Fuzzy Logic Control for Vehicle Suspension Systems

Chuanyin Tang[1], Lijie Yue[2], Lixin Guo[1], Shuwen Zhou[1], Wei Zhou[3],
and Zhaowen Wang[4]

[1] School of Mechanical Engineering and Automation, North Eastern University,
Shenyang, China
[2] School of Material Science and Engineering, Shandong University of Science and
Technology, Qingdao,China
[3] China Ministry of Communications Research Institute of Highway, Beingjing, China
[4] School of Materials and Metallurgy ,North Eastern University, Shenyang, China
monsigor@yahoo.com.cn

Abstract. An active suspension system for vehicles using fuzzy logic controls is presented in this paper. The model is described by a linear system with six degrees of freedom, subject to irregular excitations from the road surface. Based on control theory, the fuzzy control system of the active suspension is proposed. With the aid of software Matlab/simulink, a lot of simulation process is done. Simulation results indicate that the proposed active suspension system proves to be effective in the vibration isolation of the suspension system. What's more, a mechanical dynamic animation of six-degree-of -freedom half body of vehicle suspension system is obtained, with the aid of software ADAMS and media player. The animation demonstrates the mechanism principle of suspension system more vividly and visually.

Keywords: suspension system; fuzzy control; Matlab/simulink; simulation; animation.

1 Introduction

The major purpose of suspension system is to provide passenger with ride comfort while maintaining the vehicle to be safe and stable .Passive suspension system can only offer a compromise between these two conflicting criteria by providing spring and damping coefficients with fixed rates. Active suspension control system can produce corresponding control force according to running status and road oscillations to maintain the suspension system to be the optimal status.

A variety of research projects and publications deal with different types of active suspension systems have been discussed in [1], [2] and [3], .Different vehicle dynamic models have been adopted according to different study purposes during research. A two degree-of -freedom quarter body of vehicle suspension system model has been widely applied in vehicle suspension control research, while it can only indicate the vehicle body vertical movement ,not include the pitching movement of the vehicle body. Although a seven degree-of -freedom whole-body of vehicle

C. Xiong et al. (Eds.): ICIRA 2008, Part II, LNAI 5315, pp. 197–206, 2008.

suspension system model can describe not only the vertical movement of the four wheels and the body center of gravity , but also present the pitching and lateral movement, the model is too complicated to be widely applied [4]. A four degree-of -freedom half body of vehicle suspension system model can describe both the vertical movement of the body center of gravity ,and the pitching movement, which got widely used in suspension dynamic model. In our research, a six-degree-of -freedom half body of vehicle suspension system model is obtained, which has the advantages of six degree-of -freedom model, what's more, it can demonstrate the effect of seat and passenger, which makes it to be an ideal model for suspension dynamic description.

The purpose of this paper is to propose an active suspension system for vehicles, using the fuzzy-logic controller. The model is described by a linear system, subject to irregular excitation from the road surface. In particular, a mechanical dynamic animation of six degree-of -freedom half body of vehicle suspension system is obtained, with the aid of software ADAMS and media player. Finally, some concluding remarks are given.

2 Modelling

2.1 Six Degree-of -Freedom Model

The model of half-body system is shown in Figure1. The half-body suspension system is represented as a linearized six degree-of-freedom system. It consists of a single sprung mass connected to two unsprung masses ,the seat mass and the passenger mass. The sprung is free to heave and pitch while the unsprung masses and the seat and the passenger mass are free to bounce vertically with respect to the sprung mass. The suspensions between the sprung and unsprung masses are modeled as linear viscous dampers and spring elements, while the tires are modeled as simple linear springs without damping. For simplicity, all pitch angles are assumed to be small.

After applying a force- balance analysis to the model in Fig.1.,the equations of motion are given as

$$
\begin{gathered}
m_1\ddot{z}_1 = k_{21}(z_1' - z_1) + c_1(\dot{z}_1' - \dot{z}_1) - k_{11}(z_1 - r_1) - u_1 \\
m_2\ddot{z}_2 = k_{22}(z_2' - z_2) + c_2(\dot{z}_2' - \dot{z}_2) - k_{12}(z_2 - r_2) - u_2 \\
m_p\ddot{z}_p = -k_p\left(z_p - z_c\right) - c_p\left(\dot{z}_p - \dot{z}_c\right) \\
m_c\ddot{z}_c = k_p\left(z_p - z_c\right) + c_p\left(\dot{z}_p - \dot{z}_c\right) - k_c\left(z_c - z_3 + \theta e\right) - c_c\left(\dot{z}_c - \dot{z}_3 + \theta e\right) \\
m_3\ddot{z}_3 = -k_{21}(z_1' - z_1) - c_1(\dot{z}_1' - \dot{z}_1) - k_{22}(z_2' - z_2) - c_2(\dot{z}_2' - \dot{z}_2) + k_c\left[z_c - \left(z_3 - \theta e\right)\right] \\
+ c_c\left[\dot{z}_c - \left(\dot{z}_3 - \dot{\theta} e\right)\right] + u_1 + u_2 \\
J\ddot{\theta} = a[k_{21}(z_1' - z_1) + c_1(\dot{z}_1' - \dot{z}_1) - u_1] - b[k_{22}(z_2' - z_2) + c_2(\dot{z}_2' - \dot{z}_2) - u_2] \\
- e\left\{k_c\left[z_c - \left(z_3 - \theta e\right)\right] + c_c\left[\dot{z}_c - \left(\dot{z}_3 - \dot{\theta} e\right)\right]\right\}
\end{gathered}
\tag{1}
$$

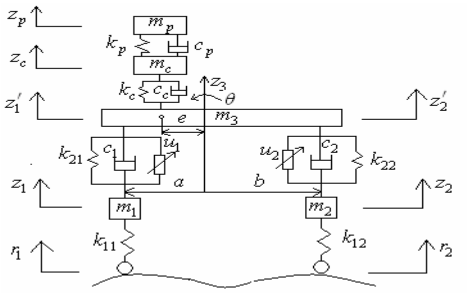


Fig. 1. Model of half body six degree-of -freedom of suspension

The system states are assigned as

$x_1 = z_1$ front wheel unsprung mass height
$x_2 = z_2$ rear wheel unsprung mass height
$x_3 = z_3$ heave position
$x_4 = \theta$ pitch angle
$x_5 = \dot{z}_1$ front wheel unsprung mass velocity
$x_6 = \dot{z}_2$ rear wheel unsprung mass velocity
$x_7 = \dot{z}_3$ heave velocity
$x_8 = \dot{\theta}$ pitch angular velocity
$x_9 = z_c$ seat heave height
$x_{10} = \dot{z}_c$ seat heave velocity
$x_{11} = z_p$ passenger heave height
$x_{12} = \dot{z}_p$ passenger heave velocity

This results in the system state equations below:

$$\mathrm{X} = (x_{1,} x_{2,} x_3 \ldots\ldots x_{12})^T \tag{2}$$

And

$$\begin{aligned} &x_1 = z_1, x_2 = z_2, x_3 = z_3, x_4 = \theta, x_5 = \dot{z}_1, x_6 = \dot{z}_2 \\ &x_7 = \dot{z}_3, x_8 = \dot{\theta}, x_9 = z_c, x_{10} = \dot{z}_c, x_{11} = z_p, x_{12} = \dot{z}_p \end{aligned} \tag{3}$$

The state space equations in matrix are given by

$$\dot{X} = AX + BR + CU \tag{4}$$

With the control input defined as the force generated at the front, rear suspensions respectively as $U = \begin{bmatrix} u_1 & u_2 \end{bmatrix}^T$, and the disturbance input defined as $R = \begin{bmatrix} r_1 & r_2 \end{bmatrix}^T$. Signals z_1, z_2, z_3, z_c and z_p are the heave height at the front wheel , rear wheel ,vehicle body, seat and passenger ,respectively.

The model parameters selected for this study are given in Table 1.

Table 1. Parameters of system model

model parameters	value	model parameters	value
font wheel mass **m_1/kg**	40	rear wheel mass**m_2/kg**	36
Sprung mass **m_3/kg**	730	front tire spring stiffness **k_{11}/ N/m/** N/m	17500
rear tire spring stiffness **k_{12}/ N/m**	17500	front suspension spring stiffness **k_{21}/ N/m**	19960
rear suspension spring stiffness **k_{22}/ N/m**	17500	front suspension damping c_1 /N.s/m	1290
rear suspension damping c_2/ N.s/m	1290	pitch axis moment of inertia/ kg.m^2	1230
Passenger mass **m_p/ kg**	60	Seat spring equivalent stiffness k_c / N/m	2400
seat mass **m_c /kg**		Cushion equivalent damping c_p N.s/m	152.8
seat equivalent damping c_c N.s/m	875	Cushion equivalent stiffness k_p / N/m	8228.7
distance between front of vehicle and center of gravity of sprung mass a/ m			1.0
distance between rear of vehicle and center of gravity of sprung mass b/ m			1.8
distance between the passenger-seat system action point and the center of gravity of vehicle body **e**/ m			0.1

2.2 Road Input Modelling

The road irregularity formula is defined as

$$\dot{x} = -2\pi f_0 x + 2\pi\sqrt{G_0 v}\omega(t) \tag{5}$$

Using Grade B road surface, which is the common road ,gives the irregular road coefficients as $G_0 = 64\times10^{-6}\ \mathrm{m}^2/\mathrm{m}^{-1}$ and velocity v is $20\mathrm{m/s}$.

3 Control System Design

The fuzzy logic control is one of the most attractive parts where fuzzy theory can be effectively applied. The fuzzy logic translates the mathematical control strategy into the linguistic control strategy. The fuzzy logic controller is usually based on the operator's knowledge, fuzzy modelling of the operator's control actions and fuzzy modelling of the process.

The fuzzy logic controller consists of three steps. The first step is the fuzzification, the second step is the reasoning using the fuzzy rule base, the last step is the defuzzification.

In the paper, the fuzzy logic controller is used to adjust the vehicle's body acceleration errors[5].

Figure 2 shows the block diagram of the vehicle body acceleration used. The error block is the difference between the actual body acceleration and the reference body acceleration.The fuzzy logic controller is designed to reduce the acceleration errors.

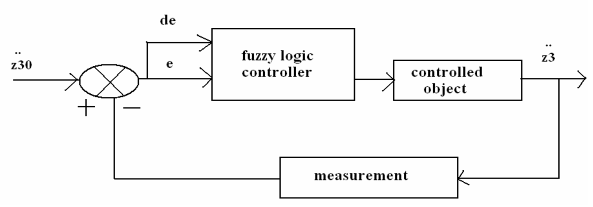


Fig. 2. Block diagram of the acceleration control system

3.1 Fuzzification

The first step in making a fuzzy logic controller is to take the inputs and determine the degree to which the inputs belong to each of the appropriate fuzzy sets via fuzzy membership function. The input is always a numerical value and must be fuzzified. The fuzzification of the input becomes a membership function to be evaluated. The fuzzy membership function can be described as follows:

$$K=\left\{\left(x,u_k\left(x\right)\right)\ \left|x\in K,u_k\left(x\right)\in\left[0,1\right]\right.\right\} \tag{6}$$

where $u_k\left(x\right)$ is the membership function specifying the grade of degree for any element in K which belongs to the fuzzy set K . The lager values of $u_k\left(x\right)$ indicate the higher degrees of membership.

The trimf membership is selected. It has seven grades; negative big (NB), negative medium (NM) ,negative small (NS) ,zero (ZE), positive small (PS), positive medium (PM), and positive big (PB).

3.2 Fuzzy Rule Base

The Mamdani fuzzy logic type is used. The fuzzy control rules are designed by a collection of if-then rules. In general, the input to an if-then rule is the current value for the input variable and the output is an entire fuzzy set. This set will later be defuzzified, assigning one value to the output.

In the paper, the fuzzy controller controls the multi-input-single-output system and the fuzzy rule is described as follows:

$$Ri: IF\ \ x\ is\ A_i...,\ AND\ \ y\ is\ B_i...,THEN\ \ z=C_i,D_i,E_i,F_i$$

A total of seven grades are used for the control force in the paper: negative big(NB),negative medium (NM), negative small (NS),zero (ZE), positive small (PS), positive medium (PM), and positive big (PB).The fuzzy rule table shown in Table 2 is developed based on these if-then rules.

3.3 Defuzzification

The defuzzification is a mapping from a space of the fuzzy control action into a space of the non-fuzzy control action. This process is necessary because the action is required to actuate the controller in many practical applications. In most cases , the

Table 2. The rules of fuzzy control

u		*de*						
		NB	NM	NS	ZE	PS	PM	PB
e	NB	NM	NS	NS	NS	ZE	PS	PM
	NM	NM	NM	NM	NS	PS	PM	PM
	NS	NB	NM	NM	NS	PM	PB	PB
	ZE	NB	NB	NM	ZE	PM	PB	PB
	PS	NB	NB	NM	PS	PM	PM	PB
	PM	NM	NM	NS	PS	PM	PM	PM
	PB	NM	NS	ZE	PS	PS	PS	PM

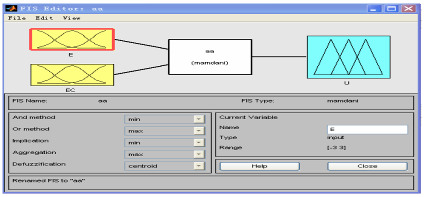
input for the defuzzification process is a fuzzy set and the output is a single number. The famous defuzzification method is the centroid calculation, which returns to the centre of the area under the curve.

In this paper, the defuzzification using the centre of area method is used and the defuzzification outputs are the controlled force. This method yields

$$Z^* = \sum_{j=1}^{n} \mu_c\left(z_j\right) z_j \Big/ \sum_{j=1}^{n} \mu_c\left(z_j\right) \tag{7}$$

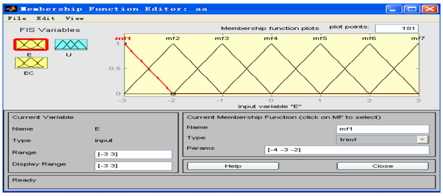
where n is the number of quantization levels of the output, z_j is the amount of control output at the quantization level j, $\mu_c\left(z_j\right)$ represents its membership value in the fuzzy output set.

The fuzzy inference system compiler and membership function complier are showed in figure 3 and figure 4.



FIS Name: aa
FIS Type: mamdani
And method: min
Or method: max
Implication: min
Aggregation: max
Defuzzification: centroid
Current Variable
Name: E
Type: input
Range: [-3 3]
Renamed FIS to "aa"

Fig. 3. Fuzzy inference system complier



plot points: 181
Current Variable
Name: E
Type: input
Range: [-3 3]
Display Range: [-3 3]
Current Membership Function (click on MF to select)
Name: mf1
Type: trimf
Params: [-4 -3 -2]
Ready

Fig. 4. Membership function complier

4 Results and Discussion

The established dynamic vehicle model and the fuzzy logic controller are used to improve the vehicle comfort using Matlab/Simulink.

Figure 5shows the corresponding Simulink block diagram of six degree-of -freedom half-body of vehicle suspension system.This Simulink block calculates the vehicle body acceleration, velocity, displacement, pitch acceleration, front and rear suspension displacement ,tire travel, seat movement and passenger movement as well. The E(acceleration error) and dE (variation acceleration error) are the input of the fuzzy logic controller. The control forces are the output of the fuzzy controller.

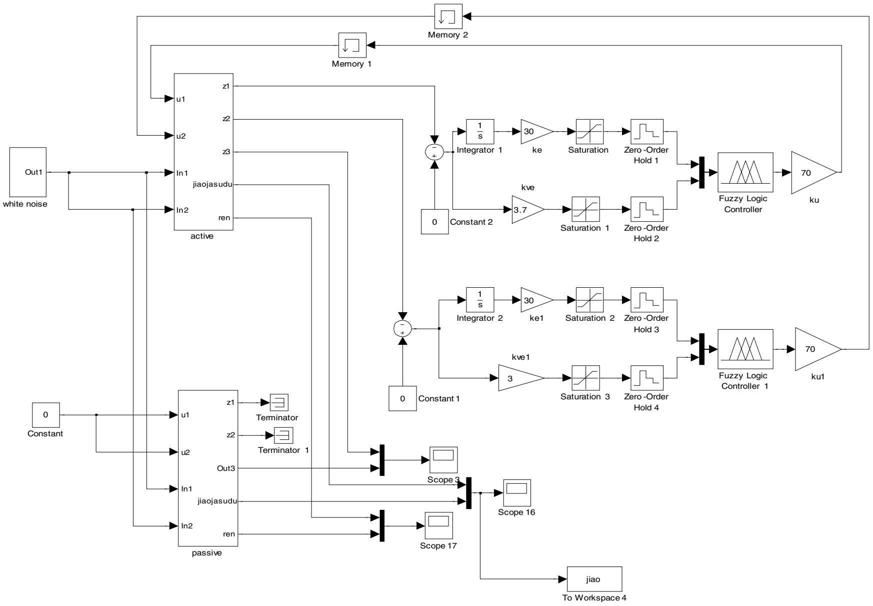
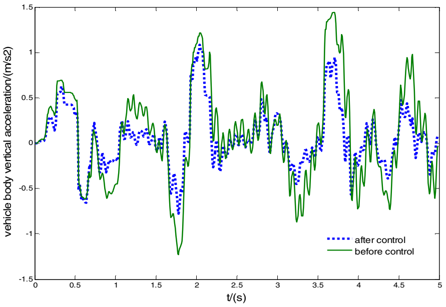


Fig. 5. Simulink block diagram of suspension system



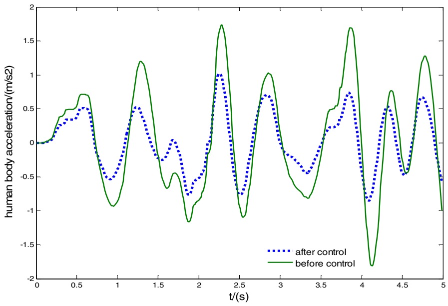


Fig. 6. Vehicle body vertical acceleration

Fig. 7. Human body vertical acceleration

Figure 6 shows the controlled body acceleration response and the passive system body acceleration response. For compare, they are out in one figure. The solid line indicates the passive system, the dash line presents the controlled system. Figure 7 shows the controlled acceleration response and the passive system acceleration response of passenger. Figure 8. shows the controlled pitching angular acceleration response and the passive system pitch angular acceleration response. Figure 9and Figure 10 show the front suspension displacement and rear suspension displacement response, respectively. Figure 11and Figure 12 show the front tire dynamic load and rear tire dynamic load response, respectively.

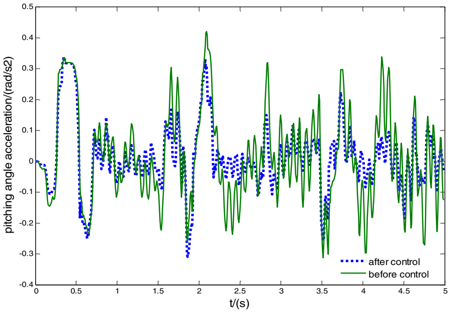


Fig. 8. Pitching angular acceleration

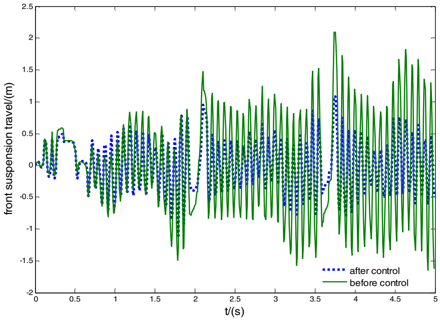


Fig. 9. Front suspension displacement

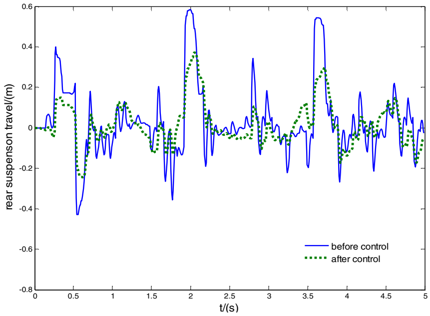


Fig. 10. Rear suspension displacement

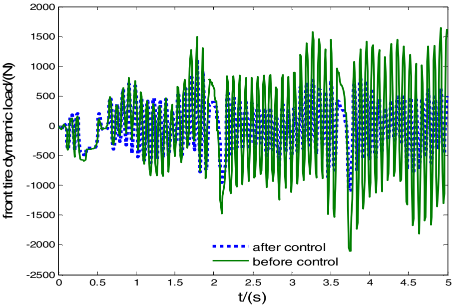


Fig. 11. Front tire dynamic load

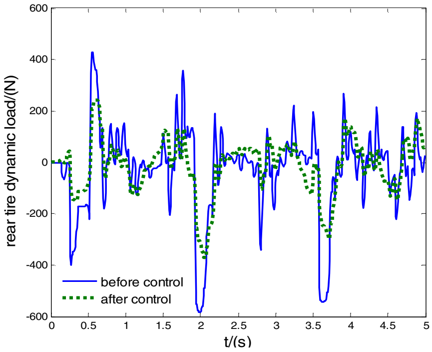


Fig. 12. Rear tire dynamic load

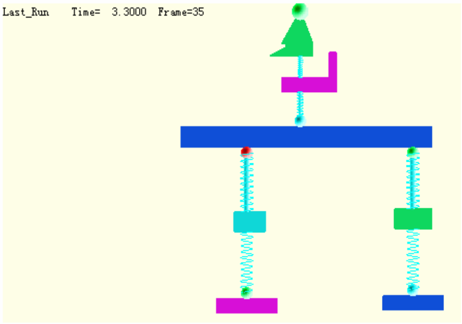


Fig. 13. Mechanical dynamic animation

Table 3 includes the performance analysis of the system. The body acceleration and pitching angular acceleration is greatly decreased, and the passenger acceleration is reduced dramatically ,which indicates that the fuzzy controller is effective in improving the system riding comfortability [6]. The compare of tire dynamic loads show that the system holding ability is meliorated.

Table 3. Performance analysis

	Body acceleration (m/ s^2)	Pitching angular acceleration (rad/s^2)	Passenger Acceleration (m/ s^2)	Front suspension displacement (m)
passive	0.4722	0.1760	0.2205	0.7503
controller	0.2679	0.1212	0.1126	0.3947
Performance melioration	43.3%	31.1%	49%	47.4%
	Rear suspension displacement (m)	Front tire dynamic load (N)	Rear tire dynamic load (N)	
passive	0.1756	750.36	175.18	
controller	0.1124	394.7	111.98	
Performance melioration	35.9%	47.9%	36.6%	

5 Animation Demonstration

First, using the software ADAMS ,the mechanical model of six degree-of-freedom half body suspension system is established, then with the aid of software media player ,a mechanical dynamic animation of six degree-of -freedom half body of vehicle suspension system is obtained, as demonstrated in figure 13. The animation illustrates the mechanism principle of suspension system more vividly and visually.

6 Conclusions

In this paper an active suspension control approach combining a fuzzy control scheme was proposed for a six degree-of-freedom half body suspension system in order to reduce heave, pitch acceleration and improve ride comfortability of the vehicle. The percentage improvement for the ride comfort compared with a passive suspension averages around 40%. It was shown that the proposed fuzzy controller proves to be effective in the vibration isolation of the suspension system. Performance of the suspension system was demonstrated in simulations.

Acknowledgement

Thanks for the fund supported by National Basic Research Program of China(973 Program)(Number: 2007CB210305) & Western Communications Construction Project ,Research Institute of Highway (Number:2005031822322). And thanks for Northeastern university for providing us with the resources.

References

1. Lin, Y.J., Lu, T.Q., Padovan, J.: Fuzzy Logic Control of Vehicle Suspension Systems. Int. J. Vehicle Design 14, 457–470 (1993)
2. Liu, H., Nonami, K., Hagiwara, T.: Active Following Fuzzy Output Feedback Sliding Mode Control of Real-Vehicle Semi-active Suspensions. Journal of Sound and Vibration 314, 39–52 (2008)

3. Sung, K.G., Han, Y.M., Cho, J.W., Choi, S.B.: Vibration Control of Vehicle ER Suspension System Using Fuzzy Moving Sliding Mode Controller. Journal of Sound and Vibration 311, 1004–1019 (2008)
4. Chen, S.A., Liu, H.G., Lu, S.L., He, R., et al.: Fuzzy Control System for Automobile Active Suspension with Four Dimensions. Automobiles Engineering 23, 375–380 (2001)
5. Chen yang.:Active Suspension System of Automobile Based on Fuzzy Control, Northeastern University (2007)
6. Yu, Z.s.: The Theory of Automobiles. The Press of Industrial Mechanics (2000)