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HALF CAR ACTIVE SUSPENSION SYSTEM

Amat A. Basari[1,a*], Saifullah Salam[2,b] Khairul A.A. Aziz[3,c] and Redzuan A. Manap[4,d]

[1,2,3,4] Universiti Teknikal Malaysia Melaka, Durian Tunggal, Malaysia. [a]Email: amat@utem.edu.my, [b]Email: saifullah@utem.edu.my, [c]Email: khairulazha@utem.edu.my, [d]Email: redzuan@utem.edu.my

***Abstract*— This paper presents a new method in modeling an active suspension system for a half-car model in state space form and develop a robust control strategy in controlling the active suspension system. Fuzzy logic is used to control the system. Velocity and displacement of front wheels are taken as input variables of the fuzzy logic controller. Active forces improving vehicle driving, ride comfort and handling properties are considered to be the controller outputs. The controller design is proposed to minimize chassis and wheels deflection when uneven road surfaces, pavement points, etc. are acting on the tires of running cars. Comparison of performance of active suspension fuzzy control system with passive suspension system is shown using Matlab/Simulink simulation. From the result, it shows that active suspension system has better performance than the passive suspension system.**

***Keywords*—Half Car Suspension System, Fuzzy Logic Controller, Active Suspension System, Passive Suspension System**

O. INTRODUCTION

Suspension system of a vehicle is critical in many aspects. Its main function is to isolate the body of the vehicle from the road surface. Besides that, it also determines the vehicle handling performance as well as the comfortness. Typical design of suspension system will consist of shock absorber and coil spring that will minimize the shock absorbsion by passenger of the vehicle. In order to obtain a good response, we may vary the constant value of each element stated above. Over pass decades, many researchers have come with an idea of active suspension system. The idea is that, we may add another control element to the system. By adding the third control element to the system, we can create a closed loop system as the input from reflection of the tire will feedback to the system.

There are many works have been done in designing an active suspension system. The dynamic of suspension system is nonlinear. Therefore, the use of robust controller in the system is needed. Uncertainty from road surface is another factor that contributes to the need of a robust control scheme for the system. [1] and [2] have obtained a good performance of the active system by utilizing the PI sliding mode control and nonlinear optimal control scheme. Both papers present the effectiveness of the control scheme in giving better performance of the system. Application of fuzzy logic controller in the active suspension system is presented by many researchers. In [3], the writer has proposed the fuzzy control for a quarter car suspension system using the mamdani model approach. While in [4], the writer has work on the Takagi-Sugeno model. The ability of fuzzy control in controlling a nonlinear dynamic system is well known. It posses nonlinear mapping capabilities, and do not require an analytical model. One of the example is in [5]. It presents induction motors control using the fuzzy logic controller.

II. HALF CAR SUSPENSION MODEL

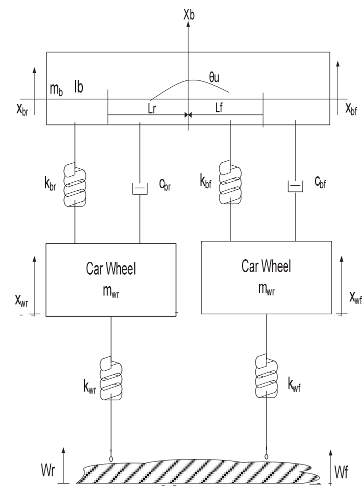

Figure 1. A half-car active suspension model.

Consider the half car suspension system as shown in Fig. 1. The systems' equation is represented by

$$M\ddot{X} + S\dot{X} + TX - Df - Ew = 0 \qquad (1)$$

Where the state vector, active control vector and excitation vector are respectively given by,

$$X = \begin{pmatrix} x_{bf} & x_{wf} & x_{br} & x_{wr} \end{pmatrix}^T,\ f = \begin{pmatrix} f_f & f_r \end{pmatrix}^T,$$
$$w = \begin{pmatrix} \dot{w}_f & w_f & \dot{w}_r & w_r \end{pmatrix}^T$$

The matrices M, S, T, D and E respectively are given as follows:

$$M = \begin{bmatrix} L_r m_b / L & 0 & L_f m_b / L & 0 \\ I_b / L & 0 & -I_b / L & 0 \\ 0 & m_{wf} & 0 & 0 \\ 0 & 0 & 0 & m_{wr} \end{bmatrix}$$

$$S = \begin{bmatrix} c_{bf} & -c_{bf} & c_{br} & -c_{br} \\ L_f c_{bf} & -L_f c_{bf} & -L_r c_{br} & L_r c_{br} \\ -c_{bf} & c_{bf} & 0 & 0 \\ 0 & 0 & c_{br} & -c_{br} \end{bmatrix}$$

$$T = \begin{bmatrix} k_{bf} & -k_{bf} & k_{br} & -k_{br} \\ L_f k_{bf} & -L_f k_{bf} & -L_r k_{br} & L_r k_{br} \\ -k_{bf} & k_{bf}+k_{wf} & 0 & 0 \\ 0 & 0 & -k_{br} & k_{br}+k_{wr} \end{bmatrix}$$

$$D = \begin{bmatrix} 1 & 1 \\ L_f & -L_r \\ -1 & 0 \\ 0 & 0 \end{bmatrix} \qquad E = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ k_{wf} & 0 & 0 & 0 \\ 0 & 0 & k_{wr} & 0 \end{bmatrix}$$

I_b is the mass moment of inertia for the vehicle body, m_b is the mass for the vehicle body, m_{wf} and m_{wr} are the mass of the front and rear wheels, respectively, x_c is the vertical displacement of the vehicle body at the centre of gravity, x_{bf} and x_{br} are the vertical displacements of the vehicle body at the front and rear suspension locations, respectively, x_{wf} and x_{wr} are the vertical displacement of the vehicle body at the front and rear wheels, respectively, θ_o is the rotary angle of the vehicle body at the centre of gravity, f_f and f_r are the active controls at the front and rear suspensions, respectively, w_f and w_r are the irregular excitations from the road surfaces, L_f and L_r are the distances from the front and rear suspension locations, respectively, with reference to the centre of gravity of the vehicle body and $L_f + L_r = L$.
Define the N state variables for the system as $x_i = X$ and

$$x_{i+\frac{N}{2}} = \dot{X}$$

where i = 1, 2, 3,…, N/2, the half-car model in equation (1) can be rewritten in state space form as follows:

$$\dot{x}_i(t) = Ax_i(t) + B_p z(t) \qquad (2)$$

where

$$A = \begin{bmatrix} 0_{\frac{N}{2}\times\frac{N}{2}} & I_{\frac{N}{2}\times\frac{N}{2}} \\ A_{p1} & A_{p2} \end{bmatrix}$$

$$A_{p1} = -M^{-1}T, A_{p2} = -M^{-1}S$$

$$B_p = \begin{bmatrix} 0_{\frac{N}{2}} \\ M^{-1}E \end{bmatrix} \qquad B = \begin{bmatrix} 0_{\frac{N}{2}} \\ M^{-1}D \end{bmatrix}$$

u(t) and *z(t)* are the control input and the disturbance input respectively.
The state equations of the half-car model in the state space form with the state variables defined as $x_1 = x_{bf}, x_2 = x_{wf}, x_3 = x_{br}, x_4 = x_{wf}$, $x_5 = \dot{x}_{bf}, x_6 = \dot{x}_{wf}, x_7 = \dot{x}_{br}$, and $x_8 = \dot{x}_{wr}$. The state equation shows that the disturbance input is not in phase with the actuator input, $rank[B] \neq rank[B \quad B_{p2}]$, therefore the system does not satisfying the matching condition. The analysis with rewriting equation (2) into the following form,

$$\dot{x}(t) = Ax(t) + Bu(t) + f(t) \qquad (3)$$

Where $x(t) \in \Re^n$ is the state vector, $u(t) \in \Re^m$ is the control input, and the continuous function $f(t)$ represents the uncertainties with the mismatched condition, $rank[B \quad f(t)] \neq rank[B]$.

III. Fuzzy Logic Controller Design

There are specific components characteristic of a fuzzy controller to support the design procedure. They are fuzzification, rule base determination and defuzzification. For this particular system two inputs was used: front body velocity $\dot{x}_{bf}$ and front body displacement x_{bf}. The output of the fuzzy control f_f and f_r are the desired actuator force. A possible choice of the membership functions for the three mentioned variables of the active suspension system represented by a fuzzy set is as shown in Fig. 2, Fig. 3 and Fig. 4.

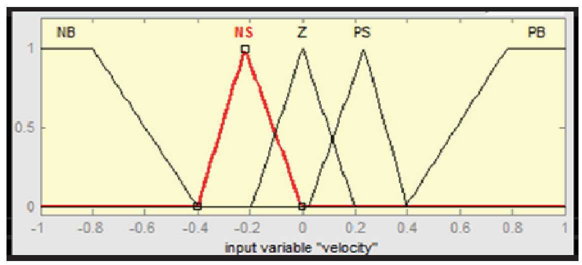

Figure 2. Membership function of body velocity

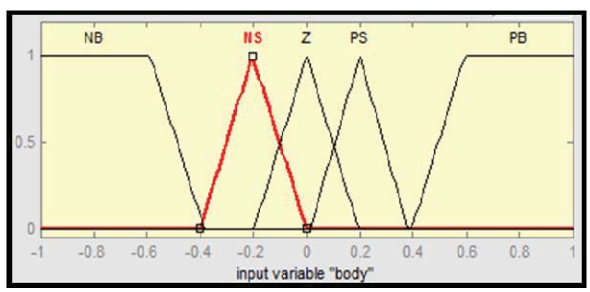

Figure 3. Membership function of body velocity

The abbreviations used correspond to:
NB is negative big, NS is negative small, Z is zero, PS is positive small and PB is positive big. The rule base determined by the designer based on the knowledge and experience. The target of the rule base is to minimize the vehicle body acceleration and it is directly will minimize the body displacement as well.

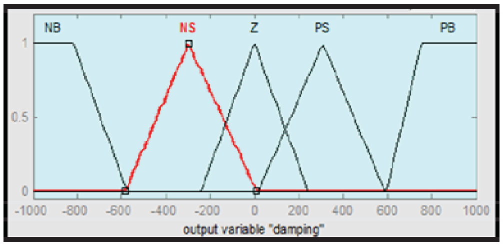

Figure 4. Membership function of desired actuator force

Table 1. Rule Base for Membership Function

No	Body velocity	Body displacement	Desired output
1	PB	PB	NB
2	PB	PS	NS
3	PB	Z	NS
4	PB	NS	NS
5	PB	NB	NS
6	PS	PB	NS
7	PS	PS	NS
8	PS	Z	NS
9	PS	NS	NS
10	PS	NB	NS
11	Z	PB	NB
12	Z	PS	NS
13	Z	Z	Z
14	Z	NS	Z
15	Z	NB	PS
16	NS	PB	NS
17	NS	PS	NS
18	NS	Z	PS
19	NS	NS	PS
20	NS	NB	PB
21	NB	PB	NB
22	NB	PS	NS
23	NB	Z	Z
24	NB	NS	PS
25	NB	NB	PB

For example, the linguistic control rules of the fuzzy logic controller obtained from the table above used in such case are as follows:

No 1: IF body velocity is PB AND body displacement is PB THEN desired output is NB

No 20: IF body velocity is NS AND body displacement is NB THEN desired output is PB

The output of the fuzzy controller is a fuzzy set of control. Thus, non fuzzy value must be obtained and in defuzzification step, the method called "centre of gravity (COG)" is used.

$$f = \frac{\int f * \mu_D(f)df}{\int \mu_D(f)df} \qquad (4)$$

where $\mu_D(f)$ is corresponding membership function

IV. Simulation and Discussion

The mathematical model of the system as defined in equation (3) and the proposed fuzzy logic controller were simulated on computer. For comparison purposes, the performance of the fuzzy logic controller is compared to the half-car passive suspension system approach. The numerical values for the model parameters are taken from [2], and are as follows:

Table 2. Vehicle Parameter

Car body, m_b	575 Kg
Centroidal moment of inertia for the car body, I_b	769 Kg/m^2
Front wheel mass, m_{wf}	60 Kg
Rear wheel mass, m_{wr}	60 Kg
Front spring coefficient, k_{bf}	1682 N/m
Rear spring coefficient, k_{br}	16812 Nm
Front tire spring coefficient, k_{wf}	190000 N/m
Rear tire spring coefficient, k_{wr}	190000 N/m
Front damping coefficient, C_{bf}	1000 N/m
Rear damping coefficient, C_{br}	1000 N/m

Road Profile

Road profile is represented by a single bump on the road with 5 cm bump height. The mathematical model of the bump for such road is given by as:

$$w(t) = \begin{cases} a(1-\cos(8\pi t))/2 & 2.25 \le t \le 2.5 \\ 0 & elsewhere \end{cases}$$

Where $a = 5cm$ and lower time limit is 2.25 *second* and the upper time limit is 2.5 *second.* Fig. 6 shows the road profile.

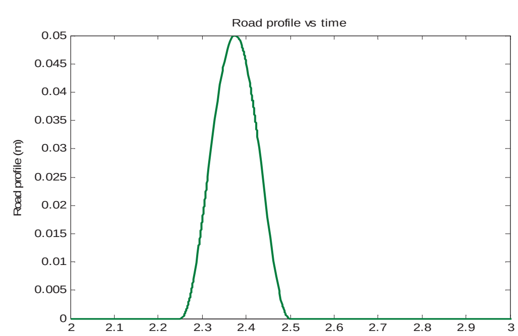

Figure 6. Road profile presented a 5 *cm* bump

The performance of the ride comfort may be observed through the car body acceleration and the performance of road handling may be observed through the wheel deflection. There are 4 parameters to be observed in the simulation. The parameters are the body acceleration, wheel deflection, suspension travel and the force to the passenger. In order to make necessary comparison between passive and active suspension, simulations have been carried using MATLAB/Simulink. The results of compared simulated values is discussed.

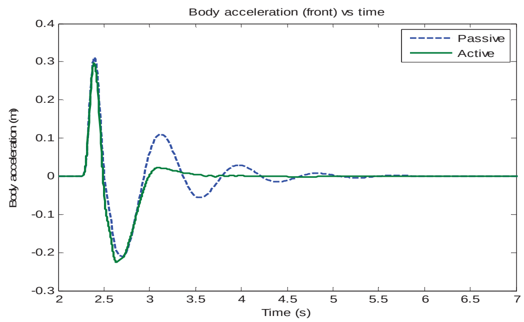

Figure 7(a). Body acceleration comparison between active and passive suspension system (front)

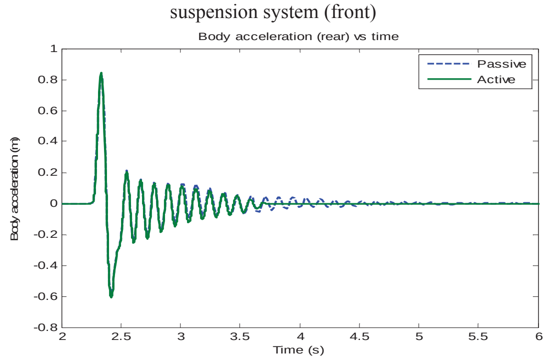

Figure 7(b). Body acceleration comparison between active and passive suspension system (rear)

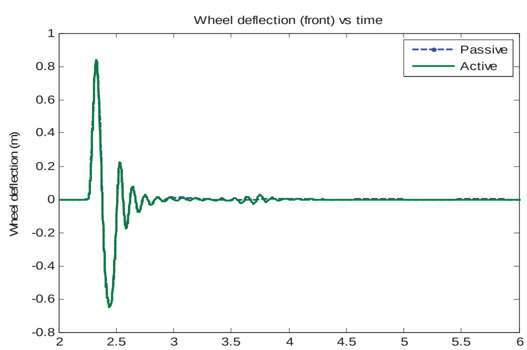

Figure 8(a). Wheel deflection comparison between active and passive suspension system (front)

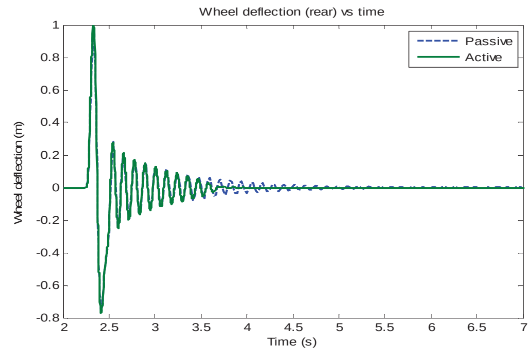

Figure 8(b). Wheel deflection comparison between active and passive suspension system (rear)

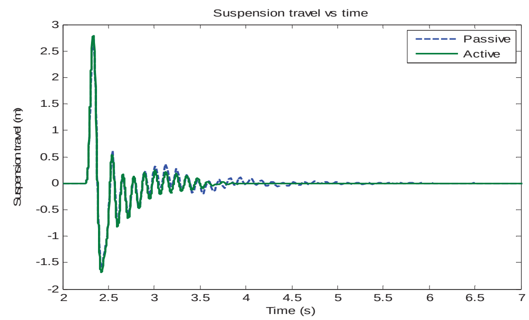

Figure 9. Suspension travel comparison between active and passive suspension system

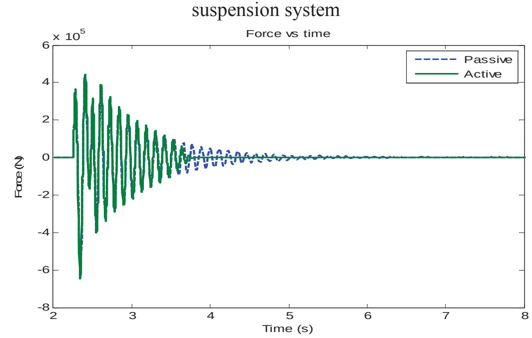

Figure 10. Force to the passenger comparison between active and passive suspension travel

Fig. 7(a) and (b) illustrates clearly the performance of the ride comfort for passive and active suspension system of the car. It is relates with the height bump. If the height bump is high, the body acceleration also increases. It also shows the comfort ability sensed by the passengers. This will explain that it control the spring motions of the car suspension system. An undamped car will oscillate up and down. But, with proper damping levels, the car will settle back to a normal state in minimal amount of time. Notice that if the damping is too strong, it also has negative consequences to the ride comfort ability. Fig. 8(a) and (b) shows the wheel deflection of the passive and active suspension system which is related to the road handling. It is associated with the contact forces of the tires and the road surfaces. In this situation, we figured that the spring is used to absorb the impact of the car. If the suspension system of the car is too hard, it will results in throwing the car on unevenness of the road. While the system is too soft, it will swing the car hence lost the contact between wheels and the road.

Fig. 9 illustrates the suspension travel for the passive and active suspension system. It is relatives to sprung and unsprung masses. It should always be lesser than the rattle space. Fig. 10 shows the response of force that given to the passenger in passive and active suspension system. From the graph, we can say that the body acceleration is directly proportional to the force. Hence, to reduce the force to the passenger we must decrease the body acceleration. As a result, since we apply the Fuzzy Logic controller into the active suspension system, the force to the passenger is reduce as well as the body acceleration.

V. Conclusion

The performances of the active suspension system under the Fuzzy Logic Controller techniques have been evaluated. Comparisons in terms of body acceleration and deflection of such systems have been carried out using the computer simulation works. In overall, the Fuzzy Logic Controller has shown better performances as compared to the conventional half-car passive suspension system.

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