Puneet Gandhi et al. [6] used a half car active suspension model with 4 degrees of freedom with different controllers such as proportional, integral and derivative, linear quadratic regulator, fuzzy and adaptive neuro fuzzy inference system. Sangzhi Zhu et al. [7] developed a new hydraulically interconnected suspension with the using of fuzzy, PID and optimal linear quadratic regulator controllers to control vehicle body's roll motion. Daniel Rodriguez - Guevara et al. [8] proposed a novel linear parameter varying (LPV) state-space (SS) model with a fictional input to represent nonlinear half-car active suspension system. H. Khodadadi and H. Ghadiri [9] used PID, fuzzy logic and H controllers to control the car suspension system based on half car. Also, a self-tuning PID controller based on fuzzy logic is developed to improve the performance of the system. J. E. Ekoru and J. O. Pedro [10] used an inner PID hydraulic actuator force control loop, in combination with an outer PID suspension deflection control loop, to control a nonlinear half-car. Yanghai Nan et al. [11] proposed a fuzzy logic control strategy for active half car suspension system which is utilized to generate counterforce to isolate vibration from the rough ground. Ahmet Yildiz [12] considered a non-linear suspension design for half vehicle model by using particle swarm optimization technique for optimizing the vehicle vibrations. Y. Susatio et al. [13] utilized direct synthesis method for tuning PID controller gains to reduce vibration on passenger seat caused by change in road surface profile or disturbance. L. C. Felix-Herran et al. [14] designed and applied a fuzzy-H control, improved with weighting functions to a novel model of a one-half semi active suspension. G. I. Mustafa et al. [15] presented an optimized sliding mode controller for vibration control of active half-car suspension systems. Muhammad A. Khan et al. [16] used feedback linearization and linear quadratic regulator controller with a half-car model. Jimoh O. Pedro and Nyiko Baloyi [17] designed a direct adaptive neural network controller to control a nonlinear half car suspension system and improve ride comfort. Mohammed H. Abushaban et al. [18] proposed a new fuzzy control strategy for a half-car active suspension system. M. Avesh and R. Srivastava [19] proposed using PID controller with an active half car suspension system to improve ride comfort to passengers and improve the stability of vehicle. L.V. Gopala Rao and S. Narayanan [20] studied the performance of half car model with optimal sky-hook damper suspension and compared it with the performance of half car model with LQR control. Wenkui Lan and Erdong Ni [21] applied fuzzy-PID controller to a half car suspension system to enhance ride comfort by reducing the body acceleration and pitch angle. Ayman H. Mohamed et al. [22] designed and studied linear quadratic regulator optimal control and PID classic control to achieve half car performance such as ride comfort and road stability. Jian Wu and Zhiyuan Liu [23] presented a novel controller design for half-cars suspension magneto-rheological by introducing a piecewise control approximation model.

In this study, a linear half-car model in MATLAB/ Simulink software is being used to compare the performance of a modified PID controller versus conventional PID controller. The model dynamic characteristics, stability of the vehicle, the quality of shock absorption and the reduction of oscillations and vibrations have been investigated.

II. Mathematical modeling

The half-car model used in this study is a four degree of freedom system. The model consists of a vehicle body with front and rear suspension elements and wheels as shown in Fig. 1. To derive equations of motion for the passive half-car suspension system, the following assumptions are considered, as below:

Pitch angle (θ) is small, springs and dampers are linear, tires have only stiffness with no damping property, the effect of friction is neglected, and the tires are always in contact with the road surface.

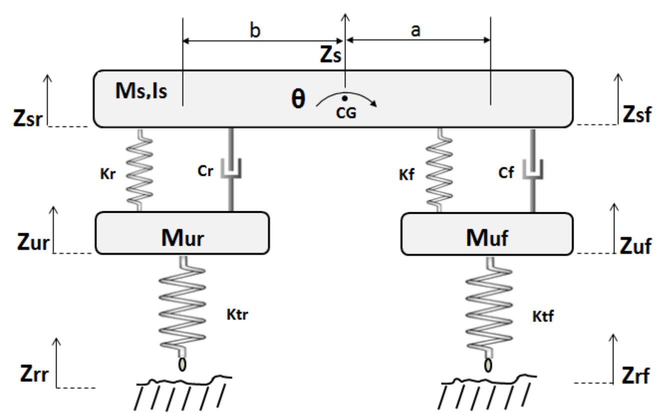

Fig. 1. Passive half-car model

By applying Newton's second law of motion, dynamic equations for the half-car model can be represented as follow, taking into consideration that the static equilibrium point is the origin for the displacement of the mass center and the angular displacement of the car body:

$$\begin{aligned} M_s\ddot{Z}_s = &-K_f(Z_s - Z_{uf} - a\theta) - C_f(\dot{Z}_s - \dot{Z_{uf}} - a\theta) \\ &- K_r(Z_s - Z_{uf} + b\theta) - C_r(\dot{Z}_s - \dot{Z_{ur}} - b\dot{\theta}) \end{aligned} \quad (1)$$

$$I_s\theta = aK_f(Z_s - Z_{uf} - a\theta) + aC_f(\dot{Z}_s - \dot{Z_{uf}} - a\theta) - bK_r(Z_s - Z_{ur} + b\theta) - bC_r(\dot{Z}_s - \dot{Z_{ur}} + b\dot{\theta}) \quad (2)$$

$$M_{uf}\ddot{Z_{uf}} = K_f(Z_s - Z_{uf} - a\theta) + C_f(\dot{Z}_s - \dot{Z_{uf}} - a\theta) - K_{tf}(Z_{uf} - Z_{rf}) \quad (3)$$

$$M_{ur}\ddot{Z_{ur}} = K_r(Z_s - Z_{ur} + b\theta) + C_r(\dot{Z}_s - \dot{Z_{ur}} + b\theta) - K_{tr}(Z_{ur} - Z_{rr}) \quad (4)$$

For active suspension system shown in Fig. 2, the hydraulic actuator forces F_{a1} and F_{a2} at the front and rear suspension system are generated respectively. Therefore, equations of motion can be written [8]:

$$M_s\ddot{Z}_s = -K_f(Z_s - Z_{uf} - a\theta) - C_f(\dot{Z}_s - \dot{Z_{uf}} - a\theta) - K_r(Z_s - Z_{uf} + b\theta) - C_r(\dot{Z}_s - \dot{Z_{ur}} - b\dot{\theta}) + F_{a1} + F_{a2} \quad (5)$$

$$I_s\theta = aK_f(Z_s - Z_{uf} - a\theta) + aC_f(\dot{Z}_s - \dot{Z_{uf}} - a\theta) - bK_r(Z_s - Z_{ur} + b\theta) - bC_r(\dot{Z}_s - \dot{Z_{ur}} + b\dot{\theta}) - aF_{a1} + bF_{a2} \quad (6)$$

$$M_{uf}\ddot{Z_{uf}} = K_f(Z_s - Z_{uf} - a\theta) + C_f(\dot{Z}_s - \dot{Z_{uf}} - a\theta) - K_{tf}(Z_{uf} - Z_{rf}) - F_{a1} \quad (7)$$

$$M_{ur}\ddot{Z_{ur}} = K_r(Z_s - Z_{ur} + b\theta) + C_r(\dot{Z}_s - \dot{Z_{ur}} + b\theta) - K_{tr}(Z_{ur} - Z_{rr}) - F_{a2} \quad (8)$$

Where;

Z_{sf} : sprung mass displacement at front body$(Z_s - a\theta)$
Z_{sr} : sprung mass displacement at rear body$(Z_s + b\theta)$
Z_s : sprung mass displacement
Z_{uf} : unsprung mass displacement at front body
Z_{ur} : unsprung mass displacement at rear body
Z_{rf} : road input to front wheel
Z_{rr} : road input to rear wheel
θ : vehicle rotational movement (rad)
F_{a1} : front actuator force (Newton)
F_{a2}: rear actuator force (Newton)

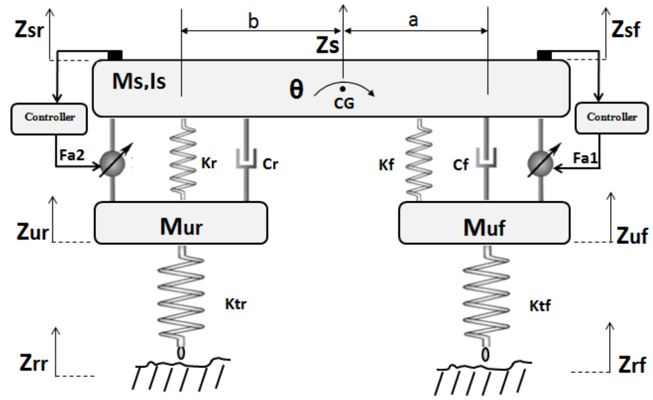

Fig. 2. Active half-car model

III. Control method

The suspension system electronic control was designed in a way that the damping energy can be improved against different road disturbances like bumps or potholes. The suspension control system relies on a control unit to control the operation of the hydraulic actuator. In Fig. 2, suitable sensors located on the car body send signals reflected road disturbances and then the signals feedback to a reference point to compare a specific actual signal with the set point. Accordingly, the control unit can moderate the output variables to acceptable ranges. In this study, two different types of PID (Proportional-Integral-Derivative) controllers are used to control a half-car model. The first type of PID controller is a conventional PID controller as shown in Fig. 3. This controller can be represented in time domain by the following equation [24]:

$$u(t) = K_p e(t) + K_i \int te(t)dt + K_d(de(t)/dt) \quad (9)$$

The second type of PID controller is shown in Fig. 4. This controller is a modified form of the conventional PID controller and can be formed by moving the proportional and derivative parts from the main forward path to the feedback path to operate on a measured signal y(t). In the meantime, keeping the integral part in the main forward path to operate on error e(t). In result, the controller is converted to an I-PD controller form [25].

The modified PID controller has the ability to eliminate the kick of error or impulse function that associated with using the conventional PID controller generated at the time the set point change [24],[26]. Thus, a preferred tracking reference can be obtained by using the modified PID controller [27]. The time domain equation of the modified PID controller is given by [24]:

$$u(t) = -K_p y(t) + K_i \int te(t)dt - K_d(de(t)/dt) \quad (10)$$

Where; u(t): Control force, e(t): Tracking error, r(t): Desired output, e(t) = r(t) – y(t), y(t): Actual output, Kp: Proportional gain, Ki: Integral gain, and Kd: Derivative gain.

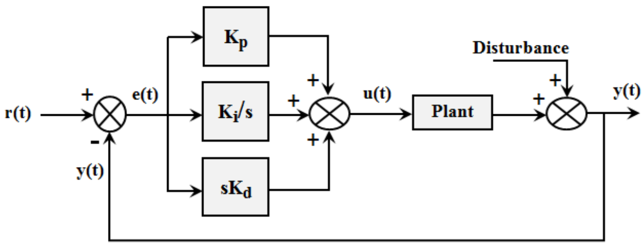

Fig. 3. Block diagram of conventional PID controller [24]

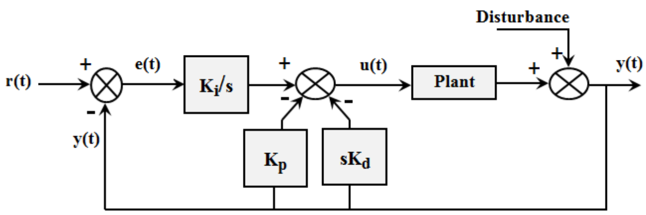

Fig. 4. Block diagram of modified PID controller [24]

IV. SIMULINK MODELING

The block diagram of a passive half car model is shown in Fig. 5. This model was built in the MATLAB/Simulink environment. A step function is used to excite the system as an external source to represent the road profile. The forward linear velocity (v) of the vehicle when it crossed over a road profile of 0.1 m height was 45 km/h. The time delay between the front and the rear wheels is calculated using the following formula as 0.225 second [6]:

$$Timedelay = (a+b)/v \tag{11}$$

When the system is activated, the hydraulic actuator forces are generated and applied to the passive system with the implementation of both modified and conventional PID controllers. In this study, the vertical displacement of the front body (Z_{sf}) and rear body (Z_{sr}) are used as feedback signals to the controllers, the desired performance of these variables are set to improve system dynamic response.

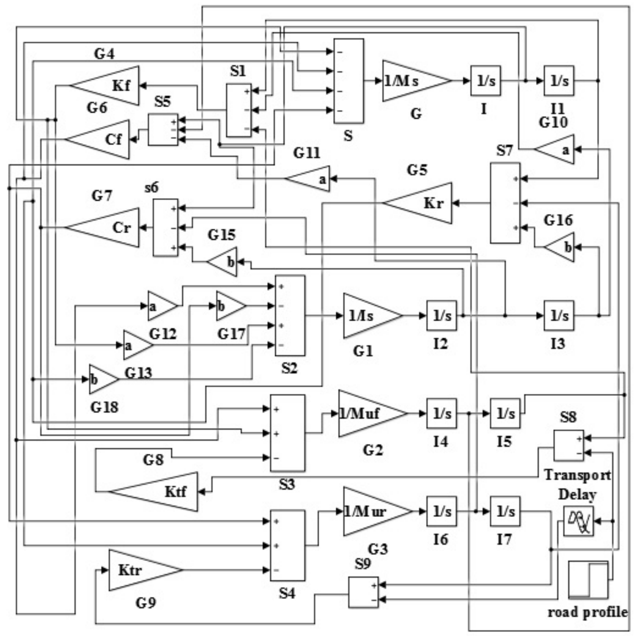

Fig. 5. Simulink block diagram of passive linear half-car model

Referring to the Simulink block diagram of passive half-car model, the vertical and angular displacement of the sprung mass as well as the displacement of the unsprung masses and the suspension deflection are obtained. With the assist of MATLAB/Simulink software environment, the mathematical model of the entire active half-car model with the conventional and modified PID controllers are shown in Figures 6 and 7 respectively. The controller gains Kp, Ki and Kd are found by applying trial-and-error tuning method and Table I. The parameters of the half-car model used for simulation are listed in Table II.

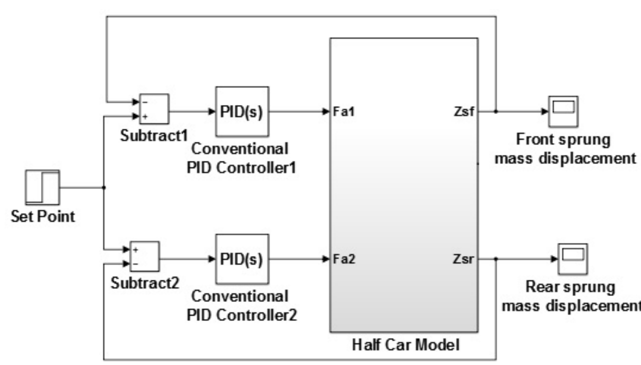

Fig. 6. Active half-car model with conventional PID controller

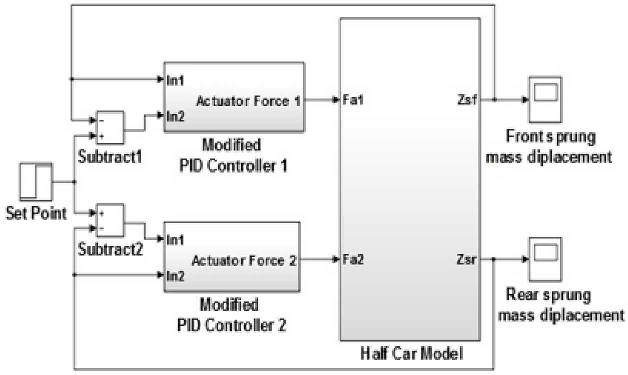

Fig. 7. Active half-car model with modified PID controller

TABLE I.
RESPONSE OF PROPORTIONAL, INTEGRAL AND DERIVATIVE CONTROLLER GAINS [19]

Closed loop response	Rise time	Over- shoot	Settling time	Steady state error
Kp	Decrease	Increase	Small change	Decrease
Ki	Decrease	Increase	Increase	Eliminate
Kd	Small change	Decrease	Decrease	Small change

TABLE II.
HALF-CAR PARAMETERS [3]

Parameter Description	Value
M_s :Body mass (sprung mass)	730 Kg
Is :Body pitch moment of inertia	2460 Kgm2
M_{uf} :Front wheel mass (front unsprung mass	40 Kg
M_{ur} :Rear wheel mass (rear unsprung mass)	35.5 Kg
K_f :Front suspension stiffness	19,960 N/m
K_r :Rear suspension stiffness	17,500 N/m
C_f :Front suspension damping coefficient	1290 Ns/m
C_r :Rear suspension damping coefficient	1620 Ns/m
K_{tf} :Front tire stiffness	175,500 N/m
K_{tr} :Rear tire stiffness	175,500 N/m
a :Distance from vehicle center of gravity (C.G.) to front axle	1.011 m
b :Distance from vehicle center of gravity (C.G.) to rear axle	1.803 m

V. SIMULATION RESULTS AND DISCUSSION

In this section, a half-car model has been simulated in three cases (passive, active with conventional PID controller and active with modified PID controller) according to the mathematical modeling implemented in MATLAB/ Simulink software to present the dynamic response. Tables III and IV show PID gains for conventional and modified PID controllers respectively. After running the simulation for five seconds, system dynamic behavior and response to a step input reference or set point signal can be found to show the main properties such as rise time, overshoot, settling time and steady state error. Using modified PID controller, the set value does not influence the proportional and derivative parts as in the conventional PID controller while the controller action is still affecting.

TABLE III.
CONVENTIONAL PID CONTROLLER GAINS

Controller	Kp	Ki	Kd
Conventional PID controller 1	175	12	234
Conventional PID controller 2	198	5	215

Therefore, only proportional and derivative actions pick up system variables while the integral action works on the error and the system response occurs accordingly. Also, by this way the noise caused by the error signal is eliminated. Figures 8 – 15 show the dynamic responses of the half-car's outputs for the three cases. The responses show that the settling time is very short which ensures high and fast passenger comfort. In addition, the peak overshoots are clearly damped, and good reference tracking are obtained, which reflect appropriate dynamic response. As a result, sufficient shock absorption and vehicle vibration reductions are achieved. From the results, the modified PID controller has a better performance to suppress both car body oscillations and suspension deflection in comparison with conventional PID controller.

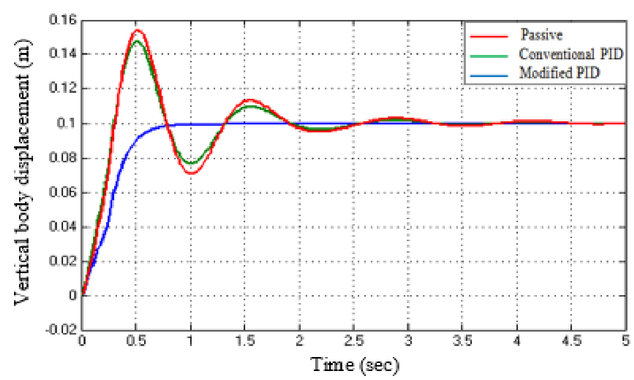

Fig. 8. Vertical body displacement

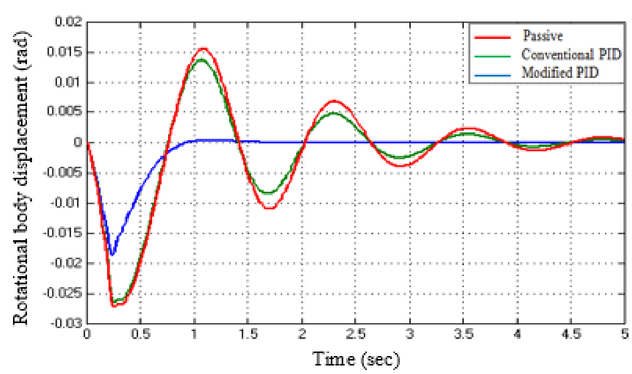

Fig. 9. Rotational body displacement

TABLE IV.
MODIFIED PID CONTROLLER GAINS

Controller	Kp	Ki	Kd
Modified PID controller 1	85	40	4250
Modified PID controller 2	100	35	3000

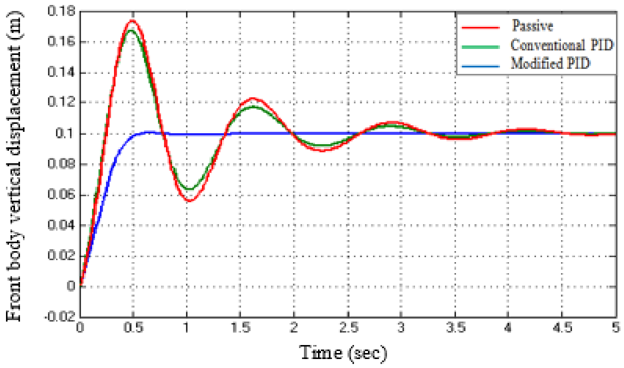

Fig. 10. Front body vertical displacement

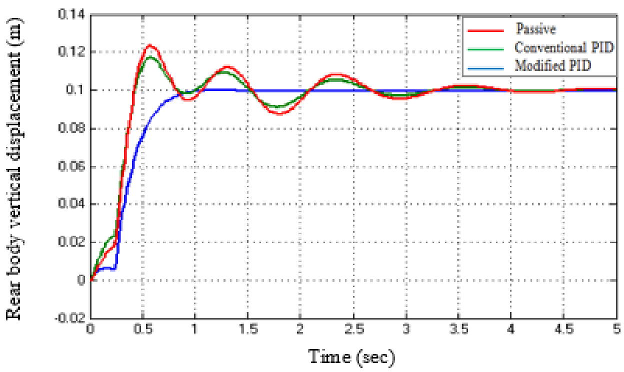

Fig. 11. Rear body vertical displacement

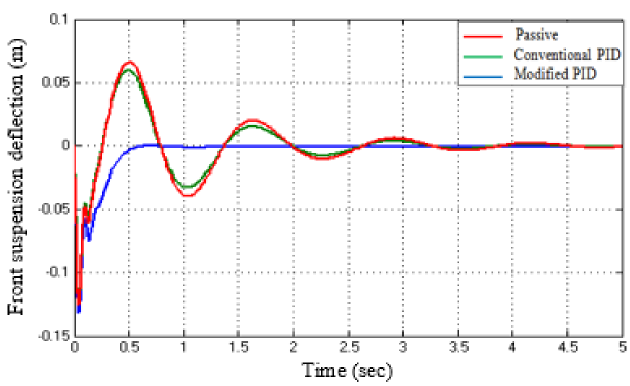

Fig. 12. Front suspension deflection

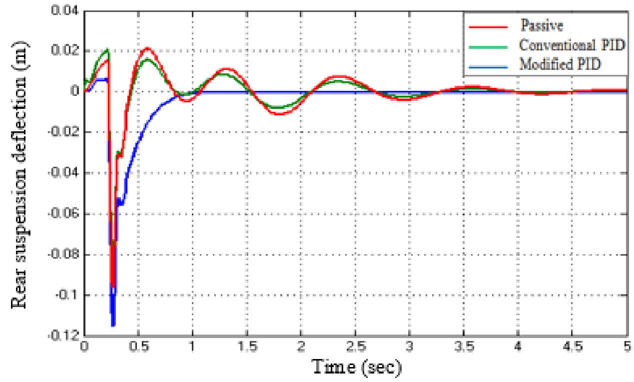

Fig. 13. Rear suspension deflection

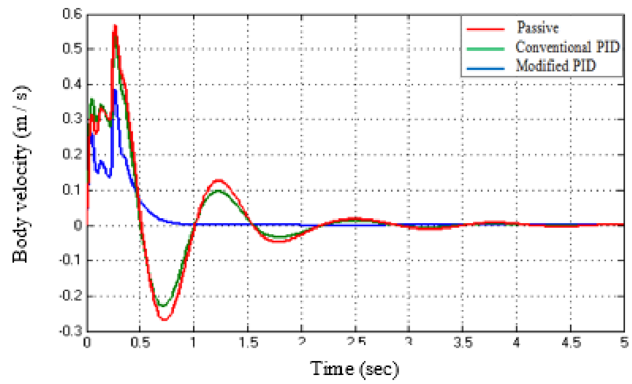

Fig. 14. Half-car body velocity

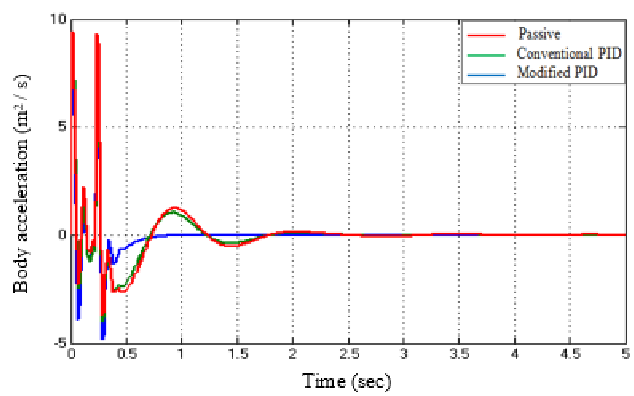

Fig. 15. Half-car body acceleration

VI. Conclusion

The proposed modified PID controller has a better performance to suppress vehicle body vibrations resulted from road disturbances. Also, the modified PID controller has a better reference tracking than conventional PID controller. In addition, the advantage of using modified PID controller is the simplicity and the short time required to calculate controller gains. Thus, in turn this advantage makes it easier to use when compared with Fuzzy, Fuzzy-PID and other types of controllers. The proposed controller offers a clear reduction in vehicle body displacement as well as suspension deflection reflected the good behavior and efficient performance. All of the above can give stability, ride comfort and good handling of the vehicle.

CONFLICT OF INTEREST

The authors declare that there is no conflict of interest in the study.

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