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(RESEARCH ARTICLE)

Performance analysis and control of an agricultural tractor suspension system

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Abstract

Suspension systems play a vital role in providing comfortable and safe vehicle ride. This paper aims to improve the passenger ride comfort, vehicle stability, safety, road holding in an active quarter tractor model. the main objective is to obtain a stable, robust, and controlled system. It is necessary to use controller to increase the stability and performance of the system. the controller selection and design aimed to achieve good passenger ride comfort and health, stability, and passenger body acceleration and displacement response under uneven road excitations. PID and LQR controllers are developed, and compare their performances against the road disturbances. The performance of the designed controllers evaluated using simulation work in MATLAB. Simulation results show that the proposed LQR control scheme can successfully achieve the desired ride comfort and passenger safety compared to PID control scheme

Keywords: Suspension system; Quarter Model; Tractor suspension Linear Quadratic Regulator; PID Control.

1. Introduction

Suspension systems for automobiles have been a balance between three opposing criteria: road handling, weight vehicle carrying, and passenger comfort. The suspension system must sustain the vehicle, offer direction control by means of handling maneuvers, and effectively isolate passengers and loads from disturbance. The main function of vehicle suspension system is to minimize the vertical acceleration transmitted to the passenger which directly provides road comfort. There are three types of suspension system; passive, semi-active and active suspension system. Traditional suspension consists springs and dampers are referred to as passive suspension, then if the suspension is externally controlled it is known as a semi-active or active suspension. [1] In the active suspension system, a force actuator controlled by the feedback controller is placed between the vehicle body. A controlled suspension system permits forward compensation between the riding comfort and the performance criteria of the suspension deviations [4-5]. Suspension functions: The automotive suspension on a vehicle typically has the following basic tasks:

i. To isolate a vehicle body from road disturbances in order to provide good ride quality.
ii. To keep good road holding.
iii. To provide good handling.
iv. To support the vehicle static weight.

Control strategy is a very importance part for the active suspension system. With the correct control strategy, it will give better compromise between ride comfort and vehicle handling. Nowadays there a lot of researches have been done to improve the performance of active suspension by introducing various control strategies. [2]

To date, many control methods such as Network Approximate Dynamic Programming, higher-order sliding mode control (SMC), H_∞ Method, Fuzzy Logic and the neural network method were used in the active vehicle suspension field.

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In this paper, the performance of the active suspension system of the tractor has been compared using two different control strategies, the first strategy is using of the Linear Quadratic Regulator Control Method and the second is the using of PID Controller, this comparison will show a good contribution to the field of suspension systems.

This work is arranged as follows: Section 2 which is devoted to the mathematical model of the agricultural tractor suspension system in addition to the road profiles used in this paper. This section also presents an open-loop Simulink model of an active suspension system. Section 3 introduces the control strategy and gives a brief explanation about the PID controller and the linear quadrature regulator approach. Section 4 covers some simulations for comparing a closed-loop Simulink tractor suspension system using the two controllers. Section 5 gives the results and discussion of this work and section 6 of this paper presents the conclusion of the performed work.

2. Mathematical Model

The main focus of this section is to provide background for mathematical model of a suspension system of quarter tractor. The dynamic model, which can describe the relationship between the input and output, enables ones to understand the behavior of the system. [2]

The 2-DOF quarter tractor model and the physical parameters used in this paper are shown in Figure 1 and Table 1 respectively, this model is one of the most widely used suspension models, which is very important when studying vehicle dynamics especially ride comfort and road handling characteristics. It represents the vibration behavior of the tractor body and the wheel.

The main components of the suspension system are damper bs, springs Ks and Kus, in addition to force actuator *Fc*. The value of the actuator force must be zero in the case of passive suspension systems. The symbol m_s denotes the sprung mass, which indicates a quarter of the total tractor mass and the unsprung mass m_{us} represents the mass of the wheel assembly system. The damping coefficient is denoted by the symbol b_{us}, while the vertical stiffness of the tire is denoted by the symbol K_{us}. The vertical displacement of the road profile, unsprung mass, sprung mass indicated by z_r, z_{us}, and z_s respectively.

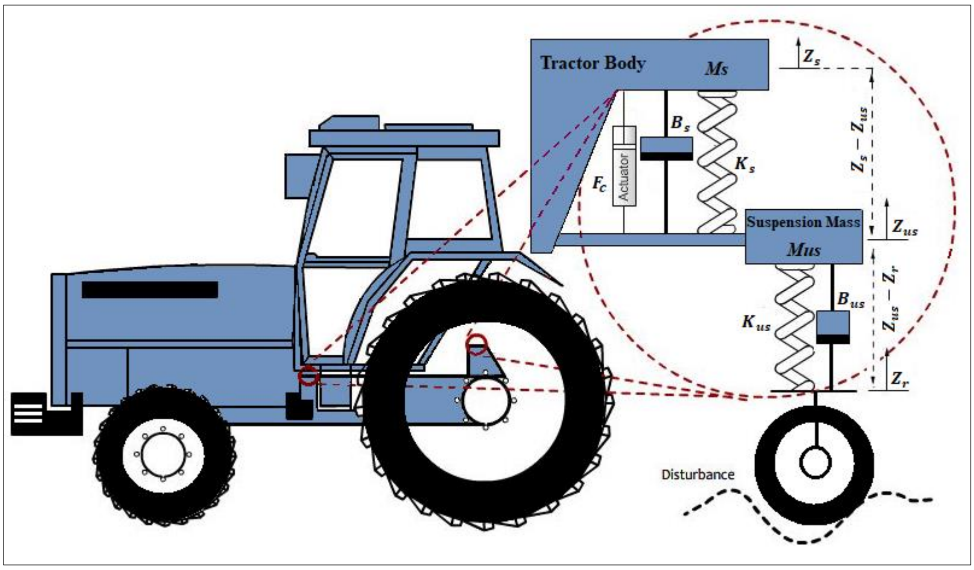

Figure 1 Active Suspension system of Quarter Tractor Model.

To find out equations of motion (EOM) for this system, the free body diagram for each mass should be determined. There are two masses in the system and the forces applied to each mass should be drawn on the diagrams. There will be two equations of motion. All the initial conditions are assumed to be zero. The forces applied to the masses are due to the spring force, damping force, active suspension force.

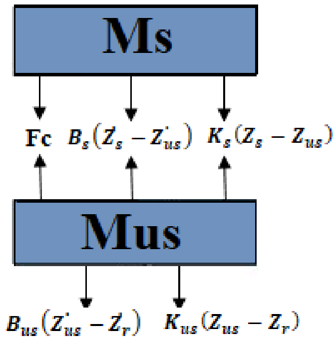

Figure 2 Free Body Diagram of the System.

Based on the free body diagram of the quarter tractor model shown in figure 2 and Newton's second law, it is very easy to write the equations of motion for the system as follows:

$$M_s\ddot{Z}_s = B_s(\dot{Z_{us}} - \dot{Z}_s) + K_s(Z_{us} - Z_s) + F_c \tag{1}$$

$$M_{us}\ddot{Z_{us}} = B_s(\dot{Z}_s - \dot{Z_{us}}) + B_{us}(\dot{Z}_r - \dot{Z_{us}}) + K_s(Z_s - Z_{us}) + K_{us}(Z_r - Z_{us}) - F_c \tag{2}$$

Where:
$z_s - z_{us}$ Indicates the suspension deflection, $\dot{z}_s$ represents the tractor body or the sprung mass speed (comfortable comfort indicator), $\ddot{z}_s$ represents the acceleration of the tractor body, $z_{us} - z_r$ represents the tire deflection (road-handling indicator), and $\dot{z}_{us}$ indicates the tire velocity.

The following data shown in the table below represents the tractor parameters and values used in the simulation

Table 1 A quarter tractor model parameters

Parameters	Values
M_s	290 kg
M_{us}	59 kg
K_s	16182 N/m
K_{us}	190000 N/m
B_s	1000 N.s/m
B_{us}	230 N.s/m

The figure below shows the complete Simulink model for an active suspension system, and this model was built based on equations 1 and 2.

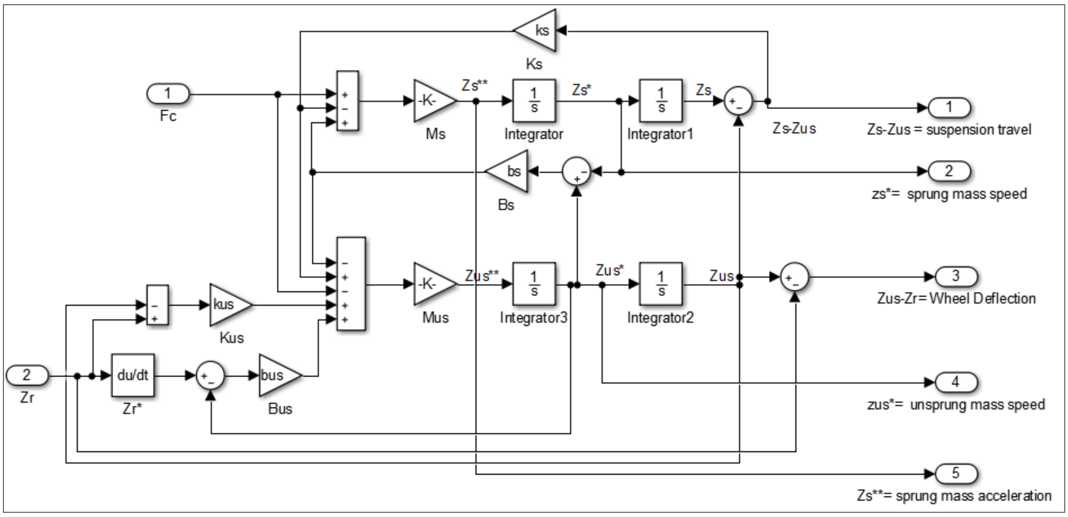

Figure 3 The Simulink model of the model

2.1. Road Profile

There are two types of disturbances introduced to the tractor suspension system in this study. First, the road profile A is a single bump as shown in Fig. 4. Second, the road profile B is two bumps as shown in Fig. 5. The first input is a step variation in which the wheel is exposed to a positive 0.1 m step, then after 1 seconds, the wheel settles down on the road. The second input is a pulse variation in which the wheel is exposed to a positive 0.1 m step then after 1 seconds, the wheel is exposed to another 0.05 m pulse.

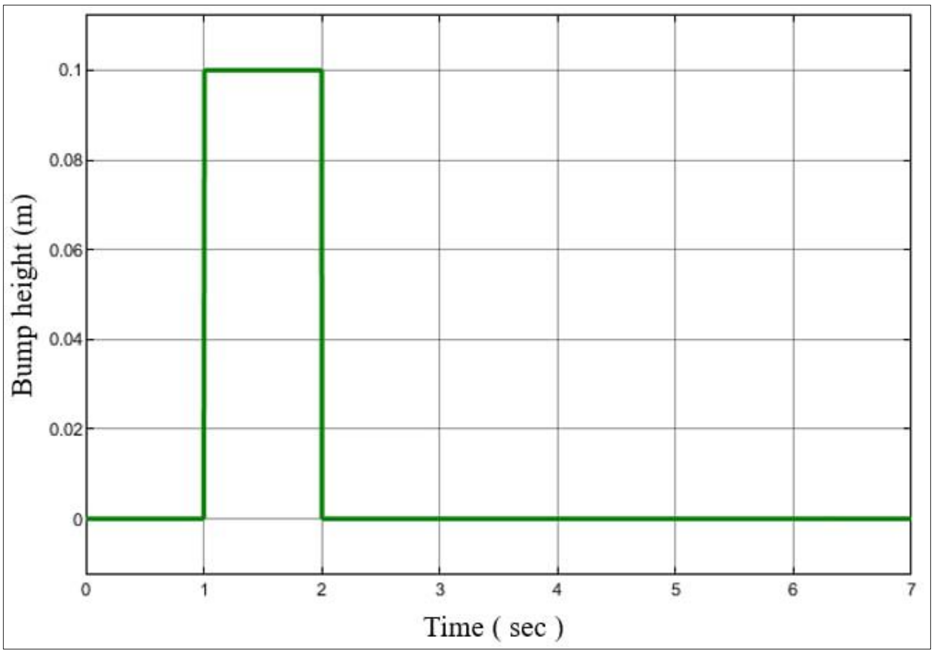

Figure 4 Road Profile A.

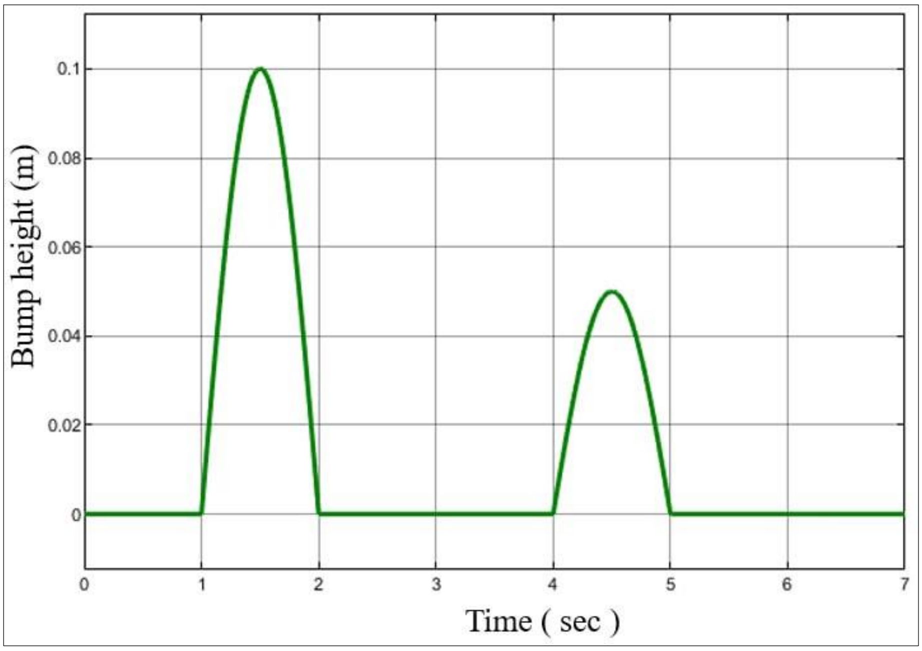

Figure 5 Road Profile B

3. Control techniques

This part can be divided into two sections. The first section provides an overview of the two controllers used in this work, while the second section shows the Simulink model structures implemented for the active suspension system with Quadratic Regulator linear controller and PID controller.

3.1. Controller Design Using Linear Quadratic Regulator (LQR)

The Linear Quadratic Regulator (LQR) is one of the most studied control problems in the literature, and it has many applications. LQR is a specified form of state feedback control method, and LQR technique makes optimal control decisions considering the states and control input of the dynamical system. The structure of the LQR control technique is same with the structure of the state feedback control; however, the design method of LQR differs from state feedback.

Consider a state variable feedback regulator for the system given as:

$$u(t) = -Kx(t) \tag{3}$$

Where: K is the state feedback gain matrix.

So, the main difference between LQR and state feedback is the calculation of the gain matrix K.

The configuration of the state variable feedback is shown in the figure 6.

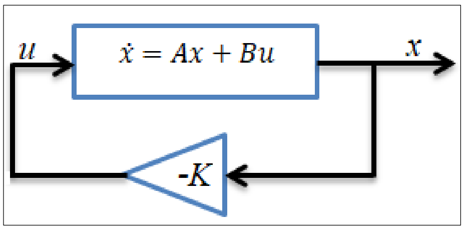

Figure 6 The configuration of the state variable feedback.

LQR algorithm solves the equation (4) to find an optimum gain matrix that minimizes the quadratic cost function below:

$$J(u) = \int_0^\infty x^T Qu + u^T Ru)dt \tag{4}$$

The LQR gain vector K can be calculated as in equation (5) which it is called Ricati Equation.

$$K = R^{-1}B^T P \tag{5}$$

P is a positive definite symmetric constant matrix which can be obtained from the solution of the Algebraic Riccati Equation as shown in equation (6).

$$A^T P + PA - PBR^{-1}B^T P + Q = 0 \tag{6}$$

Then the feedback regular u:

$$u(t) = -(R^{-1}B^T P)x(t) \tag{7}$$

In the design of LQR controller, the key point is to determine Q and R structures. Basically, there are numerous methods to determine the Q and R matrices. [3]

The selection of Q and R determines the optimality in the optimal control law. The choice of these matrices depends only on the designer. Generally, preferred method for determining the values for these matrices is the method of trial and error in simulation. As a rule of thumb, Q and R matrices are chosen to be diagonal. [7]

By selecting the matrix Q as follow:

$$Q = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 1000 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

And R=0.0001.

Therefore, the value of gain K is given by:

K= [-0.0000 0.7565 -30.6425 -1.1314]

The Simulink model for the control system includes the LQR controller is shown below in Figure 7.

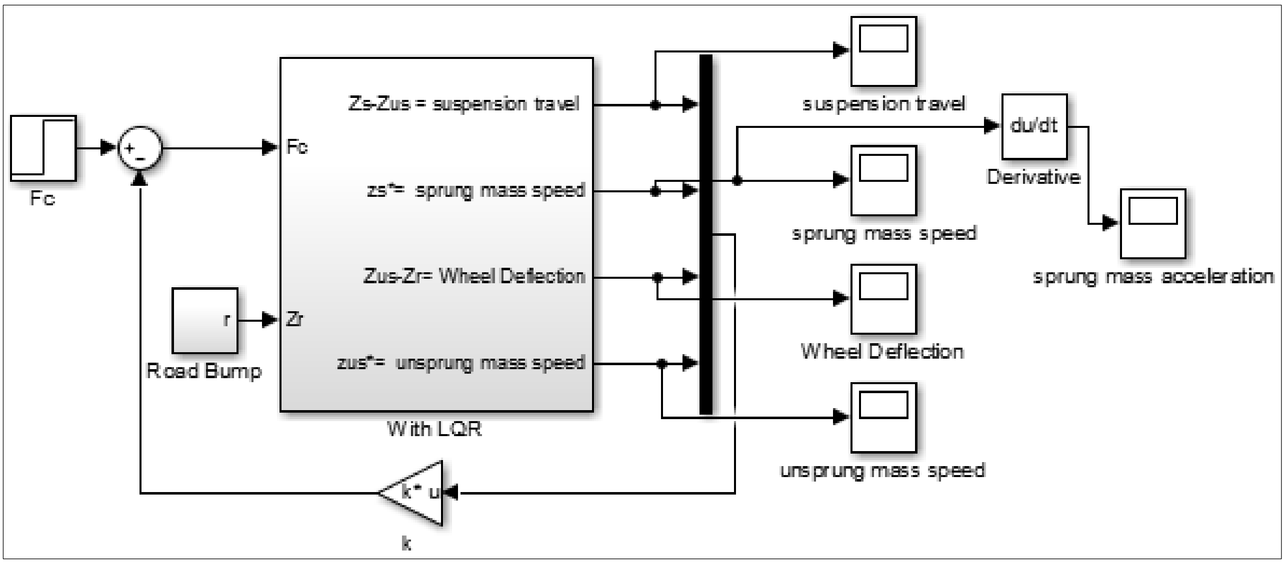

Figure 7 Simulink model for the system with LQR controller

3.2. PID Controller

PID control is a particular control structure that has become almost universally used in industrial control. PID stand for Proportional, Integral and Derivative. They have proven to be quite robust in the control of many important applications for specific operating conditions. Its structure is simple but very effective feedback control method applied to dynamical systems. [35]

The structure of PID controller is demonstrated in Figure 8

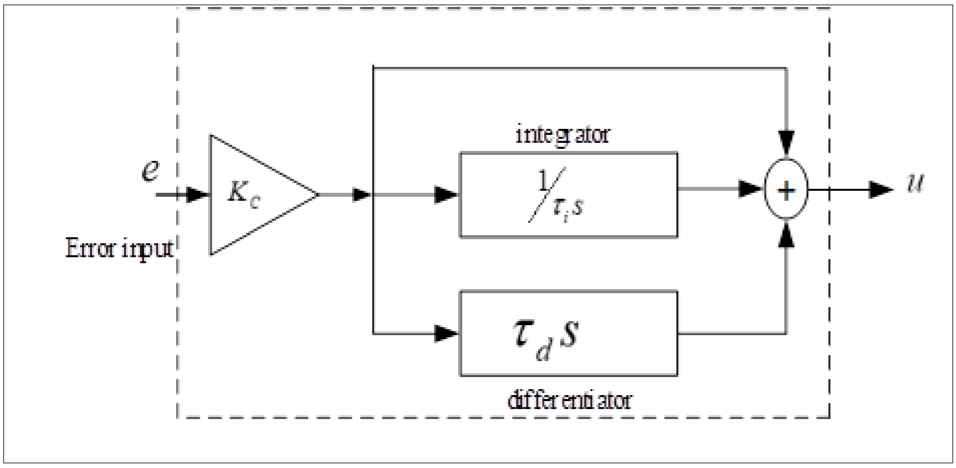

Figure 8 The basic structure of PID controller

The transfer function of PID controller is described as below, the combination of three terms is a controller output:

$$G(s) = k_p(1 + \frac{1}{T_i s} + T_d s) \quad (8)$$

Where T_I, T_D are a constant and represents an integral time and the derivative time constant respectively, K_P is gain of proportional.

The structure of the controller and the block diagram of the desired control system are implemented as shown in figures 8 and 9, which consists of the reference model, actual model of the tractor suspension and PID controller.

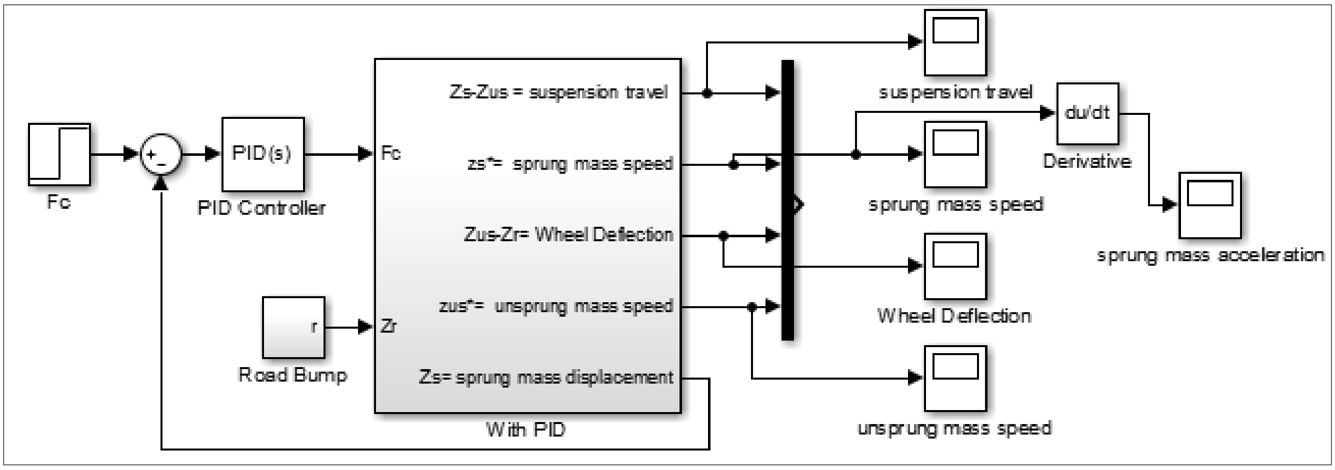

Figure 9 Simulink model for the system with PID controller.

4. Simulations

The simulation results of the quarter tractor model using MATLAB / SIMULINK show a comparison of the vehicle's performance and behavior with and without control. Figures 9 and 10 illustrate the results of body displacement of the quarter tractor model in three scenarios: one without any controller, another with an LQR controller, and the other with a PID controller. Figures 11 to 18 display the body acceleration, tire deflection, suspension deflection and velocity of the sprung mass of the vehicle under different road profiles as shown in figures 4 and 5.

4.1. Body Displacement

Figures 9 and 10 show the body displacement for the profile road disturbances A and B. The simulation results indicated that stability of suspension system without any controller takes very long-time which results in the discomfort of the passengers and poor road handling capacity of the vehicle and produces a significant jerk on the tractor chassis and introduces undesired accelerations into the system and degrades the ride characteristics of the vehicle. The LQR and PID Controller was implemented to our system; it showed a decrease in the magnitude of the body displacement.

It is shown that the LQR controller allows a fast rise time and quick settling time without oscillatory behavior. Simulation shows that the LQR controller gives suitable results for the profile road disturbance A and B effectively.

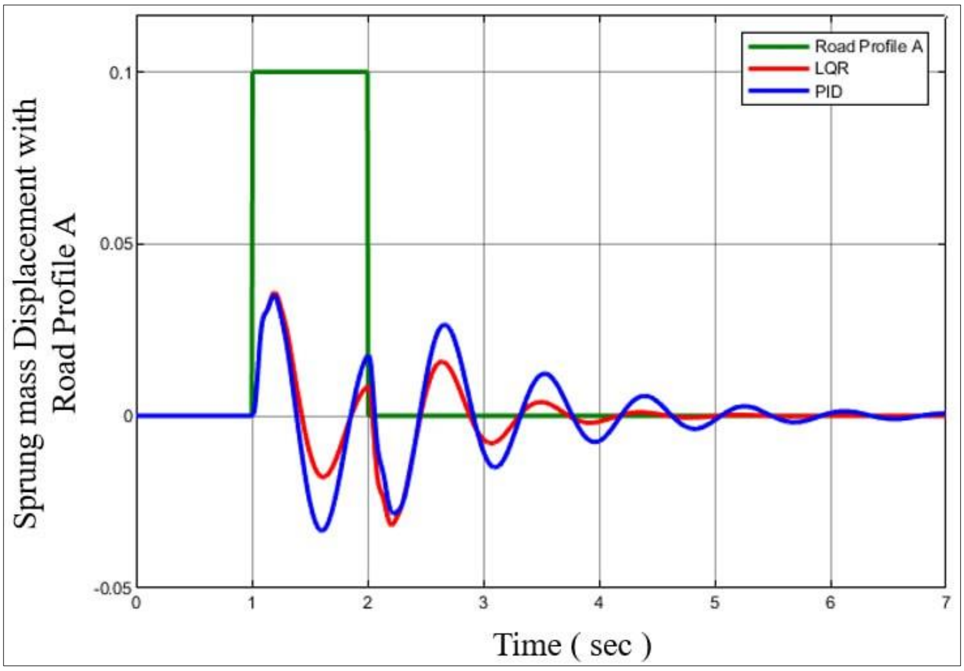

Figure 10 Body Displacement for The Profile Road Disturbances A

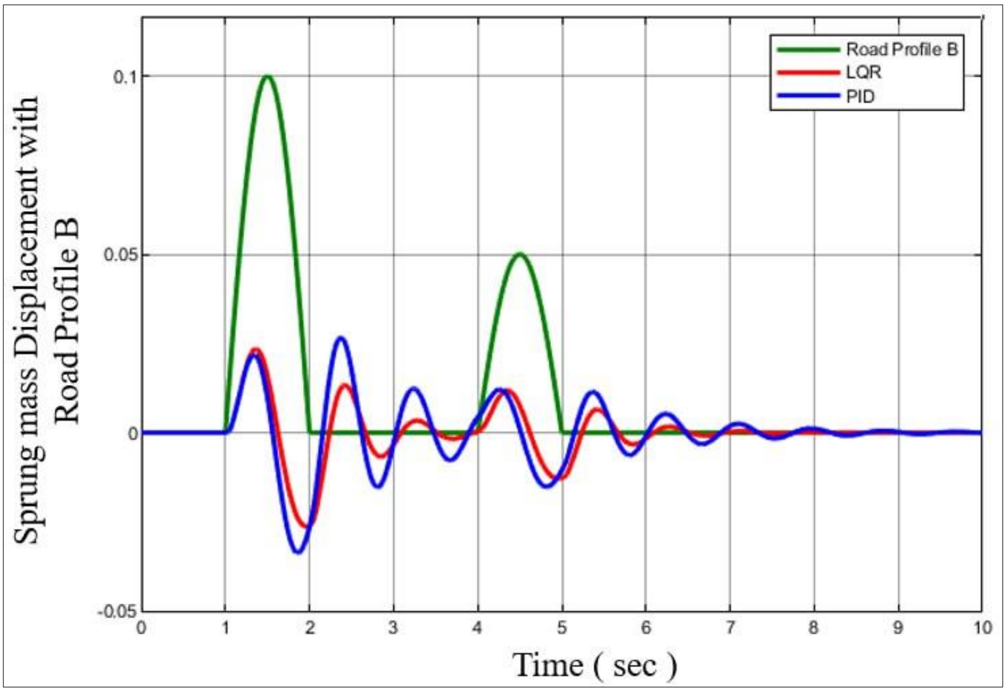

Figure 11 Body Displacement for The Profile Road Disturbances B

4.2. Sprung Mass Velocity

The simulation results for vehicle's body velocity are compared for the profile road disturbances A and B, (shown in Figure 11 and Figure 12 respectively). The active suspension system with LQR Controller reduced the velocity magnitude at road profile A and B peaks with reduction in settling time in comparison to suspension system with PID.

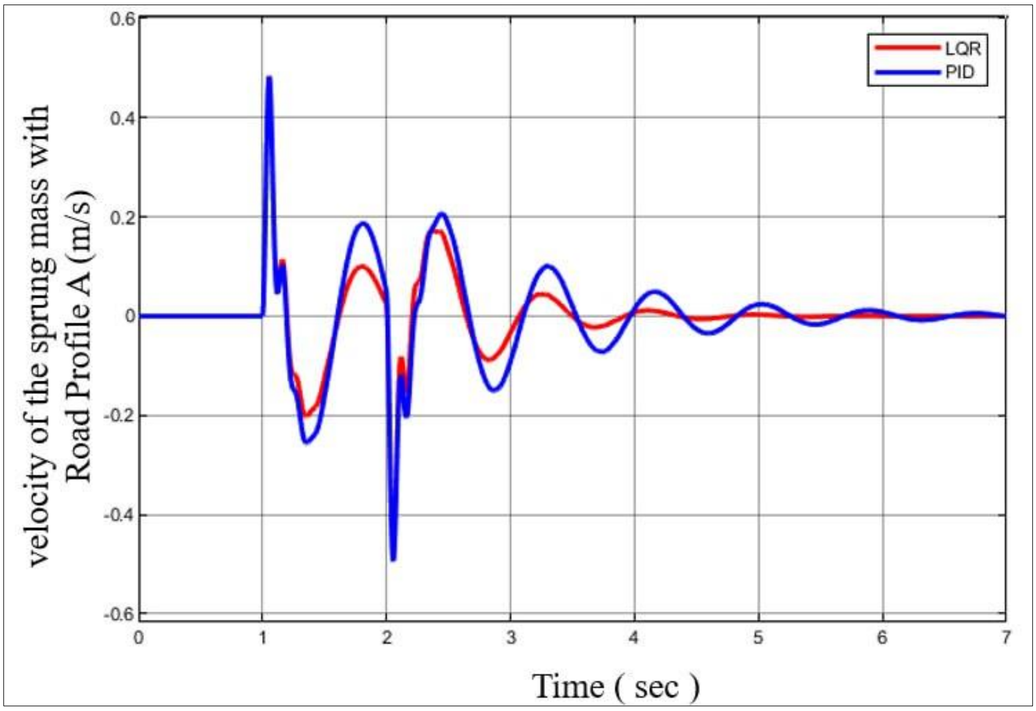

Figure 12 Body Velocity with Road Profile A

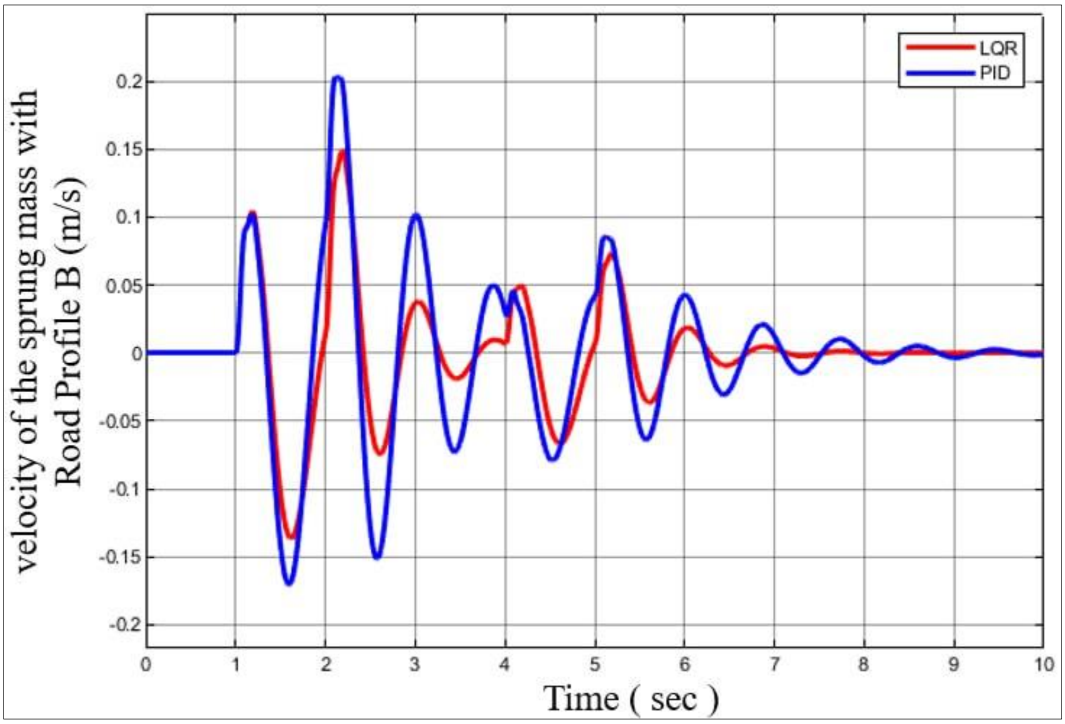

Figure 13 Body Velocity with Road Profile B

4.3. Body Acceleration

Figures 13 and 14 of body acceleration show that very close responses have been achieved by the LQR and PID at road profile A. On the other hand, significant improvement has been achieved using the LQR for road profile B, where their performance is better than PID scheme control. LQR Controller reduced settling time with the reduction of overshoot at road profile B and demonstrates the effectiveness of it in suppressing the vehicle vibration as compared to PID controller.

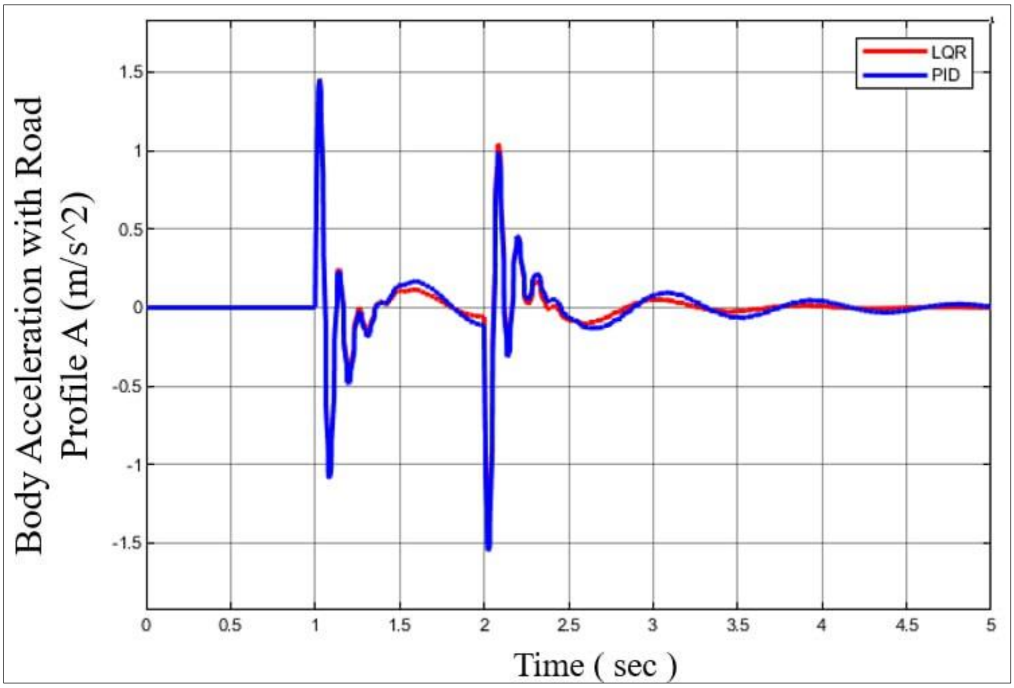

Figure 14 Body Acceleration with Road Profile A

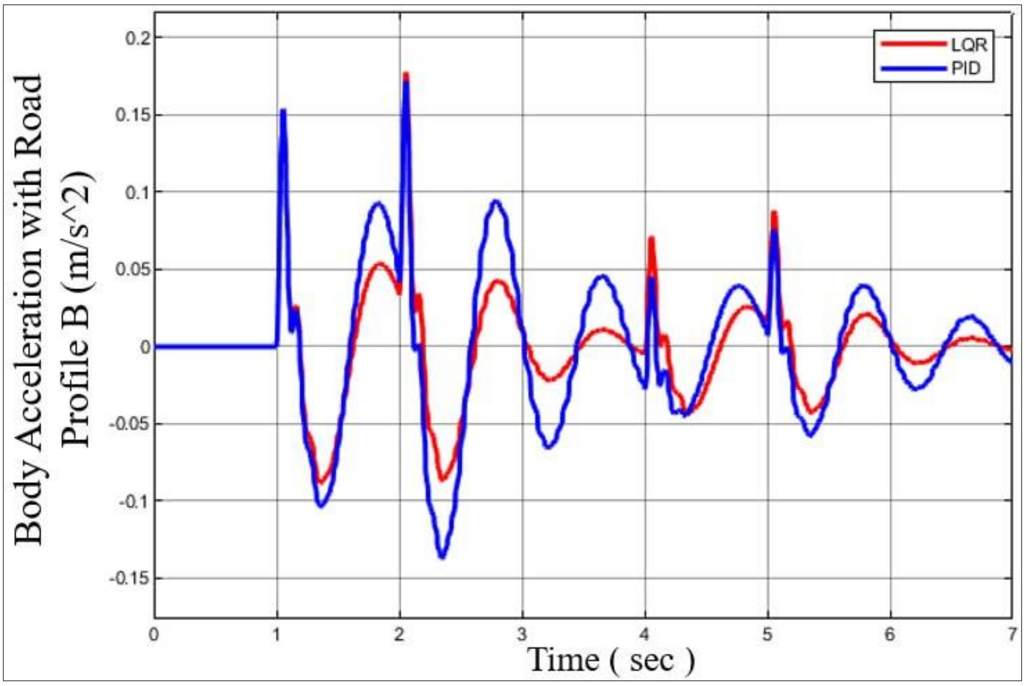

Figure 15 Body Acceleration with Road Profile B

4.4. Tire Deflection

Figures 15 and 16 of tire deflection show that very close responses have been achieved by the LQR and PID at road profile A. On the other hand, significant improvement has been achieved using the LQR for road profile B even though the amplitude is slightly high, but their performance is better than PID scheme control which its performance show that the settling time of the wheel deflection for the system with LQR controller is very fast as compared to the system with PID controller.

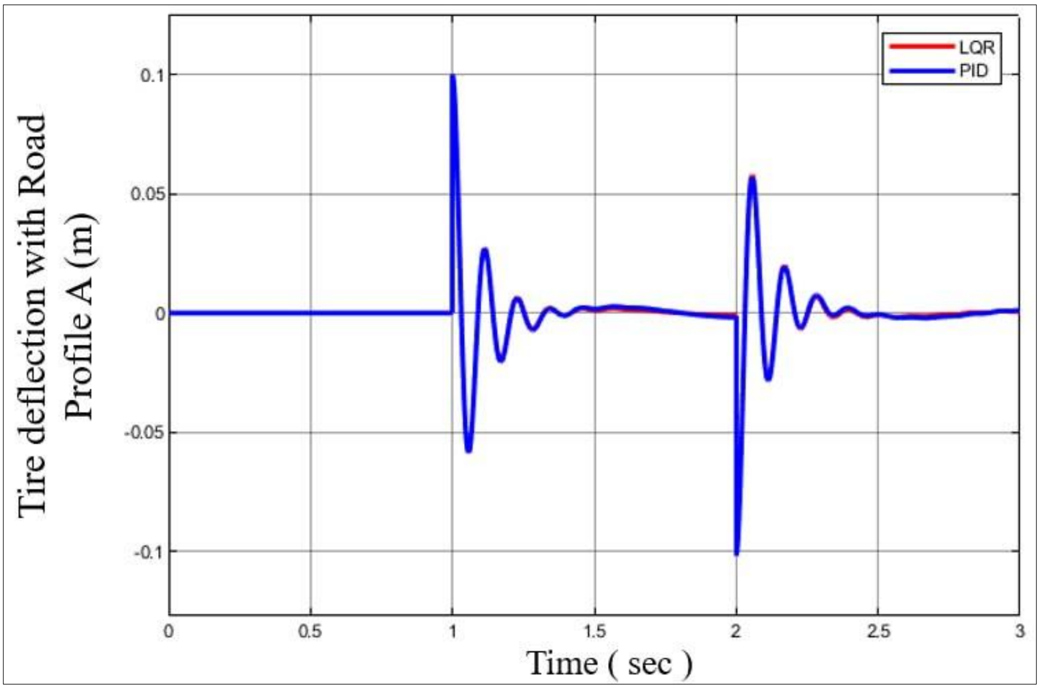

Figure 16 Tire Deflection with Road Profile A

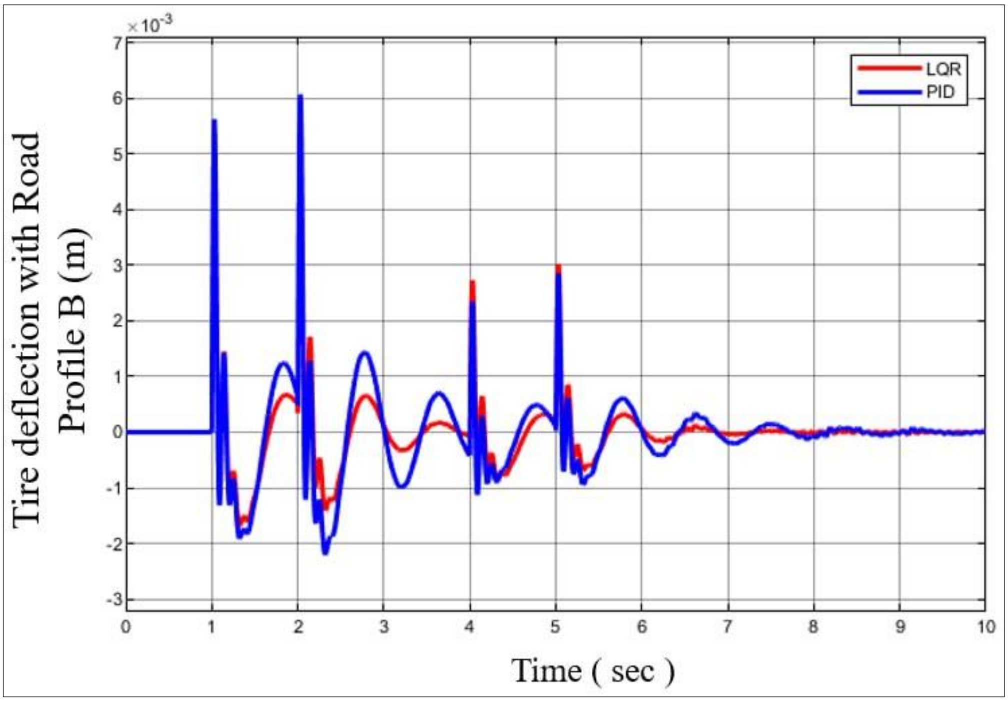

Figure 17 Tire Deflection with Road Profile B

4.5. Suspension Travel

For comparison purpose, suspension travel for LQR and PID controllers are presented in figures 17 and 18 for both controllers. The result illustrates demonstrates the effectiveness of LQR in reducing the suspension travel as compared to PID controller, which guarantee better road holding.

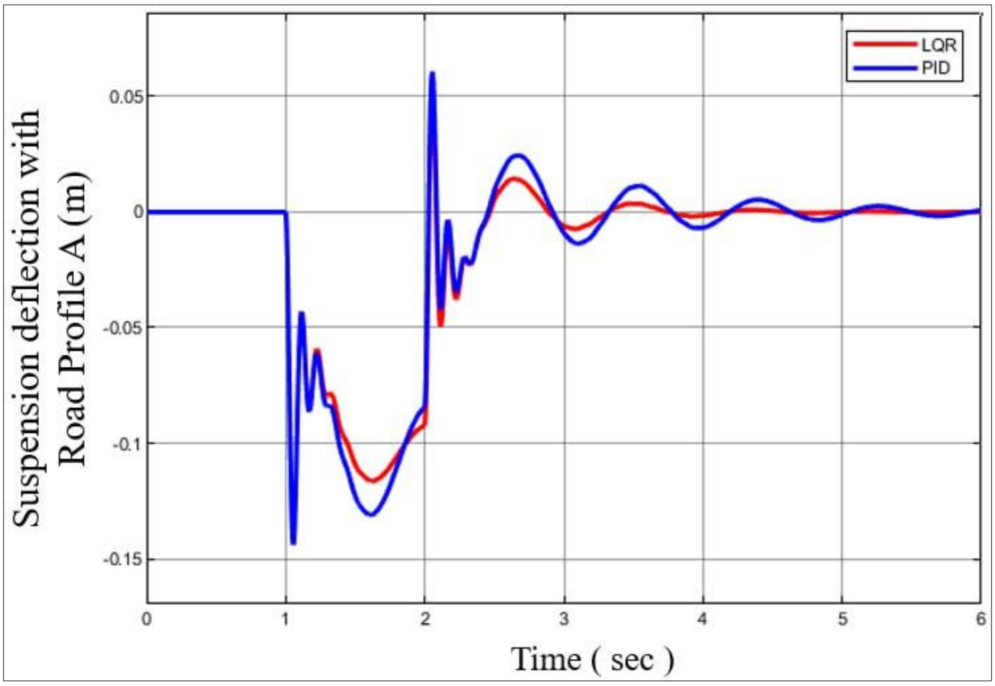

Figure 18 Suspension Travel with Road Profile A

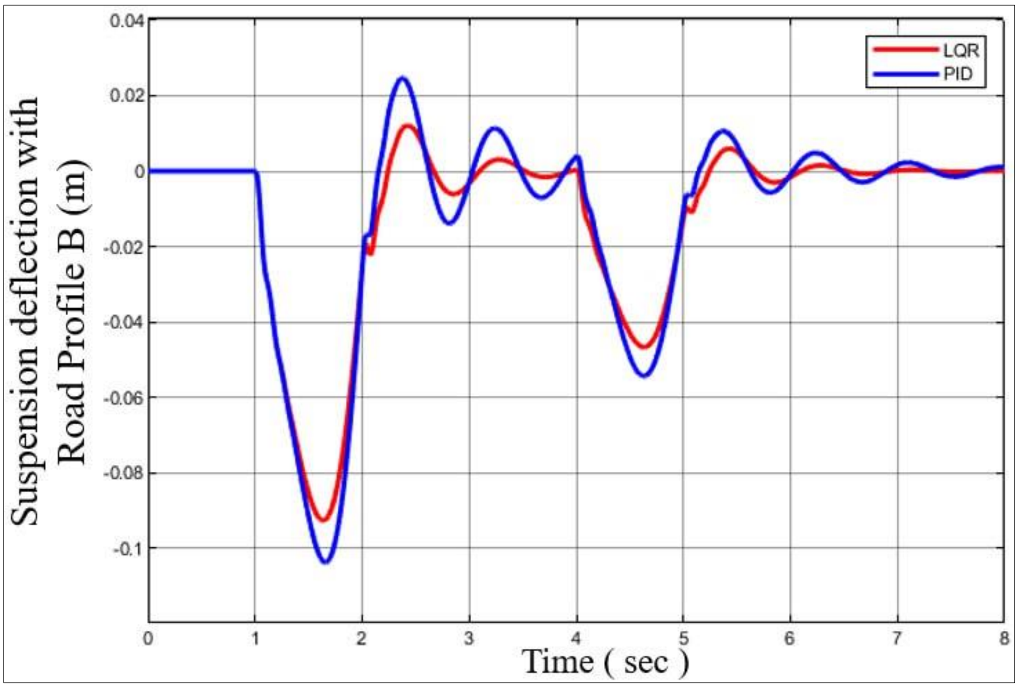

Figure 19 Suspension Travel with Road Profile B

5. Conclusion

In this paper Proportional-Integral-Derivative (PID) and linear quadratic regulator (LQR) controllers are successfully designed using MATLAB/SIMULINK. Both controllers are capable of stabilizing the suspension system very effectively as compared to passive suspension system. Based on the results discussed in previous section, it accomplished that LQR control scheme gives much better results compared to PID control scheme and passive suspension systems as far as ride comfort, road holding and suppression of vibrations are concerned. Simulation shows that the LQR controller gives suitable results for two types of disturbances, and hence it can be said that this controller could handle other real road situations.

Compliance with ethical standards

Disclosure of conflict of interest

There are no conflicts of interest.

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