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Signal frequency based self-tuning fuzzy controller for semi-active suspension system*

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Abstract: A new kind of fuzzy control scheme, based on the identification of the signal's main frequency and the behavior of the ER damper, is proposed to control the semi-active suspension system. This method adjusts the fuzzy controller to achieve the best isolation effect by analyzing the main frequency's characters and inspecting the change of system parameters. The input of the fuzzy controller is the main frequency and the optimal damping ratio is the output. Simulation results indicated that the proposed control method is very effective in isolating the vibration.

Key words: Semi-active suspension, ER damper, Frequency identification, Fuzzy controller
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INTRODUCTION

Road vehicle suspension design has received great attention in recent years. The performance of the suspension system plays an important role in achieving good handling and riding comfort. Practically, the simplest and most common types of suspensions are passive in the sense that no extra energy is required. But further development is held down because the spring and damper elements cannot be dynamically adjusted.

Active suspensions with their spring and damper elements replaced by active force actuators can remove the inherent restrictions of passive suspensions and offer better riding performance. But due to manufacturability, reliability and economy concerns, it appears that only a few types of active suspension systems have been realized so far. Semi-active suspension plays an important role gradually because it is more agile than passive suspension and cheaper than active suspension. With the development of electrorheological (ER) fluid in recent years, the ER damper replaces the hydraulic shock absorber or the adjustable valve and becomes the executive element of the suspension system (Gavin *et al*., 1996a; 1996b). The ER damper can change its damping coefficient rapidly when the voltage changes, so it shortens the control time and increases the reliability of the control system greatly.

Most control strategies for semi-active suspension systems are based on optimal control algorithms. Besides this, the application of adaptive control such as model reference adaptive control and nonlinear self-turning control for vehicle suspension systems were investigated by Sunwoo *et al*. (1990a; 1990b). Generally speaking, these controllers can minimize a defined performance index but do not have good capability for adapting to significant changes of the road and system parameters. Fuzzy inference, one of the knowledge-based approaches, has been recently applied to the design of semi-active suspensions of such complicated systems to achieve improved performance because it is easy to construct the suspensions without considering the nonlinearity and uncertainty. Lin *et al*. (1993) built a model following fuzzy controller of vehicle suspension systems. He used the error and change in error between the reference model acceleration and that of the active suspension as

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the two input variables in the fuzzy inference mechanism. Yoshimura *et al*. (2000) used the displacements and velocity of the sprung mass and unsprung mass as the input of the fuzzy controller, and damper force as the output, to control the system by 49 fuzzy rules. Foda (2000) and Rao *et al*. (1997) took the relative displacement and velocity as the input of the fuzzy controller, and active force as the output. All these methods have provided improved performance to the semi-active suspension system. If we think more about the road input of the semi-active suspension system, we can achieve better performance.

This work aimed to develop a new fuzzy logic controller that can be used for semi-active suspension system that is based on the identification of the road input signal's main frequency. Based on the behavior of the ER damper, this method adjusts the fuzzy controller to achieve the best isolation effect by analyzing the main frequency's characters and inspecting the change of system parameters. The input of the fuzzy controller is the main frequency and the optimal damping ratio is the output. Simulation and experimental results indicated that the proposed control method is very effective in isolating vibration.

DYNAMIC MODEL OF SEMI-ACTIVE SUSPENSION SYSTEM

A 2 DOF quarter vehicle suspension is employed (Fig. 1) and modeled by the linear spring coefficient k_1, tire spring coefficient k_2, and damping coefficient c. The active model consists of the sprung mass m_1, the unsprung mass m_2, and displacements for the sprung mass and unsprung mass, z_1 and z_2, respectively. And r is the road disturbance. The system dynamics is described by:

$$\begin{aligned} m_1\ddot{z}_1 &= -k_1(z_1 - z_2) - c(\dot{z}_1 - \dot{z}_2) \\ m_2\ddot{z}_2 &= k_1(z_1 - z_2) + c(\dot{z}_1 - \dot{z}_2) - k_2(z_2 - r) \end{aligned} \tag{1}$$

Here c can be changed with the voltage of the ER damper.

From Eq. (1), we can get the sprung mass' frequency response in Fig. 2. Here f_0 is the resonant frequency and f_c is the intercross frequency. For the road input signal with single main frequency, we can easily choose the best damping ratio ξ_{OPT} to achieve the best isolation effect. When the main frequency is less than f_c, we can use the maximal damping ratio ξ_{MAX}; and ξ_{MIN} is used when the main frequency is greater than f_c. But according to the complex input signal, we could not use the above methods to achieve best control effects because the signal has different main frequencies of different amplitude and angle. A best damping ratio cannot be found easily because some frequencies are less than f_c, and the others are greater than it at the same time. So we develop an adjustable fuzzy controller to choose the best damping ratio ξ_{OPT} according to the complex signal's main frequencies.

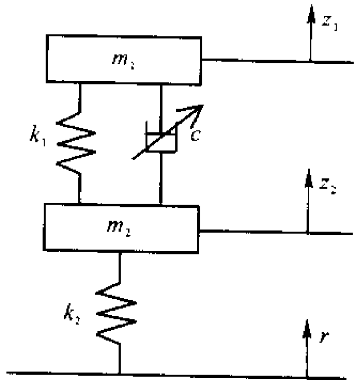

Fig. 1 The semi-active suspension system

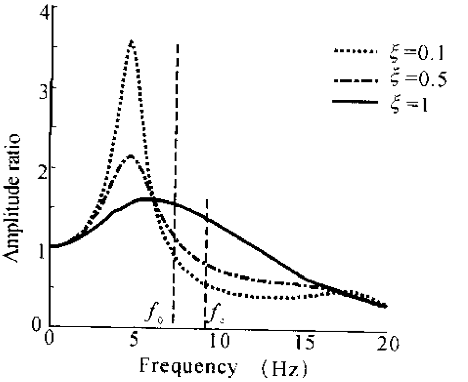

Fig. 2 Dynamic response of the system

ADJUSTABLE FUZZY CONTROL SYSTEM

Fig.3 shows the structure of the control system. It is divided into three parts: the identification of signal main frequencies, adjustable fuzzy controller and the output of control voltage. When the suspension system works, the first part will find the characters of the signal's main frequency and send them to the fuzzy controller. At the same time, the sensors will inspect the parameters of the suspension system and send them to the fuzzy controller also. Then the fuzzy controller can make decision according to the rules and choose the best damping ratio ξ_{OPT} from the optimal damping ratio table. Finally, the third part will transform ξ_{OPT} into the appropriate voltage in order to put it on the ER damper.

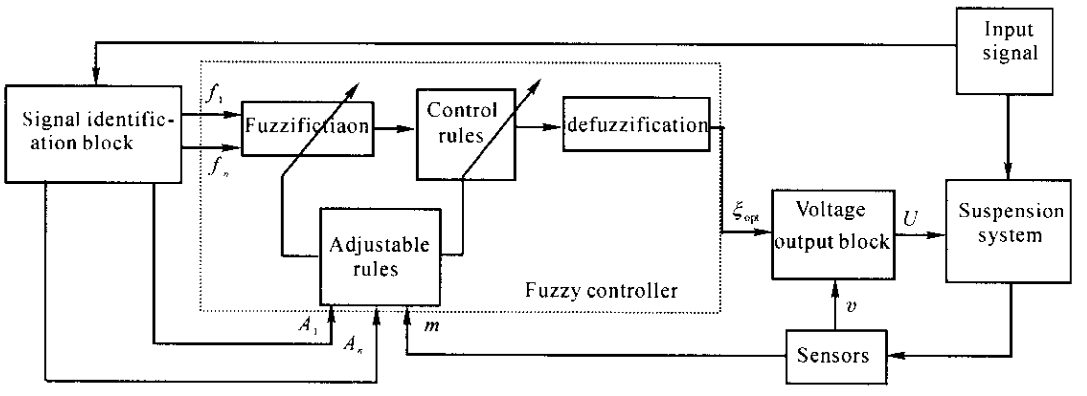

Fig.3 The block diagram of control system

1. The identification of signal main frequency

The vibratory signals in engineering often include many stable signals and concordant frequency changed signal. And they often have many main frequencies of different changing trend synchronously. There are three main characters to describe the main frequency: position, amplitude and angle. The main frequency's position indicates that the signal's power is strong. And the main frequency's angle shows the degree of its influence on the system. So how to distinguish the characters of the main frequency from the input signal, especially identify the main frequency's position, amplitude and angle, is very important for realizing excellent control of the semi-active suspension system.

Fig.4 shows how to identify the characters of the main frequency. At first, we class the signal into two groups: the stable signal and the frequency changed signal. The method to analyze

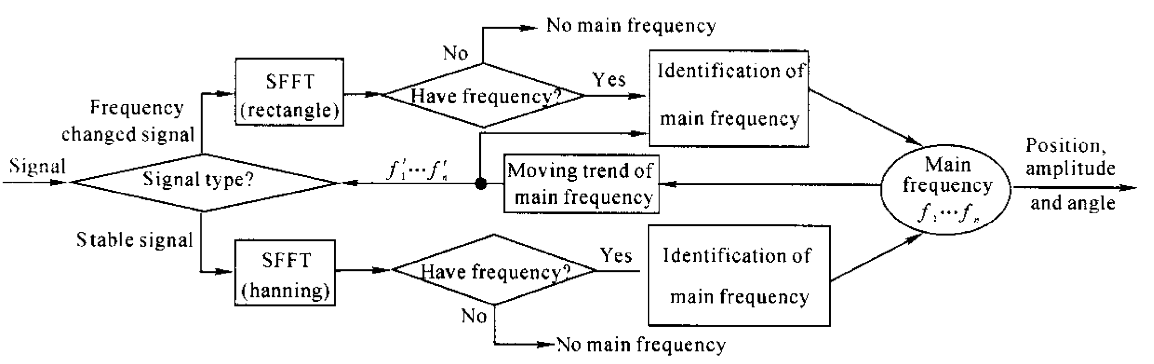

Fig.4 Identification of the signal's main frequency

the signal is Short Fast Fourier Transform Algorithm (SFFT). We use the hanning window to analyze the stable signal because of its main frequency's stable changing trend. While the rectangular window is used to handle the frequency changed signal because the rectangular window's central lobe is small, this can avoid the mistiness of the hanning window. In the process of identification, we use the main frequency's changing trend and the width of the window to search the position, amplitude and angle of the main frequency.

2. Adjustable fuzzy controller

The structure of the fuzzy logic control system for the vehicle semi-active suspension is shown by the dotted rectangle in Fig.3. The main frequencies ($f_1, f_2 \cdots f_n$), which are identified from the first part, are the input variables to the fuzzy logic controller while the optimal damping ratio ξ_{OPT} is its output. The universe of discourse for input is taken as [0 20] according to the frequency response figure (Fig. 2). And [0 1] is the output's universe of discourse because the damping ratio ξ_{OPT} is between this range in general. Triangular membership functions are chosen because they are very basic and widely used (Fig.5). The universe of discourse for the input variables is divided into five sections using the following linguistic variables, negative big (NB), negative small (NS), zero (Z), positive small (PS), positive big (PB).

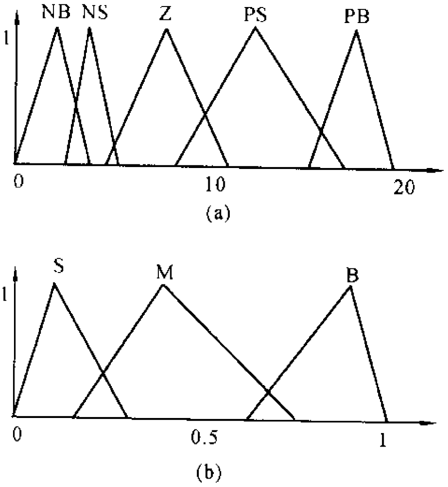

Fig.5 Membership function
(a) input; (b) output

The membership function curve is a little dense near the resonant frequency and the intercross frequency because in this region the main frequency has more vibration effect on the suspension system. The universe of discourse for the output variable is only divided into three sections because the voltage of the damper cannot be changed continuously. The defuzzification procedure employed is the centroid of area method. This method is sufficient for the controller and is one of the most popular ones available.

According to empirical knowledge and simulation results, we use the following principles to build and adjust the fuzzy control rules:

(1) If all the main frequencies are less than the intercross frequency f_c, we can use the maximal damping ratio ξ_{MAX}; and ξ_{MIN} is used when all the main frequencies are greater than f_c. This rule is easy to get from Fig.2.

(2) If the amplitude (or angle) of the main frequencies is the same size, we can adjust the fuzzy control rule with the main frequency's position. For example, if the main frequency f_1 is near the resonant frequency f_0, its weight factor will be enhanced. And the numbers of its control rule will be added in order to realize this function. When the main frequency is far away from f_1, such as in the high frequency band, its weight factor will be depressed.

(3) If the amplitude (or angle) of the main frequencies is not constant, we will adjust the fuzzy control rule according to the value of the amplitude (or angle). When the amplitude (or angle) is large, its weight factor will be enhanced because it will cause great vibration effect on the system.

(4) If the amplitude of one main frequency is too large or too small, this main frequency is regarded as invalid and we will abandon it. For example, if the main frequency f_1 is more than five times the main frequency f_2, we will abandon f_2 because compared with f_1, it is too small to cause effect.

(5) If the position of the main frequencies is near the intercross frequency f_c, the number of the membership function curve will be increased in order to increase the control precision of this region.

(6) If the sprung mass is changed, we will adjust the input membership function curve be-

cause the resonant frequency and intercross frequency change also. In practice, the sprung mass is often changed because of the different mass of the cargo. To solve this problem, we use the mass sensor to measure the real sprung mass and shift the input membership curve accordingly. Because each sprung mass has its own frequency response, this will result in a great deal of membership curves and waste much time in the control. So we class the sprung mass into several groups at first and use the same frequency response in the same kind. For example, if the sprung mass is 30 kg – 36 kg, we use the same input membership function curve.

According to the above principles we can get the fuzzy control rules. Table 1 shows the control rules in the form of the linguistic variables for two input main frequencies. And we change Table 1 into the final optimal damping ratio table in the program at first in order to save control time.

Table 1 Fuzzy rules

Output ξ		Input f_1				
		NB	NS	Z	PS	PB
Input f_2	NB	B	B	S	S	B
	NS	B	B	M	B	B
	Z	S	M	S	S	S
	PS	S	B	S	S	S
	PB	B	B	S	S	B

3. Output of the voltage

The optimal damping ratio ξ_{OPT} which we get from the adjustable fuzzy controller must be converted into the voltage of the ER damper. From experiment we have the following empirical equation that describes the relation between the damper force F and the voltage U.

$$F = av + b_2 U^2 + b_1 U + b_0 \qquad (3)$$

where a, b_0, b_1, b_2 are constants related with the structure parameters of the ER damper, v is the relative velocity of the damper's piston.

And

$$c = 2\xi\sqrt{m_1 k_1} = F/v \qquad (4)$$

From Eqs. (3) and (4), we can get the ER damper's voltage as:

$$U = \begin{cases} \dfrac{-b_1 + \sqrt{b_1^2 - 4b_2(b_0 + av - 2\xi\sqrt{m_1 k_1}\,v)}}{2b_2} & 0 \leqslant U \leqslant U_{\max} \\ U_{\max} & U_{\max} \leqslant U \end{cases} \qquad (5)$$

Where $U_{\max}$ is the highest voltage that the ER damper permitted. Eq. (5) shows the relation between the voltage U and the damping ratio ξ. The damper piston's relative velocity can be measured by the sensors. If the voltage transferred was higher than the permitted voltage $U_{\max}$, we use $U_{\max}$ to replace it in order to protect the ER damper.

SIMULATION AND RESULTS

For the quarter-car suspension system given in Section 1, the parameter values used for the simulation are: $m_1 = 48$ kg, $m_2 = 16$ kg, $k_1 = 33$ kN/m, $k_2 = 185$ kN/m, $U_{\max} = 3.5$ kV. From these parameters we can get the resonant frequency and the intercross frequency: $f_0 = 3.7$ Hz, $f_c = 5$ Hz. In the following simulation figures, the dotted line denotes the passive suspension system with damping ratio $\xi_P = 0.707$, while the dash-dotted line and the solid line denotes respectively the easy fuzzy control and the adjustable fuzzy control of the suspension system.

Fig. 6b shows the time response of the sprung mass position with the input signal in Fig. 6a. The signal is a sine wave with frequency of 3 Hz and amplitude of 1 mm before 1s and adds a sine wave with frequency of 10 Hz and amplitude of 5 mm after 1s. We can find easily that the easy fuzzy controller and the adjustable fuzzy controller have the same response line with $\xi_{OPT} = 1$ when the main frequency only was 3 Hz before 1s. And the easy fuzzy controller is better than the passive one after 1s because it adopts $\xi_{OPT} = 0.5$ through its fuzzy rules. We can find that the adjustable controller has the best effect after 1s because it takes the main frequency's amplitude into account at this time and adopt $\xi_{OPT} = 0.3$. In this way, the main frequency of 10 Hz is considered with greater emphasis because of its bigger amplitude. The result shows that the sprung mass of the passive suspension system decreases by nearly 50% when the adjustable controller is used.

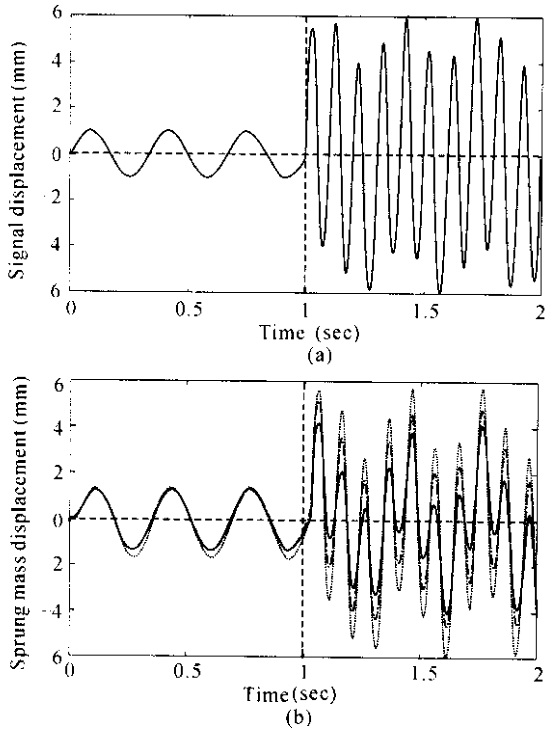

Fig.6 Response of sprung mass displacement
(a) input signal; (b) response of sprung mass
passive; easy fuzzy; adjustable fuzzy

Fig.7 and Fig.8 show the sprung mass position with the signal of two main frequencies 4 Hz and 9 Hz. The difference of the signals in Fig.7a and Fig.8a is the amplitude vibration of the two main frequencies. In Fig.7a, the amplitude of the main frequency 4 Hz is 5 times that at 9 Hz and it is opposite in Fig.8a. Apparently, the adjustable fuzzy controller has the best sprung mass position in both two cases. It takes 1 as the optimal damping ratio in Fig.7b and 0.3 in Fig.8b after using its fuzzy rules and adjustable rules because of the different amplitudes of the two main frequencies. The easy fuzzy controller just finds optimal damping ratio according to the main frequency's position and takes 0.6 as ξ_{OPT} in both two cases. Especially in Fig.7b, its effect is even worse than the passive suspension system with the damping ratio of 0.707. An important finding is that the sprung mass position in Fig.7b is higher than that in Fig.8b in general. This is because the main frequency 4 Hz is near the resonant frequency f_0, so its effect is greater than that of the main frequency 10 Hz.

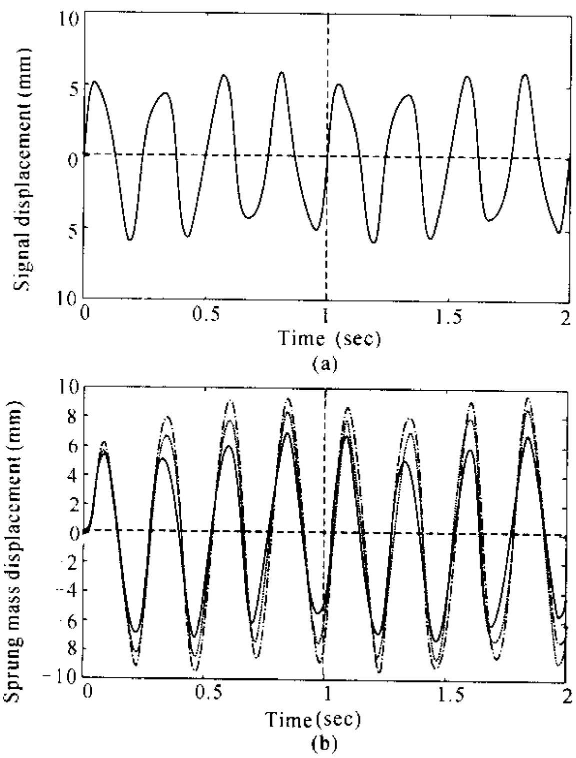

Fig.7 Response of sprung mass displacement
(a) input signal; (b) response of sprung mass
passive; easy fuzzy; adjustable fuzzy

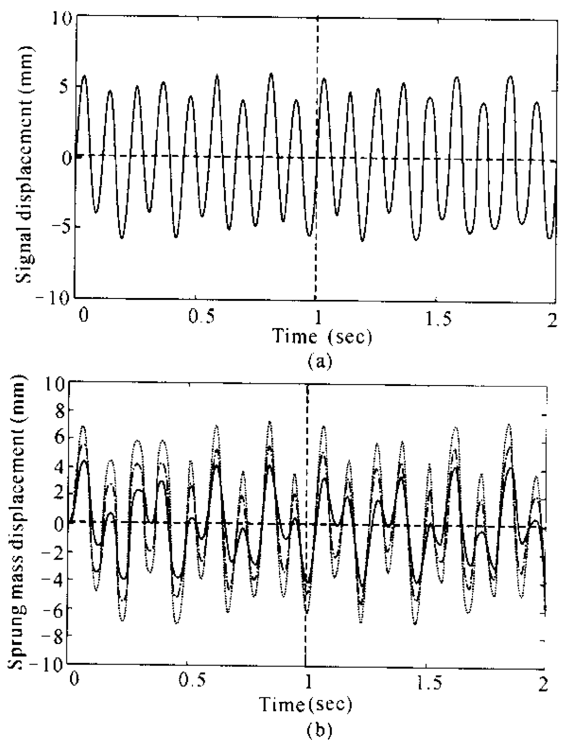

Fig.8 the response of sprung mass displacement
(a) input signal; (b) response of sprung mass
passive; easy fuzzy; adjustable fuzzy

CONCLUSIONS

An adjustable fuzzy controller for semi-active suspension system is presented. There are two most important parts of the proposed control method: one is the exact identification of the signal's main frequency; the other is the adjust rules of the fuzzy controller. In fact, this method can be used in many situations where the stimulating signal is not changed frequently, such as in the case of a submarine, whose vibration is mainly caused by the engines. The proposed adjustable fuzzy controller was found to bring down the displacement of the sprung mass in many kinds of signals. It was observed that the controller copes with the wide variations of the system's parameters, such as the change of the sprung mass. The simulation results showed that the proposed semi-active control, compared with the passive control, improved the performance better.

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