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Modeling and Analysis of an Active Suspension System using Half Vehicle Model with Road Disturbance

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Abstract

This research focuses on the development and analysis of a mathematical model for a half vehicle suspension system. The model incorporates a six degree of freedom system, including the heave motion of wheels, car body, and seats with a seated passenger, as well as the pitch motion of the car body. The effect of road disturbances on the seat and the resulting impact on the passenger are also considered. Using Newton's second law of motion, the characteristic equation of the system is derived, and a Simulink model is constructed based on the derived equations. The model is then tested under two different scenarios: a low residential speed limit of 20 km/hr and a highway speed limit of 80 km/hr. Various characteristics such as settling time, rise time, and percent overshoot are observed and compared. It is found that when the speed of the vehicle increases, such as when passing over speed bumps or encountering road disturbances, the settling time and percent overshoot for all components of the vehicle (wheels, chassis, and seat) also increase. This can lead to riding discomfort, road accidents, and adverse effects on the vehicle's health, including permanent damage and deformation of the shock absorber system. To address these issues, an analysis of both active and passive suspension systems is conducted. The research demonstrates significant improvements when an active system is controlled with a PID controller. By optimizing the system parameters, such as decreasing the proportional gain for higher speeds, the settling time and percent overshoot can be effectively reduced, mitigating riding discomfort and enhancing overall system performance.

Keywords: *Half Car Model, Active Suspension System, PID Controller, Ziegler Nichols Method.*

Introduction

A vehicle suspension system refers to the collection of components and mechanisms that connect a vehicle's frame or body to its wheels. The primary purpose of a suspension system is to provide a smooth and comfortable ride for the occupants of the vehicle by minimizing the impact of bumps and vibrations from the road surface. It also helps maintain optimal tire contact with the road, ensuring better handling and

control. The suspension system provides improved riding comfort, enhances road handling, and contributes to the stability of the vehicle (Kashem et al., 2015).

The design and arrangement of suspension components can vary depending on factors such as the vehicle type, intended use, and manufacturer preferences. However, in general, a suspension system consists of springs, dampers (shock absorbers), and linkages that connect the vehicle body to the wheels. In essence, the suspension system functions as a mechanism that isolates the car body from the wheels. Its main objective is to minimize the acceleration transmitted to the passengers, thereby enhancing riding comfort (Al-Ghanim & Nassar, 2018).

There are three main types of motion in a vehicle: pitch motion, heave motion, and roll motion, as depicted in Figure 1. Heave motion refers to the vertical up and down motion caused by road disturbances. Roll motion pertains to the angular motion in the forward direction, while pitch motion is the angular motion resulting from the coupled effect of axle forces from the front and rear wheels. Figure 1 illustrates the motion of a car.

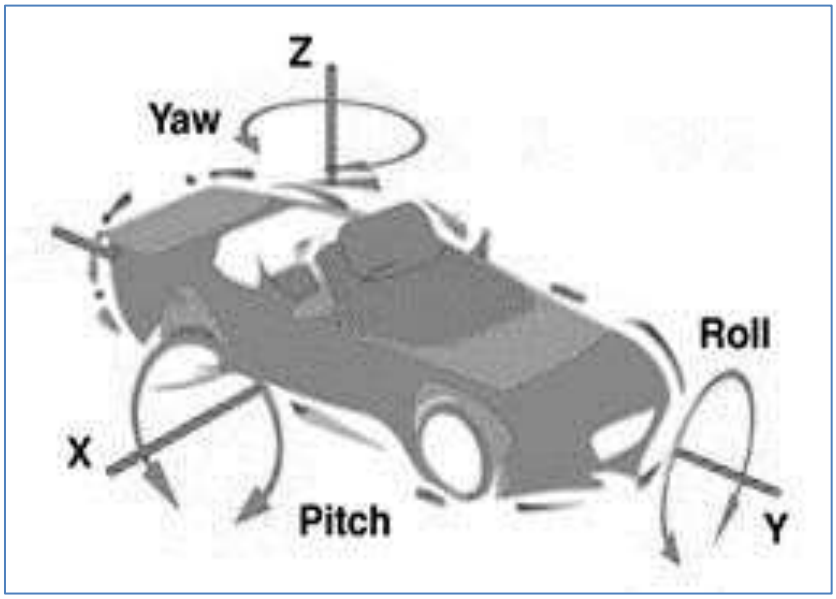

Figure 1: Different Type of Motion in a Vehicle (R.N, 2014)

Similarly, vehicle suspension systems can be broadly categorized into three types: passive suspension systems, semi-active suspension systems, and active suspension systems (Avesh & Srivastava, 2012).

A passive suspension system consists of a conventional shock absorber that comprises springs and dampers. The springs store energy, while the dampers dissipate it. The passive suspension system operates as an open loop, and its parameters are fixed.

The characteristics of a passive system are fixed and do not involve any active mechanical components. It provides a certain level of compromise between road handling, riding comfort, and load carrying. The main disadvantage of this system is that if the suspension is too firm, it may not perfectly stabilize the vehicle according to road inputs. On the other hand, if it is too flexible or soft, the vehicle may become unstable and exhibit swing motion (Gowda & Chakrasali, 2014).

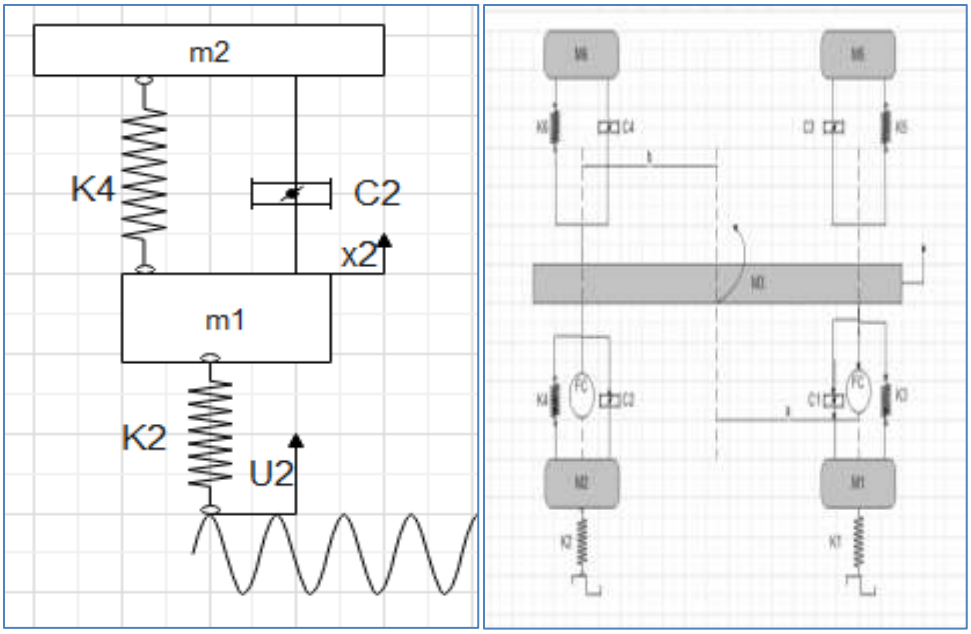

Figure 1: (a) Quarter Vehicle Suspension System Model, and (b) Half Vehicle Active Suspension System Model

A semi-active suspension system consists of a conventional spring and a variable damper. Two types of variable dampers are commonly used: Magnetorheological dampers and Orifice-controlled dampers. In Magnetorheological dampers, variable damping is achieved by exposing the fluid to a variable magnetic field. In Orifice-controlled dampers, variable damping is obtained by electronically controlling the solenoid valve in the orifice (Bakar et al., 2015).

Lastly, the active suspension system consists of springs, dampers, and an external actuator that induces a controlling force when needed. The active suspension system operates in a closed-loop system that senses road inputs and provides actuation or controlling force as required. The active suspension system can sometimes store energy, dissipate energy, or induce energy into the system (Bhanu Chander, 2009).

Mathematical Modelling of Vehicle Suspension Systems

Vehicle suspension systems require three different types of vehicle modeling: quarter, half, and full vehicle modeling. The Quarter Car model is frequently used to model the suspension system as it captures many characteristics of the vehicle body. However, it only considers heave motion while neglecting roll and pitch motion of the body, limiting the model to a two-degree-of-freedom system (Ghazaly et al., 2014). Half Car modeling is widely utilized as it provides a closer approximation to full car modeling. The model treats each wheel as a separate entity and the car body as a single entity. It considers heave motion for both wheels and incorporates both heave and pitch motion for the body, resulting in a four-degree-of-freedom system. Full Car Modeling provides a comprehensive depiction of vehicle motion. It introduces roll, pitch, and heave motion for the vehicle body, as well as roll and heave motion for the wheels. However, analyzing this system can be tedious and time-consuming. Therefore, researchers often prefer the half vehicle modeling approach to represent the vehicle's motion (Darus & Sam, 2009).

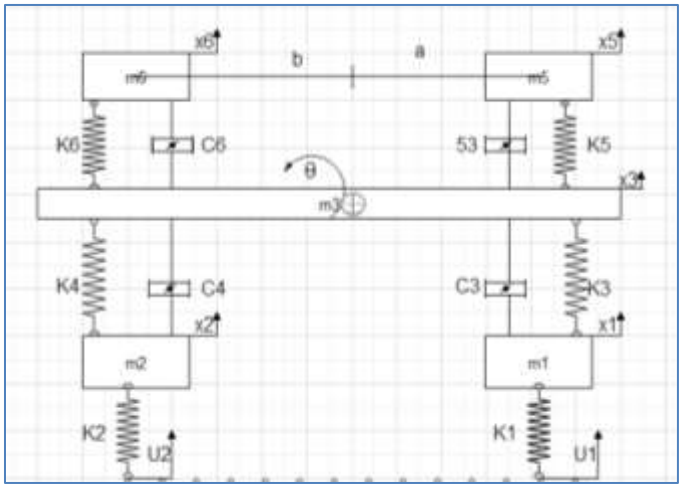

Figure 3: Half Vehicle Passive Suspension System Model Including Seat Model

Half Passive Suspension System Model Including Seat Model

The mathematical model of half vehicle model passive suspension system is shown in Figure 3 including weight of passenger as derived according to the newton second law of motion given below (Rao, 2001);

1. $m_1\ddot{X}_1 + K_1(X_1 - U_1) - K_3(X_3 - X_1 + a\theta) - C_3(\dot{X}_3 - \dot{X}_1 + a\dot{\theta}) = 0$
2. $m_2\ddot{X}_2 + K_2(X_2 - U_2) - K_4(X_3 - X_2 - b\theta) - C_4(\dot{X}_3 - \dot{X}_2 - b\dot{\theta}) = 0$
3. $m_3\ddot{X}_3 + K_3(X_3 - X_1 + a\theta) + C_3(\dot{X}_3 - \dot{X}_1 + a\dot{\theta}) + K_4(X_3 - X_2 - b\theta) + C_4(\dot{X}_3 - \dot{X}_2 - b\dot{\theta}) + K_5(X_3 - X_5 - a\theta) + C_5(\dot{X}_3 - \dot{X}_5 - a\dot{\theta}) + K_6(X_3 - X_6 + b\theta) + C_4(\dot{X}_3 - \dot{X}_6 + b\dot{\theta}) = 0$
4. $J_3\ddot{\theta}_3 - K_3a(X_3 - X_1 + a\theta) - C_3a(\dot{X}_3 - \dot{X}_1 + a\dot{\theta}) - K_5a(X_3 - X_5 - a\theta) - C_5a(\dot{X}_3 - \dot{X}_5 - a\dot{\theta}) + K_4b(X_3 - X_2 - b\theta) + C_4b(\dot{X}_3 - \dot{X}_2 - b\dot{\theta}) + K_6b(X_3 - X_6 + b\theta) + C_6b(\dot{X}_3 - \dot{X}_6 + b\dot{\theta}) = 0$
5. $m_5\ddot{X}_5 + K_5(X_3 - X_5 - a\theta) + C_5(\dot{X}_3 - \dot{X}_5 - a\dot{\theta}) = 0$
6. $m_6\ddot{X}_6 + + K_4(X_3 - X_6 + b\theta) + C_6(\dot{X}_3 - \dot{X}_6 + b\dot{\theta}) = 0$

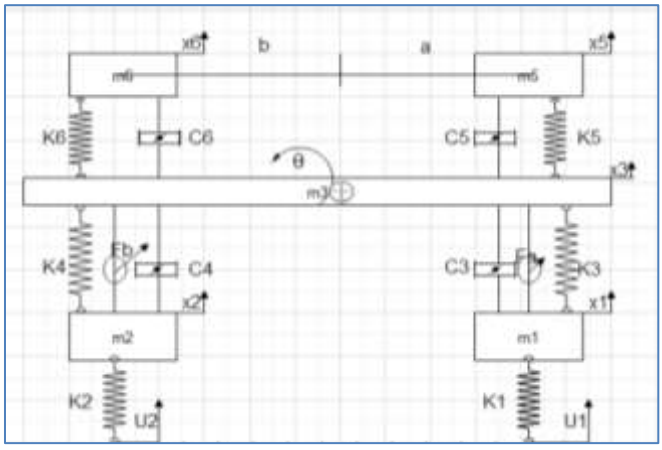

Figure 4: Active suspension Model of half vehicle including seat model also.

Whereas the mathematical model of active suspension system of half vehicle including weight as shown in Figure 4 of passenger is given below;

1. $m_1\ddot{X}_1 + K_1(X_1 - U_1) + K_3(X_3 - X_1 + a\theta) + C_3(\dot{X}_3 - \dot{X}_1 + a\dot{\theta}) - F1 = 0$
2. $m_2\ddot{X}_2 + K_2(X_2 - U_2) + K_4(X_3 - X_2 - b\theta) + C_4(\dot{X}_3 - \dot{X}_2 - b\dot{\theta}) - F2 = 0$
3. $m_3\ddot{X}_3 + K_3(X_3 - X_1 + a\theta) + C_3(\dot{X}_3 - \dot{X}_1 + a\dot{\theta}) + K_4(X_3 - X_2 - b\theta) + C_4(\dot{X}_3 - \dot{X}_2 - b\dot{\theta}) + K_5(X_3 - X_5 - a\theta) + C_5(\dot{X}_3 - \dot{X}_5 - a\dot{\theta}) + K_6(X_3 - X_6 + b\theta) + C_4(\dot{X}_3 - \dot{X}_6 + b\dot{\theta}) - F1 - F2 = 0$
4. $J_3\ddot{\theta}_3 - K_3a(X_3 - X_1 + a\theta) - C_3a(\dot{X}_3 - \dot{X}_1 + a\dot{\theta}) - K_5a(X_3 - X_5 - a\theta) - C_3a(\dot{X}_3 - \dot{X}_5 - a\dot{\theta}) + K_4b(X_3 - X_2 - b\theta) + C_4b(\dot{X}_3 - \dot{X}_2 - b\dot{\theta}) + K_6b(X_3 - X_6 + b\theta) + C_6b(\dot{X}_3 - \dot{X}_6 + b\dot{\theta}) - aF1 + bF2 = 0$
5. $m_5\ddot{X}_5 + K_5(X_5 - X_3 + a\theta) + C_5(\dot{X}_5 - \dot{X}_3 + a\dot{\theta}) = 0$
6. $m_6\ddot{X}_6 + K_4(X_6 - X_3 - b\theta) + C_6(\dot{X}_6 - \dot{X}_3 - b\dot{\theta}) = 0$

The system parameters are defined in Table 1.

System Modeling

Pressurized fluid source is model for the active suspension system (Kumar, 2008), which consists of PRV (Pressure relief valve) to control fluid pressure, direction control valve for directing the flow of fluid within the system and hydraulic actuator to convert hydraulic pressure into force that will be used to control sprung mass and all other masses motion.

The following Figure 5 shows various components in working.

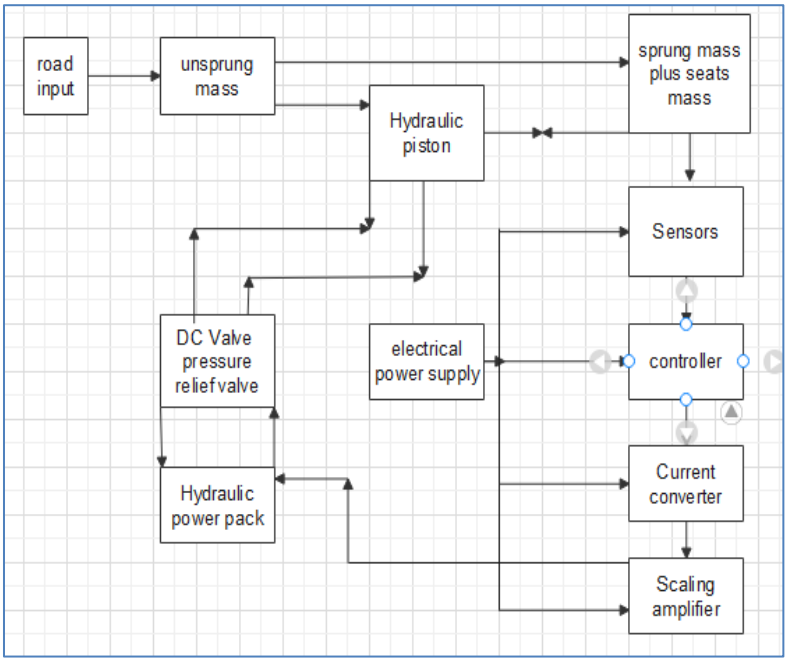

Figure 5: Half Vehicle Including Seats Mass Control Model

Passive suspension system Simulink Model

After defining the systems parameters and their essential components, an active and passive models are design base on the following values;

Table 1: Parameters value for which system model has been designed (J.A.Calvo, 2005),(Ammar Majid Hameed Al-Ghanim, 2016)

Parameters	**Symbol**	**Values**	**Unit**
Sprung mass of the vehicle Chassis	m3	1795	Kg
Moment of inertia of the vehicle	J	3443.1	Kg-m2
Unsprung mass of the front axle	m1	87.15	Kg
Unsprung mass of the rear axle	m2	140.4	Kg
mass of the front seats and passengers	m5	125	Kg
mass of the back seats and passengers	m6	125	Kg
stiffness of the front seats	k5	5000	N/m
stiffness of the back seats	k6	5000	N/m
stiffness of the front tire	K1	120	kN/m
stiffness of the rare tire	K2	210	kN/m
Damping coefficient of the front seats	C5	1000	Ns/m
Damping coefficient of the back seats	C6	1000	Ns/m
Spring constant of the front axle	K3	36350	N/m
Spring constant of the rear axle	K4	26530	N/m
Damping coefficient of the front axle	C3	1200	Ns/m
Damping coefficient of the rear axle	C4	1100	Ns/m
Front body length from the CG	a	1.32	m
Rear body length from the CG	b	1.46	m
State Variables			
Vehicle vertical displacement	X3	------	m
Vehicle rotational movement	Θ	------	rad
Front axle vertical displacement	X1	------	m
Rear axle vertical displacement	X2	------	m
mass of the front seats and passengers	X5	------	m
mass of the back seats and passengers	X6	------	m
Inputs (road excitation)			
Road Excitation at the front axle	u(t)	------	m
Road excitation of the rear axle	u(t+td)	------	m

Figure 6 below shows the Simulink model based on the parameters in the above table and according to the flow diagram of Figure 5.

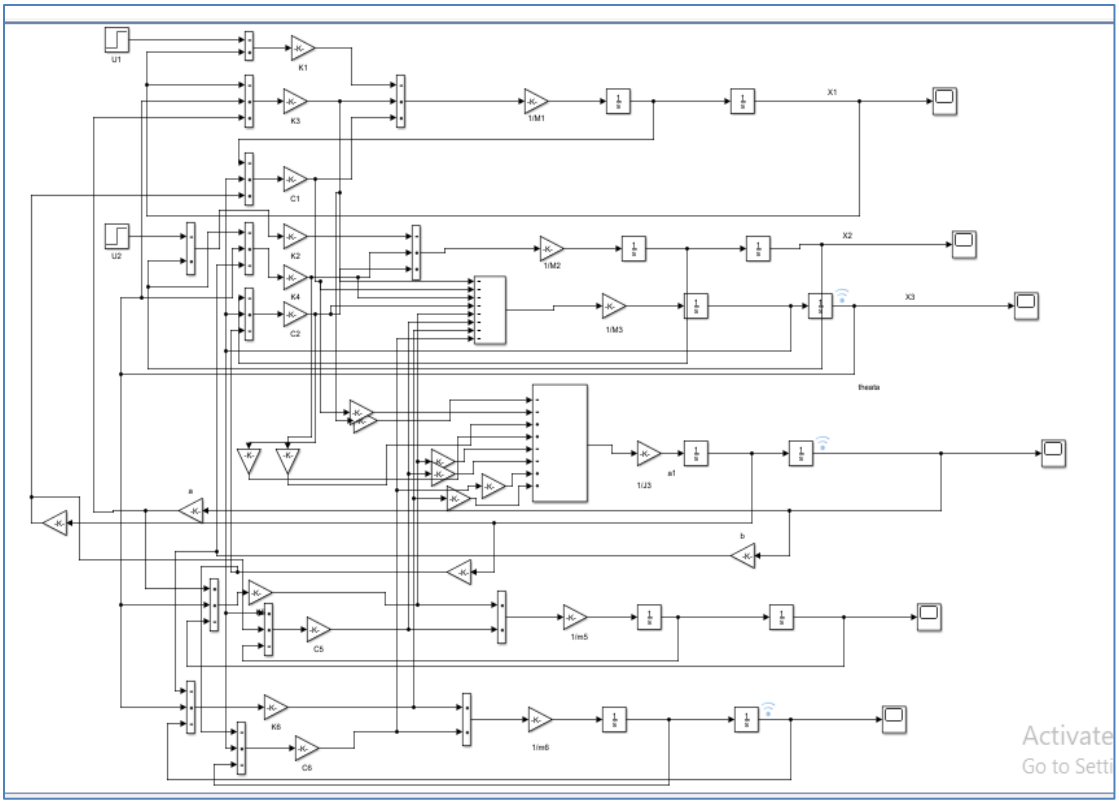

Figure 6: Simulink Model for the System without Controller and with Step Input

PID Controller Design

The PID or Proportional Integral Derivatives Controller is the most used feedback control design. PID has three terms operating on the error signal to produce a control signal (Rao et al., 2014). If U(t) is the control signal which sent to the system, Y(t) is the actual output and r(t) is the desired output and tracking error e(t) = r(t) – y(t), a PID controller has the next form (A.Leva, 2010).

$$U(t) = K_p * e(t) + K_i * \int e(t)dt + K_d * \frac{de(t)}{dt}$$

The Desired closed loop dynamics can be obtained by adjusting the three parameters K_p, K_i *and* K_d. iteratively with tuning and without specific knowledge of a plant model. Stability can be often obtained using only the proportional term. The Integral term permits the rejection of a step disturbance (Shamsuzzoha & Lee, 2008). The derivatives term provides damping or shaping of the response. PID controller are the most well established class of control systems However they cannot be used in several complicated cases, almost in the MIMO systems. Figures 7 represent the Simulink model of active suspension system using PID controller (Farag, 2020).

Tuning of PID Controller

The Ziegler-Nichols PID tuning technique is used in current research with trial-and-error methods. The technique is easy and provides quick insights into system stability (Bishop, 2011; Hari Om Bansal, 2017) were applied. The Ziegler-Nichols method relates to a step input, in which the K_p value is increased to induce system oscillations. Then, the Ziegler-Nichols criteria are used to derive the values for, K_p K_i and

K_d since the system was already oscillating, the Ziegler-Nichols criteria were directly employed to approximate the gains of the PID controllers. Consequently, adjustments to the K_p K_i, andK_d values were made based on the system's response, resulting in a perfectly stable response. The system response for the step input, both with and without the controller, is shown in Figure 7.

The first starting point was determined by Ziegler Nichols method then different gain were tuned differently according to speed limit. When speed of car start increasing its settling time also increases, and the rise start decreasing with percent overshoot increasing. In PID controller proportional gain deals with how quickly the system reaches to the desired input, while the derivative gains deals with the percent overshoot, while settling time has been adjusted using integral gains. There is always compensation between different parameters control as decreasing one the other might increase with the same set of PID gains. So the best way is to look to the condition, and to control that parameter mostly while not improving the rest of parameter significantly.

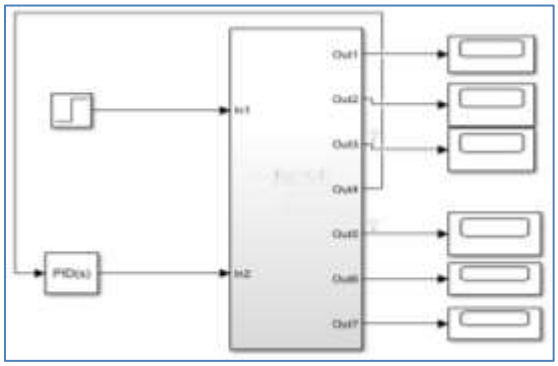

Figure 7: PID Controller Simulink Model

The PID Controller characteristic in closed loop system are shown in the Table 2

Table 2: PID Controller characteristic in closed loop system

Response	Rise Time	Settling time	% Overshoot	Steady-state error
K_p	Decrease	No effect	Increase	Decrease
K_i	Decrease	Increase	Increase	Finished
K_d	No effect	Decrease	Decrease	No effect

Result and Discussion

Result of Passive Suspension System and Active Suspension System for Speed Bump as Road Input to Wheel at Speed of 20 km/hr.

Speed bumps are parabolic vertical calming devices intended to slow traffic speeds on low volumes, low speed roads. Speed bump used in the case studies is 3 meter and has height of 0.5 meter. The standard equation of vertical displacement from speed bump to road is (Das et al., 2017).

$$Y(t) = \frac{A}{2}\left(1 - \cos\left(\frac{2\pi u t}{L}\right)\right) \quad (1)$$

Where A is the vertical height of Speed Bump, u is speed of the vehicle while L is length of the speed bump.

When the vehicle speed is 20 km/hr, the duration of the road excitation is calculated as t=3 ×3.6/20 = 0.54 sec. This is the time for which a vehicle remains in contact with the bump, or the time required to pass over a speed bump. A time delay of one second is given, and after that, the front wheel starts passing over the speed bump. It will take 0.54 seconds for it to pass through. The second wheel will pass over the bump after a time equal to $t = a + b/u = (1.32 + 1.46) \times 3.6/20 = 2.5804\ sec.$ Speed bump diagram with its Simulink model is given in Figure 8

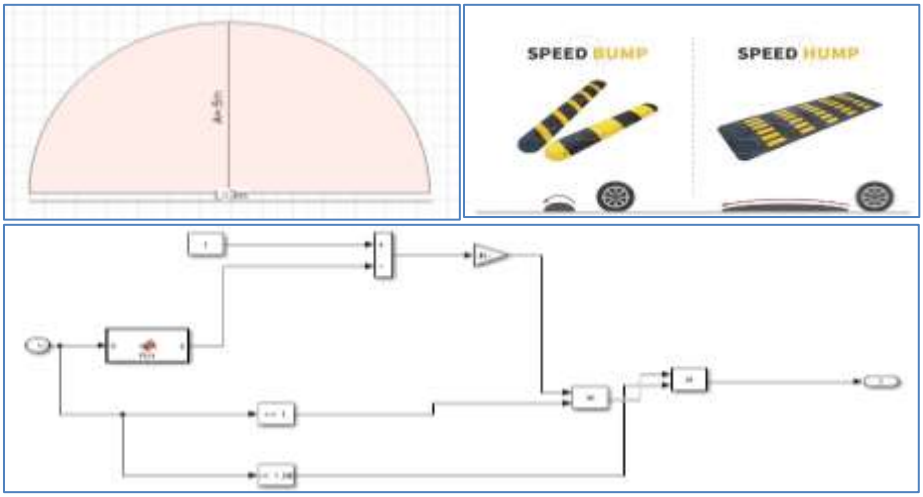

Figure 8: Speed bump and hump

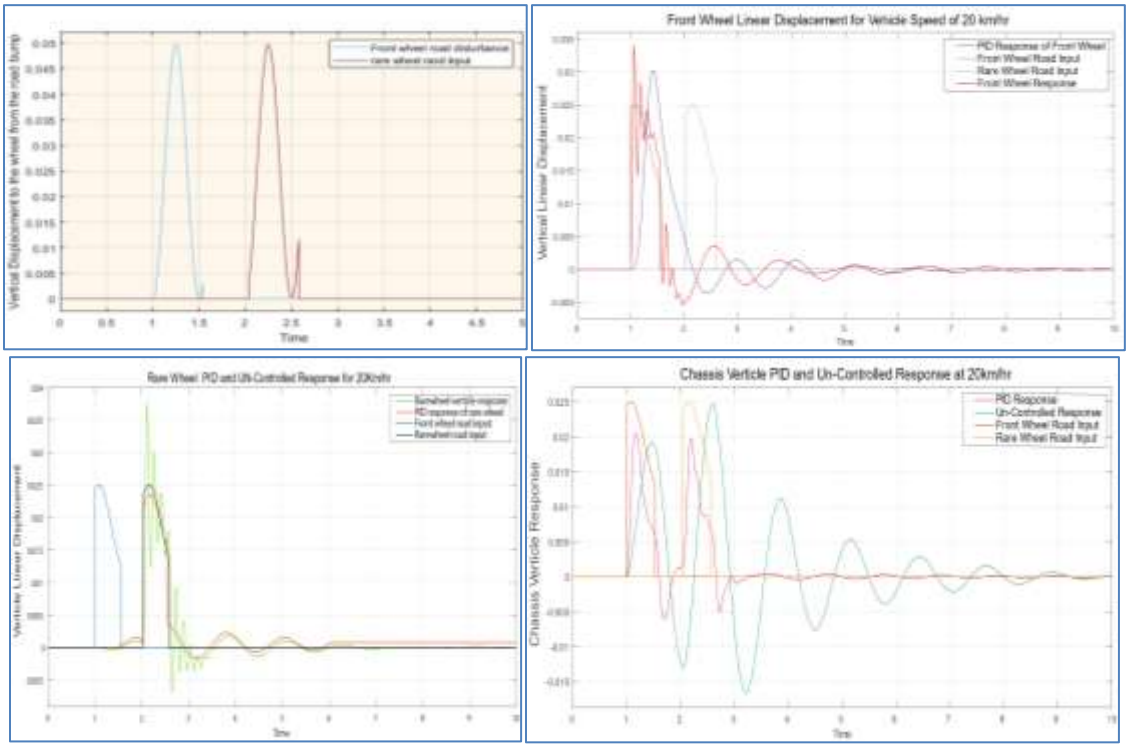

Figure 9 (a) Speed Bump Excitation for Vehicle Speed of 20 km/hr, (b) Front Wheel Response, (c) Rare Wheel Response, (d) Vehicle Chassis Response

Figure 9 (a) shows the speed bump excitation signal for a vehicle speed of 20 km/hr. The response of the front wheel's linear motion over the speed bump, as defined in equation (1), at the given road speed is shown in Figure 9 (b). The red line represents the response of the passive suspension system, and the yellow line represents the response of the PID system. The system characteristics for the front wheel mass of 1 are compared for both the PSS and ASS in Table 3. The blue lines represent the base excitation applied to both wheels by the speed bump, with an amplitude of 0.5 m and a length of 3 m.

Figure 9 (c) depicts the responses of the PID and PSS controllers when a rear wheel encounters a speed bump defined by Equation (1) at a linear speed of 20 km/hr. The red lines represent the linear displacement of the rear tire without a controller, which exhibits a final percent overshoot of 56.61% and takes approximately 0.563 seconds to settle within a 2% tolerance zone. The yellow line illustrates the response of the rear tire with the PID controller. With the application of the PID controller, the settling time decreases, and the percent overshoot reduces to 15%. A comparison of these results is provided in Table 4.

Figure 9 (d) depicts the response of the vehicle chassis linear motion over a speed bump at a speed of 20 km/hr. The blue line represents the response of the passive suspension system (PSS), while the red line represents the response of the system with the PID controller. The characteristics of the system for the vehicle chassis with a mass of 3 are compared for both the PSS and the active suspension system (ASS) in Table 4.

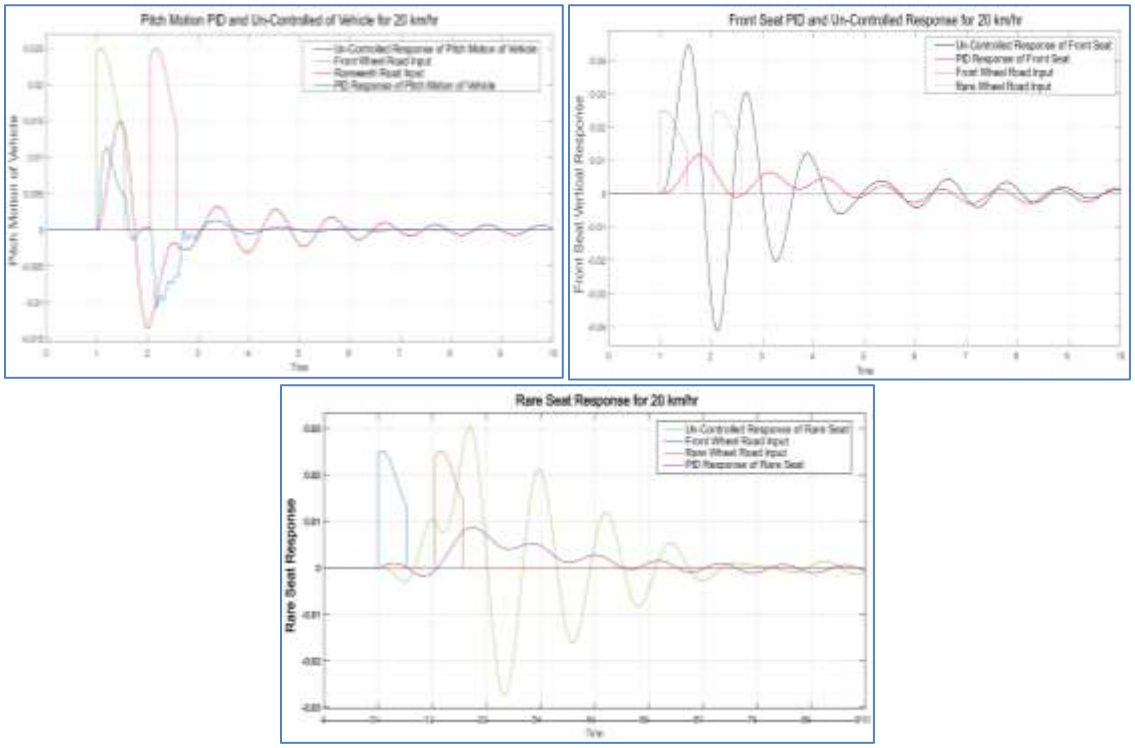

Figure 10 (a) Pitch Motion of Chassis, Response (b) Front Seat Response, and (c) Back Seat Response for speed of 20km/hr

Pitch motion response for speed bump of height 0.5 m and length 3m at speed of 20 km/hr is shown in Figure 10 (a). The Red and violet lines represent active and passive suspension response of chassis pitch motion is indicated in Table 3. PID and Un-controlled responses are compared in Table 4.

Front seat active and passive suspension system response are shown in Figure 10 (b) by green and violet lines respectively. In Table 3 comparison of settling time, percent overshoot and rise time for Active and passive suspension system for speed of 20 km/hr over speed bump is shown.

Rare seat response of is shown in Figure 10 (c). Blue and violet lines represent passive and active suspension system response respectively over speed bump at 20 km/hr. The system characteristic are shown in Table 3.

At 20 km/hr from Fig 9 – 10, the most affected mass from speed bumps is front seat mass because it reaches to 45mm while pitch motion has low amplitude which shows that at low speed there is small difference between front and rare seat seats height with respect to ground.

Table 3: System characteristic for 20km/hr

Passive Suspension system characteristic for 80km/hr speed over speed bump						
System Characteristic	Front Wheel mass	Rare Wheel mass	Chassis Mass	Moment of inertia	Front seat mass	Rare seat mass
Rise time(sec)	0.0214	0.0262	0.208	0.172	0.139	0.137
settling time(sec)	0.563	1.13	6.4	5.6	6.234	10.2
percent overshoot(%)	56.62	60.23	61.8	70.56	24	20
Active Suspension system characteristic for 80km/hr speed over speed bump						
System Characteristic	Front Wheel mass	Rare Wheel mass	Chassis Mass	Moment of inertia	Front seat mass	Rare seat mass
Rise time(sec)	0.37	0.016	0.01	0.09	0.03	0.003
settling time(sec)	4.14	2.09	1.12	1.27	3.89	2.5
Percent overshoot	15	12	20	25	16	15

Result of Passive Suspension System and Active Suspension System for Speed Bump as Road Input to Wheel at Speed of 80 km/hr.

When vehicle speed is 80 km/hr, then road excitation has duration of t=3 ×3.6/80 = 0.135 sec. This is the time for which a vehicle remain in contact with the bump or time required to pass over speed bump. A time delay of one second is given, and after that front wheel start passing over speed bump then in 0.54 sec it will passed through it. The second wheel will pass over after time equal to$t = a + b/u = (1.32 + 1.46) \times 3.6/80 = 1.2634\ sec.$

Remembers that Simulink Model will remain same except time duration for which Y(t) will not be equal to zero changes.

Front wheel linear motion response over speed bump defined in equation 1 at vehicle road speed of 80 km/hr is shown in Figure 11 (b). The red line represents passive suspension system response and yellow line represent PID response of the system. System characteristic for front wheel mass (m1) is compared for both PSS and ASS in Table 4. Blue lines represent base excitation to both wheels provided by speed bump of amplitude 0.5 m and length 3 m.

Figure 11 shows the front wheel, rare wheel, chassis, and its pitch motion response for the vehicle speed of 80 km/hr.

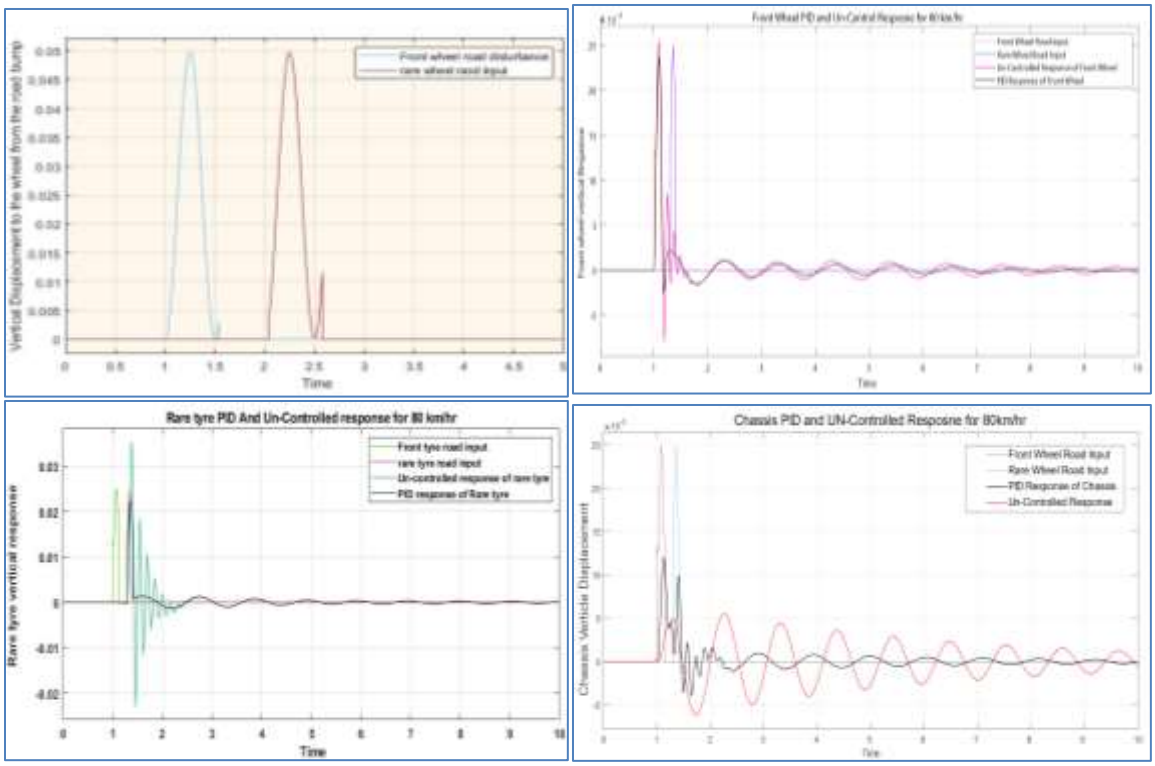

Figure 11 (a) Speed Bump Excitation for Vehicle Speed of 80 Km/Hr, (b) Front Wheel Response, (c) Rare Wheel Response, (d) Vehicle Chassis Response

Figure 11 (c) shows the PID and Passive suspension system response of rare wheel passing over speed bump of the contour define by equation 1 at 80 km/hr linear speed. In the figure Red lines represent linear displacement of rare tyre without controller which reaches to a final value with percent overshoot of 68.23 %. This response almost take 1.2 sec to settle down to 2% tolerance zone. The yellow line indicates the PID controller response of rare tyre. After applying PID controller settling time reduces and percent overshoot also decreased down to 20.1%. Comparison are carried out in Table 4.

Figure 11 (d) show response of vehicle chassis linear motion over speed bump at 80 km/hr speed. The red line represents passive suspension system response and violet line represent PID response of the system. System characteristic for vehicle chassis mass 3 is compared for both PSS and ASS in Table 4.

Pitch motion response for speed bump of height 0.5 m and length 3 m at speed of 80 km/hr is shown in Figure 12 (a). The red and green lines represent active and passive suspension response of chassis pitch motion which is indicated in Table 5. Also, the PID and Un-controlled responses are compared in the same table.

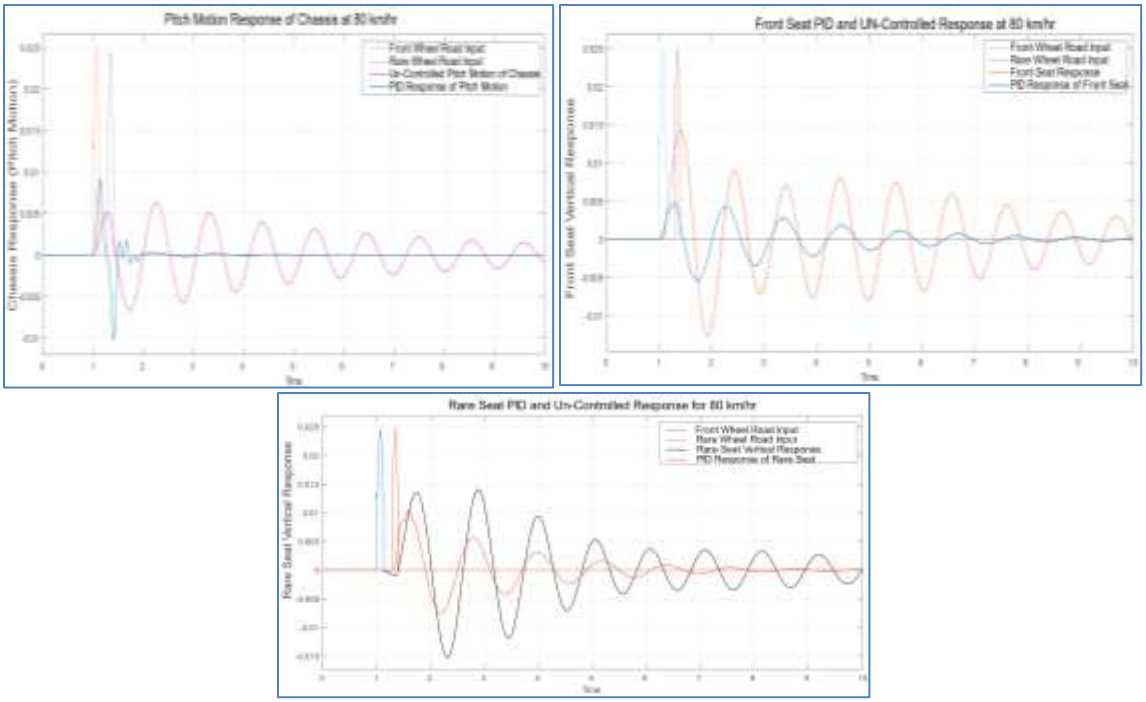

Figure 12 (a) Pitch Motion of Chassis, Response (b) Front Seat Response, and (c) Back Seat Response for speed of 80 km/hr

Front seat active and passive suspension system response are shown in Figure 12 (b) by blue and violet lines respectively. In table 5 comparison of settling time, percent overshoot and rise time for Active and passive suspension system for speed of 80km/hr over speed bump is shown. The rare seat response of is shown in Figure 12 (c). Red and green lines represent passive and active suspension system response respectively over speed bump at 80 km/hr. The system characteristic are shown in Table 4.

System characteristic comparison for PID and UN-controlled response are listed in Table 5 below.

Table 4 : System characteristic for 80 km/hr for PSS and ASS

Passive Suspension System Characteristic for 80km/hr Speed Over Speed Bump						
System Characteristic	Front Wheel mass	Rare Wheel mass	Chassis Mass	Moment of inertia	Front seat mass	Rare seat mass
Rise time(sec)	0.021	0.011	0.168	0.16	0.19	0.119
settling time(sec)	1.2	2.2	9.6	8.2	8.34	16.1
percent overshoot (%)	68.23	74.21	78.12	77.12	33	33.32
Active Suspension System Characteristic for 80km/hr Speed Over Speed Bump						
System Characteristic	Front Wheel mass	Rare Wheel mass	Chassis Mass	Moment of inertia	Front seat mass	Rare seat mass
Rise time(sec)	0.37	0.001	0.007	0.09	0.03	0.003
settling time(sec)	4.14	0.008	0.159	1.27	3	2.5
Percent overshoot (%)	20.1	15.7	17.1	14.6	8.6	10

At 80 km/hr from Figure 11 and 12, the front seat mass is affected more compared to others but at the same time its amplitude is low at this speed compared to 20 km/hr. It is observed that when speed increases the

vehicle amplitude of every single entity decreases but its oscillation increases which means that vehicle remain unstable for longer duration.

Conclusion

The effect of vehicle speed on the response of wheel suspension, chassis, and seats to external road disturbances, such as a standard dimension speed bump, was studied. A mathematical model of a half vehicle suspension system was created, incorporating a six-degree-of-freedom system. This system included the heave motion of the wheels, car body, and seats with a seated passenger, as well as the pitch motion of the car body. The model considered the impact of road disturbances on the seats and the resulting effect on the passenger. By applying Newton's second law of motion, the characteristic equation was derived, and a Simulink model was developed to simulate the system. A road bump Simulink model was integrated into the half vehicle model.

The initial testing of the model was conducted at a low residential speed limit of 20 km/hr, and characteristics such as settling time, rise time, and percent overshoot were observed. The process was then repeated at the highway speed limit of 80 km/hr. In the latter case, issues related to percent overshoot and settling time were identified, and an optimum set of parameters was determined. The proportional gain was decreased for the speed of 80 km/hr because the system quickly reached the desired input due to the higher speed. However, this adjustment affected the settling time and percent overshoot, so the percent overshoot was reduced in consideration of the settling time.

Different characteristics were compared for different speeds, revealing that as the vehicle speed increased when encountering speed bumps or other road disturbances, the settling time and percent overshoot for various vehicle components such as wheels, chassis, and seats also increased. This increase in settling time and percent overshoot could lead to riding discomfort, road accidents, and adverse effects on vehicle health, including permanent disorder and deformation of the shock absorber system. The analysis also considered active suspension systems and passive suspension systems, highlighting significant improvements when the active system was controlled with a PID controller.

References

A.Leva, S. N., A.V.Papadopoulos. (2010). PI/PID Autotuning with Contectual Models parametriasation. *Journal of Proess Control*, 463.

Al-Ghanim, A. M. H., & Nassar, A. A. (2018). Modeling, simulation, and control of half car suspension system using Matlab/Simulink. *International journal of science and research*.

Ammar Majid Hameed Al-Ghanim, D. A. A. N. (2016). Modeling, Simulation, and Control of Half Car. *International Journal of Science and Research (IJSR)*.

Avesh, M., & Srivastava, R. (2012). Modeling simulation and control of active suspension system in Matlab Simulink environment. 2012 Students Conference on Engineering and Systems,

Bakar, S. A., Samin, P., Jamaluddin, H., Rahman, R., & Sulaiman, S. (2015). Semi active suspension system performance under random road profile excitations. 2015 International Conference on Computer, Communications, and Control Technology (I4CT),

Bhanu Chander, V. (2009). Modelling and Analysis of Half Car Vehicle Suspension System Using Fuzzy Logic. *B. Tech Thesis, Univeristy College of Engineering, Tirupati, May*, 1-101.

Bishop, R. C. D. R. H. (2011). *Modern control systems*.

Darus, R., & Sam, Y. M. (2009). Modeling and control active suspension system for a full car model. 2009 5th international colloquium on signal processing & its applications,

Das, D. D. D., Kumar, M. S., & Gampa, S. R. (2017). Optimum pid controller design using pso for vehicle active suspension system considering matlab simulink modeling based road profiles. *Journal of Electrical Engineering*, *17*(2), 10-10.

Farag, W. (2020). Track maneuvering using pid control for self-driving cars. *Recent Advances in Electrical & Electronic Engineering (Formerly Recent Patents on Electrical & Electronic Engineering)*, *13*(1), 91-100.

Ghazaly, N., Kamel, A., & Mousa, M. (2014). Influence of misalignment and backlash on spur gear using fem. *International Journal of Mechanical and Production Engineering*, *2*(12), 2320-2092.

Gowda, D. V., & Chakrasali, S. (2014). Comparative analysis of passive and semi-active suspension system for quarter car model using PID controller. Int. Conf. on Recent Trends in Signal Processing, Image Processing and VLSI (ICrtSIV), Bangalore, India,

Hari Om Bansal, R. S., P. R. Shreeraman. (2017). Hari Om Bansal, Rajamayyoor Sharma, P. R. Shreeraman. *Journal of Control Engineering and Technology*.

J.A.Calvo, J. L. S. R. V. D. (2005). Establishing inspection criteria to verify the dynamic behavior of the vehicle suspension sytem by a platform vibrating test bench. *International Journal of vehicle design*, *38*(2005).

Kashem, S., Mustapha, K., Kannan, T., Roy, S., Safe, A., Chowdhury, M., Choudhury, T., Ektesabi, M., & Nagarajah, R. (2015). A study and review on vehicle suspension system and introduction of a high-bandwidth configured quarter car suspension system. *Australian Journal of Basic and Applied Sciences*, *9*(30), 59-66.

Kumar, M. S. (2008). Development of active suspension system for automobiles using PID controller. Proceedings of the World Congress on Engineering,

R.N, J. (2014). Vehicle Vibration. In *Vehicle dynamic*. springer ,New York.

Rao, P. G. K., Subramanyam, M., & Satyaprasad, K. (2014). Study on PID controller design and performance based on tuning techniques. 2014 International Conference on Control, Instrumentation, Communication and Computational Technologies (ICCICCT),

Rao, S. S. (2001). *Mechanical vibrations*.

Shamsuzzoha, M., & Lee, M. (2008). PID controller design for integrating processes with time delay. *Korean Journal of Chemical Engineering*, *25*, 637-645.