Original Article

Development Of Active Anti Roll Bar Using Fuzzy-PID To Improve Ride Comfort And Handling

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Abstract

Safety aspect is one of the main factors in choosing the type of vehicle that satisfied user's interest. Many studies have been carried out to improve ride comfort including the installation of an active anti-roll bar (AARB) system. AARB works as a vehicle wheel stabilizer that can reduce vehicle body roll during cornering. This study was conducted to obtain the reduction percentage of the roll angle and roll rate for the anti-roll bar when the vehicle passes through a pothole or bump. The value of the roll angle and the roll rate between the anti-roll bar are differentiated and evaluated between passive anti-roll bar (PARB) and active anti-roll bar (AARB) to prove which is the best type of anti-roll bar. Therefore, this study was developed by designing a 4DOF half-vehicle model equipped with an AARB system. Then, this system is added with PID controller and Fuzzy-PID controller that are simulated using MATLAB Simulink software. The anti-roll bar system is tested based on real road situations by include road input disturbances such as speed bump tests and pothole test. Speed bump and pothole tests are conducted to analyze the ride quality of the maneuver single lane test. The simulation results stated that the AARB system with fuzzy-PID has the lowest value of roll angle and roll rate. This contributes to an increase in the percentage reduction of the roll angle and roll rate response. Therefore, this proves that the installation of AARB system with Fuzzy-PID controller is the most efficient in improving ride comfort and vehicle handling.

Keywords: Active anti-roll bar, Fuzzy-PID, Ride comfort, Handling, Half car model.

1. INTRODUCTION

An anti-roll bar is one of fundamental suspension component for an automobile to enhance the stability of cornering. The left and right wheels are connected by a short lever arm and a metal rod called the anti-roll bar. Hence, the anti-roll bar joins the left and right-side suspension systems to strengthen the vehicle's suspension and roll resistance. This anti-roll bar is a crucial component of the car's chassis support system and can further enhance the handling of the vehicle at high speeds, mainly while cornering. The first anti-roll bar is obtained by Canadian inventor, New Brunswick [1]. Anti-roll bar consists of passive and active. Figure 1 shown the system of anti-roll bar. The aim of the ARB is to improve the vehicles roll stiffness while keeping the vehicle's spring system the same.

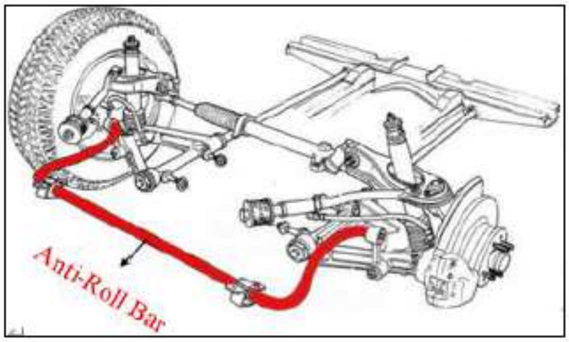

Figure 1. Anti-Roll bar (Robinson, 2018)

Anti-roll bar can enhance vehicle handling by increasing turning stability. Safety characteristics for maneuverings and steering must be the main emphasis of any automobile operation [2]. Car that equipped with anti-roll bar is to ensure the vehicle can be at the same high during high speed to reduce vehicle body roll when passing road corner or rough road. The wheel load will be transferred to opposite direction during the cornering. Therefore, the installation is important to ensure the safety during cornering. This is because, the suspension on one side of the vehicle will be compressed by the lateral force and the roll of the body resulting in the lever arm of the anti-roll bar turning up into a control arm, while the lever at the opposite end will turn down due to the suspension at the bottom of the car stick to droop [3].

Anti-roll bar can increase confident and comfort for drivers and passenger during driving. This is by increasing the stability of the vehicle. Front and rear anti-roll bar reduce body roll percentage compared to front anti-roll bar only. The spring rate will increase on front anti-roll bar that results in understeering. While oversteering happens when spring rate increase on the rear part of anti-roll bar [4]. Understeer effect can be minimise by increase the roll stiffness on rear axle.

Nowadays, many vehicles installed with roll bar or sway bar to minimise the body roll while cornering or passing through rough road. However, many researches have been done to improve and optimise the effectiveness anti roll bar. Therefore, active anti roll bar (ARB) is introduced to change the chassis stiffness and can reduce the amount of trade-off required in the ride comfort and handling characteristic.

Many researchers found that different types of controllers have been use for AARB system. The cost for active ARB system is cheap and the energy usage is small. Therefore, this paper investigated about vehicle that equipped with an active anti roll bar to improve vehicle handling without losing ride comfort. [5] Muniandy et al on 2015 used STF PI-PD as the controller which satisfy ride comfort and handling. Controller gain-schedullingLQ is used to design actuator torque and improve roll chassis dynamic [6] (Varga et al., 2015). On 2019, [7] Dawai et al used PID + Feedforward that can satisfy ride and handling characteristics. While [8] Mohamed et al on 2018 used Fuzzy-PID controller to active ARB system because the controller has advantages on self-tuning satisfy the nonlinear, time-varying and uncertainty of the system. The proposed of controller for this paper is PID controller and Fuzzy-PID controller for active anti roll bar which target to get a value for roll angle and rate angle for different type of roll bar.

2. METHODOLOGY

A. Vehicle Model

For this section, a half car model with 4DOF has been created. The model is created based on a single-track model with roll dynamics which developed from a bicycle model and half car model.

1) The Linear Single Track Model with Roll Dynamic:

The linear single-track model is a bicycle model since it can have achieved by interpreting the front and rear wheel pairs as single wheels. Figure 2 shown the single-track model. The Equation (1) until Equation (2) clarified the motion equation by let the value of angle steering is small.

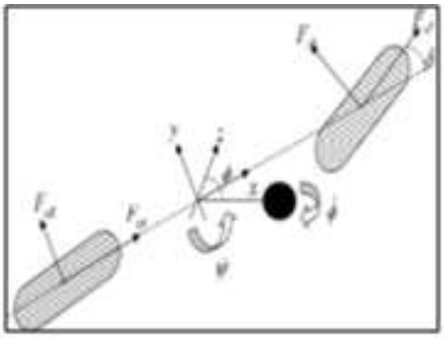

Figure 2. Single-track model (N.Zulkarnain et al., 2014)

$m(V_y + V_x\dot{\psi}) = F_yF + F_yR$ (1)

$I_{zz}\ddot{\psi} = aF_yF - bF_yR$ (2)

$I_{xx}\ddot{\phi} + C_q\dot{\phi} + K_q\,\phi = mh\left(\dot{V}_y + V_x\,\dot{\phi}\right)$ (3)

$F_yF \approx C_f\alpha_F$ (4)

$F_yR \approx C_r\alpha_R$ (5)

where:
F_yF= Force for front wheel
F_yR= Force for rear wheel
C_f= Roll stiffness for front wheel
C_r= Roll stiffness for rear wheel
I_{zz}= The inertia moment around z-axis
a and b = Distance from front and rear wheel from centre gravity
C_q= Damper coefficient
K_q= Spring coefficient
$\dot{\phi}$= Roll rate
α_F and α_R= Slips angle for front and rear wheel

The slips angle can be obtained from Equation (6) and (7).

$\alpha_F \approx \delta - \frac{1}{V_x}(V_y + a\dot{\psi})$ (6)

$\alpha_R \approx -\frac{1}{V_x}(V_y - b\dot{\psi})$ (7)

where:
δ= Angle of steering (input)
V_y = Lateral velocity (output)
$\dot{\psi}$= Yaw rate
ϕ= Roll rate

Next, the single-track model with dynamic equation is illustrated in Equation (8) until Equation (10).

Lateral acceleration:

$\dot{V}_y = \frac{C_f}{m}\left[\delta - \frac{1}{V_x}\left(V_y + a\dot{\psi}\right)\right] + \frac{C_r}{m}\left[\frac{1}{V_x}\left(V_y - b\dot{\psi}\right)\right] - \dot{V}_x\dot{\psi}$ (8)

Yaw acceleration:

$$\ddot{\psi} = \frac{ac_f}{I_{zz}}\left[\delta - \frac{1}{V_x}(V_y + a\dot{\psi})\right] - \frac{bc_r}{I_{zz}}\left[-\frac{1}{V_x}(V_y - b\dot{\psi})\right] \quad (9)$$

Roll acceleration:

$$\ddot{\phi} = \frac{mh}{I_{xx}}(\dot{V}y + Vx\dot{\psi}) - \frac{Cq\dot{\phi}}{I_{xx}} - \frac{Kq\phi}{I_{xx}} \quad (10)$$

2) Half Car Model:

Gandi et. al (2017) stated that half car model with 4DOF have the ability to provide a better knowledge of suspension performance [9]. Hence, the half car model with 4 DOF is choose in this project. The impact of anti-roll bar on both side of wheels suspension system can be studied using this model. Figure 3 shown the half car model system.

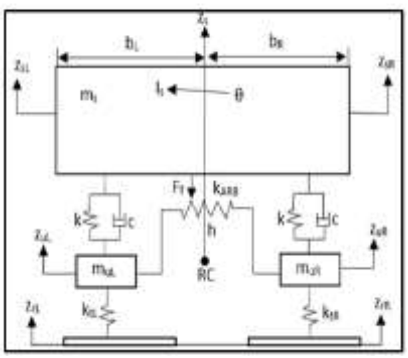

Figure 3. Half car models.

A linear differential equation in Equation (11) until Equation (14) explained the dynamic of a half car model by using Newton's Second Law method.

Body vertical acceleration:

$$\ddot{z}_S = -\frac{2c\dot{z}_S}{m_s} + \frac{c\dot{z}_{uL}}{m_s} + \frac{c\dot{z}_{uR}}{m_s} - \frac{c(b_L - b_R)}{m_s}\dot{\phi} - \frac{2kz_s}{m_s} + \frac{kz_{uL}}{m_s} + \frac{kz_{uR}}{m_s} - k\frac{(b_L - b_R)}{m_s}\phi \quad (11)$$

Left wheel vertical acceleration:

$$\ddot{z}_{uL} = \frac{kz_s}{m_{uL}} - \frac{(k_{tL}+k)}{m_{uL}}z_{u_L} + \frac{c\dot{z}_s}{m_{u_L}} - \frac{c\dot{z}_{uL}}{m_{u_L}} + \frac{kb_L}{m_{u_L}}\phi + \frac{cb_L}{m_{uL}}\dot{\phi} + \frac{k_{tL}}{m_{uL}}z_{rL} + F_L \quad (12)$$

Right wheel vertical acceleration:

$$\ddot{z}_{uR} = \frac{kz_s}{m_{uR}} - \frac{(k_{tR}+k)}{m_{uR}}z_{u_R} + \frac{c\dot{z}_s}{m_{u_R}} - \frac{c\dot{z}_{uR}}{m_{u_R}} - \frac{kb_R}{m_{uR}}\phi - \frac{cb_R}{m_{uR}}\dot{\phi} + \frac{k_{tR}}{m_{uR}}z_{rL} - F_R \quad (13)$$

Roll acceleration of single-track model and half car model:

$$\ddot{\phi} = -\frac{(b_L - b_R)kz_s}{Is} + \frac{(b_L kz_{uL})}{Is} - \frac{(b_R kz_{uR})}{Is} - \frac{c(b_L - b_R)\dot{z}_s}{Is} + \frac{b_L c\dot{z}_{uL}}{Is} - \frac{b_R c\dot{z}_{uR}}{Is} - \frac{k(b_L{}^2 + b_R{}^2)}{Is}\phi - \frac{c(b_L{}^2 + b_R{}^2)}{Is}\dot{\phi} + eF_f$$
(14)

3) Anti-Roll Bar Modeling:

Anti-roll bar (ARB) with passive suspension is combined together. In this part, the roll angle and roll rate of the vehicle will be differentiated and evaluated. This paper investigated about PARB and Active ARB with two types of controllers [10-16]. When Active ARB is combined with active force and a controller, the torque force, T_ARB is generated. Equation (15) and Equation (16) clarified the force of active anti-roll bar and the force of passive anti-roll bar respectively.

$$F_{ARB} = \frac{T_{ARB}}{L} \quad (15)$$

$$F_{ARB} = -k_{ARB}\left[\frac{(z_{SR}-z_{uR})-\left(z_{S_L}-z_{u_L}+w\Phi\right)}{L}\right] \quad (16)$$

where;

z_{rR} and z_{rL} = Different Road motion

The input of passive anti-roll bar system is z_{rR} ,z_{rL} and δ. While z_{rR} ,z_{rL}, δ and T_{ARB} are input for active anti-roll bar.

B. Controller Design

This system implemented two type of controller which are PID and Fuzzy-PID controller. The controller is proposed to reduce roll motion during cornering or passing through uneven road. Figure 4 shows the block diagram of the PID controller.

1) PID Controller

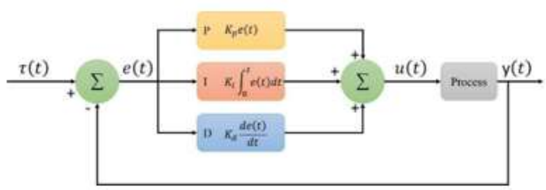

Figure 4. Block diagram of PID

The proportional, integral, and derivative (abbreviated P, I, and D) parameters are the three independent components of the three-term control used in the PID control method. The error value is represented by e(t), and the parameters are Kp, Ki, and Kd Kp is the proportional multiple. This controller's transfer function is as in Equation (17).

$$u(t) = K_\rho e(t) + K_i \int_0^t e(t) + K_d \frac{de(t)}{dt} \quad (17)$$

The simulation of PID control system gives the output control force. The function of this method to get the optimal value of Kp, K_i and K_d. The values are shown in Table I and Table II.

TABLE I. THE VALUE OF K_P, K_I AND K_D FOR N=100000

K_p	95 000
K_i	10 000
K_d	8500

Table 1. The value of $\mathbf{K_p}$, $\mathbf{K_i}$ and $\mathbf{K_d}$ for N=4000

TABLE II. THE VALUE OF K_P, K_I AND K_D FOR N=4000

K_p	65 000
K_i	560 000
K_d	8000

2) Fuzzy-PID Controller

(a) The principle of Fuzzy-PID controller

Fuzzy logic controller is one example of continuous control. The output for the control is determined by the fuzzy logic that exists between the minimum and maximum damping state. Fuzzy logic works by implementing rules that can associate control inputs with desired outputs [10]. The gain for Fuzzy-PID can changed automatically based on dynamic suspension. Figure 5 shows the design of Fuzzy-PID controller.

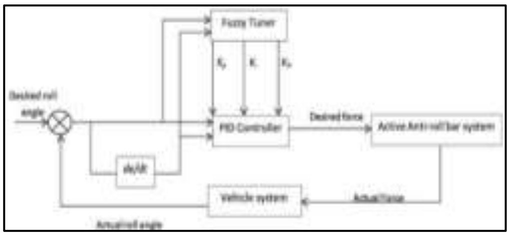

Figure 5. Fuzzy-PID controller configuration

The equation for Fuzzy-PID is defined as Equation (18).

$$T_{ARB}(t) = K_p e(t) + K_i \Sigma e(t) + K_d \Delta e(t) \quad (18)$$

where;

$e(t)$ = System error

$\Sigma e(t)$= Error sum

$\Delta e(t)$= Error variety

$e = \phi - 0$; vehicle roll angle error.

(b) Control rules

Table III shows the linguistic variable which described as NB, NS, Z, PS and PB. NB refer to negative big. NS refer to negative small, Z refer to zero, PS refer to positive small, PB refer to positive big. Figure 6 and Figure 7 illustrated the input of membership function is shown in. The graph's x-axis displays the range of input e (vehicle roll angle) and output Kp, while the graph's y-axis displays the degree of membership in the fuzzy controller (proportion coefficient).

TABLE III. FUZZY CONTROLLER RULES (MOHAMED ET. AL., 2019)

AND		u(t)				
		NB	NS	Z	PS	PB
e(t)	NB	NB	NB	NS	NS	Z
	NS	NB	NS	NS	Z	PS
	Z	NS	NS	Z	PS	PS
	PS	NS	Z	PS	PS	PB
	PB	Z	PS	PS	PB	PB

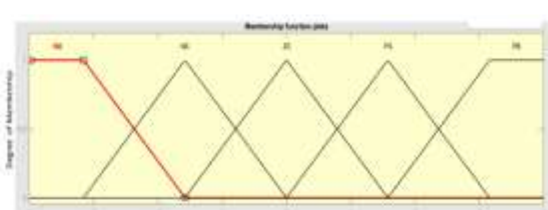

Figure 6. Membership function of input e

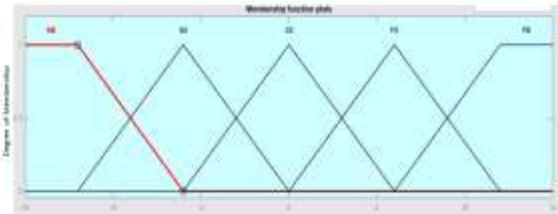

Figure 7. Membership function of output $\mathbf{K_p}$

3. SIMULATION SETUP

The simulation of the system is run to get the value of roll angle and roll rate for both controller which are PID controller and Fuzzy-PID controller. The value will be analyzed and differentiated with passive anti roll bar, active anti roll bar with PID and active anti roll bar with fuzzy-PID. For simulation, the real road situation has been implemented. The road disturbance speed bump test and pothole test are included to get the results. Table IV shows the parameters for the 4DOF half car model with single-track model used in the simulation.

TABLE IV. PARAMETERS FOR THE 4 DOF HALF CAR MODEL WITH SINGLE-TRACK MODEL

Parameter	**Symbol**	**Value**	**Unit**
Mass of car	ms	715	kg
Length between center gravity and left wheel suspension	bL	0.7	m
Length between center gravity and right wheel suspension	bR	0.75	m
Left and right tire stiffness	ktL & ktR	200000	N/m
Suspension stiffness of car	k	10000	N/m
Suspension damping of car	c	4000	N/m
Longitudinal mass moment of inertia	Is	820	kgm^2
Mass of left and right wheel	muL & muR	53	kg

C.G's height above the roll axis	h	0.36	m
Torsional stiffness	k_{ARB}	10000	N/rad
Mounting location of stabilizer bar	e	0.325	m
Length of stabilizer arm	L	0.4	m
Length of stabilizer bar	W	1.4	m
Gravitational acceleration	g	9.81	ms^2
Moment of inertia around the x axis	Ixx	602.82	kgm^2
Front cornering stiffness	Cf	18000	N/rad
Rear cornering stiffness	Cr	47000	N/rad
Forward velocity	Vx	100	km/h
Distance between the front wheel and the C.G	a	1.1	m
Distance between the rear wheel and the C.G	b	1.4	m
Moment of inertia around the z axis	Izz	2430	kgm^2
Damping coefficient	Cq	3495.7	N/m
Spring coefficient	Kq	56957	N/m

A. Passive ARB System (PARB)

passive anti roll bar (PARB) system developed using MATLAB/Simulink. The block diagram in Figure 8 shows the system. For this simulation, a few tests is analyzed to get the value for roll angle and roll rate for the PARB. The type of test speed bump test and pothole test which only affect the left wheel only. For steering input disturbance, maneuver single lane test is simulated and analyze.

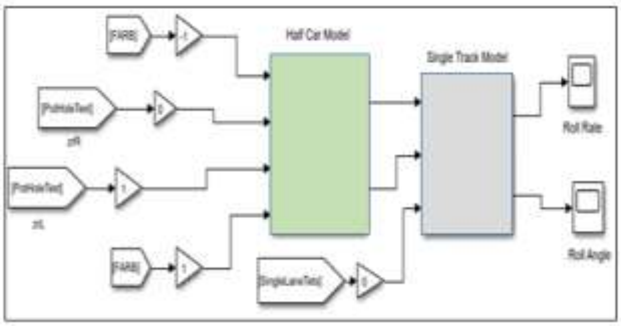

Figure 8. Block diagram for passive ARB system

The block diagram for AARB with PID controller and AARB with fuzzy-PID is shown in Figure 9 and Figure 10 respectively.

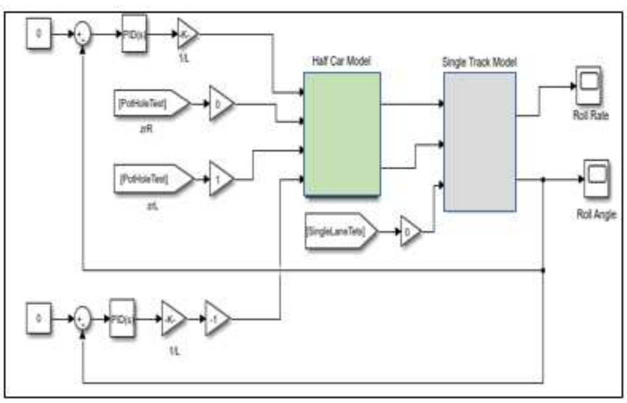

Figure 9. Active anti roll bar system with PID controller

4. SIMULATION RESULTS

The system in simulated and the result for value of roll angle and roll rate is compared between passive anti roll bar, active anti roll bar with PID and active anti roll bar with Fuzzy-PID. The ride comfort and is test using speed bump and pothole test, while handling characteristic will be test by single lane change.

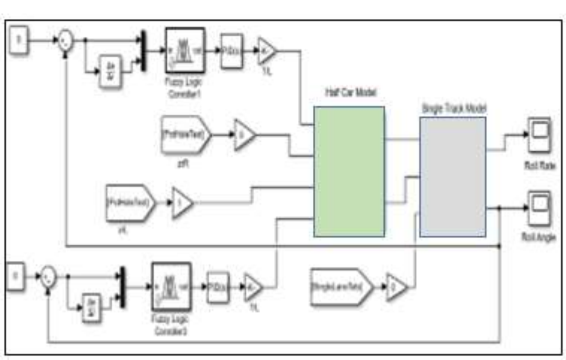
Figure 10. Active anti roll bar system with fuzzy-PID controller

A. Ride Analysis Result

1) Single Bump Test

In real driving, every road will definitely have bumps to ensure that the vehicle is driven slowly in certain areas. Therefore, Figure 11 shows the simulation results for the roll angle response in the single bump test. In the simulation, there is a comparison of roll angle values between PARB and AARB that use PID controller and Fuzzy-PID controller. The simulation results found that the roll angle value of the PARB is 0.0325 rad, which is the highest value between the PID controller and the Fuzzy-PID controller. Whereas for the AARB that uses a PID, the roll angle is 0.0212 rad lower than the roll angle of the PARB and higher than the roll angle of the active anti-roll bar with the Fuzzy-PID controller which is 0.0173 rad. The average reduction percentage for the PID controller is 34.77% while the average reduction for the Fuzzy-PID controller is 46.77%. The simulation response of roll angle illustrated in Figure 12.

2) Multiple Bump Test

The simulation results show that the roll angle value for vehicles that install PARB is 0.0313 rad, which is higher than other AARB. The AARB with PID controller recorded a roll angle value of 0.0175 rad, while the active anti-roll bar with Fuzzy-PID controller recorded a roll angle value of 0.0140 rad. Therefore, the average reduction percentage experienced by the active anti-roll bar with PID controller is as much as 44.09% while for the active anti-roll bar with Fuzzy-PID controller is as much as 55.27%.

The roll rate for the PARB is the highest with 0.316 rad/s, followed by the roll rate for the active anti-roll bar with PID of 0.157 rad/s and then the lowest value which is the AARB with Fuzzy-PID controller of 0.126 rad/ s. The average reduction percentage for AARB with PID controller is 50.31%, while for Fuzzy-PID controller is 60.13%. Figure 13 and Figure 14 stated the result simulation for roll angle and roll rate.

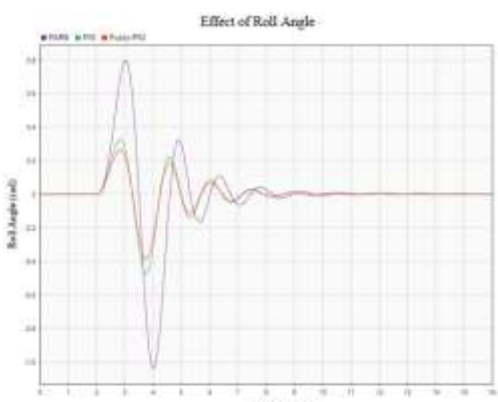

Figure 11. Response of roll angle for bump test

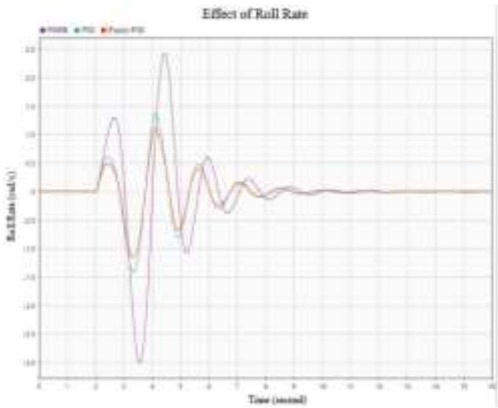

Figure 12. Response of roll rate for bump test

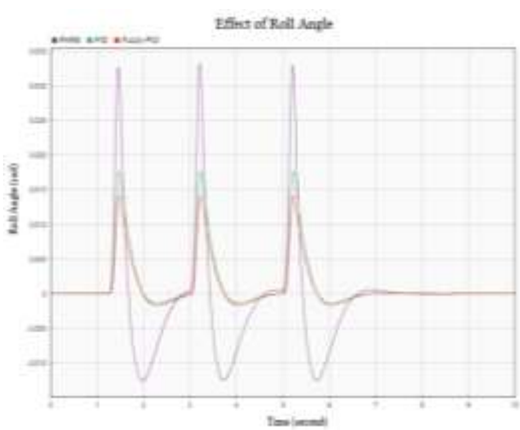

Figure 13. Response of roll angle for multiple bump test

3) Pothole Test

A pothole test is simulated to differentiate the roll angle response between PARB, AARB with PID controllers and Fuzzy-PID controller. The simulation results found that the roll angle value of the passive anti-roll bar is 0.0217 rad, which is the highest value between the PID controller and the Fuzzy-PID controller. Whereas for the AARB that uses a PID controller, the roll angle is 0.0140 rad lower than the roll angle of the passive anti-roll bar and higher than the roll angle of the active anti-roll bar with the Fuzzy-PID controller which is 0.0105 rad. The average reduction percentage for the PID controller is as much as 35.48% while the average reduction for the Fuzzy-PID controller is as much as 51.61%.

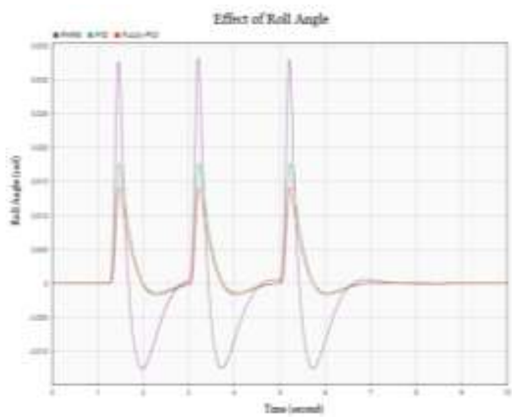

Figure 14. Response of roll rate for multiple bump test

The roll rate for the PARB is the highest with 0.210 rad/s, followed by the roll rate for the AARB with PID of 0.125 rad/s and then the lowest value which is AARB with Fuzzy-PID controller of 0.094 rad/ s. The average reduction percentage for active anti-roll bar with PID controller is 40.48%, while for Fuzzy-PID controller is 55.24%. The value of the roll angle of the passive anti-roll bar is 0.79 rad which is the highest value between the PID controller and the Fuzzy-PID controller. Whereas for the active anti-roll bar that uses a PID controller, the roll angle is 0.32 rad lower than the roll angle of the passive anti-roll bar and higher than the roll angle of the active anti-roll bar with the Fuzzy-PID controller which is 0.26 rad. The average reduction percentage for the PID controller is 59.49% while the average reduction for the Fuzzy-PID controller is 67.09%. The effect of roll angle and role rate is shown in Figure 15 and Figure 16.

4) Single Lane Change Test

The simulation results found that the roll angle value of the PARB is 0.79 rad, which is the highest value between the PID controller and the Fuzzy-PID controller. Whereas for the AARB that uses a PID controller, the roll angle is 0.32 rad lower than the roll angle of the passive anti-roll bar and higher than the roll angle of the active anti-roll bar with the Fuzzy-PID controller which is 0.26 rad. The average reduction percentage for the PID controller is 59.49% while the average reduction for the Fuzzy-PID controller is 67.09%.

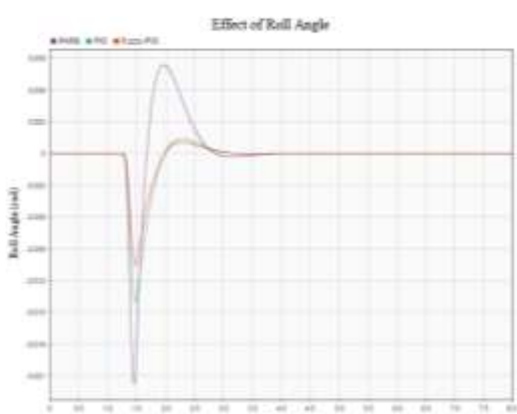

Figure 15. Response of roll angle for pothole test

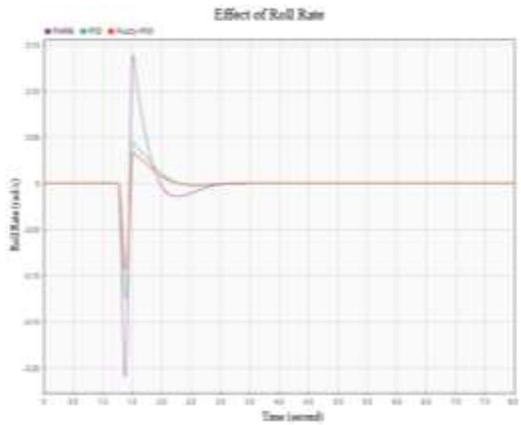

Figure 16. Response of roll rate for pothole test

The roll rate for the passive anti-roll bar is the highest with 2.43 rad/s, followed by the roll rate for the active anti-roll bar with PID of 1.37 rad/s and then the lowest value which is the active anti-roll bar with Fuzzy-PID controller of 1.12 rad/s. The average reduction percentage for active anti roll bar with PID controller is 43.62%, while for Fuzzy-PID controller is 53.91%. The effect of roll angle and role rate is shown in Figure 17 and Figure 18.

Based on the simulation, active anti roll bar with fuzzy-PID gives the lowest value for roll angle and roll rate compared to active anti roll bar with PID and passive anti roll bar. Hence, it proves that vehicle equipped with active anti roll bar with Fuzzy-PID have better ride comfort and handling. Therefore, the active anti roll bar consists of fuzzy-PID controller that will generate the needed amount of force required to balance the vehicle which can reduce the roll angle and roll rate. PID controllers and Fuzzy-PID controllers are able to change the stiffness of the anti-roll bar according to the road profile and produced the stability of the vehicle. Meanwhile, the passive anti-roll bar only depends on the stiffness of the anti-roll bar. This makes the active anti-roll bar more effective in reducing the roll angle and roll rate.

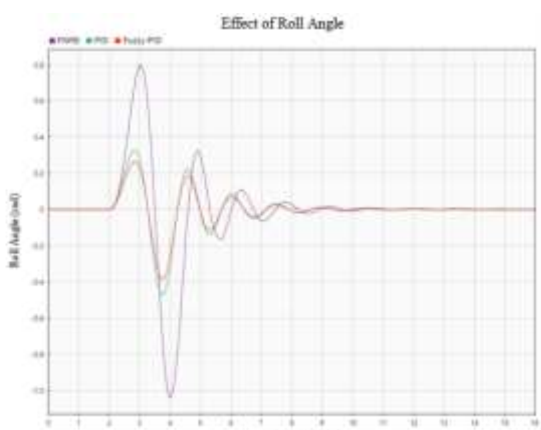

Figure 17. Response of roll angle for single lane change

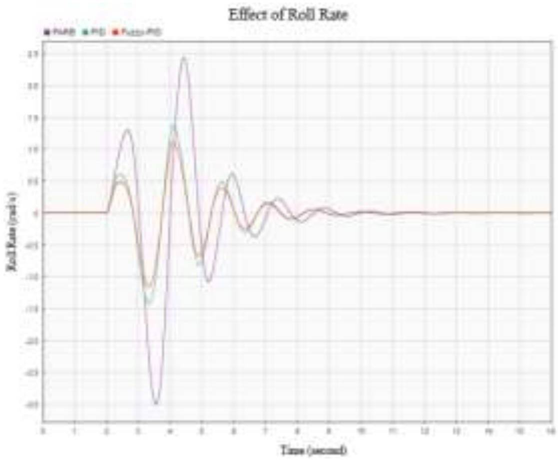

Figure 18. Response of roll rate for single lane change test

5. CONCLUSIONS AND FUTURE WORK

This paper distinguishes the performances of different roll bar such as passive anti roll bar, active anti roll bar with PID and active anti roll bar with Fuzzy-PID. A 4DOF half-car model been proposed to comprehend the study of AARB to enhance ride comfort. The MATLAB/Simulink software is used to simulate the response of roll angle and roll rate for 3 types of suspension. Based on the result, it clearly shows that AARB with Fuzzy-PID have a better performance which capable of achieving better ride and handling performance by the reduction of roll angle and roll road during passing through the road disturbance. For future work, the optimization of active anti roll bar can be investigated in order to improve vehicle stability and contribute to ride comfort.

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