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Simulation of Control System for a Half Car Model Suspension System for Passenger Car Application by Design an LQR Controller

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Abstract: - The constantly growing topic of inventive vehicle system design is of interest to researchers. The difficulty stems from the ongoing requirement for advancement in vehicle handling, ride comfort, and driving dynamics. The present study proposed a mathematical model for a 4DOF half-car active suspension system (ASS) using a LQR (Linear Quadratic Regulator) controller, based on a control approach for ride comfort and vehicle handling. The task is simulated using MATLAB/Simulink software. The unsprung masses of the wheels' heave displacements, the vehicle's pitching dynamics, and the sprung masses of its body's heave displacements are the regulated parameters. Compared to the antiquated passive suspension technology, its performance is superior (PSS). The simulation uses two bumpy sinusoidal roads and a random road input. Finally, simulation software used to show how well the recommended controller performed. The simulation's outcomes show how this study modelled and control capabilities have improved.

Keywords:- *Active Suspension, LQR, 4 DOF, MATLAB/Simulink.*

I. INTRODUCTION

The field of designing new car systems is one that is gaining popularity among researches. The ever-increasing improvement needs relating to driving dynamics, ride comfort, and vehicle handling, however, pose a difficulty. Environmental protection and fuel efficiency have also become major considerations for customers when purchasing a new vehicle. More efficient drive dynamics control technologies, such as active suspension systems, can developed to satisfy these needs. The suspension system's major job is to keep wheels in touch with the pavement at all times to provide road holding and to shield the body of the car from imperfections in the road when braking, cornering, and speeding up to increase safety and comfort while driving. [1] ,[2],[3],[4]. In order to prevent suspension movements from resulting in insufficient tire-to-road contact, the term "vehicle handling regulation" is used. A vehicle's body and wheels connected by a suspension system, which made up of a number of springs, dampers, and connections [5],[6]. The spring transfers the body's mass by storing energy and assisting in keeping the body apart from road disturbances, while the damper distributes this energy and helps to dampen the oscillations. The three primary types of damping suspension systems—passive suspension system (PSS), semi-active suspension system (SASS), and active suspension system (ASS)—are usually listed in the classification schemes for vehicles [7], [8], and [9]. The least complicated suspension system, the passive suspension has many benefits. The limitation of passive suspension, however, is its inability to eliminate undesired vibration brought on by anomalies in the road [10], [11]. The semi-active suspension thus utilizes a traditional spring and externally controlled damper to produce better performance outcomes [12], [13]. Active suspension systems offer superior handling, road feel, and responsiveness in comparison to passive systems, as well as roll stability and safety with the actuator's force. In addition, in real-time, active suspension systems can modify their dynamic properties in reaction to shifting road conditions.

Based on the information from the many sensors connected to it, the actuator force gives the system a suitable amount of control force. To improve the performance of the active suspension system, the researchers have suggested LQR control methods. Only a few researchers have focused on the entire motion control of a half vehicle, even though many independent suspension experiments have concentrated on simpler two-degree-of-freedom quarter car models. Despite the fact that there have recently been a few studies on full dynamic management of the vehicle, this study carried out using a static dynamics model, and the inputs for road disturbance are sinusoidal and random type road inputs. The objective of the automotive suspension system is to regulate the dynamic tire load with sufficient suspension working space to maximize the vehicle's stability and safety, as well as to isolate the effect of road surface disturbances on passengers to boost ride comfort. Therefore, the half-dynamic model of a vehicle and controller design were the focus of this research to address the aforementioned issues. The various vehicle speeds (20 km/h, 40 km/h, 60 km/h, and 80 km/h) used for improved accuracy. As a result, controller created for a chosen car model that has a sprung and unsprung mass dynamics suspension system.

II. MATHEMATICAL MODELING

For comparison's sake, a thorough mathematical model of the active and passive automotive suspension systems was created in this section. As a result, the mathematical equation of the vehicle model under a dynamic situation is driven in the following part. The model that chosen is described in the table below for numerical purposes.

Table 1 Lists the Model's Characteristic Variables

Model	**Measuring unit**	**Numerical value**
m_s : Vehicle body sprung mass	K g	1794.4
m_{uf} : Unsprung load at the front wheel wheel wheel vehicle body	K g	87.15
mur: Unsprung load at the rear wheel	K g	140.14
c_{sf}: Front suspension damper co-efficient	Ns/m	1190
c_{sr} :Rear suspension damper coefficient	Ns/m	1000
k_{sf} : Stiffness coefficient at the front suspension	N/m	66824
k_{sr} : Stiffness co-efficient at Rear suspension	N/m	18615
k_{uf} : Stiffness coefficient at the front wheel	N/m	101115
k_{ur} : Stiffness co-efficient at the rear wheel	N/m	101115
J_s : Pitch Axis Moment of Inertia	$K\,g\,m^2$	3443.05
L_1 : Length between rear-wheel and center of gravity	M	1.72
L_2 : Length between front-wheel and center of gravity	M	1.27
U_1: Front actuator input		—
U_2 : Rear actuator input		—

Table 2 The Variables Describing the Model Are

Model	**Measuring Unit**
x_s : Chassis vertical displacement at center of gravity	M
x_{sf} : Chassis vertical displacement at front	M
x_{sr} : Chassis vertical displacement at rear position	M
x_{uf} : Front wheels vertical displacement	M
x_{ur}: Vertical displacement of the rear wheels	M
x_{wf} : Road profile for front wheel	M
x_{wr} : Road profile for rear wheel	M

A. Active Suspension System (ASS) Mathematical Modelling

Through a controller chosen for this investigation, the state variable in an active suspension system can be regulated.

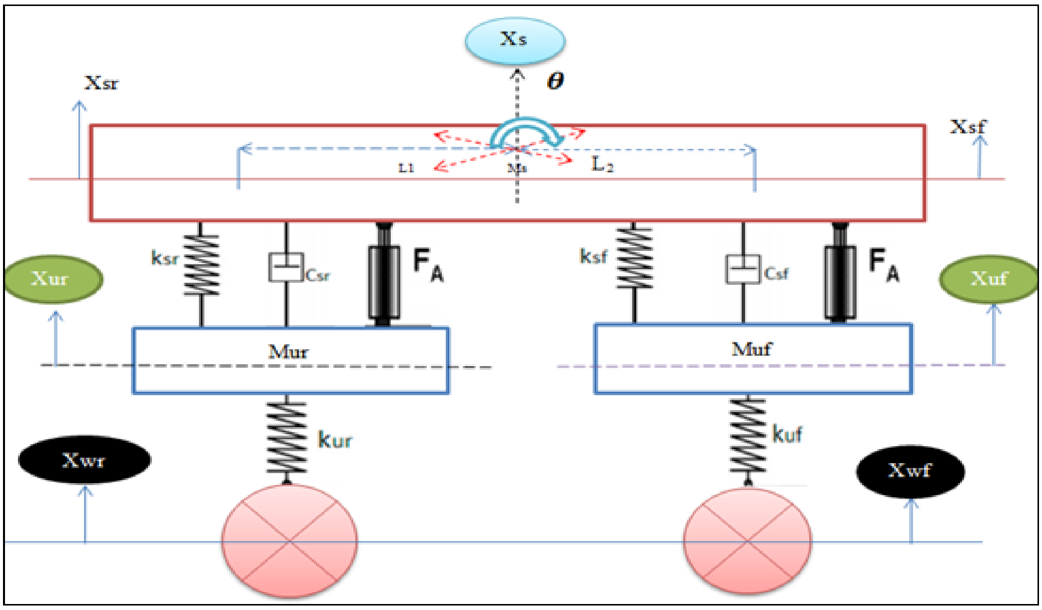

Fig 1 Model of Half Car

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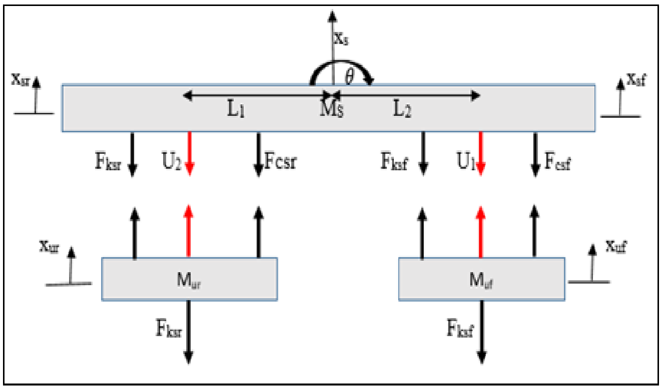


Fig 2 FBD of 4DOF Sprung

The model equation of a sprung mass on the vertical motion at the center gravity point

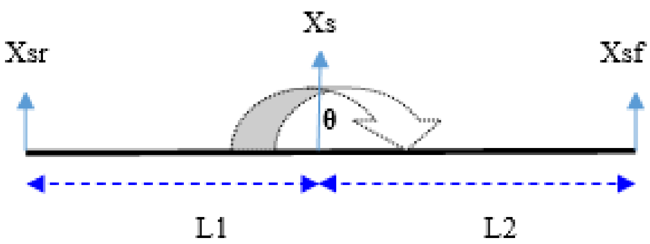
$$\ddot{x}_s = \frac{-\begin{bmatrix} c_{sf}\left(\dot{x}_{sf} - \dot{x}_{uf} - L_2\dot{\theta}\right) + k_{sf}\left(x_{sf} - x_{uf} - L_2\theta\right) \\ + c_{sr}\left(\dot{x}_{sr} - \dot{x}_{ur} + L_1\dot{\theta}\right) + k_{sr}\left(x_{sr} - x_{ur} + L_1\theta\right) \\ - F_{AF} - F_{AR} \end{bmatrix}}{m_s} \quad (1)$$

$$\ddot{\theta} = \frac{\begin{bmatrix} L_2 c_{sf}\left(\dot{x}_{sf} - \dot{x}_{uf} - L_2\dot{\theta}\right) + L_2 k_{sf}\left(x_{sf} - x_{uf} - L_2\theta\right) \\ - L_1 c_{sr}\left(\dot{x}_{sr} - \dot{x}_{ur} + L_1\dot{\theta}\right) - L_1 k_{sr}\left(x_{sr} - x_{ur} + L_1\theta\right) \\ + L_2 F_{AF} - L_1 F_{AR} \end{bmatrix}}{J} \quad (2)$$

$$\ddot{x}_{ur} = \frac{\begin{bmatrix} c_{sr}\left(\dot{x}_{sr} - \dot{x}_{ur} + L_1\dot{\theta}\right) + k_{sr}\left(x_{sr} - x_{ur} + L_1\theta\right) \\ - k_{ur}\left(x_{ur} - x_{wr}\right) - F_{AR} \end{bmatrix}}{m_{ur}} \quad (3)$$

$$\ddot{x}_{uf} = \frac{\begin{bmatrix} c_{sf}\left(\dot{x}_{sf} - \dot{x}_{uf} + L_2\dot{\theta}\right) + k_{sf}\left(x_{sf} - x_{uf} + L_2\theta\right) \\ - k_{uf}\left(x_{uf} - x_{wf}\right) - F_{A} \end{bmatrix}}{m_{uf}} \quad (4)$$

For a control purpose, the state variable should be linear independent. Assume that the pitching dynamics to the forward direction are positive.



From the geometry

$$\theta = \frac{x_{sr} - x_{sf}}{L}, \qquad x_s = \frac{L_2 x_{sr} + L_1 x_{sf}}{L}$$

Where L=L1+L2

B. *An Active Suspension System's State-Space Model*

The state-space form of the model can be expressed as follows.

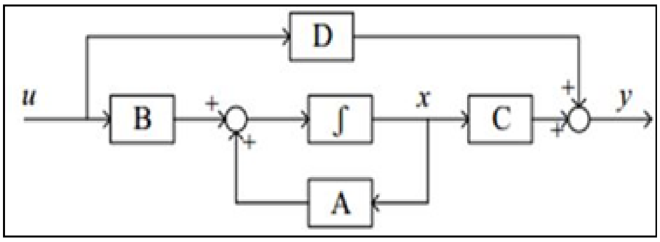


Fig 3 Block Diagram for Representing of the State Space

$$\begin{aligned} \dot{x}(t) &= Ax(t) + BU \\ y(t) &= Cx(t) + DU \end{aligned} \quad (5)$$

Let

$$x_{sr} = x_1, \dot{x}_{sr} = x_2, x_{ur} = x_3, \dot{x}_{ur} = x_4$$

$$x_{sf} = x_5, \dot{x}_{sf} = x_6, x_{uf} = x_7, \dot{x}_{uf} = x_8$$

$$y_1 = \frac{L_2 + L}{L}, y_2 = \frac{L_1 + L}{L}, y_3 = \frac{L_2}{L}, y_4 = \frac{L_1}{L}$$

$$z_1 = \frac{L_1 L_2 m_s - j - L L_2 m_s}{m_s}$$

$$z_2 = \frac{L_1 m_s L - L_1^2 m_s - j}{m_s j}$$

$$z_3 = \frac{L_1 L_2 m_s - j}{m_s j}, z_4 = \frac{L_1^2 m_s + j}{m_s j}$$

$$\dot{x}_2 = \begin{Bmatrix} z_3\begin{pmatrix} c_{sf}\left(y_1 x_6 - y_3 x_2 - x_8\right) \\ + k_{sf}\left(y_1 x_5 - y_3 x_1 - x_7\right) \end{pmatrix} \\ + z_2\begin{pmatrix} c_{sr}\left(y_2 x_2 - y_4 x_6 - x_4\right) \\ + k_{sr}\left(y_2 x_1 - y_4 x_5 - x_3\right) \end{pmatrix} \\ + z_3 u_1 - z_4 u_2 \end{Bmatrix} \quad (6)$$

$$\dot{x}_4 = \frac{\begin{Bmatrix} c_{sr}\left(y_2 x_2 - y_4 x_6 - x_4\right) \\ + k_{sr}\left(y_2 x_1 - y_4 x_5 - x_3\right) \\ - k_{ur}\left(x_3 - x_{wr}\right) - u_2 \end{Bmatrix}}{m_{ur}} \quad (7)$$

$$\dot{x}_6 = \left\{ \begin{array}{l} z_1 \left(\begin{array}{l} c_{sf}\left(y_1 x_6 - y_3 x_2 - x_8\right) \\ + k_{sf}\left(y_1 x_5 - y_3 x_1 - x_7\right) \end{array} \right) \\ + z_2 \left(\begin{array}{l} c_{sr}\left(y_2 x_2 - y_4 x_6 - x_4\right) \\ + k_{sr}\left(y_2 x_1 - y_4 x_5 - x_3\right) \end{array} \right) \\ + z_1 u_1 - z_2 u_2 \end{array} \right\} \quad (8)$$

$$\dot{x}_8 = \frac{\left\{ \begin{array}{l} c_{sf}\left(y_1 x_6 - y_3 x_2 - x_8\right) \\ + k_{sf}\left(y_1 x_5 - y_3 x_1 - x_7\right) \\ - k_{uf}\left(x_7 - x_{wf}\right) - u_1 \end{array} \right\}}{m_{uf}} \quad (9)$$

Based on the physical model of the vehicle in Fig.3, the following state-space equations relate to the suspension system for the LQR control framework.

$$\lfloor A_1 \rfloor = [0 \quad 1 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0]$$

$$\lfloor A_3 \rfloor = [0 \quad 0 \quad 0 \quad 1 \quad 0 \quad 0 \quad 0 \quad 0]$$

$$\lfloor A_5 \rfloor = [0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 1 \quad 0 \quad 0]$$

$$\lfloor A_7 \rfloor = [0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 1]$$

$$\lfloor A_2 \rfloor = \left\{ \begin{array}{c} -z_3 k_{sf} y_3 - z_4 k_{sr} y_2 \\ -z_3 c_{sf} y_3 - z_4 c_{sr} y_2 \\ z_4 k_{sr} \\ z_4 c_{sr} \\ z_3 k_{sf} y_1 + z_4 k_{sr} y_4 \\ z_3 c_{sf} y_1 + z_4 c_{sr} y_4 \\ -z_3 k_{sf} \\ -z_3 c_{sf} \end{array} \right\}^T$$

$$\lfloor A_6 \rfloor = \left\{ \begin{array}{c} z_2 k_{sr} y_2 - z_1 k_{sf} y_3 \\ z_2 c_{sr} y_2 - z_1 c_{sf} y_3 \\ -z_2 k_{sr} \\ -z_2 c_{sr} \\ z_1 k_{sf} y_1 - z_4 k_{sr} y_4 \\ z_1 c_{sf} y_1 - z_2 c_{sr} y_4 \\ -z_1 k_{sf} \\ -z_1 c_{sf} \end{array} \right\}^T$$

$$\lfloor A_4 \rfloor = \left\{ \begin{array}{c} \frac{y_2 k_{sr}}{m_{ur}} \\ \frac{y_2 c_{sr}}{m_{ur}} \\ \frac{-k_{sr} - k_{ur}}{m_{ur}} \\ \frac{-c_{sr}}{m_{ur}} \\ \frac{-y_4 k_{sr}}{m_{ur}} \\ \frac{-y_4 c_{sr}}{m_{ur}} \\ 0 \\ 0 \end{array} \right\}^T, \quad \lfloor A_8 \rfloor = \left\{ \begin{array}{c} \frac{y_3 k_{sf}}{m_{uf}} \\ \frac{y_3 c_{sf}}{m_{uf}} \\ 0 \\ 0 \\ \frac{y_1 k_{sf}}{m_{uf}} \\ \frac{y_1 c_{sf}}{m_{uf}} \\ \frac{-k_{sf} - k_{uf}}{m_{uf}} \\ \frac{-c_{sf}}{m_{uf}} \end{array} \right\}^T$$

$$[B] = \begin{bmatrix} 0 & z_3 & 0 & 0 & 0 & z_1 & 0 & \frac{-1}{m_{uf}} \\ 0 & -z_4 & 0 & \frac{-1}{m_{ur}} & 0 & -z_2 & 0 & 0 \end{bmatrix}^T$$

$$[G] = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{k_{uf}}{m_{uff}} \\ 0 & 0 & 0 & \frac{k_{ur}}{m_{ur}} & 0 & 0 & 0 & 0 \end{bmatrix}^T$$

III. CONTROL SYSTEM DESIGN

Any control system's objective is to identify deviations between the output and the target value and then make the required adjustments to get the output back on track. In this study, the an LQR controllers is used, which is based on its design specifications.

➢ *LQR Controller*

The linear quadratic regulator is a specific kind of optimum control issue that involves the production of performance indices, the minimization of costs or objective functions that are quadratic in both the state and the control [14], [15],[16]. Finding a control vector (U1 and U2) that drives the behavior of the dynamic system to the desired final state while also fulfilling the physical constraints was the aim of optimum control.

$$J = \int_0^\infty \left(x^T Q_x + U^T R U \right) dt \quad (10)$$

Q and R are Hermitian or real symmetric matrices that are both positive definite; Q is also positive-definite (or positive-semi-definite). The matrices Q and R are chosen in order to minimise the cost function "J." To specify the value of the feedback control vector 'u' as -Kx, one uses the state feedback gain matrix K. To get the optimal value of K in this study, the built-in MATLAB programme "K = lqr (A, B, Q, R)" is utilised [17]. Values for the Q and R matrices are chosen iteratively until the intended results for the specified suspension parameters are obtained. We find that the optimal.

K = 1.0e+03 *

Columns 1 Column 8

0.1866-0.0654 -1.5035 -0.0172 -2.2534 -0.9812 2.9981 -0.8697

-2.1776 -1.1260 2.8158 -0.9720 3.5098 0.0176 -1.1198 -0.0213

IV. SIMULATION AND RESULT ANALYSIS

In this section, numerical simulations on the 4 DOF half-vehicle model ASS are carried out to evaluate the effectiveness of the proposed LQR controller. The parameters of the vehicle are shown in Table1 for modeling purposes. The Matlab/Simulink software used to run the simulation.

- *Comparison of ASS and PSS Simulation Performance for Two Bumps Input and Sinusoidal Road*

These simulations intended to compare the performance of the ASS and PSS on two different bump road profiles. Two instances of bump sinusoidal road profiles are shown below, where [12], [17] is the value and h is the amplitude of the bump. It discovered that a sinusoidal bump with an 8 HZ frequency had two bumps.

$$X_{wf(t)} = \begin{cases} h(1-\cos(8\pi t)), 0.5 \le t \le 0.75\sec \\ h(\frac{1-\cos(8\pi t)}{2}), 3.5 \le t \le 3.75\sec \\ 0, otherwise \end{cases} \quad (11)$$

$$X_{wr(t)} = \begin{cases} h(1-\cos(8\pi t)), 3.0 \le t \le 3.25\sec \\ h(\frac{1-\cos(8\pi t)}{2}), 7.25 \le t \le 7.5\sec \\ 0, otherwise \end{cases} \quad (12)$$

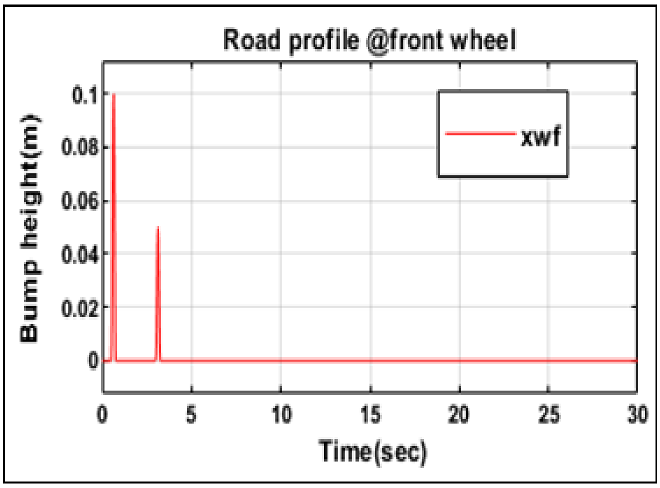


Fig 4 Front-Wheel Two Bump Expected Road Profile Simulation

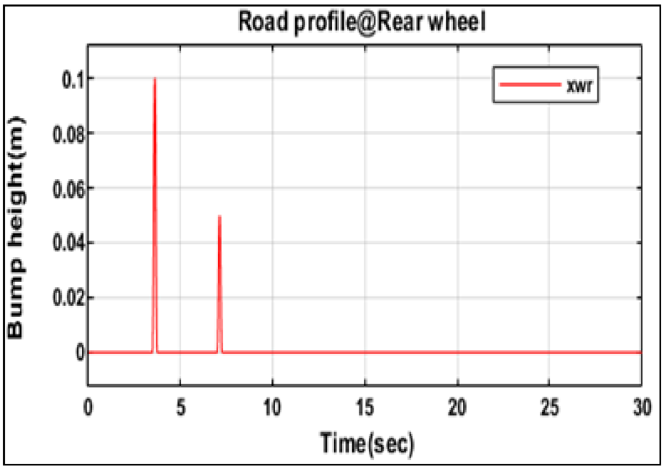


Fig 5 Rear Wheel Two Bump Expected Road Profile Simulation

The classification of road roughness for this study is based on ISO. The type B (good) and type C (average) random road inputs are designed for four-vehicle operating speeds, as it is explained below. These speeds 20 km/hr, 40 km/hr, 60 km/hr, and 80 km/hr help illustrate the effects of road roughness and vehicle speeds, as well as the increase in road handling and passenger ride quality. Using equation 13, the power spectral density for each vehicle's speed is computed.

$$X_{w(t)} = \sqrt{k}\int w(t)dt \quad (13)$$

Where v is the vehicle speed, w (t) white noise, k is the spectral density constant, $k = 4\pi^2 n_o^2 G(n_o)v$

G (no) value for B and C type road input $=16 * 10^{-6} m^3$, $64 * 10^{-6} m^3$ respectively[18] .

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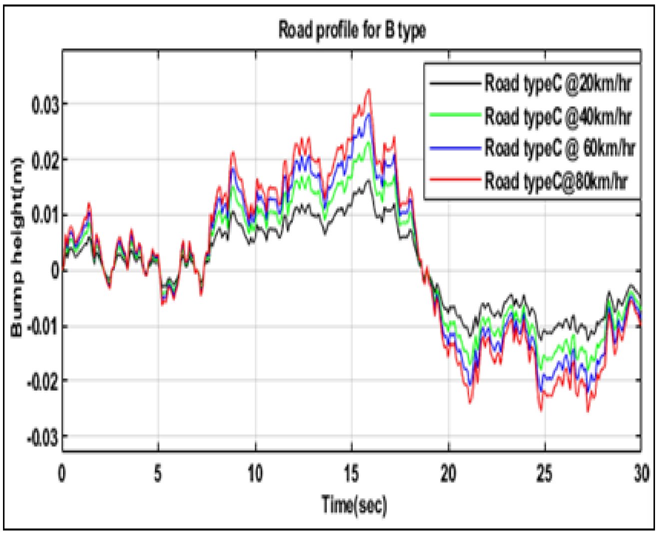

Fig 6 Random Road Profile Types B

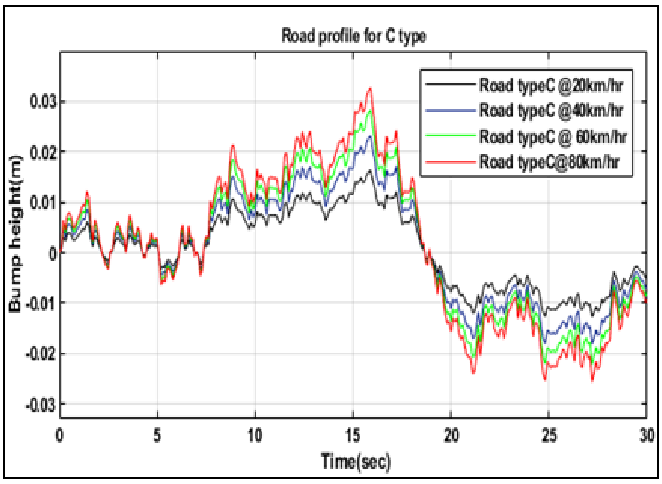

Fig 7 Random Road Profile Types C Selected Vehicle Speed Simulated.

- *Performance Analysis and Comparison the Suspension System with LQR Controller*

In this section, the LQR controller applied simultaneously and the performance comparison shown on the following graph, which obtained from Matlab Simulink.

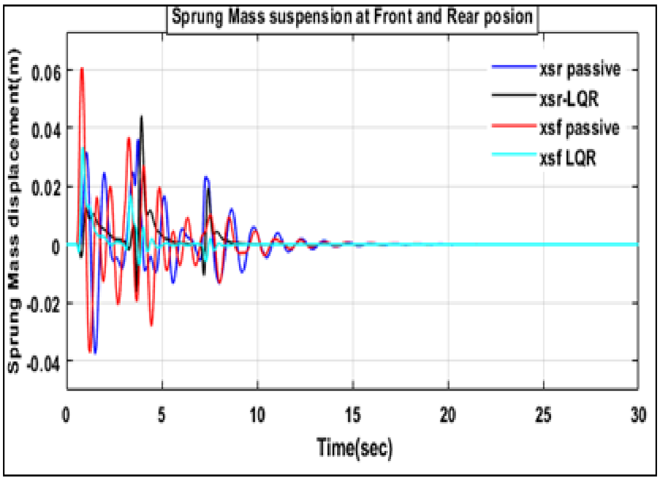

Fig 8 Displacement of the Vehicle's Body in Both its Front and Back Corners.

The outcome of the vehicle's body displacement output for the car's rear and front suspension system, both passive and active. From the simulation the vehicle vertical displacement on the positive peak values reached the passive, and LQR at 0.06 m and 0.031 m, peak value respectively .In the new design, the controller responds when the odd thing happens before it reaches the passengers.

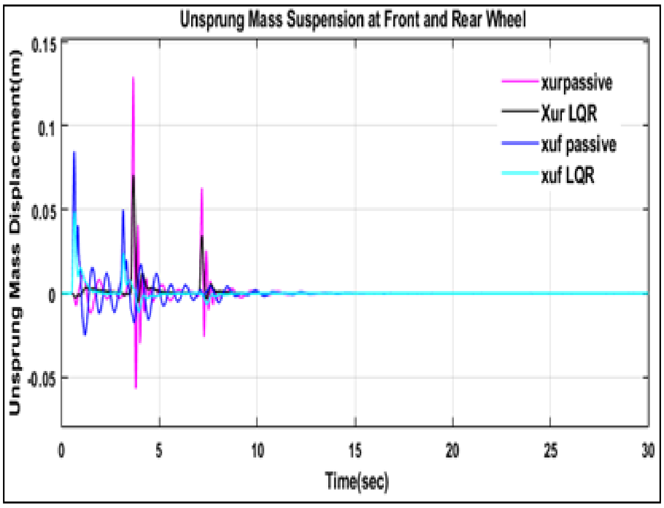

Fig 9 Rear Wheels Displacement Comparison

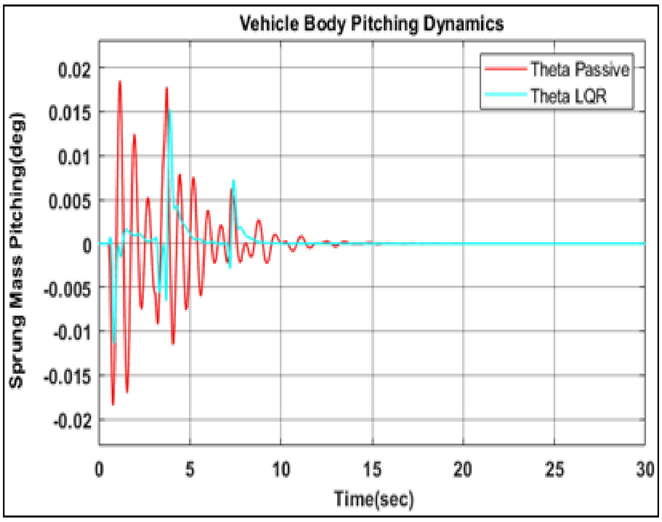

Fig 10 Vehicle Pitching Dynamics on the Lateral Direction

The pitch degree of bump disturbance for passive and active suspension systems showed in Fig 10. The active system with a LQR controller provides a better reaction to the road profile with less vibration and error. As the graph shows, the LQR controller gives a response and saves the system with small pitch in the lateral direction. Whereas in passive, the suspension reaches a maximum compare to active system.

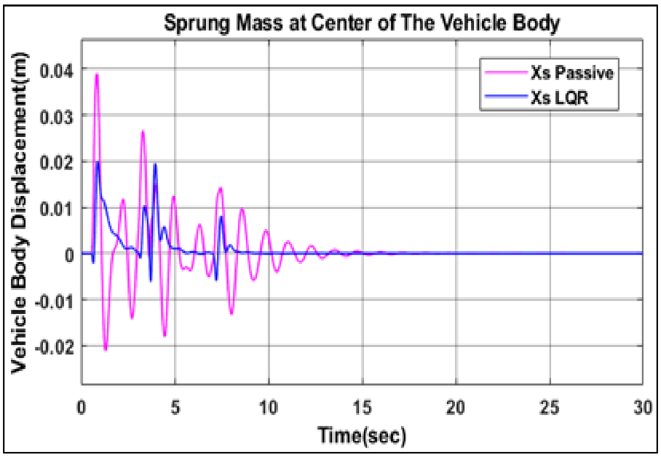

Fig 11 Sprung Mass Displacement Comparison

Fig.11. Indicate that by reducing overshoot and having a modest steady state error at the chosen road disturbance, the suspension system with LQR controller enhanced ride quality.The suspension with no control is not good traction compared with the peak-to-peak value of the new design.

➢ *Simulation Results of the Model on Band C Type Random Road Input at 20, 40, 60, and 80 Km/Hr Vehicle Speeds*

The performance of the PSS and active suspension systems with LQR controller was compared for the states of the vehicle body heave at the centre of gravity, front-wheel displacement, rear-wheel displacement, and pitching displacements on B and C class random road inputs when the vehicle was moving at 20, 40, 60, and 80 km/h. The relationship between the road's input and the segment's vehicle speed simulation is shown in the graph.

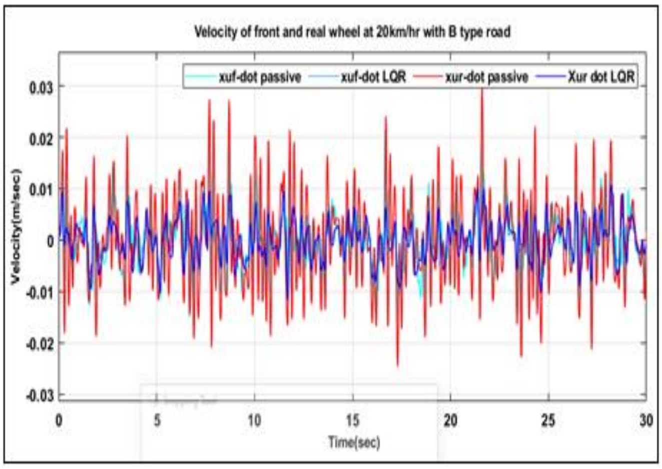

Fig 12 Front and Rear Wheels Velocity at 20km/hr

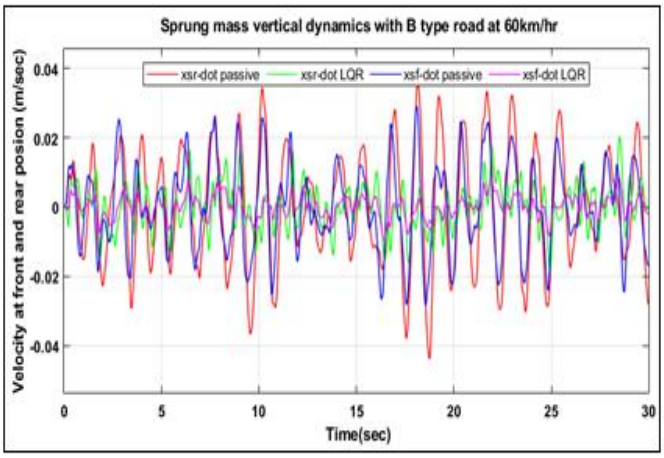

Fig 13 Vehicle Body Velocity of at 60km/hr

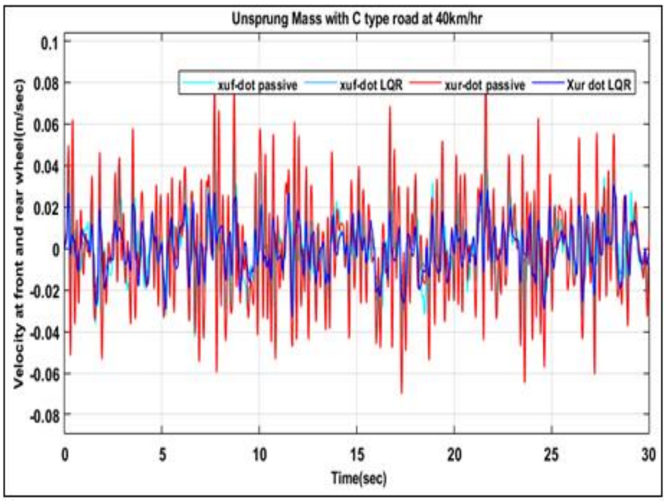

Fig 14 Front and Rear Wheels Velocity at 40km/hr

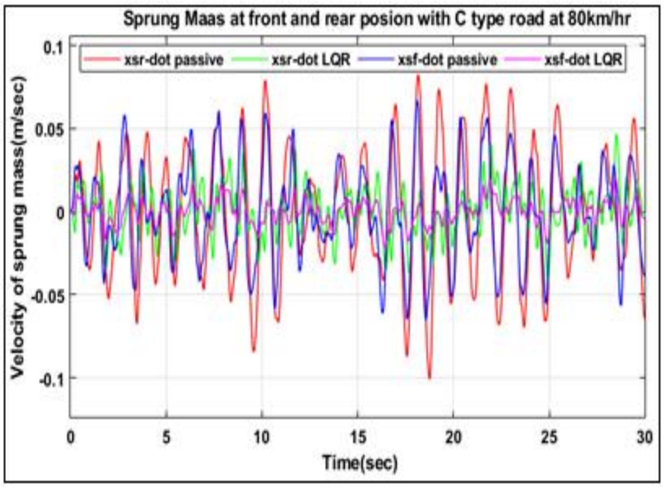

Fig 15 Vehicle Body Velocity at Front and Rear Position at 80km/hr

Fig 12, 13, 14 and 15. Show that by applying LQR controller with random type, road inputs model of half vehicle the velocity of sprung mass and unsprung mass around x and y-axis of the analyzed model was decreased on the pick-to-pick value, and settling time is faster, too.

V. CONCLUSION

The suspension system of a passenger half-car was the subject of this work's modelling and control. a dynamically constructed and derived mathematical model for linear 4DOF vehicle suspension systems. Chosen controller then used after the dynamic model has been created, and its performance is evaluated at various vehicle speeds while being subjected to input disturbances from bumpy roads. The effectiveness and performance of the controller verified by simulation using MATLAB/Simulink.

- In the vertical direction peak to peak, the ASS with LQR controller improves performance. In addition, sprung masses tilting in the lateral direction improved.
- The new design with LQR controller successfully controls the dynamic effect in both expected and random road inputs. The Simulink graph, the controller provides improvement comfort in the vertical and lateral direction, and good handling, at the two wheels.
- The research demonstrates that in controlled ASS, sprung and unsprung mass heave displacements have an amplitude and settling time that is much less than in uncontrolled PSS. Overall, the dynamic modeling of LQR controller of the ASS is efficient and produce good outcomes.

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