Active Suspension Control of a Vehicle System Using Intelligent Fuzzy Technique

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Abstract: In this paper, four degrees of freedom half body vehicle suspension system is shown and the road roughness intensity is modeled as a filtered white noise stochastic process. PID and Fuzzy logic techniques are used to control the suspension system. The desired objective is proposed as minimization of sprung mass acceleration, pitching acceleration, suspension travel and dynamic loads. The simulation results show that active suspension Fuzzy control with the body vertical acceleration and suspension dynamic deflection as comprehensive feedback parameters compared with PID control suspension can improve the ride comfort and driving stability in random excitation.

Keywords: *Vehicle, Active Suspension, Fuzzy Control, PID control, Ride Comfort.*

INTRODUCTION

Suspension is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels. Suspension systems can't only contribute to the car's handling and braking for good active safety and driving pleasure, but also keep vehicle occupants comfortable and reasonably well isolated from road noise, bumps, and vibrations, [1,2].A vehicle designer can do little to improve road surface roughness, so designing a good suspension system with good vibration performance under different road conditions becomes a prevailing philosophy in the automobile industry.

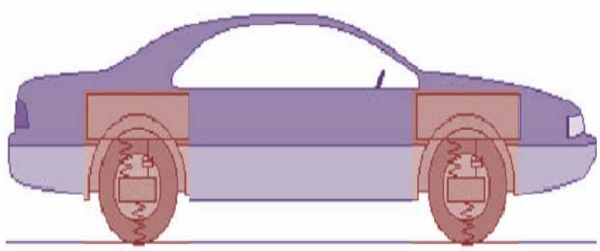
In the past three decades, there has been a considerable research activity in the field of computer controlled suspension systems for road vehicles. Passive suspension systems use conventional dampers to absorb vibration energy, the dampers and stiffness coefficients are constant. The active suspension system use extra power to provide a response-dependent damper, which is capable of producing an improved ride comfort. Over the years, both passive and active suspension systems have been proposed to optimize a vehicle's ride quality. Consequently, the traditional trade-offs between ride comfort and road holding have been relaxed to some extent [3-9].

At present, vehicle suspension has obtained high performance damping effect by using optimal control method. Industrial countries have begun to study an active, semi-active suspension system based on vibration control in the 1970s. In the 1960s, foreign scholars have proposed the concept of active suspension. Industry developed in the 70's has been of active [10-11], semi-active [12-13] suspension system based on the active vibration control. Domestic and foreign scholars have applied optimal control [14-15], adaptive control [16-18], fuzzy control [19-20], artificial neural network [24-25] to vibration control of vehicle suspension system. Modern control theory has made vibration control technology of the suspension system more perfect.

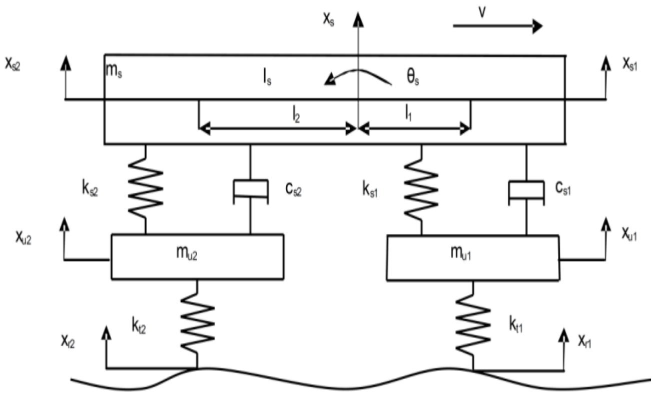
In the paper, a half body with four degrees of freedom suspension model is established as research foundation. The automobile active suspension system is manipulated as a coupled multi inputs multi output (MIMO) control problem. Based on classical PID and self-adapt parameters fuzzy controllers, active suspension systems are designed.

2. SYSTEM MODEL

Multi-body dynamics has been used extensively by automotive industry to model and design vehicle suspension. A simplified half model with four degrees of freedom is established to simulate the system as shown in Fig.1.



(a)



(b)

Fig.1 The half-car model of the vehicle, [22]

The equations of motion for the vehicle body and the front/rear wheels are given by:

$$m_s \ddot{x}_s + c_{s1}(\dot{x}_{s1} - \dot{x}_{u1}) + c_{s2}(\dot{x}_{s2} - \dot{x}_{u2}) + k_{s1}(x_{s1} - x_{u1}) + k_{s2}(x_{s2} - x_{u2}) = 0 \quad (1)$$

$$I_s\ddot{\theta}_s + I_1 c_{s1}(\dot{x}_{s1} - \dot{x}_{u1}) + k_{s1}(x_{s1} - x_{u1}) - I_2 c_{s2}(\dot{x}_{s2} - \dot{x}_{u2}) + k_{s2}(x_{s2} - x_{u2}) = 0 \quad (2)$$

$$m_{u1}\ddot{x}_{u1} + c_{s1}(\dot{x}_{s1} - \dot{x}_{u1}) - k_{s1}(x_{s1} - x_{u1}) + k_{r1}(x_{u1} - x_{r1}) = 0 \quad (3)$$

and the constraints are given by [8]

$$m_{u2}\ddot{x}_{u2} + c_{s2}(\dot{x}_{s2} - \dot{x}_{u2}) - k_{s2}(x_{s2} - x_{u2}) + k_{r2}(x_{u2} - x_{r2}) = 0 \quad (4)$$

$$x_s = (I_2 x_{s1} + I_1 x_{s2})/l \quad (5)$$

$$\theta_s = (x_{s1} - x_{s2})/l \quad (6)$$

where m_s is the mass for the vehicle body, $I_s$is the mass moment of inertia for the vehicle body, $m_{u1}$and m_{u2} are the masses of the front/rear wheels respectively, k_{s1} and k_{s2} are the damping coefficients of front/rear suspensions respectively, k_{s1} and k_{s2} are the spring stiffness of front/rear suspensions respectively, k_{t1} and k_{t2} are the stiffness of front/rear tires respectively. x_s is the vertical displacement the vehicle body at the center of gravity, θ_s is the rotary angle of the vehicle body at the center of gravity, x_{u1} and x_{u2} are the vertical displacements of the front/rear wheels, x_{r1} and x_{r2} are the irregular excitations from the road surface, x_{s1} and x_{s2} are the vertical displacements of the vehicle body at the front/rear suspension locations, $l_1$and $l_2$are the distances of the front/rear suspension locations, with reference to the center of gravity of the vehicle body, and $l_1 + l_2 = l$.

Equations (1) to (4) can be rewritten as

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = \{P\}$$

Where

$$[M] = \begin{bmatrix} \frac{l_2 m_s}{l} & 0 & \frac{l_1 m_s}{l} & 0 \\ \frac{I_s}{l} & 0 & \frac{-I_s}{l} & 0 \\ 0 & m_{u1} & 0 & 0 \\ 0 & 0 & 0 & m_{u2} \end{bmatrix},$$

$$[C] = \begin{bmatrix} c_{s1} & -c_{s1} & c_{s2} & -c_{s2} \\ l_1 c_{s1} & -l_1 c_{s1} & -l_2 c_{s2} & l_2 c_{s2} \\ -c_{s1} & c_{s1} & 0 & 0 \\ 0 & 0 & -c_{s2} & c_{s2} \end{bmatrix}$$

$$[K] = \begin{bmatrix} k_{s1} & -k_{s1} & k_{s2} & -k_{s2} \\ l_1 k_{s1} & -l_1 k_{s1} & -l_2 k_{s2} & l_2 k_{s2} \\ -k_{s1} & k_{s1} + k_{t1} & 0 & 0 \\ 0 & 0 & -k_{s2} & k_{s2} + k_{t2} \end{bmatrix}$$

$$\{P\} = \begin{bmatrix} 0 \\ 0 \\ k_{t1} x_{r1} \\ k_{t2} x_{r2} \end{bmatrix}, \{X\} = \begin{bmatrix} x_{s1} \\ x_{u1} \\ x_{s2} \\ x_{u2} \end{bmatrix}$$

The displacement x_{r1} and x_{r2} may be represented by an irregular road excitation as band limited white noise, which is determined by different road surface conditions and velocity. In this study, the vehicle travels over the bump in the *x* direction with velocity, *Va* and *b* are the distances from the body centre to the front and rear suspensions. There is a time delay of δt between front and rear wheel road inputs:

$\delta t = (a + b)/V.$

3. Active fuzzy control design for suspensions system

The proposed algorithm was applied to design a half-car fuzzy control-based active suspension system as a test bed. The closed loop model of the fuzzy logic controller for the vehicle suspension system is presented in Fig.2. In an active suspension system design exercise, the main objective is to improve the ride performance while keeping a good tire to road contact, which is essential for car handling and safety.

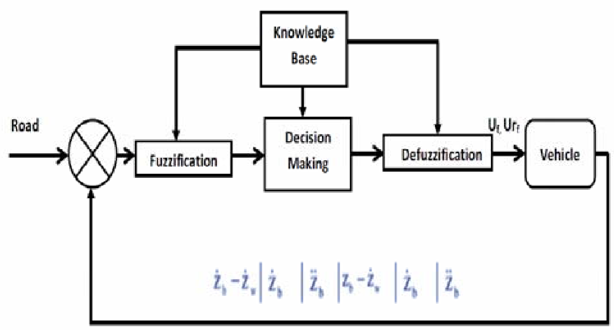


Fig.2 closed loop model of the fuzzy logic controller for the vehicle suspension system

The vertical body acceleration produced on the seats is the most common and meaningful measure of ride performance [7]. The vehicles have a longitudinal distance between the front and rear axles and they are multi-input systems that respond to the vertical bounce and the pitch motions.

Hence, in addition to the vertical body acceleration which is included in a quarter-car model, a half-car model includes also an angular movement, called pitch, around its lateral axis.
In this study the Mamdani method has been used. The algorithm of the MIMO fuzzy logic controller for the vehicle suspension system has six inputs: [body acceleration $\ddot{z}_s$, body velocity $\dot{z}_s$, body deflection velocity $\dot{z}_s - \dot{z}_u$ and one output: desired actuator force f] for both of front and rear. The control system itself consists of three stages: fuzzification, fuzzy inference machine and defuzzification. The fuzzification stage converts real-number (crisp) input values into fuzzy values while the fuzzy inference machine processes the input data and computes the controller outputs in cope with the rule base and data base. These outputs, which are fuzzy values, are converted into real-numbers by the defuzzification stage. A possible choice of the membership functions for the four mentioned variables of the active suspension system represented by a fuzzy set is as follows: for body deflection velocity $\dot{z}_s - \dot{z}_u$ .The triangular membership functions have been used as shown in Fig. 3. The input and output relations in form of Cartesian rule surfaces are presented in Fig. 4.

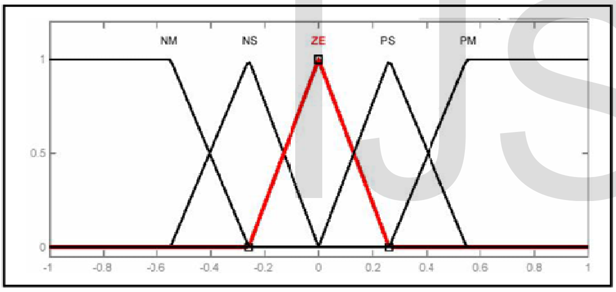


Fig. 3. (a) Membership function for body deflection velocity $\dot{z}_s$

for body velocity

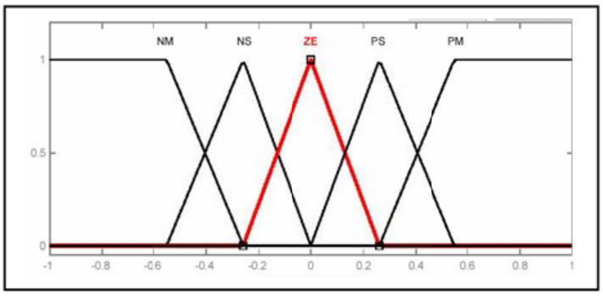


Fig. 3. (b): Membership function for body velocity

for body acceleration $\ddot{z}_s$

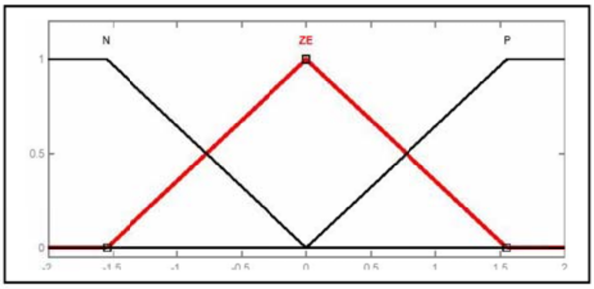


Fig. 3(c).: Membership function for body acceleration

for desired actuator force f

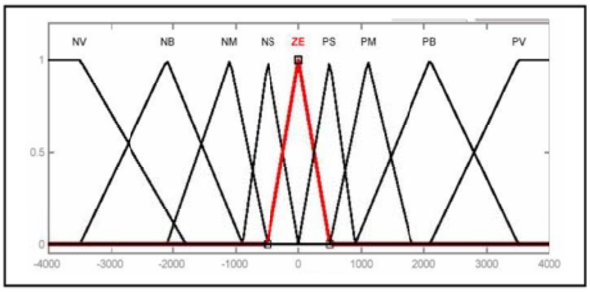


Fig. 3(d).: Membership function for desired actuator force

The abbreviations used correspond to:

NV is Negative Very Big, **NB** is Negative Big, **NM** is Negative Medium, **NS** is Negative Small, **ZE** is Zero, **PS** is Positive Small, **PM** is Positive Medium, **PB** is Positive Big and **PV** is Positive Very Big.

The example of rule base used in the active suspension system showed in Table 1 with fuzzy terms derived by the designer's knowledge and experience. The table consists of two parts; the left part has for controlling the front vehicle suspensions and the right part for the rear. We considered coupling between the two parts is as disturbance for challenge the proposed controller.

Table.1 example of rule base.

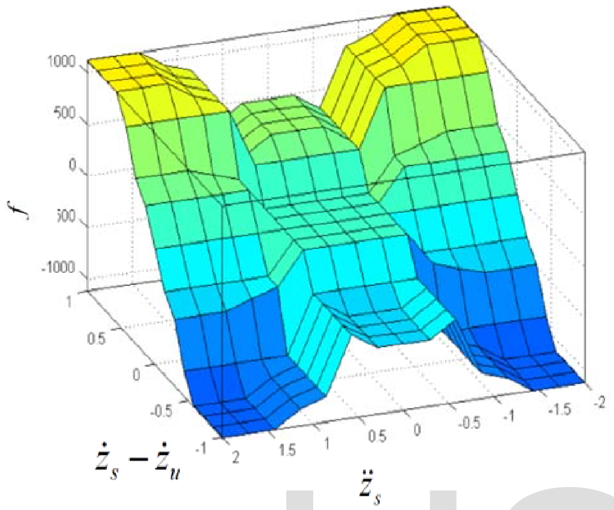
$\dot{z}_{s1}-\dot{z}_{u1}$	$\dot{z}_{s1}$	$\ddot{z}_{s1}$	$\dot{z}_{s2}-\dot{z}_{u2}$	$\dot{z}_{s2}$	$\ddot{z}_{s2}$	f_f	f_r
PS	NS	ZE	PS	NM	ZE	PM	PS

R:
IF ($\dot{z}_{s1} - \dot{z}_{u1}$ **=PS)and(** $\dot{z}_{s1}$ **=NS)and(** $\ddot{z}_{s1}$ **=ZE)and (** $\dot{z}_{s2} - \dot{z}_{u2}$ **=PS)and(** $\dot{z}_{s2}$ **=NM)and (** $\ddot{z}_{s2}$ **=ZE) THEN (**f_f**=PM), and(**f_r**=PS).**

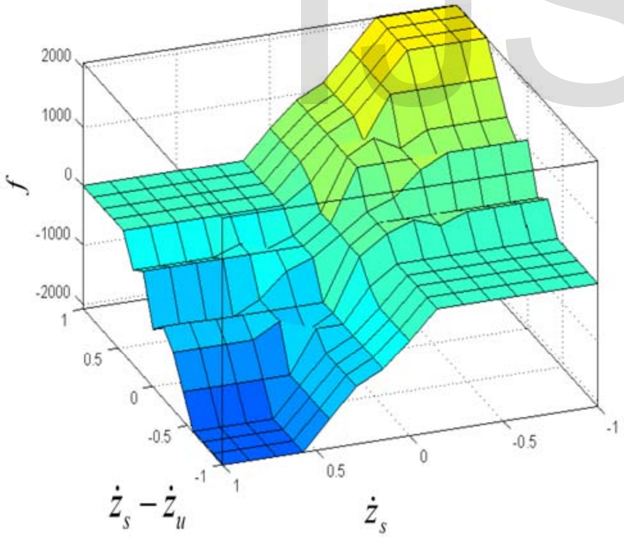
The output of the fuzzy controller is a fuzzy set of control. As a process usually requires a non-fuzzy value of control, a method of defuzzification called "center of gravity method"(COG), is used here [22]:

$$f = \frac{\int_F f * \mu_D(f).df}{\int_F \mu_D(f).df} \quad (4)$$

where $\mu_D(f)$ is corresponding membership function. The actuator force (f) is chosen to give $\pm 6\ kN$ as a maximum and minimum values, [23].



(a)



(b)

Fig. 4.Input and output relations in the form of Cartesian rule surfaces.

4. Results

In the conventional control approach, the aim is to follow zero displacement as a reference value for the body bounce motion. However, when this is realized, it is observed that the suspension working limits degenerate in order to compensate for the difference in the elevation of the vehicle body, in addition to which the amplitude of the road surface displacement changes. This causes the suspensions to malfunction after sometime, and to go out of order, resulting in reduced degrees of freedom and a harsh ride. In order to overcome this practical difficulty a new FLC approach is proposed.

The simulation results has been carried out using the mathematical model of the system using two types of controllers PID and Fuzzy control are compared for two kinds of road conditions, namely smooth and random road excitations as shown in Figs. 5 (a and b).Fig. 5.c shows the road input power spectral density. It provides information on the frequency range at which the majority of the output occurs. It clears the majority of the response occurs at around 1Hz.

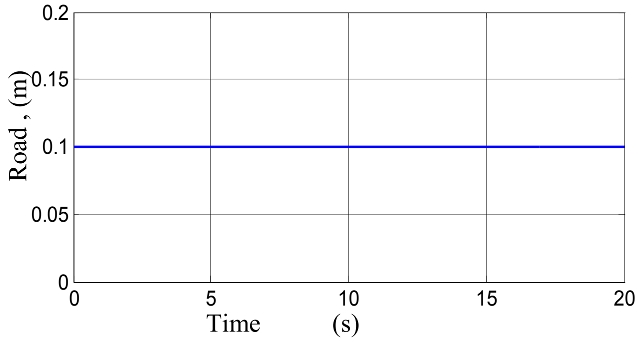
It must be considered that there are two road inputs to the system which are applicable to the front wheel and, with a time delay δt, to the rear wheel. The selection of the road input is very important to check the suspension working space loss problem. If the classical control algorithm were applied, aiming at zero displacement of the vehicle body, there would be a shortening in suspension length in order to realize zero absolute $\dot{z}_s$ reference change in vehicle body elevation.

The half car model parameters are shown in table 1 and the simulation results are shown in Figs.6 (a-e) shows comparison between the suspension working space, the body acceleration (for front, rear and center) and dynamic tire load for smooth road showing good results in ride comfort and handling characteristics for fuzzy control suspension policy. It can clear that there is an improvement in ride comfort performance.

Table 1. The mean values of vehicle system parameters

parameters	values	parameters	values
m_s	1800 kg	k_{s1}	66000 N/m
I_s	3400 kgm^2	k_{s2}	1800 N/m
m_{u1}	90 kg	c_{s1}	1200 Ns/m
m_{u2}	150 kg	c_{s2}	1000 Ns/m
I_1	1.27	k_{t1}	100000 N/m
I_2	1.73	k_{t1}	100000 N/m

Figure 6.(f)represents the controller signal for both of PID and Fuzzy controllers actions. It is apparent that, the signal amplitude of fuzzy control is less than the PID signal for the same road. Simulation results of active suspension controlled by PID and Fuzzy control are compared in Fig. 7(a-f) for real road roughness, it illustrates the comparison between the body acceleration, suspension working space and dynamic tire load results respectively. It can be noticed that the fuzzy logic control provided better results than PID. Also, Fig.7 assures the results of Fig.6 which will be effective in designing smaller actuator size when we use fuzzy control as an implemented controller in the suspension system.



(a)

Fig.5a. smooth road Surface input

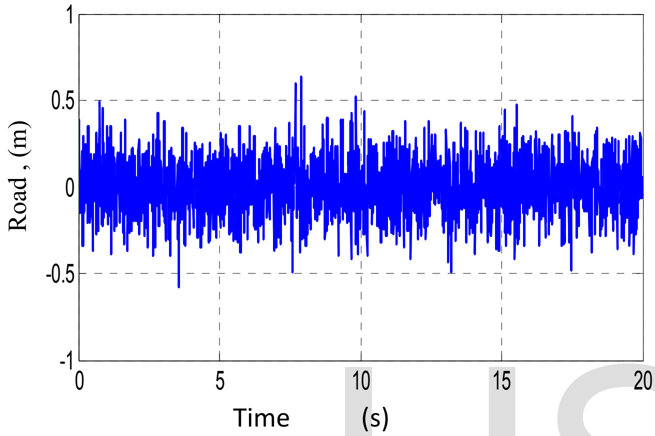


Fig.5 b. random road Surface input

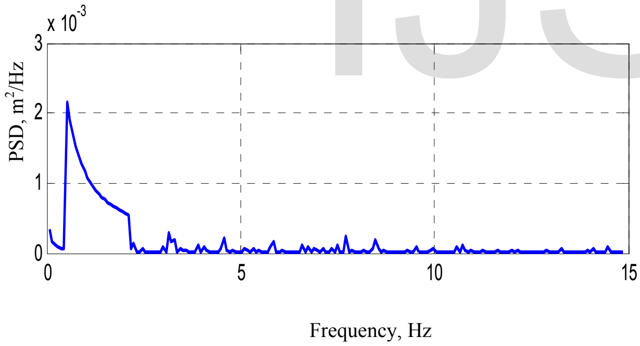
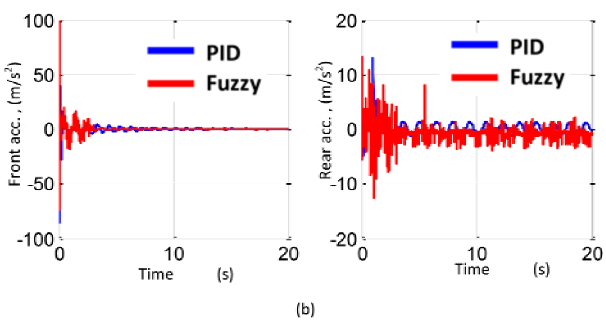


Fig. 5.cthe road power spectral density.



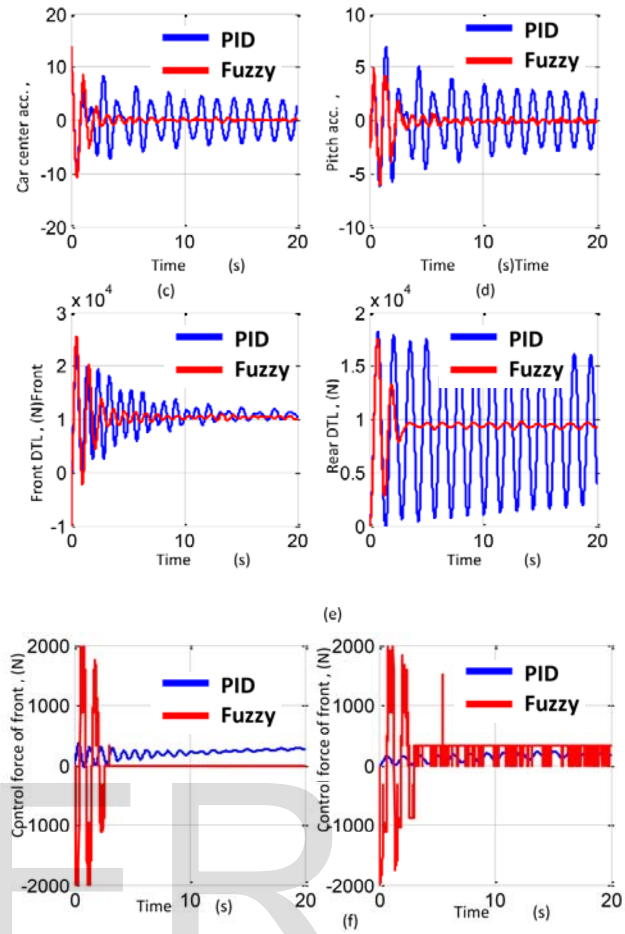
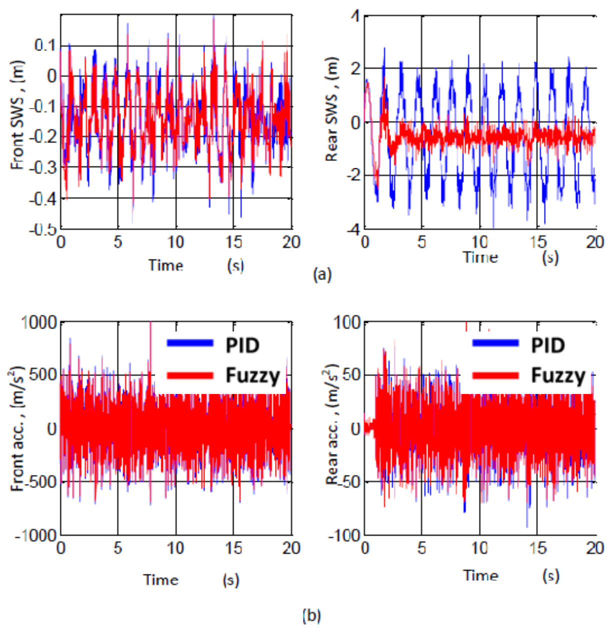


Fig.6 Time response of the vehicle body motions with soft road based on PID and fuzzy controllers; (a) SWS for front and rear; (b) acceleration for front and rear; (c) car center acceleration; (d) pitch acceleration; (e) DTL for front and rear; (f) control force for front and rear.



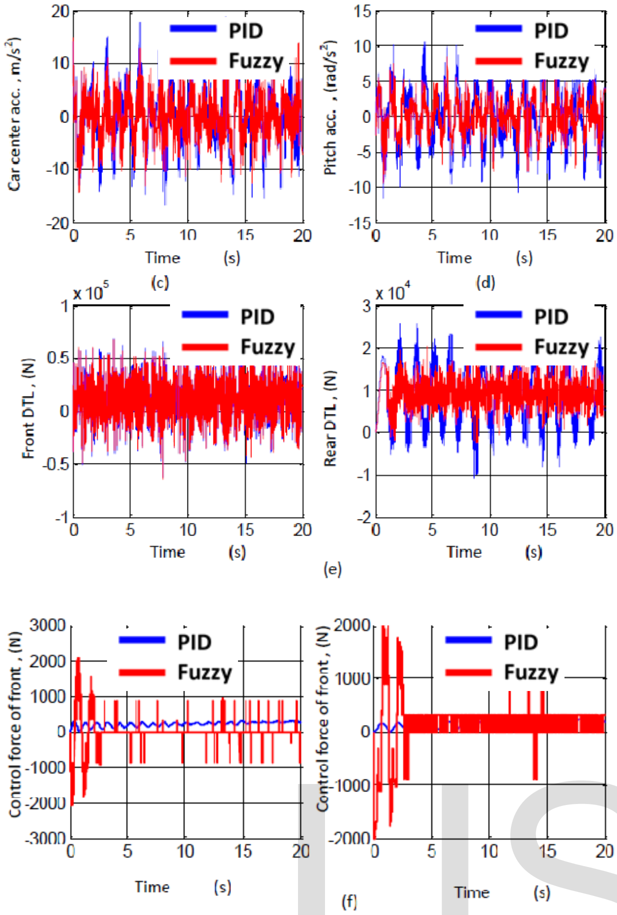


Fig.7 Time response of the vehicle body motions with soft road based on PID and fuzzy controllers; (a) SWS for front and rear; (b) acceleration for front and rear; (c) car center acceleration; (d) pitch acceleration; (e) DTL for front and rear; (f) control force for front and rear.

5. Conclusions

In this study, two fuzzy logic controllers for a MIMO system have been designed and the results have been presented. The suggested control approach does not cause any degeneration in suspension working limits, while it improves ride comfort as was the objective of this study. The fuzzy controlled vehicle body and pitch motion reach zero reference values much faster and are much more comfortable than the PID controlled ones.

The results prove that the proposed active suspension system is very effective in vibration isolation of the vehicle body, which indicates that the proposed controller proves to be effective in the stability improvement of the suspension system.

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