International Journal of Recent Technology and Engineering (IJRTE)
ISSN: 2277-3878, Volume-8 Issue-5, January 2020

Intelligent Based Vibration Control towards the Improvement of Vehicle Ride Comfort

Sharifah Munawwarah, Fitri Ykub, Pauziah Muhammad, Aminudin Hj Abu, Zainudin A. Rasid

Abstract**: *The mechanism of vehicle suspension system is splitting the car body from the tire of the car. The suspension system is an important element in vehicle design because it effects the ride comfort and passenger's safety. These two criteria are conflicting to each other. Therefore, this paper proposes an intelligent chassis control of quarter car active suspension system to enhance the vehicle ride comfort and road holding performance. The fuzzy logic control is designed to maximize the driving comfort by keeping the wheel always in contact with road surfaces. Several scenario has been created for analysis purposes. The performance of the fuzzy logic is compared with the PID control and LQR control method respectively. The novelty of this paper is, the ride comfort is measured up to higher derivative of motion. No significant literature was found on the higher derivative of motion such as jerking to improve the ride comfort. Based on the simulation results, Fuzzy logic outperforms the other two controllers on car body displacement, car body acceleration, wheel deflection and jerking with with higher percentage of improvement compare to PID control and passive suspension system.***

Keywords: ride comfort, active suspension system, quarter car model, jerking, PID control, LQR control, FLC control.

I. INTRODUCTION

The topics of vehicle suspension has been reviewed and yet remained attractive among researchers and academia [1]. The framework of the suspension system is to provide comfort not only for passenger but also for car after given road impact of particular road conditions [2]. The vehicle under vibration usually gets affected through overshoot, settling time, and steady state error for comfort passenger criteria. There are three goups mainly in vehicle suspension system namely passive, semi active and active suspension. Passive suspension system is also known as a traditional suspension system that consists of shock absorber and spring attached at each tire's parallel in the vehicle. The role of spring in a vehicle is to hold the vehicle body and used in absorbing and storing the energy. The shock absorber or damper is an element of the automobile suspension that used to get rid of the trembling energy that is kept inside the spring. Besides, the input from the road that transferred to the vehicle is being controlled. Numerous analysts have taken research at the conventional suspension framework including its structure, execution, and security [3-4] It was found from [5-6], passive control contains constant characteristic of springs and damper where it does not involve any feedback for mechanism. Semi-active suspension in other functionality can store the energy in the spring element meanwhile, controllable damper is used to dissipate the energy. Some of the semi-active suspension systems use the passive damper and the controllable spring, the controllable damper usually acts with limited capability to produce a controlled force when dissipating energy [7]. Unlike passive and semi-active suspension, active suspension system able to add energy into the vehicle dynamic system by the use of actuators and can be controlled by using the controller that will calculate either to add or dissipate energy from the system. This paper will use the active suspension system as the main system due to its advantages [8].

The vehicle suspension system can be modeled separately with a quarter car [9-10], half car [11-12] and full car [13]. However, the quarter car model is chosen in this paper due to its simplicity in designing and only include vertical movement analysis compare to half car and full car model [14]. Even though active suspension able to add and dissipate energy from the system, the conflicts in design requirements still exist. Choosing the suitable control is one of the confronts in suspension system control approach. It is crucial to implement the control device in order to prevent deterioration of passenger ride comfort and avoiding damage to the car which could improve vehicle drivability and handling performances. There are various control approaches that have been conducted in the previous study to enhance the vehicle ride comfort in half car active suspension system. These include Proportional Integral Derivative, Sliding Mode Control, Neural Network, Linear Quadratic Regulator, Linear Quadratic Gaussian, H∞, Fuzzy Logic, Model Predictive Control etc.

Manuscript published on January 30, 2020.
* Correspondence Author

Sharifah Munawwarah*, Malaysia-Japan International Institute of Technology, University Technology Malaysia Kuala Lumpur, Malaysia. Email: shmunawwarah13@gmail.com

Fitri Yakub*, Malaysia-Japan International Institute of Technology, University Technology Malaysia Kuala Lumpur, Malaysia.
Email: mfitri.kl@utm.my

Pauziah Muhammad, Malaysia-Japan International Institute of Technology, University Technology Malaysia Kuala Lumpur, Malaysia.
Email: pauziah.kl@utm.my

Aminudin Hj Abu, Malaysia-Japan International Institute of Technology, University Technology Malaysia Kuala Lumpur, Malaysia.
Email: aminudin.kl@utm.my

Zainudin bin A.Rasid, Malaysia-Japan International Institute of Technology, University Technology Malaysia Kuala Lumpur, Malaysia.
Email: arzainudin.kl@utm.my

Research conducted in [15] combined the active force control and Iterative learning together with PID based approach in a new hybrid control method for a vehicle suspension system. Besides, active force control iterative learning scheme was designed, simulated and utilized for a quarter car active suspension control. In the recent study by the author in [16] proposed a genetic algorithm to determine the optimal PID gains to acquire best ride comfort levels. The authors compared the performance of the controllers between classical PID and Passive system. It is found that optimized PID GA gives out the best ride comfort enhancement. Research on ride comfort enhancement through PID controller was conducted by Daniyan et., al [17]. The controller is proposed to increase the performance of ride comfort and stability of the car. As a result, the active suspension system with a PID controller gives the significant improvement by reducing the peak overshoot and faster settling time compare to passive suspension system. Other papers that proposed the application of PID in vibration control of quarter car active suspension system are presented in [18-19]. The application of LQR has been critically investigated in Electromagnetic Suspension (EMS) System in stabilizing the vehicle response [20]. The results revealed that controller strategy provides a satisfactory improvement for the EMS system compared to passive suspension performance. The effects of sprung mass variations on active suspension under LQR control is studied in [21]. The propose method based on the concept of Grobner bases and sum of roots allows the controller gain to be tuned whilst the sprung mass is retained as a symbolic parameter. Instead of using a trial and error method to get the optimum weighting matrix parameters of LQR, Nagarkhar and Vikhe [22] implemented the GA with the objective of minimizing the root mean square (RMS) controller force. Based on the observation of the output graph, the GA method was proven beneficial to improve the LQR controller performance in a Macpherson strut quarter car suspension system.

The MPC technique is applied by [23] where the experiment is being set up by using Quanser active suspension By changing the road characteristics and the plant, and by applying disturbance to the system, system responses of MPC is examined critically. Based on the output graph, it witnessed that MPC method are successful against disturbances in both simulation and real-time experiments in terms of minimizing the oscillations. The case study performs by [24] proved that their FL controller gives the significant impact on reducing the amplitude of vehicle body sprung mass of quarter vehicle model when compared with PID and passive suspension system. Chuanyin et.al [25] has conducted a comparison between Neural Network (NN), Genetic Algorithm (GA), and Linear Quadratic Gaussian (LQG) on active suspension system on vehicles. Results from the studies show that the LQG has been proven to be effective for the suspension system which results in excellent ride comfort and good holding ability.

Most transportation devices are designed to reduce acceleration as much as is reasonably practicable. A derivative of acceleration that is also known as a jerk, is a biomechanical effect on the human body when there is a change of motion. In physics and engineering jerk and snap should always be considered when vibration occurs and particularly when this excitation induces multi-resonant modes of vibration. Jerk is rarely stated in mechanics teaching, but it is a common everyday experience [26]. For example, when a car accelerates or decelerates, the passenger will experience the changes in motion that are likely the jerk. This motion of vibration also became a threat to a driver and passenger to get a comfortable ride. In addition, the harsh bumps may cause the suspension to hit the physical stops known as “bump stopper”. As a result, the undesired accelerations and significant jerk on the vehicle chassis due to the impact on the system will degrade the ride quality of the vehicle [27]. Therefore, minimizing jerk is an important consideration from the smooth driving of a car to the design of the vehicle suspension system.

Most of the findings among the researchers only considered up until car body acceleration. What they do not discover yet and emphasize, is the higher derivatives of motion like jerking, snap and crackle. The vehicle body vibration not only experiences the acceleration but there is the jerking involve. In order to obtain satisfactory riding comfort, the vibration in jerking also should be minimized. Therefore, the novelty of this paper are to: i) to boost the compromise ride comfort and drivability of passenger car in quarter active suspension system respectively with minimal wheel and suspension deflection via Fuzzy Logic controller ii) reject the disturbance where the controller of active suspension should maintain the steady state close to zero in which there are multiple road inputs iii) investigate the higher derivatative of motion which is jerking to obtain the minimal oscillation in order to get satisfactory comfort v) attain physical value of maximum suspension deflection that cannot exceed by the given maximum value of road disturbance.

II. MATHEMATICAL MODEL OF QUARTER CAR ACTIVE SUSPENSION SYSTEM

The two-degree-of-freedom (2DOF) system that represent the quarter car active suspension system is adopted from [18-19]is illustrated in Fig. 1. The model consist of sprung mass that represented by m_b, while m_w denotes as unsprung mass. The road disturbance signal is denoted by *w*. While x_b represents car displacement of the car body, x_w represents the displacement of the wheel. The spring constant denotes as k_b and k_w respectively whereas c_b represents dampers and control force as *u* respectively.

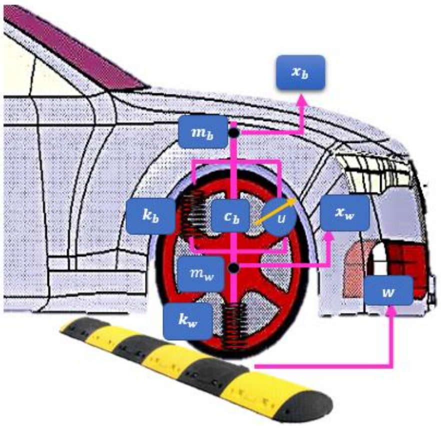


Fig. 1 Schematic diagram of quarter car model

The equations for both masses can be form as shown in the equation (1) - (2)

$$m_b\ddot{x}_b = -k_b(x_b - x_w) - c_b(\dot{x}_b - \dot{x}_w) + u \qquad (1)$$

$$m_w\ddot{x}_w = k_b(x_b - x_w) + c_b(\dot{x}_b - \dot{x}_w) + k_w(x_w - w) - u \qquad (2)$$

Equation (1) and (2) can be arranged in the state space form given as

$$\dot{X}(t) = Ax(t) + Bu(t) + w(t) \qquad (3)$$

Where, $x = [x_1 \quad x_2 \quad x_3 \quad x_4]^T$

$= [x_b \quad x_w \quad \dot{x}_b \quad \dot{x}_w]^T$

Rewrite (3) in the matrix form yields:

$$\begin{bmatrix} \dot{X}_1 \\ \dot{X}_2 \\ \dot{X}_3 \\ \dot{X}_4 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ \frac{-x_b}{m_b} & \frac{k_b}{m_b} & \frac{-c_b}{m_b} & \frac{c_b}{m_b} \\ \frac{k_b}{m_w} & \frac{-(k_b+k_t)}{m_w} & \frac{c_b}{m_w} & \frac{-c_b}{m_w} \end{bmatrix} \begin{bmatrix} x_b \\ x_w \\ \dot{x}_b \\ \dot{x}_w \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{1}{m_b} \\ \frac{-1}{m_w} \end{bmatrix} u + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{k_t}{m_w} \end{bmatrix} r \qquad (4)$$

Table- I: Passenger vehicle parameter [19]

Symbol	Description	Value
m_b [kg]	Sprung mass	290
m_w [kg]	Unsprung mass	59
k_b [n/m]	Spring constant of suspension system	16812
k_w [n/m]	Spring constant of wheel and tire	190000
c_b [n/m/s]	Damping constant of suspension system	1000

Table 1 shows the parameters car which is used to simulate the response in PID, LQR and Fuzzy controller design through Matlab and Simulink software. Before begin the controller implementation, the stability loop analysis for an open loop response is a must to make sure that the system is stable where all poles are in the left-half plane. Fig. 2 shows the quarter car pole zero map of the system that all poles are located in the left-half plane.

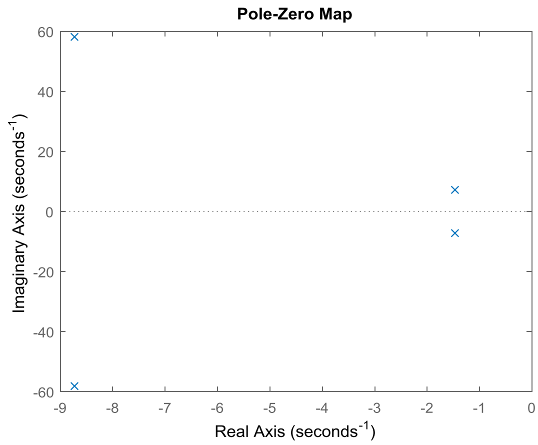


Fig. 2 Pole-zero plot of quarter car for an open loop response

The eigenvalues where the poles are located in the passive suspension system are given by;
eig (A) = -8.7293 +58.1529i, -8.7293 -58.1529i, -1.4694 + 7.1993i, -1.4694 - 7.1993

III. CONTROLLER DESIGN

This section will elaborate the control design for this paper. Firstly, the PID control is design as a benchmark control given by sub section A. Next it will compare the famous optimal control which is LQR that can deal with linear system as deliberate in sub section B. Lastly, the FLC controller is design as in sub section C to improve the ride comfort and road holding. Besides, the performances of three controllers are compared with the uncontrolled suspension system.

A. PID Controller

The ideal version of the PID controller is given by:

$$u_{PID}(t) = K_p e(t) + K_i \int_0^t e(t)\,dt + K_d \frac{de(t)}{dt} \qquad (5)$$

Where K_p =proportional coefficient K_i =integral coefficients, K_d= derivative coefficients and e= error signal

In this paper, several tunings have been measured based on settling time and percentage of overshoot. The PID tuner is used until the best response is obtained as shown in the Fig. 3.

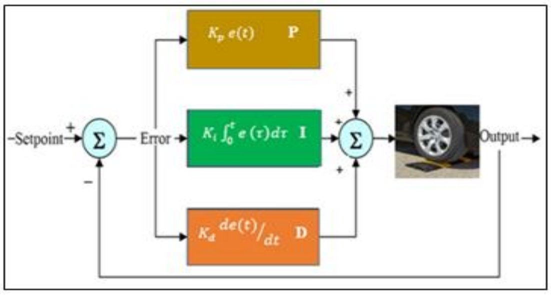


Fig. 3 PID control

Table- II: PID parameters for quarter

Gain value		
K_p	K_i	K_d
915	5	310

B. LQR Controller

The linear quadratic regulator (LQR) is another type of controller which regulate the system and drive the initial state vector to zero. The state variable regulator for the system is consider as follows [22],[29]-:

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

K is the constant state feedback gain matrix (m x n) that gives the closed loop state equation with desired performance characteristics. Basically, the main function of this controller is to be able to minimize the cost function, J by balancing the weight of matric Q and R.

$$J = \frac{1}{2}\int_0^t x^T Qx + u^T Ru \tag{7}$$

u is the input which is the multiplication of gain K and the state where K is the state feedback gain matrix.
The matrix K is represented by:

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

The P matrix must fulfil the reduced-matrix Ricatti equation given as:

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

Then, the feedback regulator u is

$$u(t) = -(R^{-1}B^T P)x(t) \tag{10}$$

$$u(t) = -Kx(t) \tag{11}$$

The value of Q and R matrices for both car model are given as follows

$$Q = \begin{bmatrix} 1000 & 0 & 0 & 0 \\ 0 & 1000 & 0 & 0 \\ 0 & 0 & 1000 & 0 \\ 0 & 0 & 0 & 1000 \end{bmatrix}, R = 0.01 \tag{12}$$

The value of feedback gained matrix K for quarter car are given by;

$$K = [2.9738 \quad -347.8456 \quad 58.1740 \quad -47.0729]$$

C. Fuzzy Logic Controller

The Fuzzy Logic Controller is one of the intelligent controllers that was a pioneer by Zadeh in 1965. Numerous research papers related to FLC in suppressing the vibration and enhancing the ride comfort have been studied for the past few decades. The fuzzy logic controller has an advantage over classical controller when it is applied to complex systems. It can be developed with minimal knowledge about the system dynamics. To designing a fuzzy logic controller consists of the following four steps [28]:

1. Fuzzification: In this step the crisp inputs are transformed to fuzzy values.
2. Rule design: In this step the fuzzy output truth values are calculated.
3. Computation: In this phase the required control actions are computed.
4. Defuzzification: In this step the fuzzy output is converted back to the crisp values.

In this paper, there are two inputs of membership functions of error, e and change in error $\dot{e}$, and control output, u. The rules are utilized to generate the system configuration given as in Table III where where abbreviations used correspond to P=Positive, Z=Zero, N=Negative. The rules are expressed by if-then rules, which extract from fundamental knowledge and experience of the system

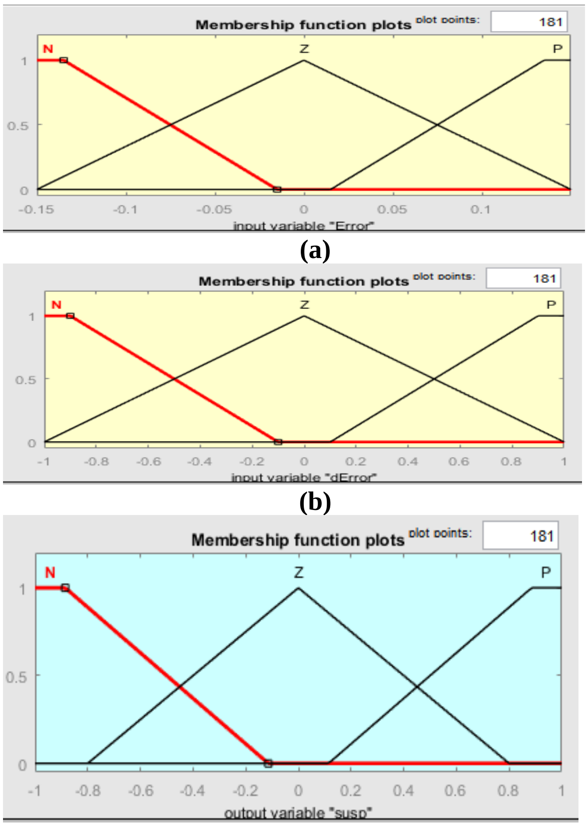
The triangular and trapezoidal membership function for e and $\dot{e}$ are shown in Fig. 4. The range for e is set to [-0.15 0.15] and range for $\dot{e}$ is [-1 1]. A method of defuzzification known as Centre of Gravity Method is used as process that requires a non-fuzzy value of control.

$$f_a = \int_{F_a} \frac{f * \mu_D(f).df}{F_a \mu_D(f).df} \tag{13}$$

Where $\mu_D(f)$ is a corresponding membership function. The actuator force (f_a) is chosen to give ± 691 N for quarter car to generate a signal to the suspension system at the same time suppress the vehicle's vibration.

Table- III: FLC rules

$\dot{e}$ \ e	**N**	**Z**	**P**
N	P	P	Z
Z	P	Z	N
P	Z	N	N



(a)

(b)

(c)

Fig. 4 Membership function for FLC

The closed loop diagram of the quarter car active suspension with FLC is illustrate in Fig. 5.

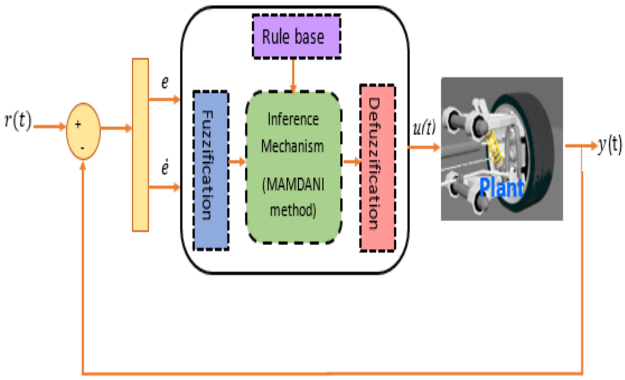


Fig. 5 FLC control

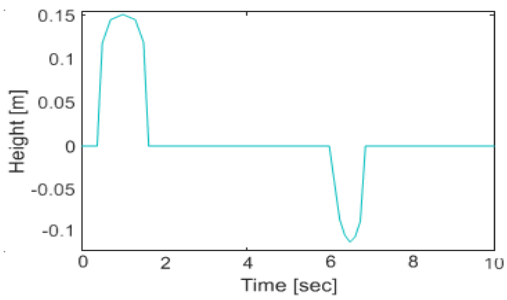
The *r(t)* input signal which is multiple road profiles, two input to the fuzzy logic control is error, e and derivative error, $\dot{e}$. The control signal value *u(t)* will produce throughout the controller in which the actuator produce force signal to the suspension system and the system will generate the desired output and suppress the vibration in quarter car active suspension system.

IV. RESULTS AND DISCUSSION

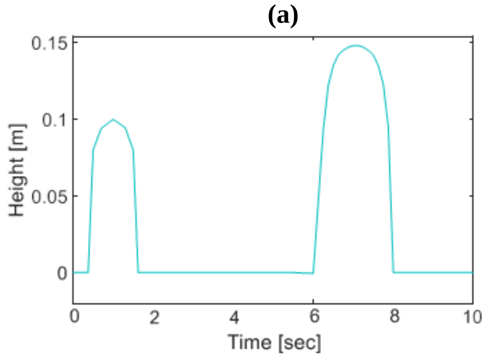
In this paper, there are two scenarios under different road profile that has been done throughout the simulation. The term of ride comfort is closely related with the reduction of car body acceleration. Meanwhile the vehicle stability indicates that the suspension deflection should be low. The goals of this paper are to minimize the car body displacement, car body acceleration, suspension deflection and jerking. Besides, it is expected that the Fuzzy logic control provide higher percentage of improvement, and settling time response should be fast for the four parameters that have been monitored. The improvement percentage is calculated as:

$$improvement(\%) = \frac{PeakValue_{uncontrolled} - PeakValue_{controlled}}{PeakValue_{uncontrolled}} \, x\,100 \quad (14)$$

Fig. 6(a) and (b) shows the road profiles used for the analysis purposes.



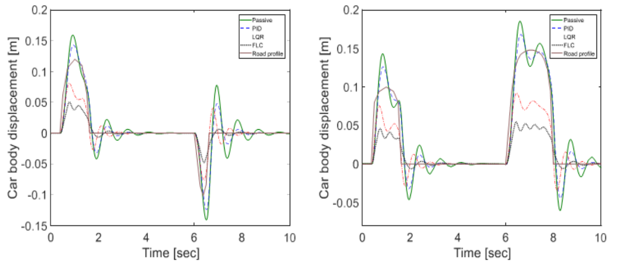
(a)



(b)

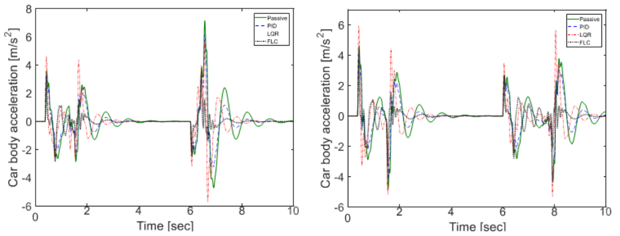
Figure 6 (a) Bump and pothole (b) Dual bumps

The quarter car model is assuming to be at the constant speed along the road within 10 seconds. The performances of PID, LQR and Fuzzy Logic Control are through Matlab and Simulink software.



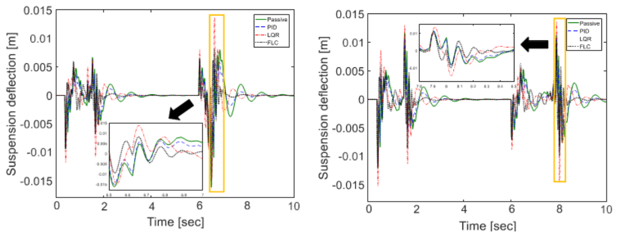
(a) (b)

Fig. 7(a) Car body displacement for bump and pothole scenario (b)Car body displacement for dual bumps scenario



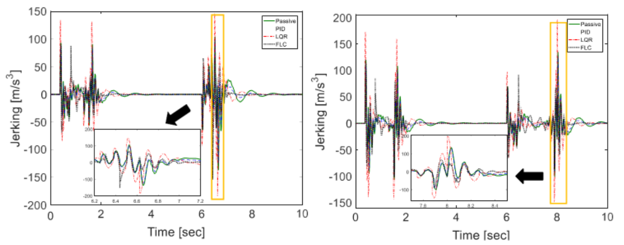
(a) (b)

Figure 8 (a) Car body acceleration for bump and pothole scenario (b) Car body acceleration for dual bumps scenario



(a) (b)

Fig. 9 (a) Wheel deflection for bump and pothole scenario (b) wheel deflection for dual bumps scenario



(a) (b)

Fig. 10(a) Jerking for bump and pothole scenario (b) Jerking for dual bumps scenario

The time domain analysis for the car body displacement is shown in Fig. 7 for both cases. For bump and pothole case that shown in Fig. 7(a), the active control system using FLC controller able to reduce the maximum peak by compared with passive, PID and LQR.

Note that the classical PID provide the reduction percentage of 38.45% followed by LQR, 47.44% and fuzzy logic, 87.18% respectively. Meanwhile, for dual bump case as shown in Fig. 7(b) illustrates that the amplitude of vibration is also effectively minimized by using the FLC approach compared to PID and LQR control. The settling time response also shows a minimal time to reach the stability state. The improvement percentage that is given by PID is 50.00% followed by LQR, 53.13% and FLC, 84.38% respectively. The overall performance of car body displacement for both cases are analyzed in detail as shown in Table IV.

The significant improvement of FLC in term of car body acceleration is found in Fig. 8(a) for bump and pothole case. The FLC control manage to suppress the vibration greatly and minimal settling time response compared to PID and LQR control. The improvement percentage given by PID, LQR and FLC control are13.54%, 20.69 and 71.55% respectively. However, for dual bump cases as illustrates in Fig. 8(b), LQR is still oscillating even after the present of second bump where it gives high overshoot compared to PID and FLC respectively. It is noticeable that the FLC technique gives less overshoot and greatly improve the ride comfort. Based on the respected Fig. 8, this phenomenon demonstrates that passengers or drivers still facing uncomfortable ride due to occurrence of the vibrations in the vehicle suspension system. The overall performance of car body acceleration for both cases are analyzed in detail as shown in Table V.

The output response for both cases in term of wheel deflection are shown in Fig. 9(a) and 9(b). The graph of wheel deflection for bump and pot hole case reveals that the PID control gives the best response compare to LQR control in term of maximum improvement percentage. It is noticeable that the maximum peak overshoot in LQR control slightly close to the uncontrolled suspension system that lead to slow response and low improvement percentage by 14.29% compared to PID control, 28.57%. In contrast, Fuzzy logic control provides the lower amplitude of vibration and settling time that leads to greater ride comfort performance of 57.14%. As illustrates in Fig. 9(b), the PID control gives the lower peak amplitude that give higher percentage of improvement by 27.27%. The maximum peak of FLC is slightly close to the peak of passive suspension system. However, it can be seen that the FLC control able to reduce the amplitude of vibration effectively with minimal time taken to reach the stability state. In contrast, the LQR gives the longest time taken to reach the stability state even though it perform well in amplitude reduction compared to FLC. The FLC give the lowest percentage of improvement compared to LQR control by 10.00% and 9.09% respectively. The overall performance of wheel deflection for both cases are analyzed in detail as shown in Table VI.

The output response in Fig. 10(a) and 10(b) demonstrate the performance of jerking in both cases. In the respected Fig. 10(a) the result reveal that PID control manage reduce the amplitude of vibration compared to LQR control. PID gives 4.79% of improvement meanwhile 1.57% in LQR control. It can be seen that the maximum peak of LQR control is close to without controller. It is clearly noticed that FLC control has lower amplitude and smaller settling time response when compared with PID and LQR control in which FLC control gives the higher percentage of improvement of 13.11%. This response implies that the proposed method achieves significantly better performance on ride comfort over the benchmark and optimal control respectively. The vibration reduction of jerking in Figure 8(b) is guaranteed by the design of FLC control. The FLC control manage to suppress the vibration greatly and minimal settling time response compared to PID and LQR control. The improvement percentage given by PID, LQR and FLC control are 4.44%, 46.99% and 55.23% respectively. The overall performance of jerking for both cases are analyzed in detail as shown in Table VII.

Table- IV: Car body displacement performances

Scenario	**Bump and pot hole**			
Control	Passive	PID	LQR	FLC
Maximum peak value (m)	0.078	0.048	0.041	0.010
Settling time (sec)	>2	1.13	0.91	0.85

Scenario	**Dual bump**			
Control	Passive	PID	LQR	FLC
Maximum peak value (m)	0.032	0.016	0.015	0.005
Settling time (sec)	>2	0.94	0.92	0.82

Table- V: Car body acceleration performances

Scenario	**Bump and pot hole**			
Control	Passive	PID	LQR	FLC
Maximum peak value (m/s^2)	7.142	6.175	5.664	2.032
Settling time (sec)	>2	1.71	1.46	1.02

Scenario	**Dual bump**			
Control	Passive	PID	LQR	FLC
Maximum peak value (m/s^2)	3.768	3.293	3.484	1.051
Settling time (sec)	>2	1.53	1.57	1.19

Table- VI: Wheel deflection performances

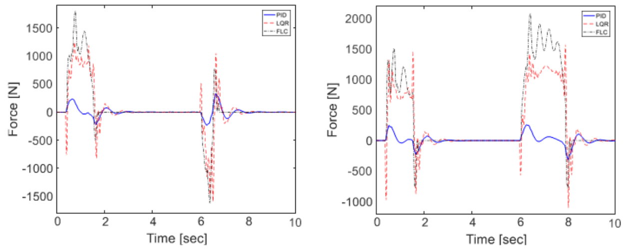
Scenario	**Bump and pot hole**			
Control	Passive	PID	LQR	FLC
Maximum peak value (m)	0.007	0.006	0.009	0.003
Settling time (sec)	>2	1.25	1.52	1.17

Scenario	**Dual bump**			
Control	Passive	PID	LQR	FLC
Maximum peak value (m)	0.011	0.014	0.012	0.010
Settling time (sec)	>2	1.43	1.81	1.09

Table- VII: Jerking performances

Scenario	Bump and pot hole			
Control	Passive	PID	LQR	FLC
Maximum peak value (m/s^3)	93.410	88.940	94.880	81.160
Settling time (sec)	>2	1.27	1.68	1.12

Scenario	Dual bump			
Control	Passive	PID	LQR	FLC
Maximum peak value (m/s^3)	133.00	127.100	195.500	59.550
Settling time (sec)	>2	1.07	1.37	1.03



(a) (b)

Fig. 11(a) Control force for bump and pothole scenario (b) Control force for dual bumps scenario

The maximum control forces acting on the three controllers for quarter car active suspension system are shown in Fig. 11(a) and 11(b). Higher control force is required for FLC control compared to LQR and PID control. Simultaneously, the FLC controller achieves best performance on the ride comfort compared to the other control methods. The overall performance of maximum force required for the three controllers in both cases are analyzed in detail as shown in Table VIII.

Table- VIII: Maximum control force

Scenario	Bump and pot hole		
Control	PID	LQR	FLC
Maximum Control force (N)	228.9	1580	1926
Settling time (sec)	1.61	1.46	0.36

Scenario	Dual bump		
Control	PID	LQR	FLC
Maximum Control force (N)	309.5	1548	2183
Settling time (sec)	1.07	1.27	0.92

To summarized, Table IX and X show that the improvement percentage via FLC control in both scenarios able to enhance the ride comfort and vehicle stability compared to PID and LQR respectively.

Table-IX: Improvement percentage based on bump and pot hole scenario

Parameters	Controller		
	PID	LQR	FLC
Car body displacement (m)	38.46 %	47.44 %	87.18 %
Car body acceleration (m/s^2)	13.54 %	20.69 %	71.55 %
Wheel deflection (m)	14.29 %	14.28 %	57.14 %
Jerking (m/s^3)	4.79 %	1.57 %	13.11 %

Table-X: Improvement percentage based on dual bumps scenario

Parameters	Controller		
	PID	LQR	FLC
Car body displacement (m)	50.00 %	53.13 %	84.38 %
Car body acceleration (m/s^2)	12.61 %	7.54 %	72.11 %
Wheel deflection (m)	27.27 %	9.09 %	10.00 %
Jerking (m/s^3)	4.44 %	46.99 %	55.23 %

Based on the comparison between PID, LQR and FLC response, it is proven that FLC control beneficial to the quarter car active suspension system in enhaancing the road holding performance and passenger comfort.

V. CONCLUSION

Three active controllers, namely PID, LQR, and FLC are designed to improve the vehicle ride comfort. The FLC provides a higher percentage of improvement compared to PID and LQR respectively. It can be concluded from the simulation results that FLC control able to reduce the vibration effectively at all parameters measured in quarter car active suspension system which are car body displacement, car body acceleration and jerking. The road holding criteria under dual bumps scenario shows that vibration not effectively reduced the vibration, but still the use of FLC is proven beneficial under bump and pot hole scenario. Through the two scenarios conducted for analysis purposes, the controller performances indicated that the ride comfort and road holding is perform well under multiple road disturbances. For future work it is recommended to validate the controller performances in real car experiment. Besides, controller is suggested to implement in more complex system in order to verify its robustness.

ACKNOWLEDGEMENT

This research was financially supported by UTM research grant under Vot number 15J97 from University Technology Malaysia

REFERENCES

1. A. Aly and F. Salem, "Vehicle suspension systems control: a review," Int. J. Control. Autom. Syst., vol. 2, no. 2, pp. 46–54, 2013.
2. G. A. Hassaan, "Car Dynamics using Quarter Model and Passive Suspension , Part I : Effect of Suspension Damping and Car Speed," Int. J. Comput. Tech., vol. 1, no. 2, pp. 1–9, 2014.
   M. C. Smith and G. W. Walker, "Performance Limitations and Constraints for Active and Passive Suspensions: a Mechanical Multi-port Approach," Veh. Syst. Dyn., vol. 33, no. 3, pp. 137–168, Mar. 2000.
3. C. Li, M. Liang, Y. Wang, and Y. Dong, "Vibration suppression using two-terminal flywheel. Part II: application to vehicle passive suspension," J. Vib. Control, vol. 18, no. 9, pp. 1353–1365, Aug. 2012.
4. M. Appleyard and P. E. Wellstead, "Active suspensions: some background," IEE Proc. - Control Theory Appl., vol. 142, no. 2, pp. 123–128, Mar. 1995.

5. W. Sun, H. Gao, and P. Shi, “Background, modelling and problem statements of active suspensions,” Studies in Systems, Decision and Control, vol. 204. pp. 1–13, 2020.
6. S. D. Nguyen, S. B. Choi, and Q. H. Nguyen, “An optimal design of interval type-2 fuzzy logic system with various experiments including magnetorheological fluid damper,” Proc. Inst. Mech. Eng. Part C J. Mech. Eng. Sci., vol. 228, no. 17, pp. 3090–3106, Dec. 2014.
7. W. Sun, H. Pan, J. Yu, and H. Gao, “Reliability control for uncertain half-car active suspension systems with possible actuator faults,” IET Control Theory Appl., vol. 8, no. 9, pp. 746–754, 2014.
8. B. Ion, C. Francesco, and A. Veronica, “Improving Driver Comfort with Linear Quadratic Controller for Active Suspension System,” 2015 Trans Tech Publ. Switz., vol. 811, pp. 199–203, 2015.
9. M. K. Shehan, B. B. Sahari, N. A. B. A. Jalil, T. S. Hong, and A. B. As’arry, “Dynamic analysis of an optimal active suspension system using global search optimization,” International Journal of Engineering and Technology(UAE), vol. 7, no. 3. pp. 43–46, 2018.
10. A. Mozaffari, A. Doosthoseini, and N. L. Azad, “Predictive Control of Suspension Systems Through Combining Dynamic Matrix and Constrained Variable Structure Controllers,” Journal of Dynamic Systems, Measurement and Control, Transactions of the ASME, vol. 138, no. 12. 2016.
11. F. León-Vargas, F. Garelli, and M. Zapateiro, “Limiting vertical acceleration for ride comfort in active suspension systems,” Proc. Inst. Mech. Eng. Part I J. Syst. Control Eng., vol. 232, no. 3, pp. 223–232, 2018.
12. S. H. Zareh, A. Sarrafan, A. A. A. Khayyat, and A. Zabihollah, “Intelligent semi-active vibration control of eleven degrees of freedom suspension system using magnetorheological dampers,” J. Mech. Sci. Technol., vol. 26, no. 2, pp. 323–334, 2012.
13. M. Shabani, S. Azadi, and R. Kazemi, “The advantages of using more complete model based controller for semi-active suspension system,” in International Conference on Mechanical Engineering and Advanced Technology, ICMEAT 2012, 2012, pp. 1–6.
14. R. Rosli, M. Mailah, and G. Priyandoko, “Active suspension system for passenger vehicle using active force control with iterative learning algorithm,” WSEAS Trans. Syst. Control, vol. 9, no. 1, pp. 120–129, 2014.
15. H. Metered, W. Abbas, and A. S. Emam, “Optimized Proportional Integral Derivative Controller of Vehicle Active Suspension System Using Genetic Algorithm,” vol. No.2018-01. p. SAE Technical Paper, 2018.
16. I. A. Daniyan, K. Mpofu, and D. F. Osadare, “Design and simulation of a controller for an active suspension system of a rail car,” vol. 5, no. 1, pp. 1–15, 2018.
17. C. Kuber, “Modelling Simulation and Control of an Active Suspension System,” Int. J. Mech. Eng. Technol., vol. 5, no. 11, pp. 66–75, 2014.
18. A. A. Abdeen, K. Ibrahim, and A. M. Nasr, “Active Suspension System Design Using Fuzzy Logic Control and Linear Quadratic Regulator,” in Proceedings of the International Conference on Advanced Intelligent Systems and Informatics 2018, 2019, vol. 1, pp. 152–166.
19. N. H. Amer, R. Ramli, W. N. L. W. Mahadi, M. A. Z. Abidin, H. M. Isa, and S. M. Foong, “Implementation of LQR controller on electromagnetic suspension system for passenger’s Car,” Int. Rev. Mech. Eng., vol. 8, no. 1, pp. 258–264, 2014.
20. A. Shavalipou, S. Mohamed Ha, and Z. Mohd Nopia, “Symbolic Parametric LQR Controller Design for an Active Vehicle Suspension System,” J. Appl. Sci., vol. 15, no. 8, pp. 1127–1132, Aug. 2015.
21. M. P. Nagarkar and G. J. Vikhe, “Optimization of the linear quadratic regulator (LQR) control quarter car suspension system using genetic algorithm,” Ing. E Investig., vol. 36, no. 1, pp. 23–30, 2016.
22. B. E. Durmaz, B. Kacmaz, I. Mutlu, and M. T. Soylemez, “Implementation and comparison of LQR-MPC on active suspension system,” in Electrical and Electronics Engineering (ELECO), 2017 10th International Conference on, 2017, pp. 828–835.
23. F. Yakub et al., “Enhancing vehicle ride comfort through intelligent based control,” in 2016 IEEE International Conference on Automatic Control and Intelligent Systems (I2CACIS), 2017, pp. 66–71.
24. T. Chuanyin and Z. Tiaoxia, “The research on control algorithms of vehicle active suspension system,” in IEEE International Conference on Vehicular Electronics and Safety, 2005., 2005, pp. 320–325.
25. D. Eager, A. M. Pendrill, and N. Reistad, “Beyond velocity and acceleration: Jerk, snap and higher derivatives,” Eur. J. Phys., vol. 37, no. 6, p. 065008, 2016.
26. B. K. Cho, “Active Suspension MPC controller.pdf,” KSME Int. J., vol. 13, no. 2, pp. 168–174, 1999.
27. M. H. Harun, A. Afhzan, A. Rahim, I. Shairah, A. Halim, and N. E. Abdullah, “Performance Comparison of Active Suspension For Ride Comfort Using Linear Quadratic Regulator (LQR) and Fuzzy Logic Controller (FLC),” in 2018 9th IEEE Control and System Graduate Research Colloquium (ICSGRC), 2018, no. August, pp. 99–104.
28. M. Kaleemullah, W. F. Faris, and F. Hasbullah, “Design of robust H∞, fuzzy and LQR controller for active suspension of a quarter car model,” in 4th International Conference on Mechatronics (ICOM), 2011, pp. 1–6.

AUTHORS PROFILE

Sharifah Munawwarah received her BEng.degrees in Electronic System Engineering from the University of Technology Malaysia in 2017. She is now with the Malaysia-Japan International Institute of Technology as a master student under supervision of Dr. Fitri Yakub. Her research interest includes intelligent control, vibration control systems.

Fitri Yakub received his Dip Eng., and BEng.degrees in Mechatronics Engineering and Electronics Engineering from University of Technology Malaysia in 2001 and 2006 respectively. He obtained MSc. in Mechatronics Engineering from International Islamic University Malaysia in 2011. He received doctorate in Automatic Control Laboratory, Tokyo Metropolitan University in 2015. He is now with the Malaysia-Japan International Institute of Technology since 2012. His field of research interest includes intelligent control, automatic and robust control, and motion control systems.

Pauziah Muhamad is a Seniour Lecturer.She completed her Diploma Mechanical Engineering (UTM), Bachelor Engineering (UTM), Msc Engineering (UTM), PhD Mechanical Engineering (Sheffield,UK). Her research interest includes Nonlinear system identification in structural Vibration, Structural Health Monitoring for damage detection, Smart material-Magnetoreological (MR) and Electrorheological (ER) fluids applications, Vibration damper and active Suspension system and Noise and Vibration Control.

Assoc. Prof. Dr Aminudin bin Hj Abu is the Head of IDS ikohza.He completed his Dip.Eng (SNPU, Seoul, Korea), B.Eng (Hanyang Univ. , Seoul, Korea), M.Sc in Mechanical Precision Eng (Hanyang Univ. Seoul, Korea), Dr. Engineering in Automotive Eng. (Hanyang Univ. Seoul, Korea).He has won numerous awards and he has numerous publications.His Research interests includes Noise and Vibration, Damage detection and Fatigue, Model updating, Sensitivity Analysis, Design of Experiment

Zainudin bin A.Rasid is a Senior Lecturer.He completed his B.Mech Eng (UTM), M.Sc (IIUM), PhD (UPM).He has won numerous awards and he has numerous publications.His Research interests including, Composite and Bio-composite materials, Smart Material, Stress and Post-buckling analysis of structures, Geometric and material non-linearities, Finite element and numerical methods, nad Stress analysis in Gears and mechanical components.