Active Suspension System Design Using Fuzzy Logic Control and Linear Quadratic Regulator

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Abstract. The motor vehicle industry has shown a mechatronics system with intelligent control systems. Mechatronics refers to a successful combination of mechanical and electronic systems. In mechatronics, traditional systems of mechanical engineering are combined together with components from computer science, mathematics and electrical engineering. This paper presents enhancing an active suspension for a quarter car model to improve its performance by applying a specific controller. Separating a vehicle's body from road abnormalities is the major purpose of a suspension system, in order to provide the maximum ride comfort for passengers and keep hold of continuous road wheel contact to provide road holding. First controller applied is fuzzy logic controller (FLC), and the second one is a Linear Quadratic Regulator, the car's behaviour such as car body displacement, suspension deflection, and wheel travel is considered to obtain maximum damping force in the actuator. A comparative study has been verified to get the best performance for comfort of passenger ride and road managing. AQ1 AQ2 AQ3

Keywords: Active suspension system · Fuzzy logic control (FLC)
Linear quadratic regulator (LQR) · Servo valve · Quarter car model
Sprung mass · Unsprung mass

1 Introduction

Luxury automobile companies like Volvo, Mercedes-Benz and BMW has been using active suspension system as a part of their designed manufacture. Because of the active suspension systems' high cost, power supply requirements and difficulty, that's why barely luxury cars manufacture it until nowadays. Allen constructed a quarter-car modeled test to apply different control methods on suspension systems [1]. Therefore, there were different designs applied since the early days of vehicles [2]. Methods of control for an active suspension system were developed by different scientists [3–8]. These research results can be classified depending on the used control methods. The definition of the word "Suspension" is referrers to the mechanical parts, that attaches a car chassis with axles and wheels. The performance of the suspension system depends on achieving passenger comfort while driving the vehicle, road holding, and handling [9].

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Passenger comfort and safety is the main task of a designed suspension system, therefore designs usually begin with this mission. In the last decades, active suspension systems have played a main task in the academic world and manufacture. Passive and active suspensions are the main types of an automobile suspension system. A Passive suspension system is reliable and easier and less expensive than active suspension, but its performance is limited, and it fails in the aspect of comfort. But it can properly be balanced through active suspension systems, because the active suspension system is able to regulate, control parameters in order to achieve road unevenness [10, 11].

Lately, a lot of theories were suggested to take control of the suspension problems In this paper, fuzzy logic control and Linear, quadratic regulator is planned as a tactic to control active suspension systems and simulation process was applied by MATLAB/SIMULINK. Numerous designers have chosen fuzzy logic control (FLC) to be an other control technique for the Active Suspension systems (ASS) [12–14]. In this paper, the considered model for the suspension system is the Quarter car model (QCM), the analysis for this system depends on two types of masses, sprung and unsprung mass [15]. Inactive suspension system Electro-hydraulic servo valves are the active parts in the system. Because of their high accuracy, they are used in applications which precise position control is occurring and they have the main effect on the whole electro-hydraulic servo systems' operation [16, 17].

This paper has studied the effect of road irregularities in order to achieve the optimal damping force as resultant with enhancement of passenger comfort, riding stability and road handling, and this is achieved by applying different control strategies which are FLC and LQR and a PID Controller.

2 Theoretical and Simulation Models

A QCM is tested in this paper as an alternative of a full car's model, two degrees of freedom is considered. The QCM shows the main characteristics of a full model and also simplifies the analysis, Fig. 1, shows a simplified block for the QCM and Table 1 shows the values for the system parameters. Assuming that the wheels have a continuous road contact and that Z_s and Z_u are calculated from static equilibrium point. We also assume that the driver is driving with constant speed, to design the road input. Equations of the systems are:

$$m_s\ddot{Z}_s = k_s(Z_u - Z_s) + b_s(\dot{Z}_u - \dot{Z}_s) + Fa \tag{1}$$

$$m_u\ddot{Z}_u = -k_s(Z_u - Z_s) - b_s(\dot{Z}_u - \dot{Z}_s) - k_t(Z_r - Z_u) - Fa \tag{2}$$

Where Fa stands for the input control variable from the Electro-Hydraulic Servo Valve and Z_r is the input disturbance from the road and it can be modelled as white noise. The state-space variables are identified as: $x_1 = Z_s - Z_u$ Sprung-Mass Displacement.

$x_2 = \dot{Z}_s$ Sprung-Mass Absolute Velocity.
$x_3 = Z_u - Z_r$ Unsprung-Mass Displacement (Tire Deflection).
$x_4 = \dot{Z}_u$ Unsprung-Mass Absolute Velocity.

Equation of motion is described as:

$$\dot{x} = Ax + Bu \tag{3}$$

$$y = Cx + Du \tag{4}$$

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -ks/ms & -bs/ms & ks/ms & bs/ms \\ 0 & 0 & 0 & 1 \\ ks/mu & bs/mu & (-kt-kt)/mu & -bs/mu \end{bmatrix}, \quad B = \begin{bmatrix} 0 & 0 \\ 1/mb & 0 \\ 0 & 0 \\ -1/mu & kt/mu \end{bmatrix}$$

$$C = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 1 & 0 & -1 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix}, \quad D = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \end{bmatrix}.$$

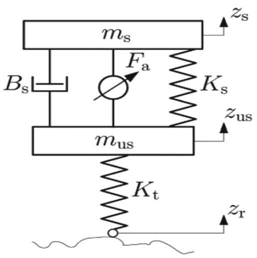

Fig. 1. Block diagram for the active suspension system of a QCM.

Table 1. Parameter values for suspension system.

Symbol	Parameter	Numerical value (unit)
M_s	Sprung-mass	250 (kg)
M_u	Unsprung-mass	50 (Kg)
K_s	Suspension stiffness	16812 (N/m)
K_t	Tire stiffness	190000 (N/m)
b_s	Damping coefficient	1000 (Ns/m)

2.1 Passive Suspension

It is mainly used in middle and low end cars. Springs and dampers are the main components of a passive system as shown in Fig. 2. No feedback or control algorithms are applied in this kind of suspensions. Passive systems are restricted systems, they have limited boundaries and do not enhance their behaviour over the environmental change.

As a result, they are generally successful with low limited input disturbances. Thus, they can't attain the optimal values for the passenger ride comfort nor road handling.

Therefore, many control theories have been applied on suspension systems to get the maximum ride comfort and road handling.

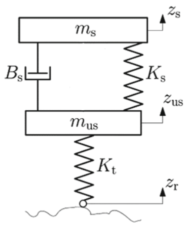

Fig. 2. A schematic diagram for passive suspension system.

2.2 Proportional–Integral–Derivative Controller (PID Controller)

PID controller depends generally on a feedback control loop theory, that is applied on many manufacturing industries. The main technique of this type of controller is that it tries to make the Error equals to a zero value, which the error is the difference between a set-point and the resultant value from the feedback of the plant.

Figure 3, shows a brief comparison between Passive suspension system and Active suspension system controlled by PID Controller, and the simulation results are shown in Figs. 4, 5, 6, 7, 8, 9 and 10, after applying different input variables. AQ4

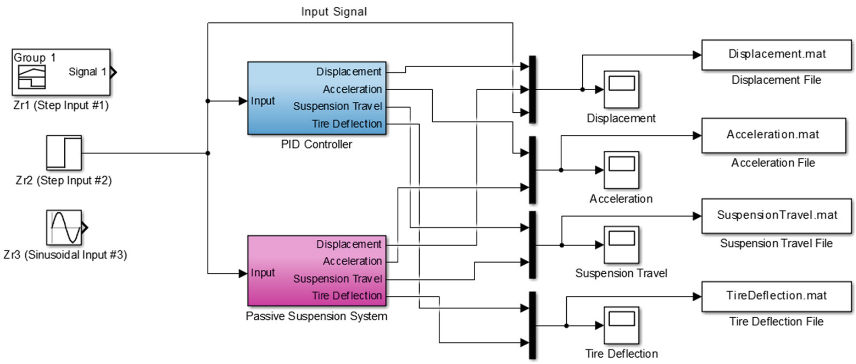

Fig. 3. Simulink model for passive suspension and active suspension system using PID.

After applying the step input, the system responce for passive suspension and PID will be shown in Figs. 4, 5, 6 and 7.

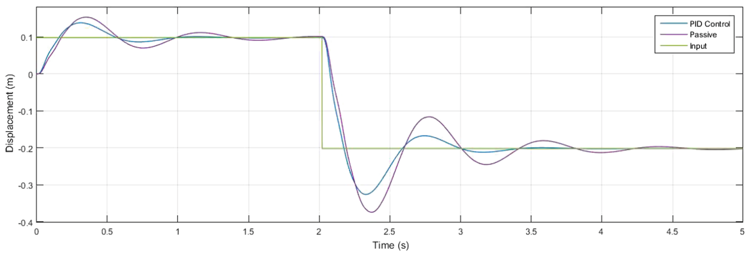

Fig. 4. Car body displacement for passive suspension system and PID controller.

Table 2 and Fig. 5, show the overshoot, settling time and rise time for two compared systems.

Table 2. Overshoot, settling time and rise time.

	Passive	PID
Overshoot	0.054	0.038
Settling time	4.6	3.7
Rise time	0.173	0.156

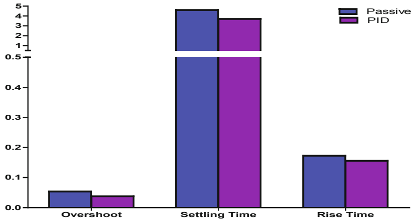

Fig. 5. Passive vs PID chart.

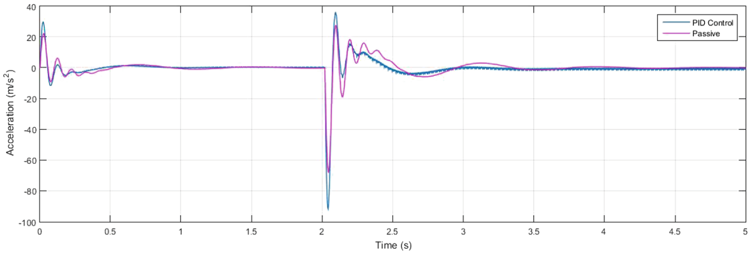

Fig. 6. Car body acceleration for passive suspension system and PID controller.

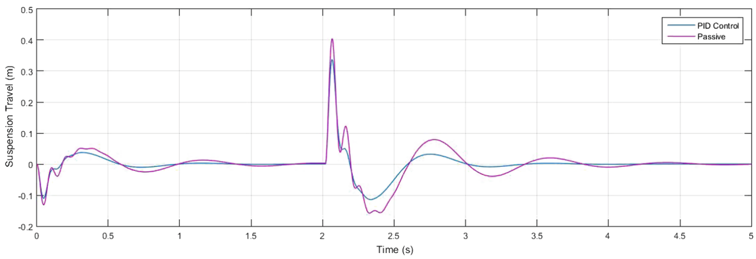

Fig. 7. Suspension travel for passive suspension system and PID controller.

Figure 8, show system responce for passive suspension and PID after applying step input step time 0.1 s and amplitude 0.5 m step.

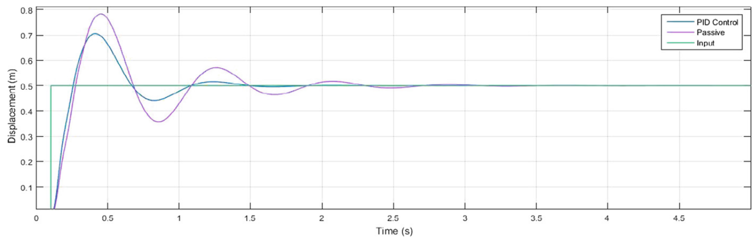

Fig. 8. Car body displacement for passive suspension system and PID controller.

Table 3 and Fig. 9, show the overshoot, settling time and rise time for two systems compared.

Table 3. Overshoot, settling time and rise time.

	Passive	PID
Overshoot	0.284	0.206
Settling time	3.9	2.3
Rise time	0.27	0.25

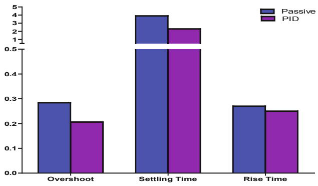

Fig. 9. Passive vs PID chart.

Figure 10, show system response for passive suspension and PID after applying sinusoidal wave frequency of 7.7 rad/s and amplitude 0.1 m.

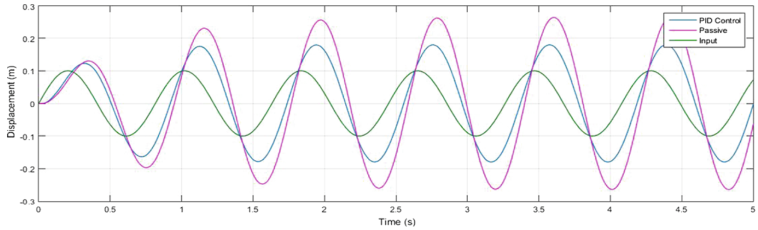

Fig. 10. Car body displacement for passive suspension system and PID controller.

2.3 Fuzzy Logic Control Design

In 1965, FLC was invented by Zadeh [19]. FLC consists of three main steps (1) **Fuzzification** (Use Membership-Functions to graphically illustrate a condition), (2) **Rule Evaluation** (Applying fuzzy rules on the system), and (3) **Defuzzification** (Achieving the crisp or actual results). The Simulink model and the designed model used in an active suspension system by applying FLC are shown in Figs. 11 and 12, respectively. In the designed Fuzzy model two input variables are obtained, which are Error and Error-Derivative, and one output variable as shown in Figs. 13, 14 and 15.

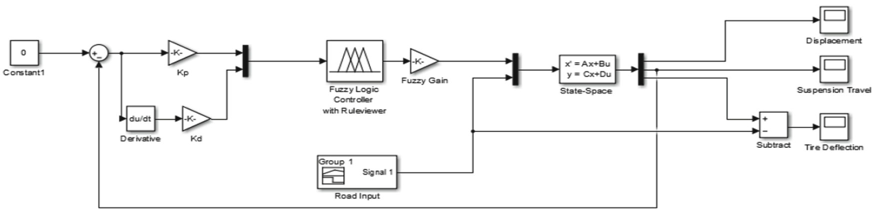

Fig. 11. Simulink model for active suspension system using FLC.

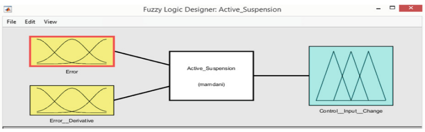

Fig. 12. FLC.

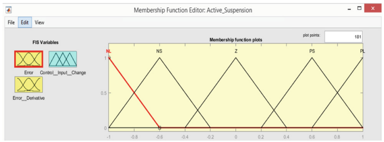

Fig. 13. Input variable “Error”.

The control system consists of three stages: falsification, fuzzy inference process and defuzzification. There are five membership functions for both input and output, the membership functions are triangular-shaped with equaled width. Universe of discourse is from [−1 1].

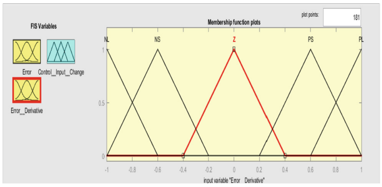

Fig. 14. Input variable “Error_Derivative”.

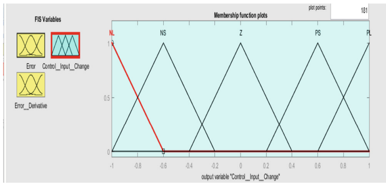

Fig. 15. Output variable “Control Output Change”.

The rule base applied in the designed system is illustrated by the Table 4 with fuzzy terms gained from the designer’s practice and knowledge. The output-surface of the FLC system is illustrated in Fig. 16.

Table 4. Rule base of FLC.

	Error-derivative					
Error		NL	NS	Z	PS	PL
	NL	NL	NL	NL	NS	Z
	NS	NL	NL	NS	Z	PS
	Z	NL	NS	Z	PS	PL
	PS	NS	Z	PS	PL	PL
	PL	Z	PS	PL	PL	PL

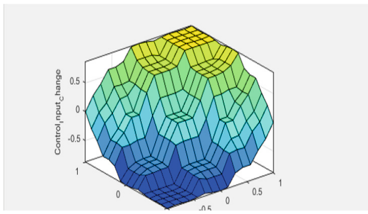

Fig. 16. The output surface of the fuzzy system.

2.4 Linear Quadratic Regulator Design

The LQR control theory is a great method for designing controllers with difficult systems that have strict performance necessities. It seeks for achieving the best controller parameters that reduces a specified cost-function. Q and R are weighs the state vector and the system input correspondingly, and they are the main vectors of the cost function. The methods used to solve the LQR parameters are the State-space model, figuring out the control law and calculating the feedback gain. The feedback gains applied in the LQR system will guide to the best results. In this research, the feedback gains and the state-feedback controller is designed using the LQR controller [16]

$$\dot{\boldsymbol{x}}(\boldsymbol{t}) = \boldsymbol{Ax}(\boldsymbol{t}) + \boldsymbol{Bu}(\boldsymbol{t}), \quad \boldsymbol{t} \geq \mathbf{0}, \boldsymbol{x}(\mathbf{0}) = \boldsymbol{x}_{\mathbf{0}} \tag{5}$$

Where: $\boldsymbol{x}(\boldsymbol{t})$ is the state-vector and $\boldsymbol{u}(\boldsymbol{t})$ is the input-vector. It is required to determine the matrix $\boldsymbol{K}$ also the static full state feedback control law,

$$\boldsymbol{u}(\boldsymbol{t}) = -\boldsymbol{Kx}(\boldsymbol{t}) \tag{6}$$

Satisfies the following criteria:

- the closed loop system is closely stable.
- the quadratic performance function is minimized.

$$\boldsymbol{J}(\boldsymbol{K}) = \frac{\mathbf{1}}{\mathbf{2}} \int_{\mathbf{0}}^{\infty} \left[\boldsymbol{x}^{\boldsymbol{T}}(\boldsymbol{t})\boldsymbol{Qx}(\boldsymbol{t}) + \boldsymbol{u}^{\boldsymbol{T}}(\boldsymbol{t})\boldsymbol{Ru}(\boldsymbol{t})\right] \boldsymbol{dt} \tag{7}$$

Q is a nonnegative definite matrix that penalizes the departure of system states from the equilibrium and R is a positive definite matrix that penalizes the control input [15, 18]. LQR problem can be solved by achieving lagrange-multiplier method and is specified by:

$$\mathbf{K} = \mathbf{R}^{-1}\mathbf{B}^{\mathbf{T}}\mathbf{P} \tag{8}$$

where P is a nonnegative definite matrix satisfying the Riccati-matrix Equation,

$$\boldsymbol{A}^{\boldsymbol{T}}\boldsymbol{P} + \boldsymbol{PA} + \boldsymbol{Q} - \boldsymbol{PBR}^{-1}\boldsymbol{B}^{\boldsymbol{T}}\boldsymbol{P} = \mathbf{0} \tag{9}$$

The following LQR design algorithm is used to determine the optimal state feedback. To define the best possible state feedback, a designed LQR algorithm is applied as shown:

- Riccati matrix equation is first solved

$$-\boldsymbol{A}^{\boldsymbol{T}}\boldsymbol{P} - \boldsymbol{PA} - \boldsymbol{Q} + \boldsymbol{PBR}^{-1}\boldsymbol{B}^{\boldsymbol{T}}\boldsymbol{P} = \mathbf{0}$$

- Optimal state $\boldsymbol{x}^*(\boldsymbol{t})$ is determined from

$$\dot{\boldsymbol{x}}^*(\boldsymbol{t}) = \left[\boldsymbol{A} - \boldsymbol{BR}^{-1}\boldsymbol{B}^{\boldsymbol{T}}\boldsymbol{P}\right]\boldsymbol{x}^*(\boldsymbol{t}) \tag{10}$$

  Starting with $\boldsymbol{x}(\boldsymbol{t}_0) = \boldsymbol{x}_0$ as initial conditions
- The optimal control $\boldsymbol{u}^*(\boldsymbol{t})$ is obtained from

$$\boldsymbol{u}^*(\boldsymbol{t}) = -\boldsymbol{R}^{-1}\boldsymbol{B}^{\boldsymbol{T}}\boldsymbol{Px}^*(\boldsymbol{t}) \tag{11}$$

- The optimal performance index is obtained from

$$J^* = \frac{1}{2} x^*(t) P x^*(t) \tag{12}$$

The weighting matrices Q and R are the main parameters developing any LQR system. The structure of Q and R values have a huge effect on systems reaction. The number of elements of Q depends on the number of state variable n, while the number of elements of R depends on the number of state variables of m [2, 14].

In the designed LQR system the state-space model is based on:

$$x_1 = \theta, x_2 = \dot{\theta}, x_3 = x, x_4 = \dot{x} \tag{13}$$

$$\dot{x}_1 = x_2$$

$$\dot{x}_2 = 62.43x_1 + 0.91x_4 - 9.09u$$

$$\dot{x}_3 = x_4$$

$$\dot{x}_4 = -2.67x_1 - 0.18x_4 + 1.8u \tag{14}$$

where θ is the pendulum angle measured from vertical reference, $\dot{\theta}$ is the rotational speed of pendulum, x is the cart position, $\dot{x}$ is the cart speed, and u is the input force. Matlab was used to solve Riccati's Equation to get the best gain vector. Simulink model for the designed LQR control system is shown in Fig. 17.

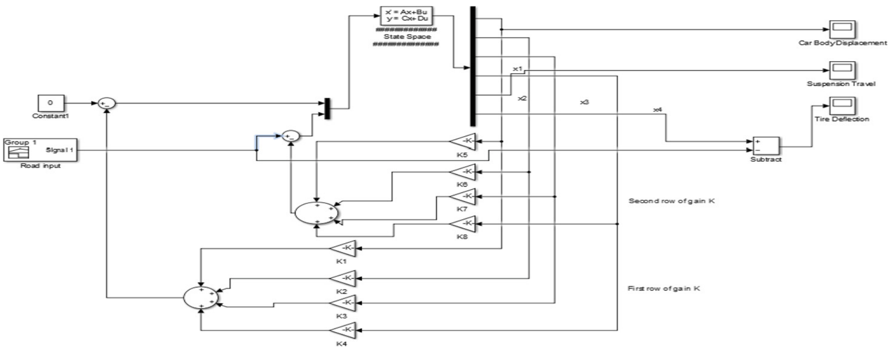

Fig. 17. Simulink model for active suspension system using LQR control system.

After applying the LQR equations on the designed system, these are the applied matrices for the designed system:

A = [0 1 0 0; [−ks −bs ks bs]/mb; 0 0 0 1; [ks bs −ks−kt −bs]/mw];
B = [0 0; 1/mb 0; 0 0; −1/mw kt/mw]; C = [1 0 0 0; 0 1 0 0; 0 0 1 0; 0 0 0 1];
D = [0 0; 0 0; 0 0; 0 0]; Q = [120 0 0 0; 0 1 0 0; 0 0 15 0; 0 0 0 0.06];
R = [100,0; 0,500];

3 Results and Discussion

The aim of this work is obtaining an optimal controller that achieves the best ride comfort and road handling qualities. In last few years, researchers have been trying to use different controllers instead of the passive system. The following results show an efficient active suspenstion system technique that uses different control theories for obtaining the best damping force and least settling time.

A brief comparison is described in this section between PID Controller, fuzzy logic controller and Linear Quadratic Regulator. Simulation results are introduced to evaluate the applied controllers. Different road profiles are applied such as step and sinusoidal disturbances, to check the performance of the controllers and make a comparison between them. Figure 18, shows a block diagram for the three control systems.

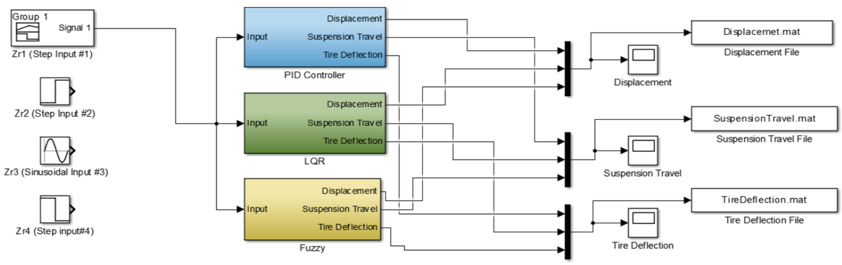

Fig. 18. Simulink model for active suspension system using PID, LQR and FLC.

a. Step Disturbance

Assuming the road disturbances are a step or Sinusoidal disturbances, The effect of car body displacement and suspension travel respectively in quarter car model after applying the FLC, LQR and PID controller with different inputs will be represented. After applying step input as 0.1 for 2 s and −0.2 after 2 s. The system response for LQR, FLC and PID will be shown in Figs. 19, 20 and 21.

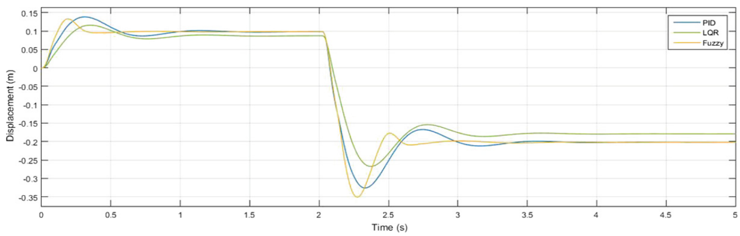

Fig. 19. Body displacement for a QCM controlled by LQR, FLC and PID.

Table 5 and Fig. 20, show the overshoot, settling time and rise time for two systems compared.

Table 5. Overshoot, settling time and rise time.

	PID	LQR	FLC
Overshoot	0.0385	0.015	0.033
Settling time	0.16	0.24	0.12
Rise time	4.4	4.5	3.8

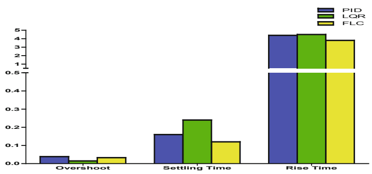

Fig. 20. PID vs LQR and FLC chart step input.

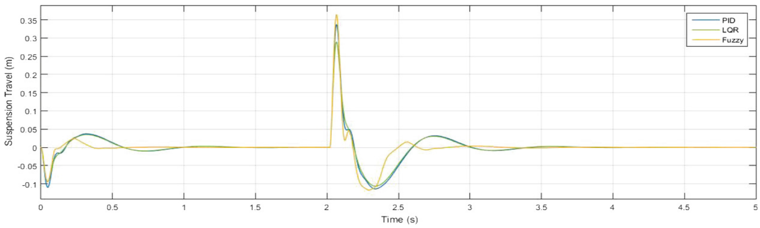

Fig. 21. Suspension travel for a QCM controlled by LQR, FLC and PID.

Figures 22, 23 and 24, show system response for LQR, FLC and PID after applying step input step time 0.1 s and amplitude 0.5 m step.

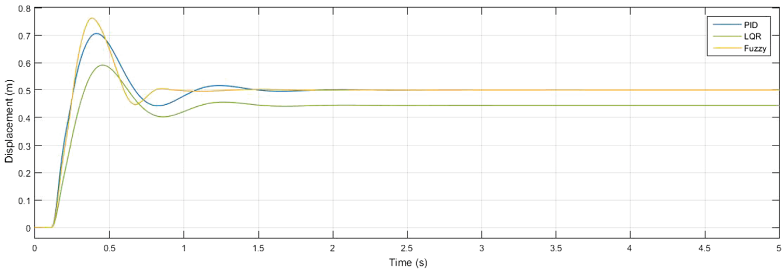

Fig. 22. Body displacement for a QCM controlled by LQR, FLC and PID.

Table 6 and Fig. 23, show the overshoot, settling time and rise time for two systems compared.

Table 6. Overshoot, settling time and rise time.

	PID	LQR	FLC
Overshoot	0.206	0.09	0.262
Settling time	1.8	2.4	1.6
Rise time	0.256	0.33	0.254

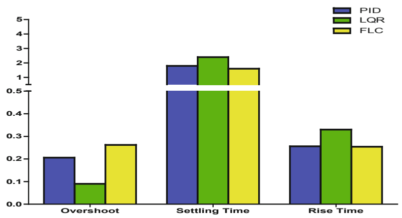

Fig. 23. PID vs LQR and FLC chart with step input 0.5.

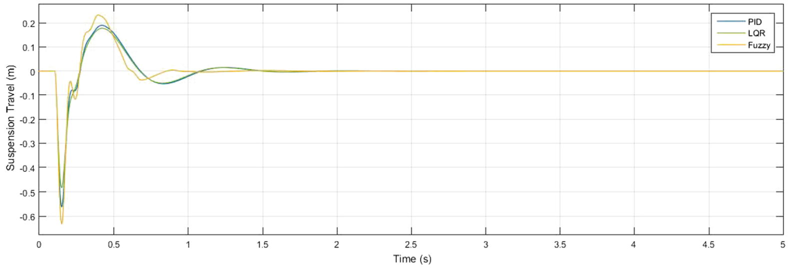

Fig. 24. Suspension travel for a QCM controlled by LQR, FLC and PID.

Figures 25, 26 and 27, show system response for LQR, FLC and PID after applying step input with a negative amplitude 0.1 m step.

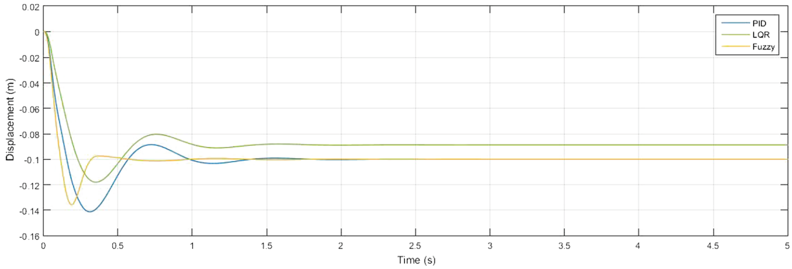

Fig. 25. Body displacement for a QCM controlled by LQR, FLC and PID.

Table 7 and Fig. 26, show the overshoot, settling time and rise time for two systems compared.

Table 7. Overshoot, settling time and rise time.

	PID	LQR	FLC
Overshoot	0.041	0.018	0.036
Settling time	1.6	1.5	1
Rise time	0.16	0.23	0.12

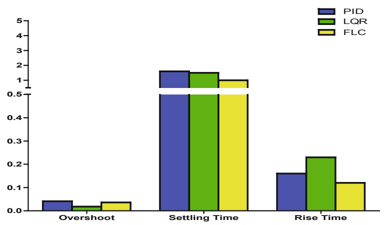

Fig. 26. PID vs LQR and FLC chart with step input 0.5.

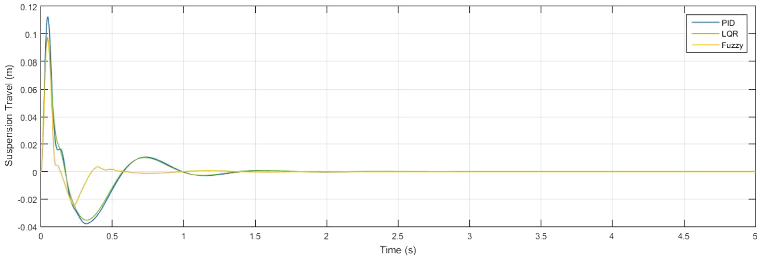

Fig. 27. Suspension travel for a QCM controlled by LQR, FLC and PID.

b. **Sinusoidal Disturbance**

Figures 28 and 29, show system response for LQR, FLC and PID after applying sinusoidal wave frequency of 7.7 rad/s and amplitude 0.1 m.

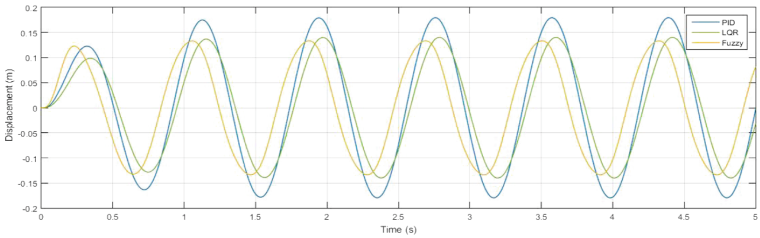

Fig. 28. Body displacement for a QCM controlled by LQR, FLC and PID.

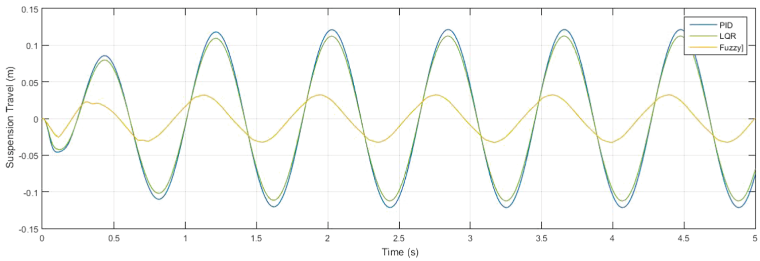

Fig. 29. Suspension travel for a QCM controlled by LQR, FLC and PID.

4 Conclusion

This paper has provided three design methods of controlling an active suspension system and a brief comparison between passive suspension system and PID Controller. The main three controllers applied in this paper are FLC, LQR and PID Controller. All methods were modelled and simulated using a Matlab/Simulink. After comparing a passive suspension system with an active suspension system controlled by PID controller, simulation results showed that the PID controller has shown better values in overshooting, settling time and rise time. Meanwhile, after comparing the FLC, LQR and PID together, LQR recorded the best overshoot values, while FLC recorded greatest results in settling time and rise time. So automobiles that will be designed in the future should apply active suspension controlled by FLC to achieve the best values for passenger ride comfort and road handling.

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