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Design of an Active Suspension System using PD Controller

Ahmad Alsultan, Ammar Mohammad, Aryana Ranjbar, Ebrahem Alsaleh, Murat Otkur, Yousef Bosakher
College of Engineering and Technology,
American University of the Middle East, Kuwait
53645@aum.edu.kw, 53468@aum.edu.kw, 38388@aum.edu.kw, 57110@aum.edu.kw,
Murat.Otkur@aum.edu.kw, 61250@aum.edu.kw

Abstract – Since the beginning of the transportation era, whether it was passenger or commercial vehicles, comfort of the passengers and stability of the vehicle have always been a crucial concern. With the evolution of modern vehicles, suspension systems that are responsible for providing passengers with comfortability and stability also evolved. Initially, they were purely mechanical, depending solely on the working principles of springs and dampers. However, the advancement of automotive technology has led to the introduction of more advanced systems like the Active Suspension Systems (ASS). In this study, a model-based design and tuning of an ASS was conducted. The main goal of the study was to achieve the minimum pitch angle, vertical acceleration, and vertical displacement of the car's body for a bump passing manoeuvre. A Half Car Model (HCM) was developed using MATLAB/Simulink software and validated using literature data. Using MATLAB mobile software installed at an iPhone brand mobile phone, a 2020 model year Ford F-150 vehicle was tested considering a bump passing manoeuvre, and the suspension system parameters of the vehicle were determined via matching the vehicle response with the simulation results. A Proportional Derivative (PD) controller was implemented in the model, and its parameters were tuned. A 93.54% improvement in terms of maximum pitch angle, a 96.69% improvement in maximum vertical acceleration, and a 25.17% improvement in the maximum vertical displacement of the vehicle chassis were achieved as a result of this study. To standardize the achievement of the study, a second bump profile that is based on a standard was also implemented in the model. A 94.07% improvement in terms of maximum pitch angle, a 95.99% improvement in maximum vertical acceleration, and a 24.51% improvement in the maximum vertical displacement of the vehicle chassis were achieved as a result the standardized bump passing manoeuvre.

Keywords: Active Suspension System, Half Car Model, MATLAB/Simulink, PD, Pitch Angle, Vertical Acceleration, Vertical Displacement

1. Introduction

The suspension system is a crucial structure that links a vehicle's body to its wheels, allowing controlled movement between them. Its primary function is to prevent distortions caused by road defects, such as potholes and bumps, enhancing ride comfort and handling. A well-designed suspension system significantly impacts passenger comfort and vehicle safety by maintaining stability, reducing rollovers, and improving control during unexpected driving manoeuvres. Conventional suspension systems consist of two key components: a spring, an elastic mechanical component that stores potential energy when deformed, and a damper, also known as a shock absorber, which uses fluid-filled chambers to control the spring's movement and prevent extended oscillations by dissipating kinetic energy.

When modelling a vehicle's suspension system, usually, one of three different models is used: Quarter-Car Model (QCM), Half-Car Model (HCM), or Full-Car Model (FCM). With different numbers of wheels and components included in the model, each of these models reflects a various degree of complexity and enables different degrees of analysis and accuracy depending on the design and goal of the study. Andronic et al. observed the suspension behaviour of a QCM that is developed to study the vehicle's ride behaviour under various road disturbances [1]. The QCM is a basic model, as it includes one sprung mass for the mass of the vehicle body and one unsprung mass for mass of the suspension system and wheel assembly; it is often used during early vehicle development stages. It has two Degrees of Freedom (DOF), with both motions of the masses being translational in the vertical direction. Rahman and Tanvir conducted a study on a HCM, which includes both front and rear axles of the vehicle, meaning that it includes one sprung mass for the mass of the vehicle body and two unsprung masses for mass of the front and rear suspension system and wheel assembly; this introduces pitch motion and provides a 4-DOF, three translational motions in the vertical direction and one rotational motion (pitch). The authors explored the vehicle pitch and vertical behaviour across different road conditions [2]. Rababah and Bhuyan explored the FCM, as it is the most advanced model in which it captures the whole vehicle and includes the roll motion as well, and the authors tested how a vehicle responds to step, bump, and sinusoidal road profiles [3]. This model consists of one sprung mass and four unsprung masses, and it has 7-DOF: five translational motions in the vertical direction for the four unsprung masses and one sprung mass, and two rotational motions (pitch and roll).

Passive, semi-active, and active suspension systems (ASS) are three main types of vehicle suspension systems with respect to ride comfort via control system. A passive suspension system (PSS) relies on a spring and damper, with no adaptability to different road and vehicle conditions. Semi-active systems (SASS) allow real-time changes to damping characteristics, improving ride comfort and handling. Riduan et al. focused on the ASS, which is an electromechanical device responsible for increasing the ride quality by adding an independent force between the sprung and unsprung masses, enhancing stability, passenger comfort, and vehicle handling [4]. This independent force is generated by the actuators and has been applied to the vehicle wheels to control the vertical and the horizontal motion of the vehicle to obtain its stability. Additionally, along with the spring and damper, the ASS uses controls to predict the road conditions and performs the necessary adjustments [5].

There are many control strategies and algorithms that can be implemented in the ASS, such as Proportional-Integral-Derivative (PID), Model Predictive Control (MPC), and so on. Each of these strategies has a different and unique way to enhance the performance of the vehicle depending on the employed algorithm. Talib and Mat Darus examined the effect of implementing a PID controller on a suspension system. The system uses P (proportional) to deal with present errors, I (integral) for accumulated past errors, and D (derivative) for predicting future error trends. They found that this controller enhanced the performance of the suspension by using the three modes to calculate the error between the desired setpoint and the process variable used as feedback. The result of the calculation has been applied to a FCM to calculate the force needed to generate better system stability [6]. Göhrle et al. investigated the applications of an MPC controller for an ASS. The controller uses a mathematical model to predict the future behaviour of the system to apply the required inputs to the system in a short period of time. Where the system applies and repeats an approach called the receding horizon approach. This approach will minimize the cost function of the prediction horizon by calculating the best set of actions and applying only the first control input to the real system. [7].

Over the past decades, engineers and researchers have conducted numerous studies on ASS. Some of the studies were conducted on a QCM, such as the study that was conducted by Kuber in 2014, where a PID controller was implemented in a QCM for an ASS that was developed using MATLAB/Simulink software. The implementation of PID control resulted in a 92% reduction in setting time and a 55% decrease in vertical body displacement compared to the PSS, which indicates the effectiveness of the PID controller in providing ride comfort and vehicle stability [8]. In a later study, Rodriguez-Guevara et al. implemented a dual-controller strategy by combining MPC and LQR within a Linear Parameter Varying (LPV) framework for an ASS employed at a QCM. The authors observed a 22.89% decrease in maximum chassis displacement, a 23.49% decrease in maximum suspension displacement, and a 9.61% decrease in maximum chassis acceleration under different road disturbances [9]. Other studies were conducted on an HCM, such as the study that was conducted by Avesh and Srivastava in 2012, where a PID controller was employed on an HCM using MATLAB/Simulink software to improve the performance of the suspension system. The authors observed an improvement in performance, with vertical displacement, settling time, and velocity responses improved by 89%, 92%, and 58%, respectively, between PSS and ASS [10]. In 2018, Al-Ghanim and Nassar developed an HCM and implemented three control strategies: PID, FLC, and Fuzzy-PID using MATLAB/Simulink software. The study indicated that all active controllers are better than the PSS in reducing body displacement, acceleration, and overshoot. Among them, the Fuzzy-PID controller provides the best performance, achieving better dynamic response [11].

The paper is structured as follows: the first section comprises the introduction and literature review. The second section contains the methodology employed in this study. The third section includes the modelling process. The fourth section presents the study's findings and discussions, while the fifth section offers conclusions and recommendations for further studies.

2. Methodology

The methodology flow chart for the project is illustrated in Figure 1. An HCM using MATLAB/Simulink software was developed. A validation of the model was performed using step input response data obtained from a previous academic study [12]. MATLAB mobile software was installed on an iOS mobile phone, and the gyroscopic sensor readings of the mobile phone provided by the app was used to obtain the vehicle measurements during bump passing manoeuvre. Thereafter, the model parameters, such as spring and damping coefficients, were tuned to match the vehicle measurements in order to establish the parameters of the vehicle's suspension. A PD controller was implemented, and the controller's parameters were tuned to minimise the vehicle pitch angle, vertical acceleration, and vertical displacement.

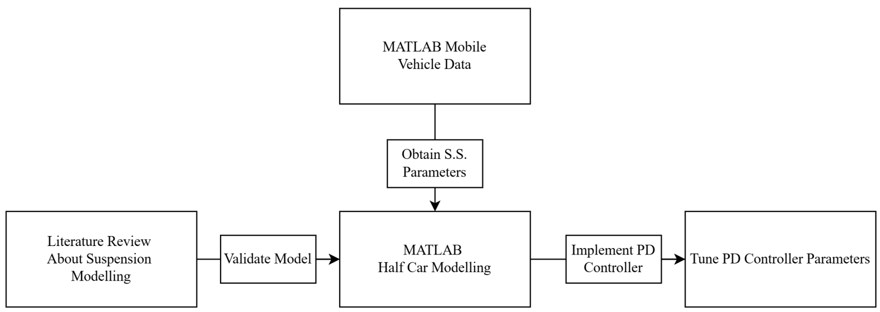


Fig. 1: Flow Chart of the Conducted Study

3. Modelling Process

3.1. Schematic Diagram, Free Body Diagrams, and Equations of Motion

The schematic diagram of the HCM that is shown in Figure 2 comprises three masses: two unsprung masses (representing the suspension systems and wheel assembly for front and rear axles) and one sprung mass (representing the vehicle body and chassis). The geometry of these masses was neglected. The system incorporates two force actuators, represented in red, along with four dampers and four springs. The definition of each parameter used at the model is provided in Table 1. The model comprises 4-DOF, three of which denote the translational motion in the vertical direction of the masses, and one relates to the rotational motion (pitch). Both translational and rotational Free Body Diagrams (FBDs) are provided in Figure 3 and the Equations of Motions (EOMs) that were developed using the FBD are provided in Equations (1) - (4). The FBD was sketched, and the EOM were developed under the assumption that the displacement of the vehicle body mass is greater than the displacement of both unsprung masses which is greater than the the displacement of the road surface.

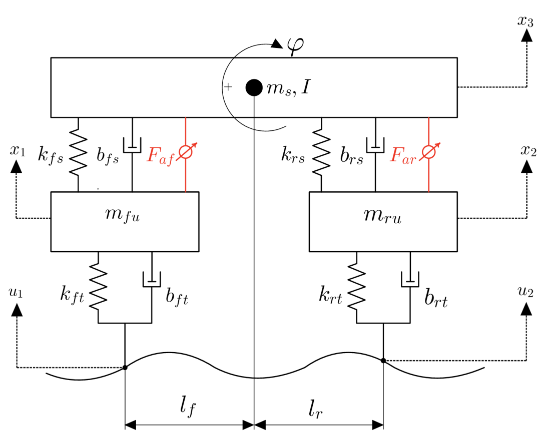


Fig. 2: Schematic Diagram of the Half-Car Model

Table 1: Definition of the Parameters used in the Model

Parameter	Unit	Definition	Parameter	Unit	Definition
m_s	kg	Sprung mass (chassis)	b_{fs}	N.s/m	Front suspension system damping coefficient
m_{fu}	kg	Unsprung mass of the front axle	b_{rs}	N.s/m	Rear suspension system damping coefficient
m_{ru}	kg	Unsprung mass of the rear axle	b_{ft}	N.s/m	Front tire damping coefficient
I	$Kg.m^2$	Moment of inertia of the vehicle body and chassis	b_{rt}	N.s/m	Rear tire damping coefficient
k_{fs}	N/m	Front suspension system spring coefficient	l_f	m	Distance between center of gravity and front axle
k_{rs}	N/m	Rear suspension system spring coefficient	l_r	m	Distance between center of gravity and rear axle
k_{ft}	N/m	Front tire spring coefficient	F_{af}	N	Front actuator force
k_{rt}	N/m	Rear tire spring coefficient	F_{ar}	N	Rear actuator force

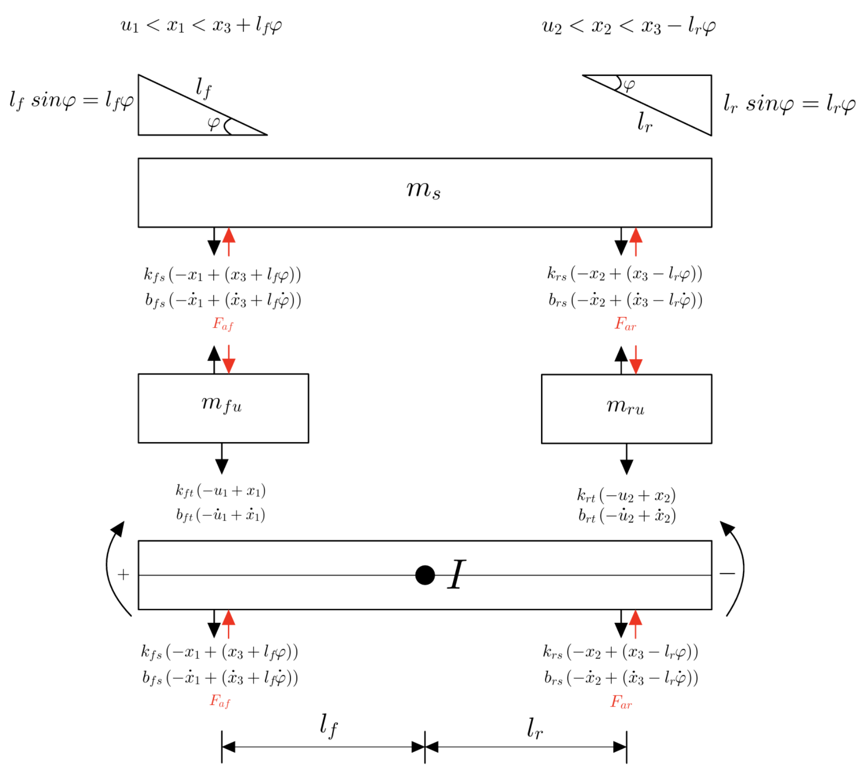


Fig. 3: Free Body Diagrams of the Half-Car Model

$$m_s \times \ddot{x}_3 - k_{fs}(x_3 + l_f\varphi - x_1) - b_{fs}(\dot{x}_3 + l_f\dot{\varphi} - \dot{x}_1) + F_{af} - k_{rs}(x_3 - l_r\varphi - x_2) - b_{rs}(\dot{x}_3 - l_r\dot{\varphi} - \dot{x}_2) + F_{ar} = 0 \quad (1)$$

$$m_{fu} \times \ddot{x}_1 + k_{fs}(x_3 + l_f\varphi - x_1) + b_{fs}(\dot{x}_3 + l_f\dot{\varphi} - \dot{x}_1) - F_{af} - k_{ft}(x_1 - u_1) - b_{ft}(\dot{x}_1 - \dot{u}_1) = 0 \quad (2)$$

$$m_{ru} \times \ddot{x}_2 + k_{rs}(x_3 - l_r\varphi - x_2) + b_{rs}(\dot{x}_3 - l_r\dot{\varphi} - \dot{x}_2) - F_{ar} - k_{rt}(x_2 - u_2) - b_{rt}(\dot{x}_2 - \dot{u}_2) = 0 \quad (3)$$

$$I\times \ddot{\varphi} - k_{fs}\times l_f\left(x_3 + l_f\varphi - x_1\right) - b_{fs}\times l_f\left(\dot{x}_3 + l_f\dot{\varphi} - \dot{x}_1\right) + F_{af}\times l_f + k_{rs}\times l_r\left(x_3 - l_f\varphi - x_2\right) + b_{rs} \times l_r\left(\dot{x}_3 - l_f\dot{\varphi} - \dot{x}_2\right) - F_{ar}\times l_r = 0 \quad (4)$$

where $\dot{x}$, $\ddot{x}$, $\dot{\varphi}$, and $\ddot{\varphi}$ are vertical velocity, vertical acceleration, angular velocity, and angular acceleration respectively.

3.2. Final Model

Figure 4 illustrates the final model developed via MATLAB/Simulink software. The model was constructed as subsystems that represent distinct elements. For example, each mass in the model is illustrated as a subsystem. In addition, all connections are made as a labeled signal, and each color represents a distinct group of elements.

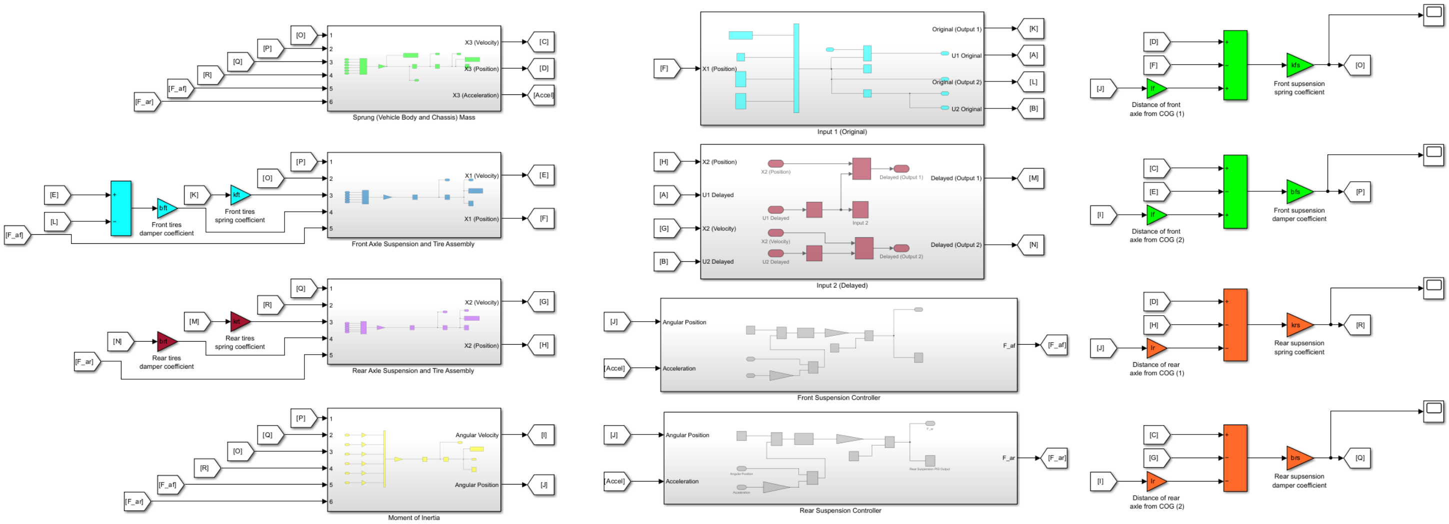
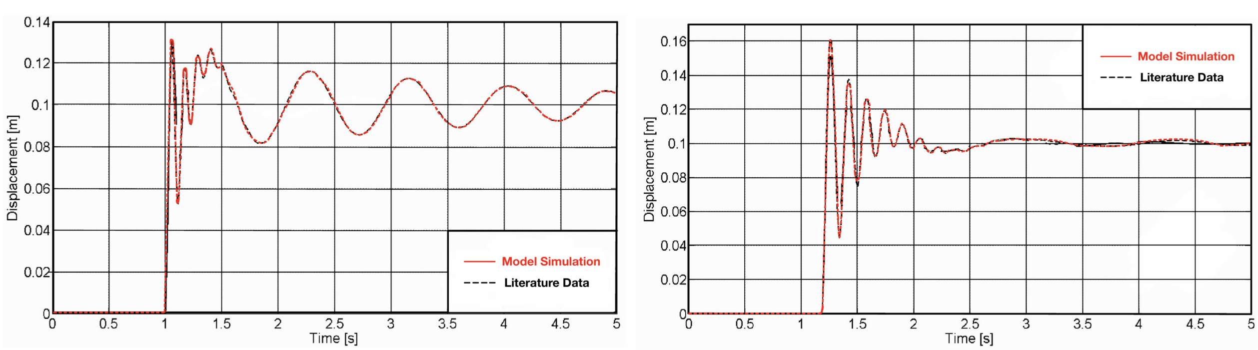


Fig. 4: Final MATLAB/Simulink HCM

3.3. Model Validation

The HCM was developed using MATLAB/Simulink software to solve the EOM (1) - (4). The HCM was validated using data obtained from a previous academic study [12]. The study involved passing the vehicle over a bump with a height of 0.1 m (step input / input 1) and a speed of 60 km/h. Figures 5 (a and b) show the displacement of the front and rear axle masses with respect to time for both the literature data (black) and the model simulation (red) results. Model simulation results match exactly with the literature date proving the validation of the model.



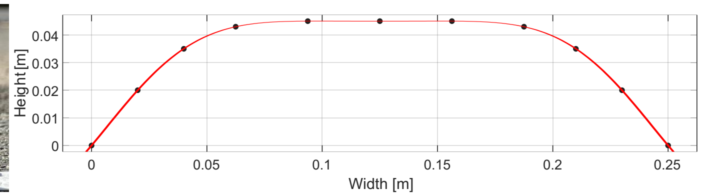
a (Front Axle Mass) b (Rear Axle Mass)

Fig. 5: Displacement of Axle Masses vs. Time (Literature Data & Model Simulation)

3.4. Vehicle Measurements & Specifications

The goal of the study is to design an ASS for a passenger car that will minimize pitch angle, vertical acceleration, and vertical displacement for a bump passing manoeuvre for the bump depicted in Figure 6 (a). A 2020 Ford F-150 SuperCrew (short bed) vehicle has been used for the bump passing manoeuvre measurement with MATLAB mobile software sensor installed on an iOS mobile phone with a sampling rate of 50 Hz. Important vehicle parameters and speed bump specifications are shown in Table 2. The speed bump profile measurements, including height and width at various points, were taken manually. By using MATLAB's "Curve Fitter tool" and Fourier series approximation, a curve that represents the speed bump sectional profile was obtained as shown in Figure 6 (b). To increase the accuracy of the approximation, a 4-term Fourier series was used. The mathematical equation that represents the curve is provided with Equation (5).

a (Physical Representation of the Speed Bump)



b (Graphical Representation of the Speed Bump)

Fig. 6: Speed Bump Sectional Profile (Input 3)

Table 2: Important Vehicle Parameters and Speed Bump Specifications

Parameters	Unit	Value
Vehicle Total Length	m	5.89
Vehicle Width	m	2.03
Vehicle Height	m	1.96
Vehicle Wheelbase	m	3.69
Vehicle Total Weight	kg	2500
Speed Bump Height	m	0.045
Speed Bump Width	m	0.25

$$\text{Bump Height}\ (y) = 0.0368 - 0.0119\sin(12.5664x) - 0.0435\cos(25.1328x) + 0.0302\sin(37.6992x) + 0.0068\cos(50.2656x) \tag{5}$$

where x is the horizontal position from the reference point.

3.5. Determining Suspension System Parameters

Given a driver mass of 100 kg, the vehicle's total weight during the test was 2600 kg. The front suspension and wheel assembly was assumed to be 120 kg, while the rear one is 180 kg, meaning that the sprung mass was equal to 2300 kg. The distance between the front axle and the center of gravity of the vehicle where the mobile phone was placed (l_f) is 1.29 m, while the distance between the rear axle and the center of gravity of the vehicle (l_r) is 2 m. Suspension system parameters including kf_s, k_{rs}, k_{ft}, k_{rt}, b_{fs}, b_{rs}, b_{ft}, b_{rt}, and I were tuned to match the pitch angle simulation with vehicle's pitch angle measurements within an acceptable limit of accuracy as shown in Figure 7 where the black line represents the vehicle measurements and the red line represent the simulation. Table 3 provides the tuned parameters for the suspension system and vehicle inertia.

3.6. Controller Parameters Tuning

MATLAB PID controller block is used for the ASS. A setpoint value of zero radian for pitch angle and zero m/s^2 for vertical acceleration of the vehicle body and chassis mass is used as the reference for the PID. Vehicle acceleration is divided by 100 to reduce the magnitude signal to a reasonable order that is comparable with the pitch angle value in radian. The (I) gain of the PID is set to zero as through the trials even without the (I) gain, no steady state error is observed for the simulation. The tuning process of the PID controller parameters was made manually, and the Table 4 summarizes the results in terms of the maximum pitch angle (M.P.A.) with the unit of degrees, the maximum vertical acceleration (M.V.A.) with the unit of m/s^2, and the maximum position of the combination of the vehicle body and chassis mass with the unit of meters. The values of the controller parameters that resulted in the minimum value of the maximum vertical acceleration as it is an important factor for ensuring driving comfort was selected for the model. (P) gain was selected as 200 while (D) gain was 150.

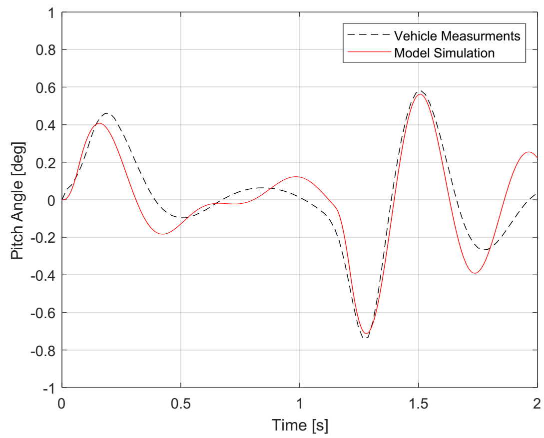


Fig. 7: Tuning of Suspension System Parameters Process

Table 3: Tuned Suspension System Parameters

Parameters	Value	Unit
I	4000	$Kg.m^2$
k_{fs}	300000	N/m
k_{rs}	250000	N/m
k_{ft}	150000	N/m
k_{rt}	150000	N/m
b_{fs}	25000	N.s/m
b_{rs}	25000	N.s/m
b_{ft}	500	N.s/m
b_{rt}	500	N.s/m

3.7. Alternative Speed Bump Profile [AS/NZS 2890.1:2004]

To test and validate the developed model under a standardized road condition, another speed bump profile (input 2) was implemented in the model. The Australian/New Zealand Standard (AS/NZS 2890.1:2004) provided the geometry for the bump profile [13], where the speed bump has a height of 0.05 [m] and a width of 0.35 [m], and to generate the curve representing the bump profile (Figure 8).

Table 4: Results of Tuning the PID Controller Parameters

	P = 50	P = 100	P = 150	P = 200
D = 50	M.P.A. = 0.1177 [deg.] M.V.A. = 0.8011$[m/s^2]$ M.P. = 0.0749 [m]	M.P.A. = 0.1141 [deg.] M.V.A. = 0.3524 $[m/s^2]$ M.P. = 0.0248 [m]	M.P.A. = 0.1107 [deg.] M.V.A. = 0.2366 $[m/s^2]$ M.P. = 0.0134 [m]	M.P.A. = 0.1075 [deg.] M.V.A. = 0.2276 $[m/s^2]$ M.P. = 0.0075 [m]
D = 100	M.P.A. = 0.0682 [deg.] M.V.A. = 0.438 $[m/s^2]$ M.P. = 0.0607 [m]	M.P.A. = 0.0669 [deg.] M.V.A. = 0.2359 $[m/s^2]$ M.P. = 0.0367 [m]	M.P.A. = 0.0657 [deg.] M.V.A. = 0.1283 $[m/s^2]$ M.P. = 0.0222 [m]	M.P.A. = 0.0645 [deg.] M.V.A. = 0.1256 $[m/s^2]$ M.P. = 0.0132 [m]
D = 150	M.P.A. = 0.0478 [deg.] M.V.A. = 0.2067 $[m/s^2]$ M.P. = 0.0212 [m]	M.P.A. = 0.04772 [deg.] M.V.A. = 0.1488 $[m/s^2]$ M.P. = 0.0167 [m]	M.P.A. = 0.0466 [deg.] M.V.A. = 0.1097 $[m/s^2]$ M.P. = 0.0134 [m]	M.P.A. = 0.046 [deg.] M.V.A. = 0.0865 $[m/s^2]$ M.P. = 0.011 [m]
D = 200	M.P.A. = 0.0368 [deg.] M.V.A. = 0.1936 $[m/s^2]$ M.P. = 0.023 [m]	M.P.A. = 0.0364 [deg.] M.V.A. = 0.1469 $[m/s^2]$ M.P. = 0.019 [m]	M.P.A. = 0.036 [deg.] M.V.A. = 0.1136 $[m/s^2]$ M.P. = 0.0158 [m]	M.P.A. = 0.0357 [deg.] M.V.A. = 0.0896 $[m/s^2]$ M.P. = 0.0134 [m]

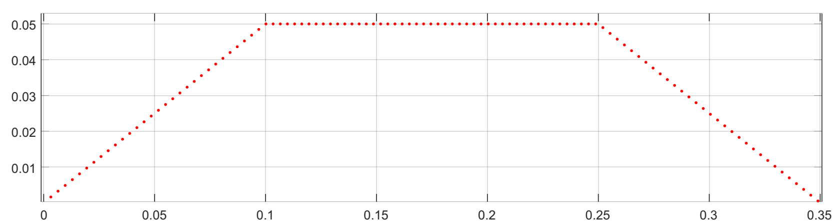
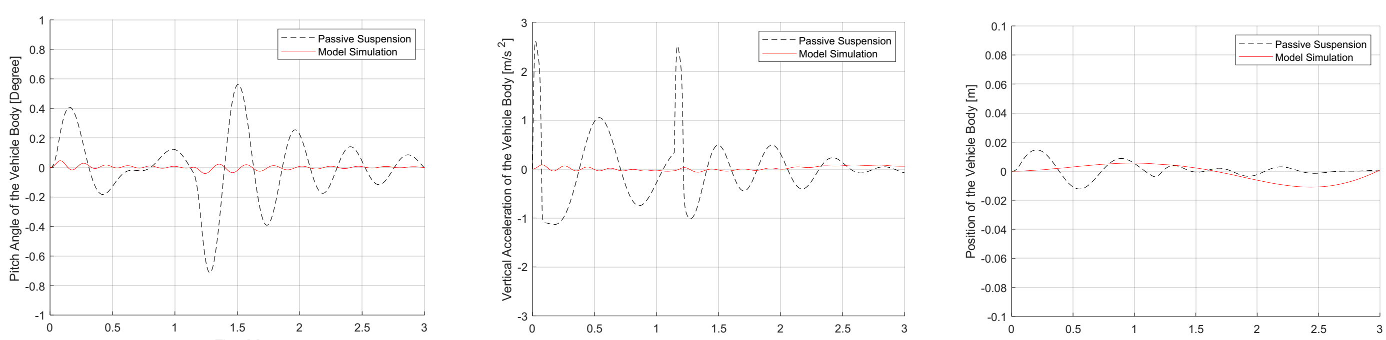


Fig. 8: Graphical Representation of the Standardized Speed Bump Sectional Profile

4. Results & Discussions

Tables 5 and 6 present a comparison between the passive suspension system and the ASS developed in the study. The comparison encompasses the maximum pitch angle, maximum vertical acceleration, and maximum vertical position of the vehicle chassis under two different bump passing manoeuvres, as these parameters are critical in terms of driving comfort. The results indicate a significant improvement in all parameters. Figures 9 (a, b, and c) and Figures 10 (a, b, and c) illustrate a graphic representation of the results. The project has achieved its goal, as evidenced by the reduction in maximum pitch angle, vertical acceleration, and vehicle body position.

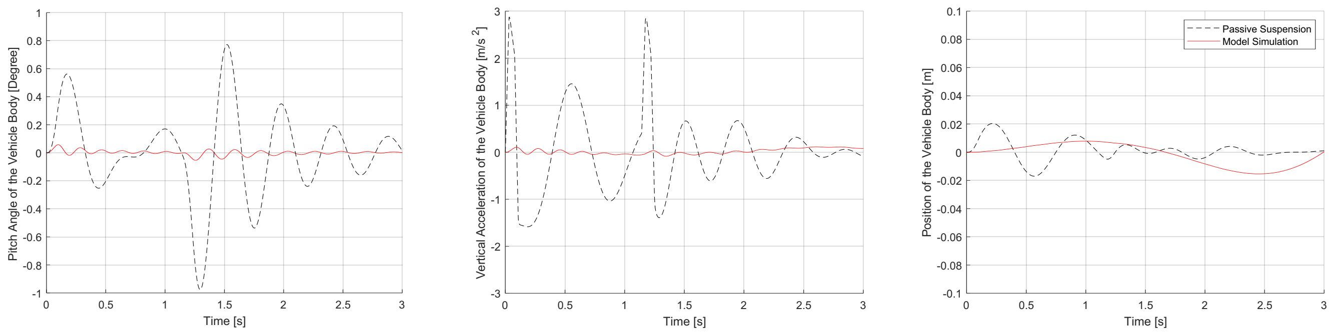


a (Pitch Angle) b (Vertical Acceleration) c (Vertical Position)

Fig. 9: Comparison between Passive Suspension System and the Results of the Model Simulation (Input 3)

Table 5: Results of the Study in Improving Driving Comfort (Input 3)

	Passive Suspension System	Model Simulation (ASS)	Improvement
Maximum Pitch Angle	0.7126 [Degree]	0.046 [Degree]	93.54%
Maximum Vertical Acceleration	2.6135 [m/s^2]	0.0865 [m/s^2]	96.69%
Maximum Vertical Displacement	0.0147 [m]	0.011 [m]	25.17%



a (Pitch Angle) b (Vertical Acceleration) c (Vertical Position)

Fig. 10: Comparison between Passive Suspension System and the Results of the Model Simulation (Input 2)

Table 6: Results of the Study in Improving Driving Comfort (Input 2)

	Passive Suspension System	Model Simulation (ASS)	Improvement
Maximum Pitch Angle	0.9763 [Degree]	0.0579 [Degree]	94.07%
Maximum Vertical Acceleration	2.8747 [m/s^2]	0.1153 [m/s^2]	95.99%
Maximum Vertical Displacement	0.0204 [m]	0.0154 [m]	24.51%

5. Conclusion & Future Work

The goal of the study was to design an ASS to provide a comfortable drive for both the driver and passengers for two different bump passing manoeuvres via implementing a PD controller at an HCM developed with MATLAB/Simulink software. The study resulted in a 93.54% reduction in terms of maximum pitch angle, a 96.69% improvement in reduction of vertical acceleration, and a 25.17% reduction in the maximum vertical position of the vehicle chassis for one of the bump passing manoeuvres, while it resulted in a 94.07% reduction in terms of maximum pitch angle, a 95.99% improvement in reduction of vertical acceleration, and a 24.51% reduction in the maximum vertical position of the vehicle chassis for the other bump passing manoeuvre, which were achieved by implementing a PD controlling strategy. For future work, other control algorithm such as MPC, will be implemented, and performance comparison of PD and MPC controllers will be analysed.

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