MODELLING AND SIMULATION OF ACTIVE SUSPENSION SYSTEM FOR QUARTER CAR MODEL WITH ANALYSIS OF VEHICLE RIDE PERFORMANCE

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ABSTRACT: The optimal level of vehicle ride performance is not promptly provided by conventional passive suspension systems. In passive suspension, there is a trade-off between control and comfort requirements. An active suspension system offers comfort and control, as well as active roll and pitch control during cornering and braking. As a result, it provides a level and comfortable ride across extremely rugged terrain. This study analyzes the vehicle ride performance as well as the modelling, simulation, and control of an active suspension system using MATLAB/Simulink software. The vehicle ride performance is determined via computer simulation using MATLAB/Simulink. A proportional-integral-derivative (PID) controller is configured properly to regulate and enhance the vehicle ride performance. From the simulation, it demonstrates the practicality of this control strategy, and the vehicle ride performance is compared with a passive suspension system.

KEYWORDS: *Active Suspension System; Quarter Car Model; Vehicle Ride Performance; Suspension Dynamics*

1.0 INTRODUCTION

With the recent developments in automotive industries, vehicle's handling and passenger's comfort have been the major focus of many researchers which is being studied in automotive engineering as vehicle dynamics. The study of vehicle's ride performance has been under serious research for years focusing on improving the passenger's comfort, the handling and stability of the vehicle during operation. The phrase "ride performance" always refers to a wide range of factors, such as handling qualities, stability, comfort level, and reaction to road disturbances. In the realm of vehicle dynamics, the ride performance of a vehicle is studied through the analysis of noise, vibration and harshness transmitted to the vehicle's occupants due to the unevenness of the road surfaces such as bumps, potholes, and so on, or from the operations of the vehicle's powertrain and drivetrain. The study of noise, vibration, and harshness (NVH) focuses on occupant comfort, especially in cars. This is because there are noise and vibration levels that, if they are experienced for an extended period of time within an eight-hour reference, can be considered dangerous for occupants [1]. However, when traveling for an extended period of time, this vibration induced from the uneven road, engine, and driveline has a significant negative impact on the driver's general health, which might also be exposing some parts of the vehicle to damage [2]. In the analysis of vehicle's vibration, the suspension system is one of the major components that are critically analyzed, being the only linkage between the road surface and the vehicle chassis, through which the vibration caused by uneven road surfaces is always passed to the passengers. The primary function of a suspension system is the regulation of ride quality and vehicle handling. By regulating the ride quality, we mean, isolating the passenger's body and cargos from road disturbances and inertia during braking, cornering, and acceleration, while regulation of the vehicle's handling means to prevent suspension movements in order to create adequate tyre-road contact [3]. Furthermore, a vehicle with a rigid suspension is always associated with good handling stability but with poor ride comfort and vice versa [4]. There are basically five categories of suspension types widely used by automakers which are passive, semi-active, slow-active, active and fully active. However, in this report, we will be discussing these mentioned categories of suspension types, with a primary focus on the active suspension system proposed here. We will also model and simulate this active suspension system using a 2-DOF quarter-vehicle model to simplify the ride performance analysis and highlight the reasons for proposing this particular suspension type for Vertiga company.

2.0 METHODOLOGY

Suspension systems are crucial components in vehicles, which are responsible for providing stability, handling, and comfort by managing the interaction between the wheels and the road. These systems can be broadly classified into several types based on their level of control and responsiveness. As a result, each type represents a different approach to managing the vehicle's dynamics, offering varying levels of adaptability and performance.

Active suspension system makes use of sensors, actuators, and a control unit to constantly monitor and modify various suspension parameters such as spring stiffness and damping coefficients in real time [5]. Also, it actively manages the vertical movement of the wheels relative to the chassis or vehicle body using an on-board system. However, active suspension systems offer superior ride comfort and vehicle stability but are more expensive than semi-active or passive suspension systems. These systems function between the vehicle's sprung and unsprung masses, reducing vertical acceleration and vibrations caused by road conditions and vehicle movement thus enhancing vehicle handling and stability [6]. Meanwhile, active suspension system is basically categorized into two types which are low bandwidth (slow active suspension) and high bandwidth (fully active suspension).

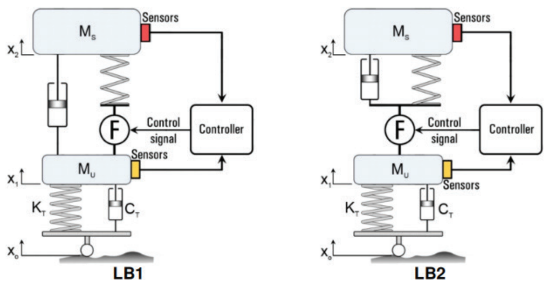


Figure 1: The slow active suspension system [6]

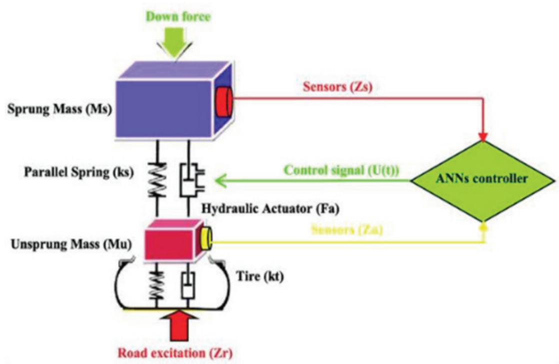


Figure 2: The Fully active suspension system [7]

3.0 MODELLING ASSUMPTIONS

This section involves the derivation of a 2-degree-of-freedom (2-DOF) mathematical equations for active suspension system for a quarter car model. In this design the actuating force as shown in figure 4 above will be also considered since that's what differentiates the model from passive system. However, figure 5 below represents the free body diagram of the model where f_s is the suspension spring force, f_d is the suspension damping force, f_a is the required actuating force, while f_t is the radial tyre force. Meanwhile, in this design an upward direction is assumed to have a positive value while downward is assumed negative. However, Z_s represents the vertical displacement of the vehicle's body while Z_u represents the vertical displacement of the wheel. And also, there are two masses involved in this design which are M_s representing the mass of the vehicle body (sprung mass) and M_u representing the mass of the wheel (unsprung mass).

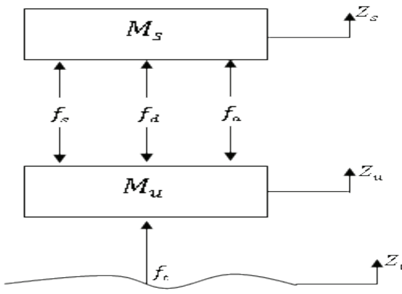


Figure 3: The Free body diagram of Active suspension system

Recalling Newton's second law of motion,

$$F = Ma \tag{1}$$

Considering the sprung mass (Ms), there are three forces acting on it which are directed upward (positive); however, the summation of these forces is shown in Eqn (2) below. Also remembering that Zs is the displacement of the sprung mass, hence, the velocity will be its derivative ($\dot{Z_s}$) where its acceleration will be its second derivative ($\ddot{Z_s}$), the same also goes to the vertical dynamic characteristics of the unsprung mass (Z_u, $\dot{Z_u}$ and $\ddot{Z_u}$). Therefore, considering only the sprung mass, Eqn (1) will now become,

$$f_s + f_d + f_a = M_s\ddot{Z}_s \tag{2}$$

Recall that

$$f_s = K_s(Z_u - Z_s)$$

$$f_d = C_s(\dot{Z}_u - \dot{Z}_s)$$

Substituting these to Eqn (2), we now have

$$K_s(Z_u - Z_s) + C_s(\dot{Z}_u - \dot{Z}_s) + f_a = M_s\ddot{Z}_s \tag{3}$$

Furthermore, considering the unsprung mass (Mu), there are total of four forces acting on the wheel. It is also worth remembering that upward forces are positive while downward forces are negative, however, the summation of forces acting on the unsprung mass is thus represented in Eqn (4) below. Where;

$$f_t = K_t(Z_t - Z_u)$$
$$[K_t(Z_t - Z_u)] - [K_s(Z_u - Z_s)] - [C_s(\dot{Z_u} - \dot{Z_s})] - f_a = M_u\ddot{Z_u} \quad (4)$$

Moreover, Eqn (3) and Eqn (4) now represent the required 2-DOF mathematical equations for the active suspension system of a quarter car model.

3.2 MATLAB and Simulink Model of Active Suspension for Quarter Car Model

Using the mathematical models derived in Eqn (3) and Eqn (4), MATLAB and Simulink are used to simulate the active suspension system for a quarter car model. A PID controller will then be used to manage the active system (actuator force) to achieve the desired vehicle vertical displacement. In this model, Zsddot represents the vehicle body's vertical acceleration, Zsdot is the vertical velocity, and Zs is the vertical displacement of the vehicle's body (sprung mass). Meanwhile, Zu, Zudot, and Zuddot refer to the vertical displacement, velocity, and acceleration of the vehicle's wheel (unsprung mass), respectively, with Zt as the road input. The parameters listed in Table 2 are used in this model. Figure 4 shows the PID controller managing the actuator force based on feedback from the vehicle's displacement.

Furthermore, the road input due to the irregularities of the road will be induced using step-input block and sine-wave block differently, to get their different plots as shown in Figure 4 and results presented in subsections 4.1, 4.2, 4.3 and 4.4. The height of the road bump is set at 0.1m while the time is set to 1 second in both step block and sine-wave block.

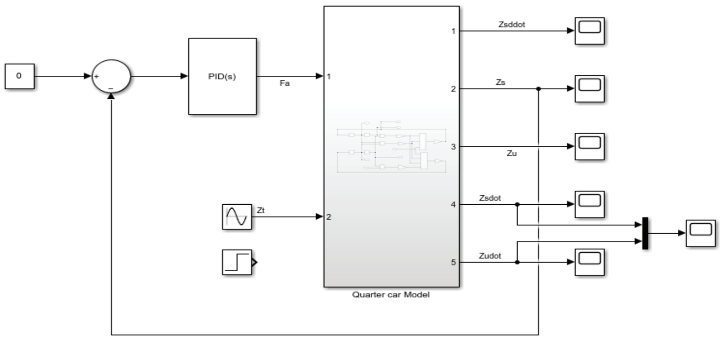
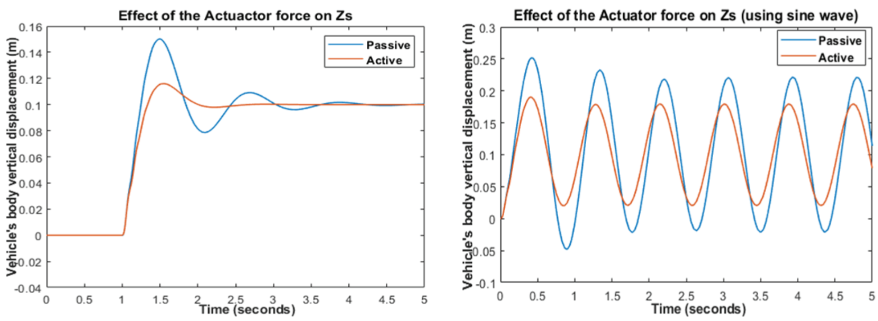


Figure 4: PID controller receiving output feedback and adjusting the actuator force

4.0 RESULTS AND DISCUSSION

4.1 The vehicle's Body displacement (Zs)

Step block and sine wave were used differently to compare the vehicle's response on its vertical displacement for both passive and active suspension system. Meanwhile, the model presented in figure 4 is an active suspension, but when we set Fa to zero, it will automatically become passive suspension. However, we will compare passive and active suspension on vehicle's body vertical displacement using step and sine wave input as shown in Figure 5(a) and (b) respectively.



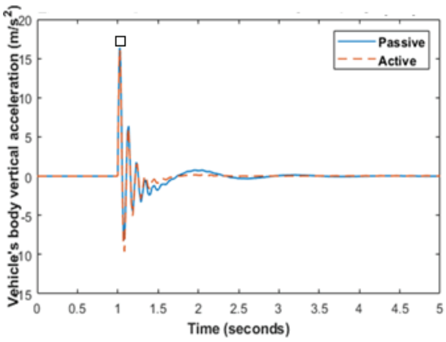
(a) (b)

Figure 5: Vehicle's body displacement using: (a) step-input, (b) sine-wave

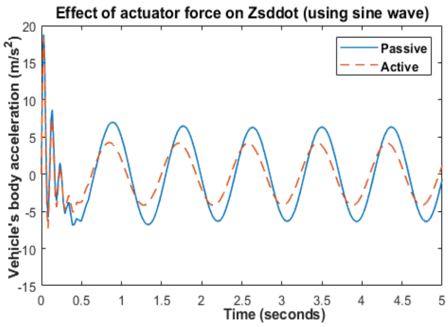
From the simulation results shown in 5(a) and (b) above, it was noted that passive suspension showcased a higher amplitude on vehicle's body vertical displacement at 0.15m and takes longer time for its oscillation time to decay. Meanwhile, the active suspension system displayed a lower amplitude at 0.116m and takes shorter time for the oscillation to decay at 2.378 seconds. This shows that passive suspension transmits larger vibration to the vehicle's body with lesser attenuation causing uncomfortable ride to the passengers, but it can actually improve the vehicle's handling especially during cornering and braking. Conversely, the active suspension shows lower amplitude with lesser time for the oscillation to decay which means that it transmits lesser vibration that does not last longer to decay, this is very suitable for a comfortable ride. Therefore, vehicle's body displacement is an important parameter when considering the vehicle's ride comfort and handling (tire grip to road surface).

4.2 The vehicle's Body Acceleration (Zsddot)

In this subsection, we study the response of the vehicle to its vertical acceleration in both passive and active suspension system when it encounters a road irregularity such as road bumps or potholes. Meanwhile, step block and sine-wave block were used as the road input parameter as shown in Figure 6(a) and (b) respectively.



(a)



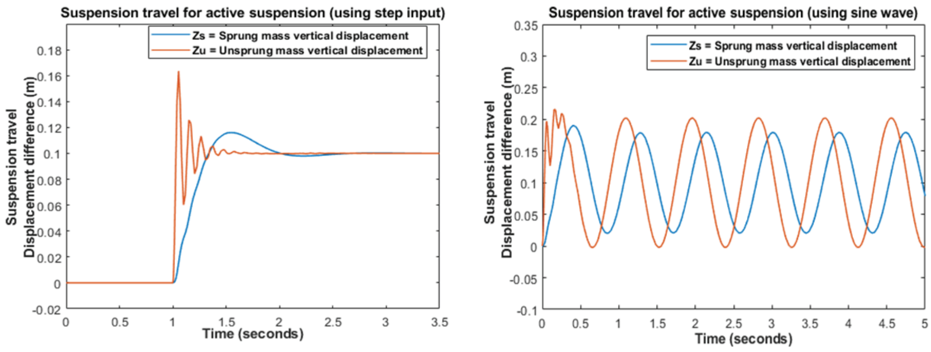
(b)

Figure 6: Vehicle's body acceleration using: (a) step-input, (b) sine-wave

It was observed from Figure 6(a) and (b) above that both the passive and active suspension shows very little difference on their amplitudes of vibration, but active suspension shows lesser time for its oscillation to decay at 1.7 seconds while passive decayed at 3.5 seconds. This means that the passengers inside the vehicle receive almost equivalent impulse during the climbing of the bump, but active suspension controlled the impact to stop within a shorter time at 1.7sec than the passive at 3.5sec. However, for vehicle's ride comfort optimization, the vehicle's vertical acceleration is also one of the major responses to consider, since it defines the measure of the impact forces experienced by the passengers.

4.3 The Suspension Travel (Zs - Zu)

Suspension travel refers to the displacement difference between the vehicle's body vertical displacement (Zs) and its wheel vertical displacement (Zu) in the context of this active suspension where Zs is the only monitored response. However, Figure 7(a) and (b) below represent the suspension travel of this active model.



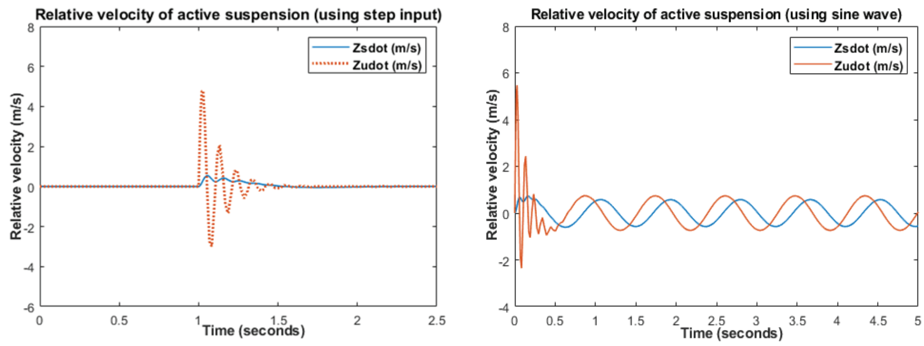
(a) (b)

Figure 7: Suspension travel using: (a) step-input, (b) sine-wave

From the simulation results shown in Figure 7(a) and (b) above, it was observed that Zu displayed a very high amplitude (at 0.164m) with so many vibrations recorded within a short period of time which is not comfortable if transmitted to the passengers. Meanwhile, Zs displayed a lower and smoother amplitude (at 0.116m) which simply implies that the actuating force Fa isolated and controlled the vibration of the suspension from transmitting to the vehicle body. However, the suspension travel is calculated as (displacement of Zs – displacement of Zu) which is given as (0.116m – 0.164m = – 0.048m). This negative sign implies that the suspension compresses 0.048m

4.4 The Relative Velocity (Zsdot - Zudot)

The relative velocity refers to the velocity difference between the vehicle's body vertical velocity (Zsdot) and its wheel vertical velocity (Zudot) in the context of this active suspension where Zs is the only monitored response. Figure 8(a) and (b) represent the relative velocity of this active model.



(a) (b)

Figure 8: Relative velocity using: (a) step-input, (b) sine-wave

The simulation results in Figure 8(a) and (b) show that there is a large velocity difference between the two vertical velocities. This velocity means the rate at which the vehicle's sprung and unsprung masses move vertically. However, higher velocity difference shows that the rate at which the unsprung mass moves vertically is very high at 4.78m/s while the rate at which the vehicle body move vertically is very low at 0.56m/s, which shows that the PID controller intervened and reduced the rate at which the road impact is sent to the passengers. This is because in our model, the controller is only focused on the body displacement (Zs) not on wheel's displacement (Zu). However, the relative velocity is calculated as (Zsdot – Zudot) which is given as (0.56m/s – 4.78m/s = – 4.22 m/s).

4.0 CONCLUSION

In order to improve vehicle ride performance on a passive suspension system, a controller was designed as part of the procedure to develop an active suspension system for a passenger car. A two-degree-of-freedom quarter car model has been used for mathematical modelling of passive and active suspension systems, concentrating on bounce motion to evaluate vehicle ride performance. For the active suspension system, the PID controller design method has been investigated. It has been discovered that suspension travel in an active suspension system is 23% less than passive suspension system. It is possible to achieve a better compromise than utilizing only passive elements by including an active element in the suspension system. PID controller design's potential to improve vehicle ride comfort is investigated and effective to use in vehicle suspension system.

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