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Modelling and Optimization of Linear Active Suspension System for Half-Vehicle Model

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ABSTRACT

Obtaining a mathematical model for the passive and active suspension systems for the entire automobile model is the objective of this project. Current automotive suspension systems exclusively use fixed-rate springs and damping coefficients as passive components. Vehicle suspension systems are frequently judged on their capacity to enhance passenger comfort and offer acceptable road handling. Only passive suspensions provide a solution to these two conflicting criteria. By directly manipulating the suspensions force actuators, the active suspension has the potential to minimize the traditional design as a compromise between handling and comfort. In this thesis, the active suspension system for a half-vehicle model was constructed using the FUZZY Controller approach. Various types of road profiles are used to compare passive and active suspension systems. In time domain evaluations using sinusoidal road input, the passive and active suspension systems are compared. Results reveal that passenger bounce, passenger acceleration, and tyre displacement decreased by 74.2%, 88.72%, and 28.5%, respectively. This suggests that an active suspension system has a greater chance of improving comfort and road holding.

Keywords: Passive and Active Suspension System, FUZZY controller, State space equation, Road Profile, MATLAB & SIMULINK.

I. INTRODUCTION

The automobile's suspension system connects the wheels to the body with such a manner that the bodywork provides protection from jolts caused by driving on inclines in the road. The performance, comfort, and safety of an automobile are all impacted by the suspension. In order to keep the body of the car from pitching and swaying, the system of suspension suspends the body of the car just above the ground and keeps it there [1-3]. The vehicle's suspension must also keep all four tyres firmly planted on the ground in order to ensure efficient acceleration, braking, and cornering—the components of good handling. The electrically controlled suspension system in automobiles is intended to strike a balance between comfort and enhanced handling on the road. that's why the suspension. This chapter provides an overview of the AVSS-related literature. starting with suspension system functions, classification, and forms of mathematical models utilised in suspension system design [3-5].

In order to reduce the sprung weight displacement for ride comfort, taking into account various road profiles, the current work aims to develop a fuzzy logic control algorithm for one of the various kinds of half car suspension systems and focus on comparing the outcomes obtained with those of all its forms passive suspension system [5-9]. It is crucial to comprehend the active suspension system for the half-car model before beginning to construct the software to mimic this system using MATLAB. The results from the simulation will be utilised afterward to analyse and support the optimal controller parameter. To develop a mathematical model for a half-vehicle Suspension System. To simulate the passive suspension model of the half-vehicle for rollover in MATLAB-SIMULINK environment. To simulate the active suspension system model of half-vehicle for rollover prevention using fuzzy controller in MATLAB-SIMULINK environment. Compare the results obtained from simulation [10-12].

Mathematical Model of The Vehicle

When there is considerable pitched and rolling motion, the quarter car model is insufficient for studying vehicle dynamics. When lateral dynamics are not taken into account, the half-car model has two suspensions, one for each front and rear wheel. The lateral motions of the vehicle are taken into account in this, therefore the half-car model, which is represented by both front wheels in Figure 1, is created.

Dynamic equations of the model are given as [13]

$$M_s\ddot{Z}_s + C_s(\dot{Z}_{sr} - \dot{Z}_{ur}) + C_s(\dot{Z}_{sl} - \dot{Z}_{ul}) + K_s(Z_{sr} - Z_{ur}) + K_s(Z_{sl} - Z_{ul}) = F_r + F_l \dots \quad (1)$$

$$I_{yy}\ddot{\varphi}_s + 0.5wC_s(\dot{Z}_{sr} - \dot{Z}_{ur}) - 0.5wC_s(\dot{Z}_{sl} - \dot{Z}_{ul}) + 0.5wK_s(Z_{sr} - Z_{ur}) - 0.5wK_s(Z_{sl} - Z_{ul}) = 0.5wF_r - 0.5wF_l \quad (2)$$

$$M_u\ddot{Z}_{ur} - C_s(\dot{Z}_{sr} - \dot{Z}_{ur}) - K_s(Z_{sr} - Z_{ur}) + K_u(Z_{ur} - Z_{rr}) = -F_r \quad (3)$$

$$M_u\ddot{Z}_{ul} - C_s(\dot{Z}_{sl} - \dot{Z}_{ul}) - K_s(Z_{sl} - Z_{ul}) + K_u(Z_{ur} - Z_{rl}) = -F_l \quad (4)$$

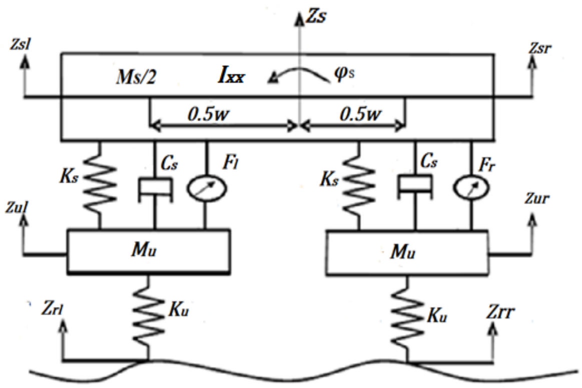


Figure 1: Half car model [14].

and the constraints are given by

$Z_s = (Z_{sr} + Z_{sl})/2;$ $\quad \varphi_s = ((Z_{sr} + Z_{sl})/w;$

Table 1: Fixed parameters of the vehicle model

Parameters	Values	Parameters	Values
Kt	200000 N/m	Iy	4140 kg-m^2
Mp	100 kg	2W	1.450 m
M	2160 kg	a	1.524 m
M1, M3	85 kg	b	1.156 m
M2, M4	60 kg	Xp	0.234 m
Ix	946 kg-m^2	Yp	0.375 m

Table 2: Variable design parameter ranges of the vehicle model

Design Parameters	Lower bound	Upper bound
Kp (N/m)	90000 N/m	120000 N/m
Cp (Ns/m)	400 Ns/m	900 Ns/m
K1, K3 (N/m)	75000 N/m	100000 N/m
C1, C3 (Ns/m)	875 Ns/m	3000 Ns/m
K2, K4 (N/m)	32000 N/m	70000 N/m
C2, C4 (Ns/m)	875 Ns/m	3000 Ns/m

1) State Space Formulation

The system can be posed in state space form by selecting state variable as follows [16]

$$y_1 = \dot{Z}_{sr}; \quad y_2 = \dot{Z}_{ur}; \quad y_3 = \dot{Z}_{sl}; \quad y_4 = \dot{Z}_{ul};$$
$$y_5 = Z_{sr}; \quad y_6 = Z_{ur}; \quad y_7 = Z_{sl}; \quad y_8 = Z_{ul};$$

Then the state space equation of half car model is given as follow:

$$\{\dot{y}\} = A\{y\} + B\{f\} + D\{r\} \tag{5}$$

where

$\{y\} = [y_1 \quad y_2 \quad y_3 \quad y_4 \quad y_5 \quad y_6 \quad y_7 \quad y_8\,]^T$, is the state vector

$\{r\} = [Z_{rr} \quad Z_{rl}\,]^T$, is the road input vector

$\{f\} = [F_r \quad F_l]^T$, is the control force vector

A, B, and *D* are invariant coefficients and their expressions are given in Appendix A-2.

2) Mathematical Modelling for Rollover

When a vehicle runs on a curved road, a centripetal force acts at the centre of gravity of the vehicle which can be given by Equation 6.

$$F_C = \frac{MV^2}{R} \tag{6}$$

where F_C, M and V are centripetal force, mass and speed of the vehicle respectively. R is the curvature radius of the road. The car should not roll over as a result of this centripetal force. Because in our case study we are making the road run instead of the vehicle, an additional input to the car is given to analyse the rollover dynamics in along with the four wheels' road input disturbances. Therefore, a comparable centrifugal force is utilised as input to the system model to make the car equivalent while it is travelling on the curving road and to integrate the rollover. Equation 7 provides the roll moment that this centripetal force will have on the vehicle.

$$M_{roll} = F_c h_{roll} \tag{7}$$

where M_{roll} is the rolling moment acting on the vehicle and h_{roll} is the distance between the roll center and center of gravity of the vehicle. The rolling moment input is applied to the Equation 7 and the converted equation is given by Equation 8.

$$I_{xx}\ddot{\varphi} = -0.25w^2(2K_{sf} + 2K_{sr})\varphi - 0.25w^2(2C_{sf} + 2C_{sr})\dot{\varphi} + 0.5wK_{sf}z_{ufl} + 0.5wC_{sf}\dot{z}_{ufl} - 0.5wK_{sf}z_{ufr} - 0.5wC_{sf}\dot{z}_{ufr} + 0.5wK_{sr}z_{url} + 0.5wC_{sr}\dot{z}_{url} - 0.5wK_{sr}z_{urr} - 0.5wC_{sr}\dot{z}_{urr} + 0.5wF_{fl} - 0.5wF_{fr} + 0.5wF_{rl} - 0.5wF_{rr} + M_{roll} \quad (8)$$

Road Profile

Road profile imperfections come in three different varieties: smooth, rough small and rough. The flat road profile represents a single bump on the road. There are both minor and harsh road profiles, as shown by the consistent peak height and variable bump height, respectively [18]. Two different types of road input excitement a left wheel input and a speed breaker, respectively are presented in this study to help understand the behaviour. The left wheel input causes both the front left and rear right tyres to suddenly jolt. In response to the velocity breaker input, both front (or rear) tyres hit the bump. The road profile is shown graphically in Figure 2.the front and rear road bump inputs:

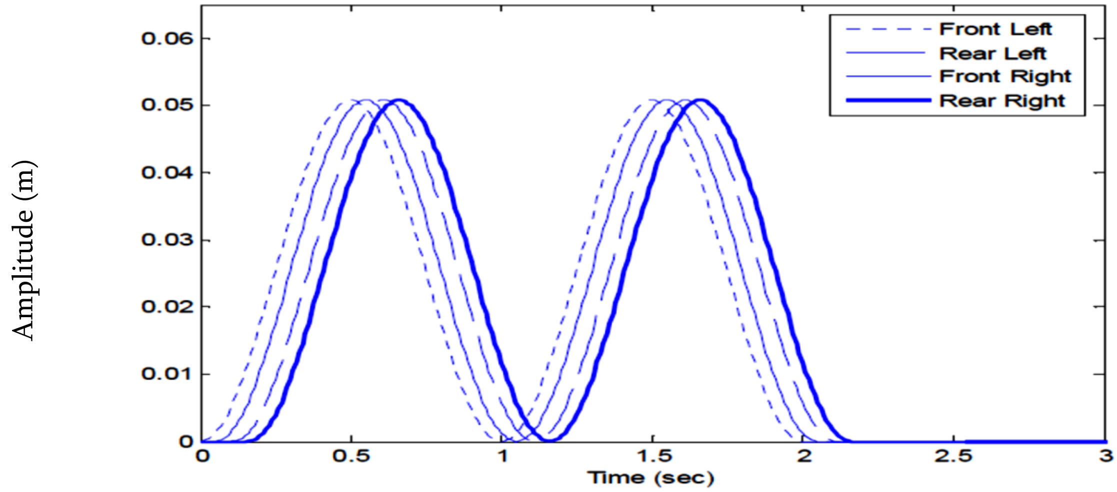


Figure 2: Bump road input disturbance for the front and rear wheels.

$$Z_{rf} = \begin{cases} \frac{a_0}{2}\left(1 - \cos\left(\frac{2\pi V t_f}{\lambda_0}\right)\right) & 0 \le t_f \le \frac{2\lambda_0}{V} \\ 0 & Otherwise \end{cases} \quad (9)$$

$$Z_{rr} = \begin{cases} \frac{a_0}{2}\left(1 - \cos\left(\frac{2\pi V (t_r - t_d)}{\lambda_0}\right)\right) & t_d \le t_r \le t_d + \frac{2\lambda_0}{V} \\ 0 & Otherwise \end{cases} \quad (10)$$

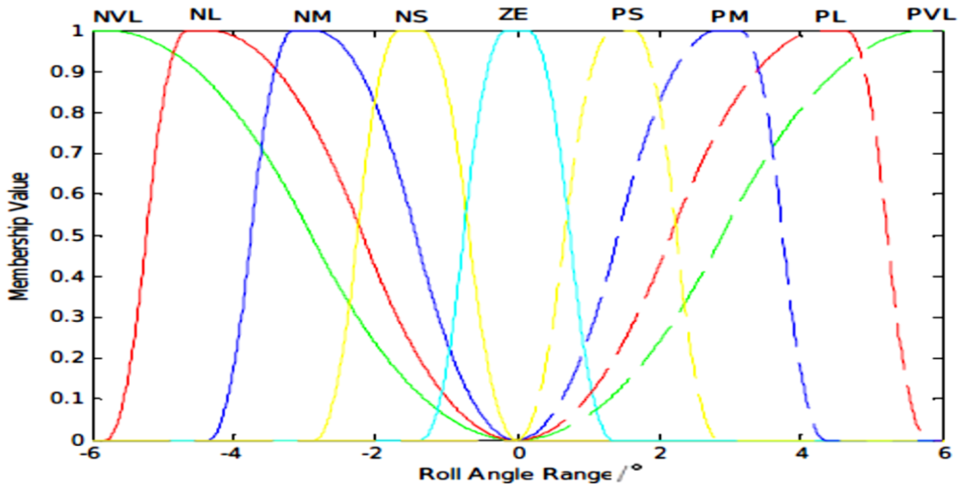
where t is the length of the simulation, a_0 denotes the size of the bump, V denotes the vehicle's forward speed, _0 denotes the wavelength of the disturbance, and the suffixes rf and rr denote the front and back wheel inputs for the suspension, respectively. tr = 1 + td, where td is the time difference between the front and back wheels.

$$t_d = \frac{a + b}{V} \quad (11)$$

For this study $a_0 = 0.05$ m; $\lambda_0 = .20$ m and $V = 20$ m/s

Fuzzy Logic Control System

The framework of fuzzy logic offers a simple approach for transforming an erroneous input into a precise input. The fuzzy logic system of control typically goes through three stages [1]. Specifically, defuzzification, control rule design, and fuzzification [4] [9] [10]. Fuzzification is the process through which input data with real numbers (which are crisp) are transformed into fuzzy data. The member function (MF) used at this step will be utilised to fuzzify the inputs and outputs of the suspension model. After that, the system for fuzzy inference is built using the IF-THEN control rules. During the defuzzification stage, the controller output is converted into fuzzy values to actual values. In this work, the input to the system's fuzzy control is the speed and acceleration of the vehicle body. The control force is the outcome of fuzzy logic control. According to some, the level surface at the tip of the trapezium in the trapezoidal MFs produces a stronger control force [4].



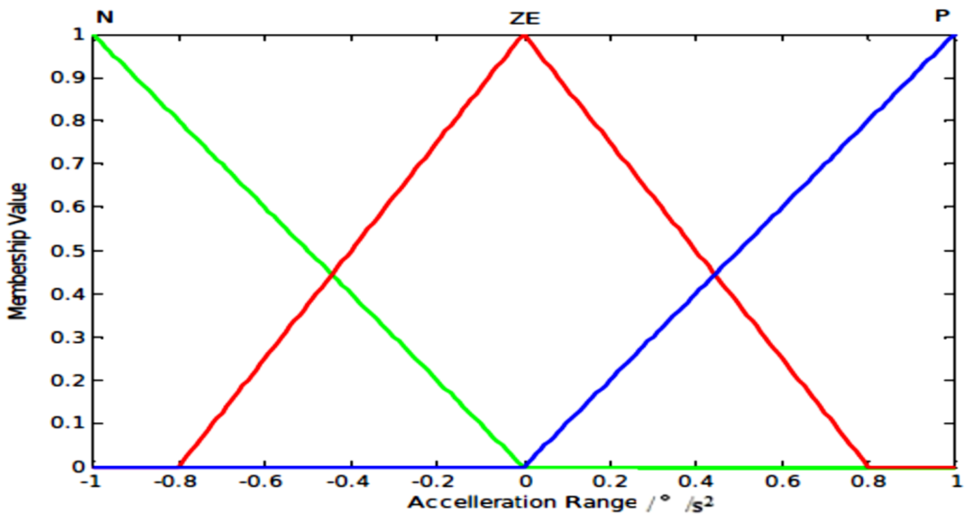


Figure 3: Mapping of membership function of FLC inputs

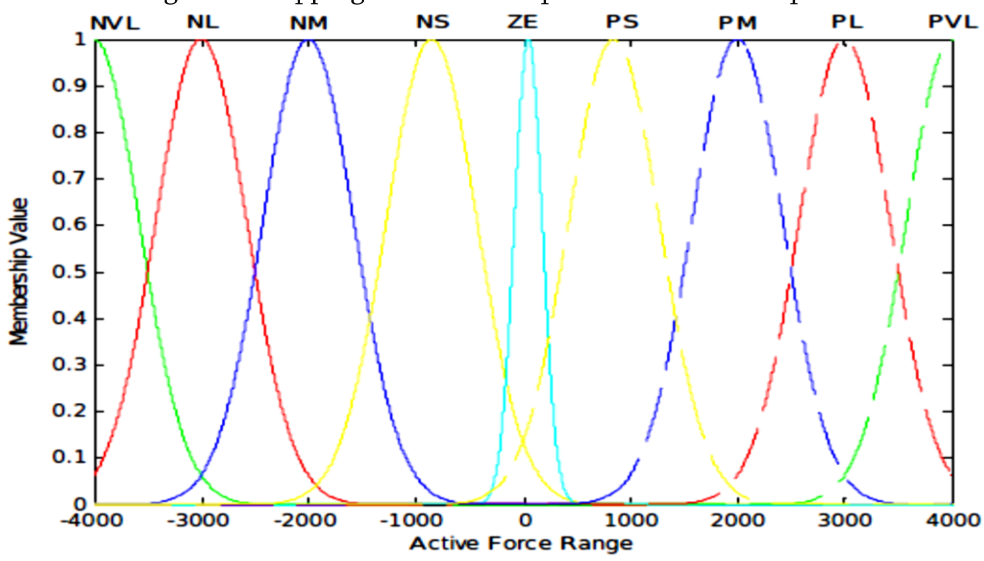


Figure 4: Mapping of output membership function of FLC

θ	$\ddot{\theta}$	f	θ	$\ddot{\theta}$	f	θ	$\ddot{\theta}$	f
NVL	N	NVL	NVL	ZE	NVL	NVL	P	NL
NL	N	NNL	NL	ZE	NL	NL	P	NM
NM	N	NL	NM	ZE	NM	NM	P	NS
NS	N	NM	NS	ZE	NS	NS	P	NS
ZE	N	ZE	ZE	ZE	ZE	ZE	P	ZE
PS	N	PM	PS	ZE	PS	PS	P	PS
PM	N	PL	PM	ZE	PM	PM	P	PS
PL	N	PVL	PL	ZE	PL	PL	P	PM
PVL	N	PVL	PVL	ZE	PVL	PVL	P	PL

Rule 1: IF (θ) = NVL AND ($\ddot{\theta}$) = N THEN (f) = NVL

Rule 15: IF (θ) = PS AND ($\ddot{\theta}$) = ZE THEN (f) = PS

The process of transforming a fuzzy output value into a crisp value that is utilised to manage the target plant is known as defuzzification. In this instance, the centroid of gravity approach is used to carry out the defuzzification stage. The following equation results from the controller's output.

$$f(x) = \frac{\int x_k \mu_k(x_k)dx}{\int \mu_k(x_k)dx} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (13)$$

II. RESULTS AND ANALYSIS

This chapter covers the operation of the linear active suspension mechanism as it is detailed in chapter two. Experiments based on a mathematical framework for quarter- and half-vehicle designs will be performed with the help of the MATLAB/SIMULINK programme. Road disturbance will be utilised as the system's input to assess the extent to which the suspension system works in terms of ride comfort and vehicle handling. Among the metrics that is going to be measured for quarter- and half-vehicle models are the suspension departure, wheel deflecting, and automobile body acceleration. The road pattern or disturbance is found. The objective is to achieve minimum amplitude values for the acceleration of the car body, suspension travel, and wheel deflection. The steady state of each component should also be quick.

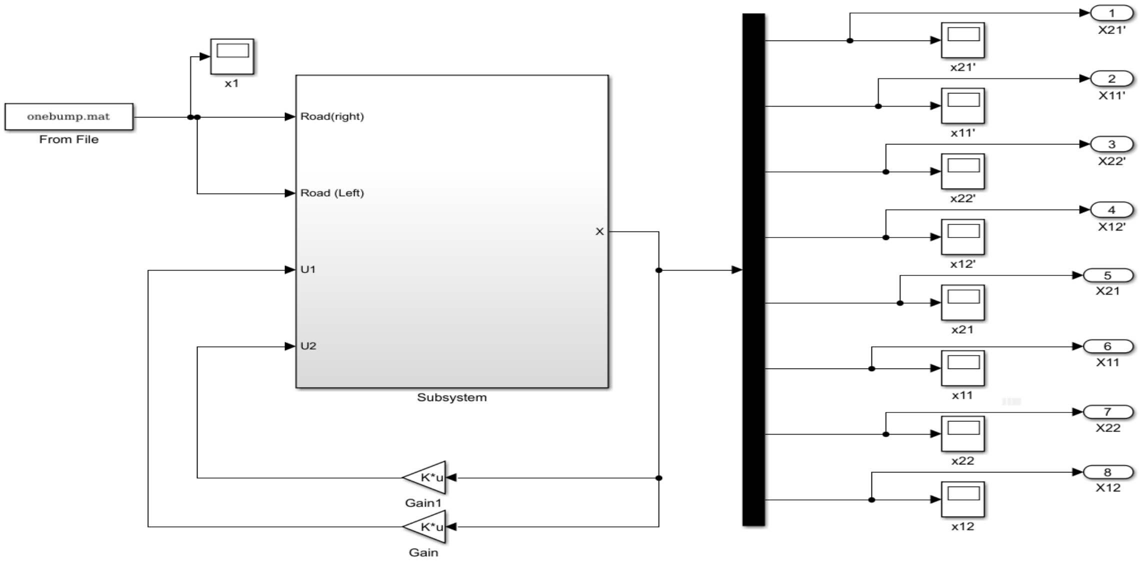


Figure 5. Half car 4 DOF passive suspension system

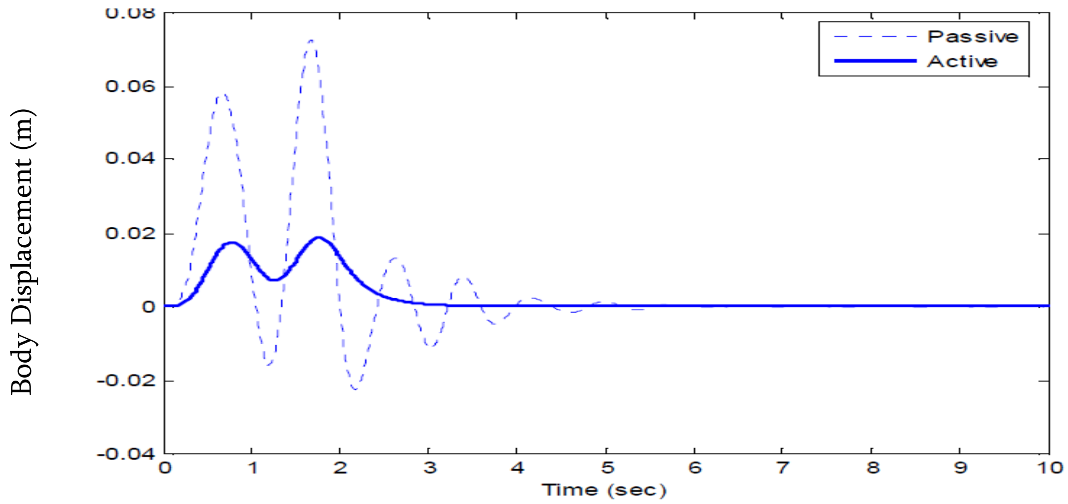


Figure 6: Body displacement (m)

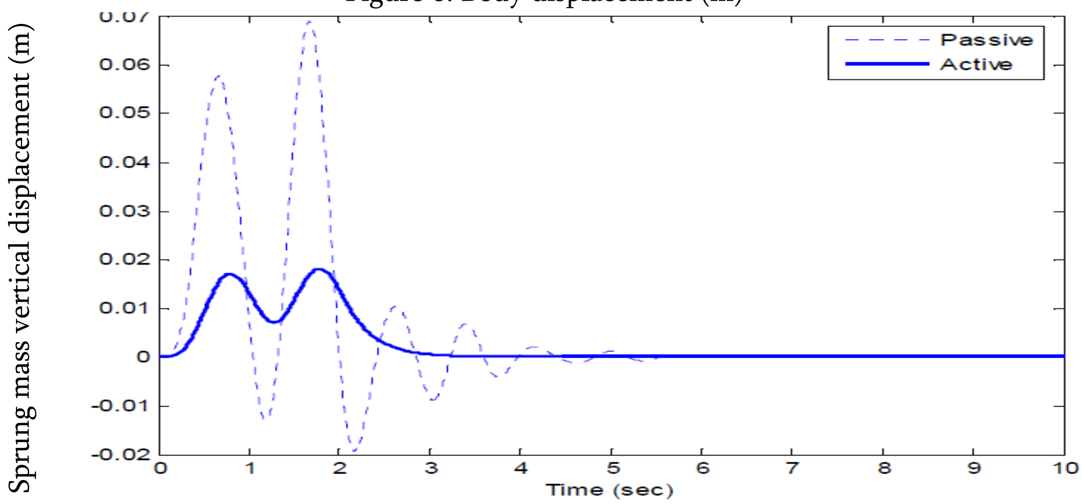


Figure 7: Vehicle sprung mass vertical displacement (m)

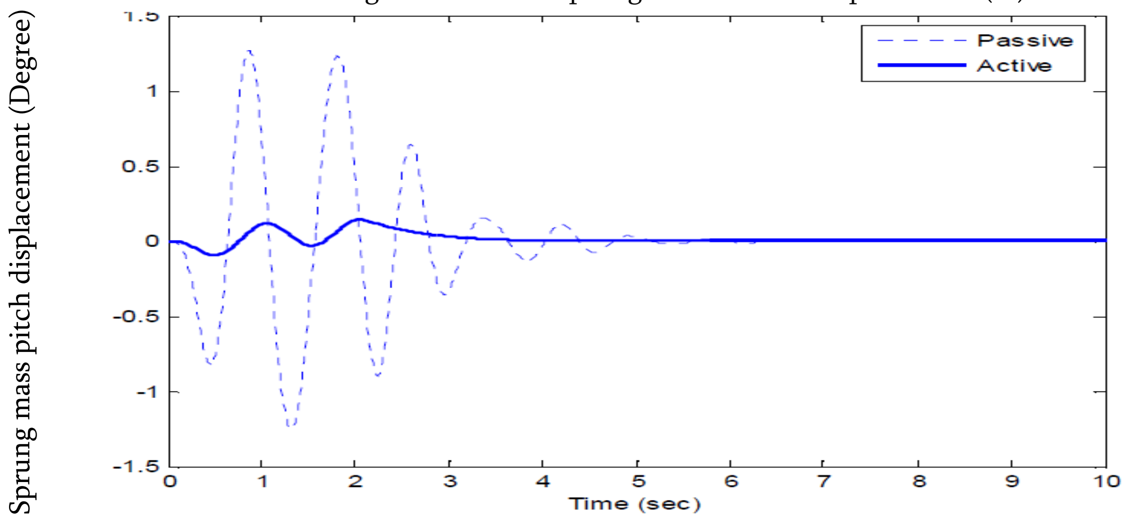


Figure 8: Sprung mass pitch displacement (Degree)

There are two parts to this section. The first section provides the ideal settings for the current models and contrasts the outcomes with simulated annealing, whereas the second half examines the modelling of the present suspension that have been constructed to achieve the greatest of their abilities. The vehicle information displayed in Table 4.1, the inputs for the road displayed in Figure 5, and the ideal suspension settings obtained using the evolutionary technique described and used in the simulation. The seats of both suspension systems exhibit pleasing natural frequencies between 0.8 and 1.5 Hz, as can be seen.

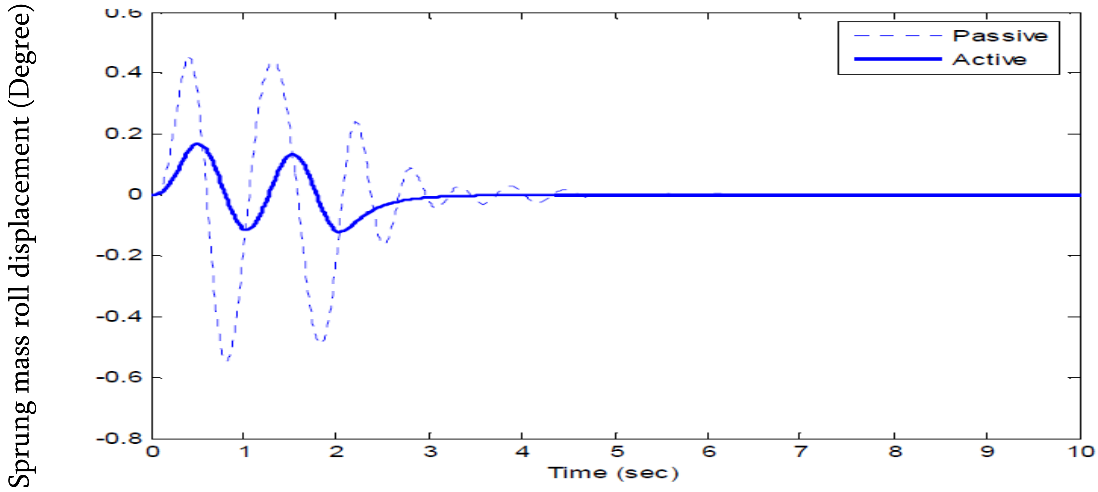


Figure 9: Sprung mass roll displacement (Degree)

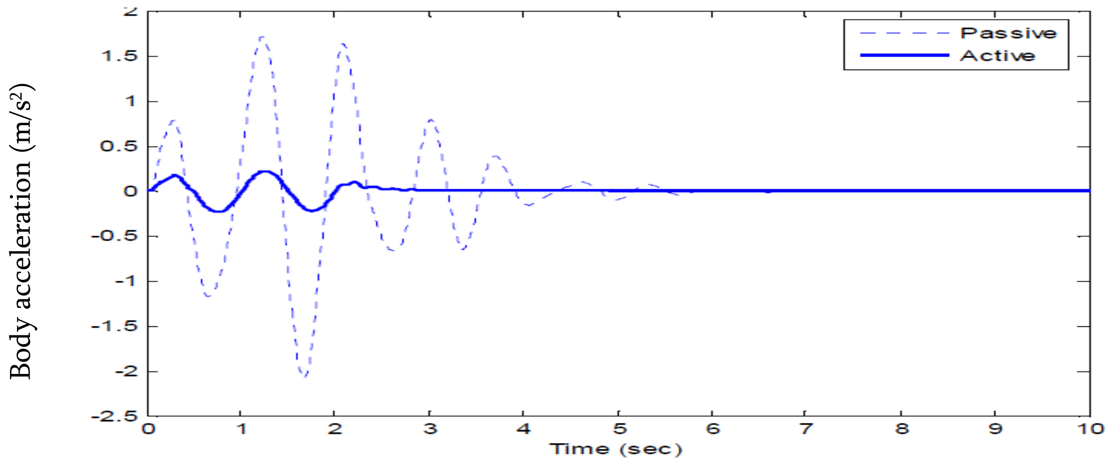


Figure 10: Vehicle body acceleration (m/s²)

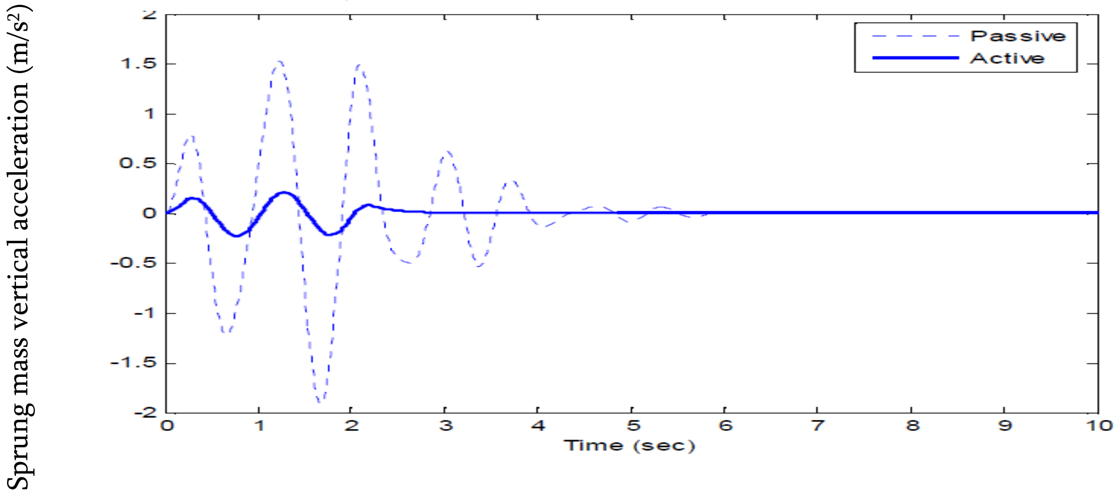


Figure 11: Sprung mass vertical acceleration (m/s²)

In comparison to passive suspension, active suspension lowers the motorist's vertical shift peak by approximately 74.2% and shortens the settling time from 6 seconds to 3.5 seconds, as seen in Fig. 6. Additionally, it can be demonstrated that, even though roll and pitch movement are exaggerated as well as a result, return to zero is comparably quick (Fig. 7 to 9), sprung mass displacement in the vertical direction is smaller with an active suspension than passive suspension. This will occur as a result of the Fuzzy controller for active suspension's design placing more emphasis on vertical movement for a comfortable ride.

Active suspension reduces seat acceleration with spring mass upwards acceleration approximately 88.17% and 88.72%, respectively, in comparison to passive suspension. Additionally, it reduces the settling time from 6.5 to 3 seconds, a 50% reduction.

Additionally, since ride comfort is given more weight in the active LQR controller design, the vertical weighted RMS acceleration of the seat and sprung mass is decreased from 0.3032 m/s² to 0.0534 m/s² and 0.2834 m/s² to 0.0492 m/s², respectively. The point of gravity of the spring's mass is where the spring mass weighed RMS acceleration is calculated despite the seat being close to the right front side of the tyres, as can also be seen.

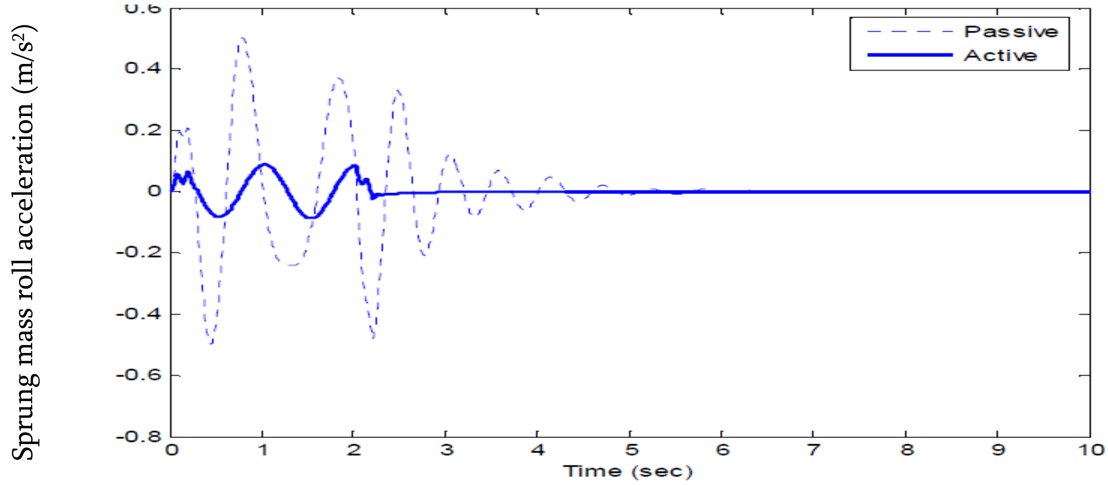


Figure 12: Sprung mass roll acceleration (m/s²)

When compared with passive suspension, the latter reduces roll acceleration by 65%, as seen in Fig. 11. As a result, it has been proven that an active suspension of action is unquestionably better than the passive situation.

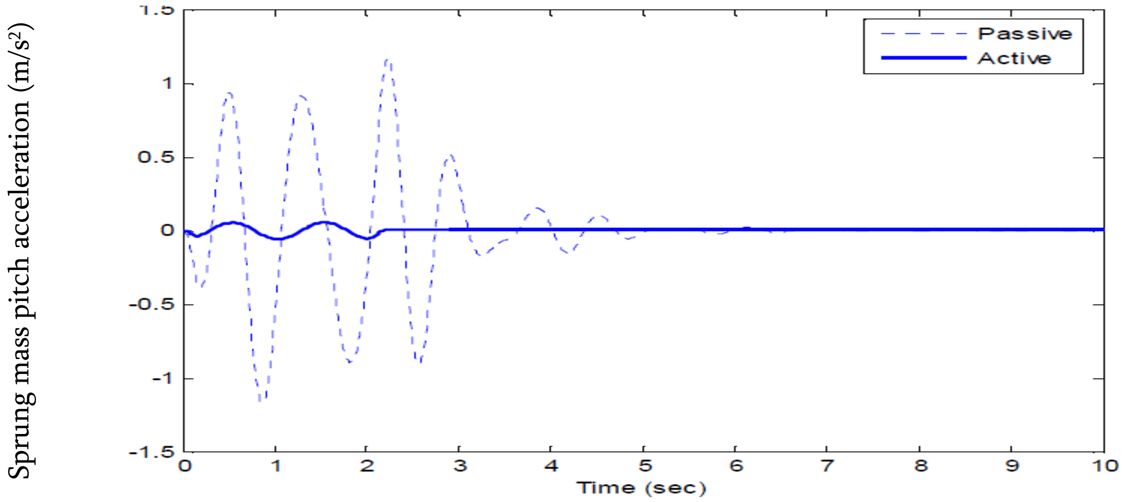


Figure. 13: Sprung mass pitch acceleration (m/s²)

Active suspension has a smaller acceleration amplitude spectrum, which results in incredibly quick returns to zero with regard to the angle of acceleration observed in Fig. 13. Greater amplitude disruptions were also found at about 0.6 and 1.6 seconds. When we look at the excitation we can see that these disturbances are most likely brought on by the wheel motion's phase angle being somewhat ahead of the disturbance.

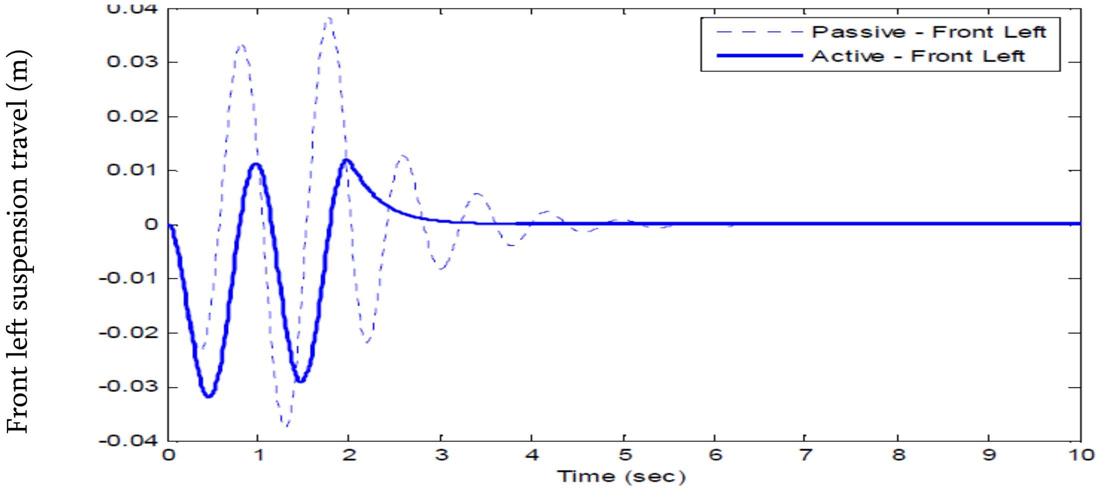


Figure 14: Front left suspension travel (m)

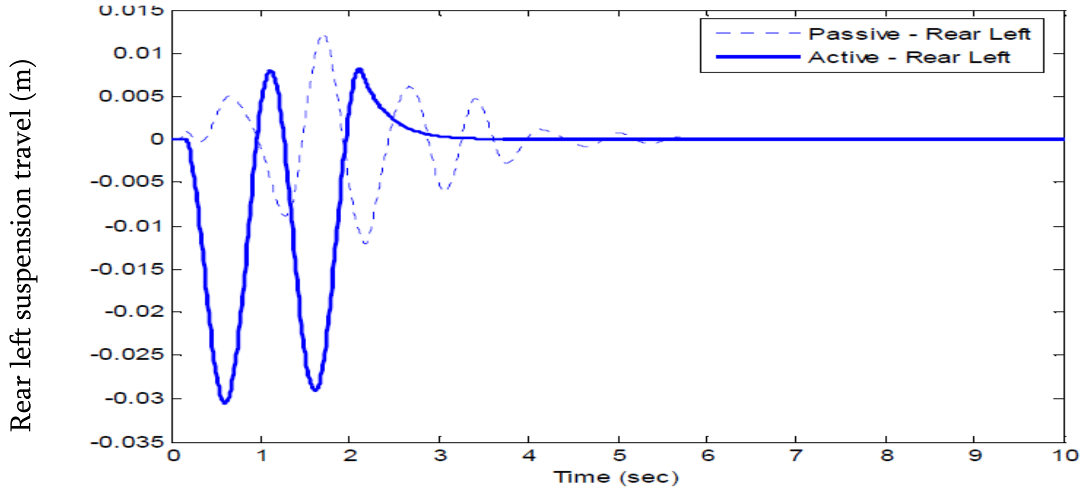


Figure 15: Rear left suspension travel (m)

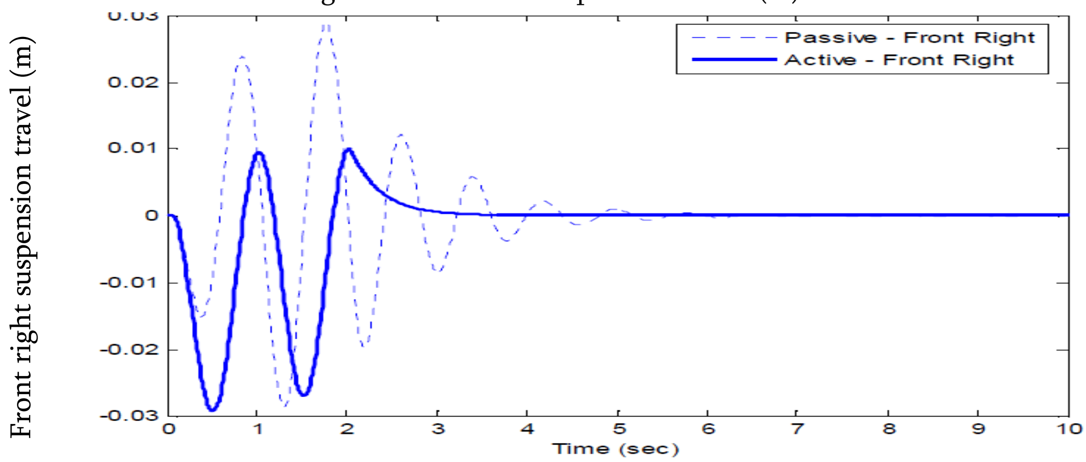


Figure 16: Front right suspension travel (m)

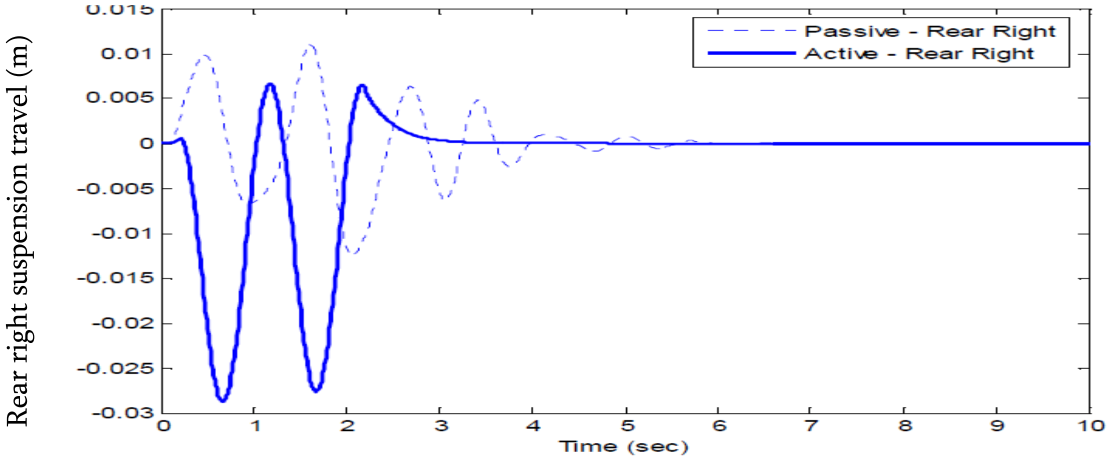


Figure 17: Rear right suspension travel (m)

In order to provide an improved ride experience with fewer movements of the sprung mass, an active suspension system's suspension travel improves by 56–60% in contrast to a passive system of suspension (Fig. 14–17).

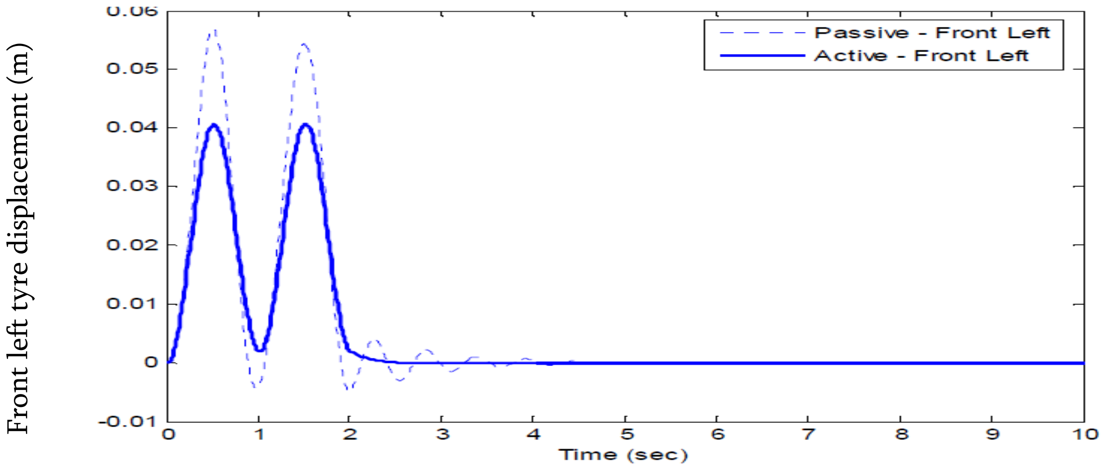


Figure 18: Front left tyre displacement (m)

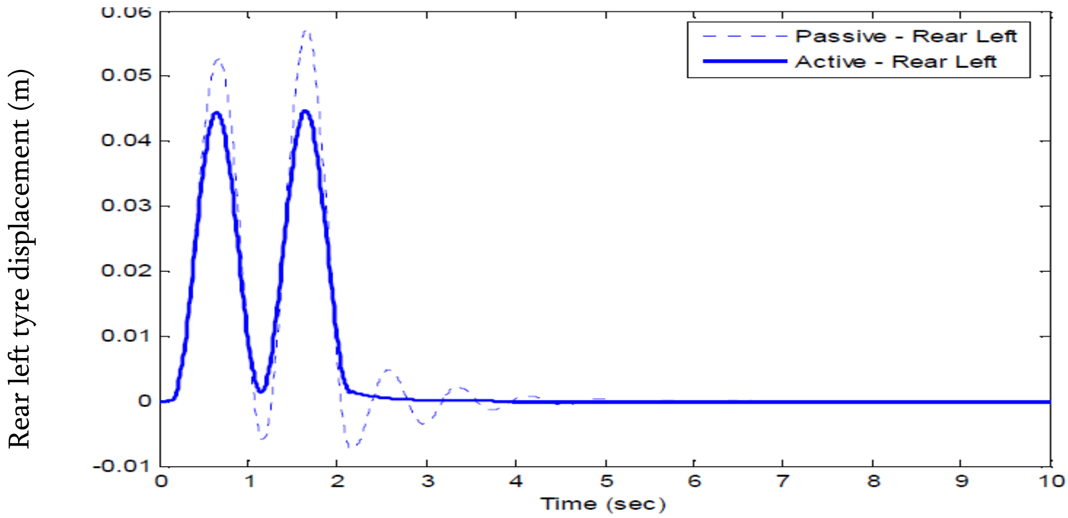


Figure 19: Rear left tyre displacement (m)

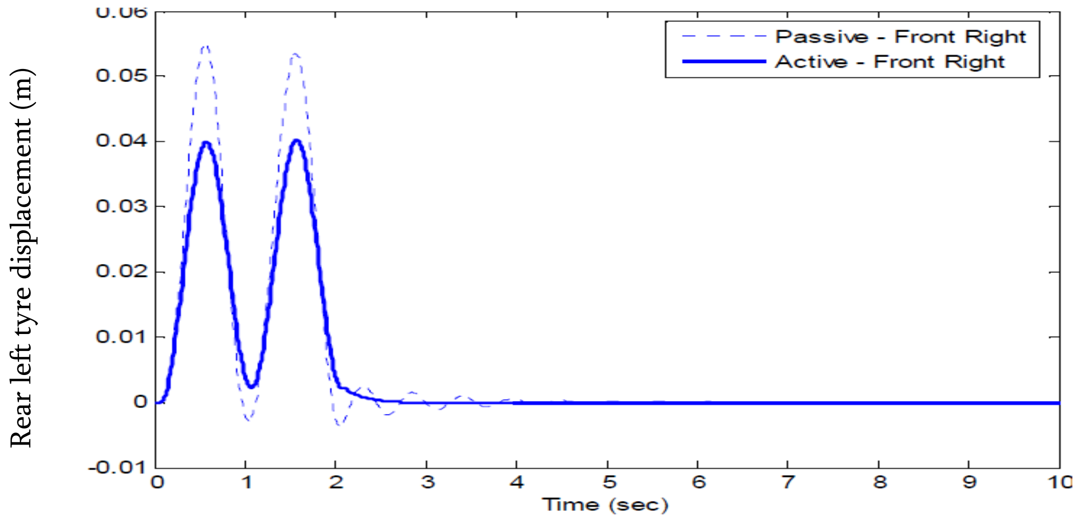


Figure 20: Front right tyre displacement (m)

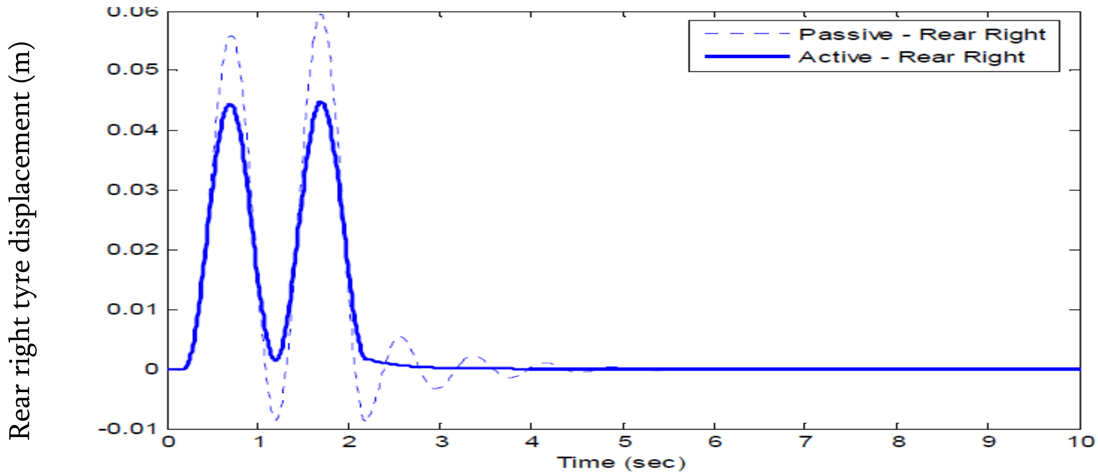


Figure 21: Rear right tyre displacement (m)

Additionally, an active suspension system improves road grip because the tyre movement is around 28.5% lower with it than with the passive suspension system. Additionally, it can be deduced that active suspension results in greater suspension movement that passive suspension since the features of travel of the suspension and road stability were mutually exclusive.

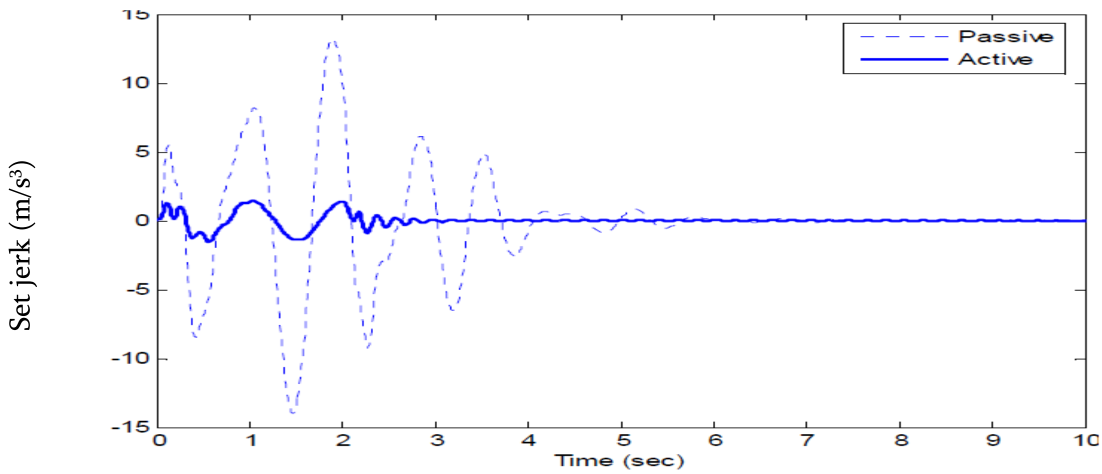


Figure 22: Set jerk (m/s³)

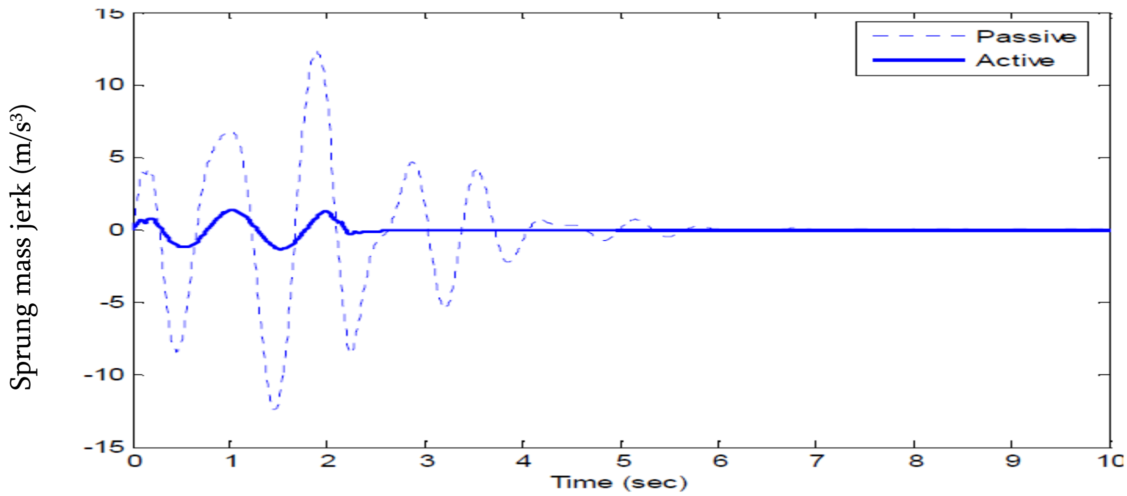


Figure 23: Sprung mass jerk (m/s³)

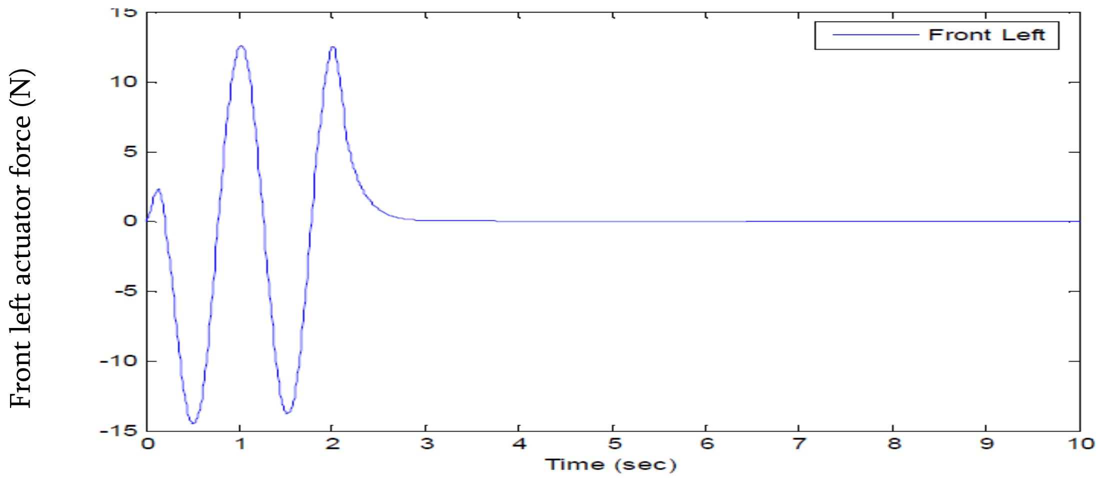


Figure 24: Front left actuator force (F)

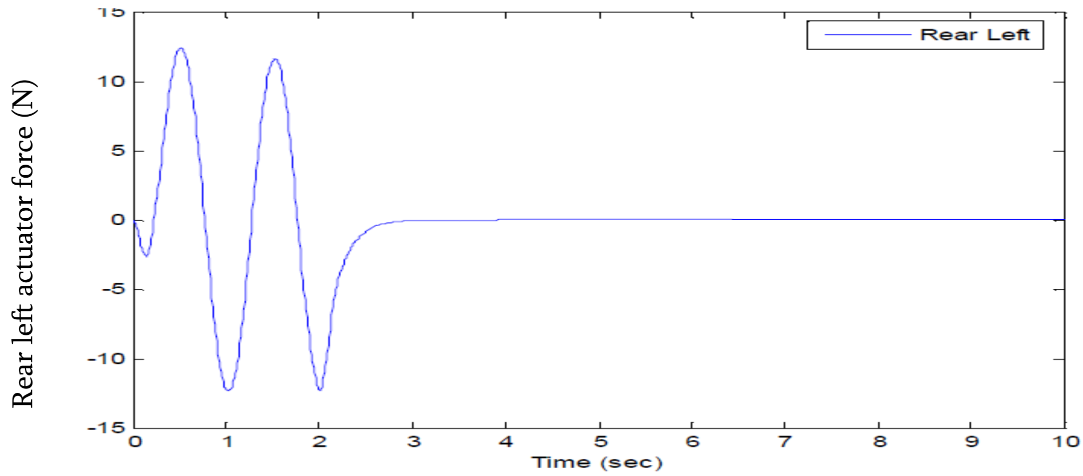


Figure 25: Rear left actuator force (F)

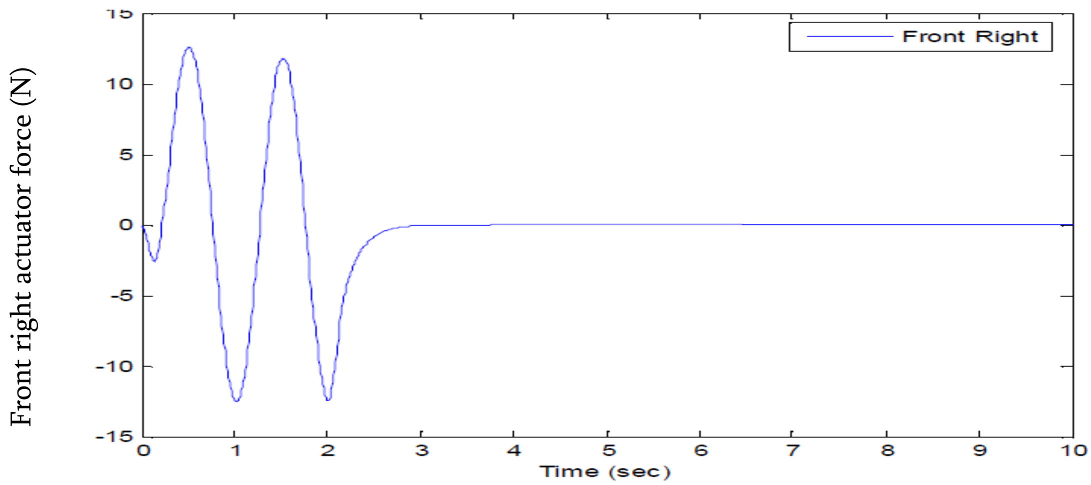


Figure 26: Front right actuator force (F)

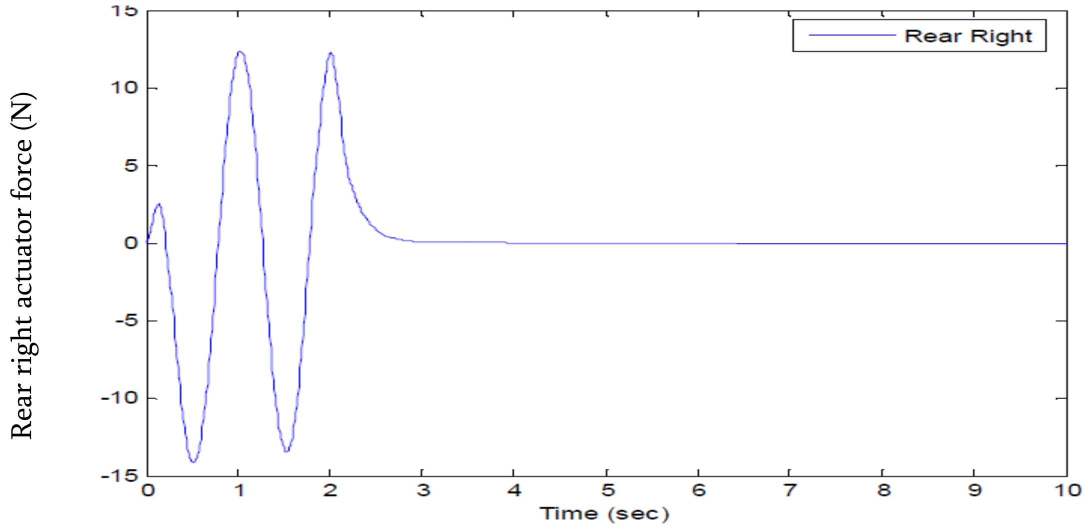


Figure 27: Rear right actuator force (F)

Additionally, active suspension causes a slower rate of shift in acceleration, as seen in Fig. 22 and 23. As a result, the driver's position and the mass of springs feel very less jerk when compared to passive suspension. Figures 25–27 depict the actuator forces required for active suspension; all of them are well within the constraints that apply and can be successfully implemented.

III. CONCLUSION

The goal of the current study was to make the greatest car suspensions using the FUZZY comptroller and taking into consideration its strengths and capabilities. As design objectives to assess the comfort of the suspension, the greatest bouncing velocity of the position and sprung weight, the roots mean square (RMS) measured acceleration of the position and sprung mass in accordance with ISO2631 requirements, a jerk the suspension travel, road a holding, as well as tyre bending are all given. Despite the enormous exploring space for the parameters, the solution area is severely constrained by a number of constraints. As an outcome, the limited optimization problem is transformed using the penalty function technique into an unconstrained one.

According to the simulation results, switching to active suspension decreases the driver's vertical shift peak by roughly 74.2% and cuts the settling time in half, between 6 seconds to three seconds. Because ride comfort is given more weight in the active LQR controller design, the vertical weighted RMS velocity of the seats and sprung mass is reduced to 0.3032 m/s2 to 0.0534 m/s2 and 0.2834 m/s2 to 0.0492 m/s2, respectively. While tyre displacement is reduced by 28.5% to provide road grip, active suspension travel increases by 56–60% over passive suspension to provide a more comfortable ride. This demonstrates how adaptive suspension systems can more effectively increase comfort and road holding.

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