Improving ride comfort of semi-active suspension with magnetorheological damper by fuzzy-PID control

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ABSTRACT

Semi-active suspension using magnetorheological (MR) damper has become one of the popular choices for improving vehicle ride comfort due to its adjustable damping, affordable cost, and quick response. MR damper has highly nonlinear and obvious hysteresis behavior, so its control becomes complicated. Fuzzy control combined with proportional-integral-derivative (PID) controller is used to study the vibration reduction control strategy of semi-active suspension with MR damper. PID is highly robust, but the control parameters cannot be changed during the control process. Therefore, it has good adjustment ability for specific situations, but cannot effectively adjust nonlinear systems. Fuzzy control performs fuzzy inference based on input variables and fuzzy rules constructed from expert knowledge, and continuously outputs effective control parameters to adjust nonlinear systems. The advantages of both are used to construct fuzzy-PID controller to control MR damper. A quarter-car semi-active suspension with MR damper model is established using MATLAB/Simulink for simulation analysis. Compared with passive and PID, fuzzy-PID reduces the root mean square (RMS) values of body acceleration, suspension travel, and tire deflection, reduces body vibration, and improves ride comfort.

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1. INTRODUCTION

When a vehicle is driving, the road profiles excite the vehicle body to vibrate. The faster the vehicle speed or the greater the unevenness of the road surface, the more obvious the vibration and the worse the ride comfort. Currently, passive suspension, active suspension, and semi-active suspension have been developed to reduce vehicle body vibrations and improve vehicle stability [1]. Passive suspension cannot change damping and has good vibration reduction effect for specific road conditions [2]. Active suspension can adjust the suspension stiffness, damping characteristics, and vehicle height in real time according to road excitation and vehicle operating conditions, thereby improving ride comfort and handling stability. However, due to its complex structure, high manufacturing cost, considerable energy consumption, and stringent requirements for control reliability and safety, it is mainly applied in high-end passenger cars and high-performance off-road vehicles [3]. Semi-active suspension changes the damping and cannot adjust the spring stiffness and vehicle height like active suspension. It has low cost, strong flexibility, and low energy consumption, so it is widely used in automobile vibration reduction [4], [5].

At present, magnetorheological (MR) damper is a commonly used semi-active suspension actuator that quickly adjusts the state of MR fluid according to the magnetic field; that is, the liquid state is converted into a solid or semi-solid state, and the magnetic particles in the liquid are arranged regularly. When the magnetic field disappears, the MR fluid returns to the liquid state, and the arrangement of the magnetic particles becomes disordered [6], [7]. MR dampers offer several advantages, including low energy consumption, fast response, and relatively low cost, making them widely used in vibration control and reduction systems [8]–[10]. However, their mechanical response exhibits strong nonlinearity and pronounced hysteresis, making effective control of MR dampers an important issue in semi-active suspension research and a continuing focus of scholarly attention [11], [12].

Owing to its simple structure, ease of implementation, and strong robustness, the proportional-integral-derivative (PID) controller has been widely adopted in industrial control systems [13], but it is not suitable for controlling nonlinear systems. Researchers propose modified PID controller, such as fractional-order PID (FOPID) controller [14], sigmoid PID controller [15], and gray PID controller [16], to control more complex systems. Tumari *et al.* [17] used a meta-heuristic algorithm (marine predators algorithm) to determine the optimal gain parameters for the FOPID controller, which requires five optimal gain parameters. Aysal *et al*. [18] compared several controllers and metaheuristic optimization algorithms for half-car active suspension systems. The optimized PID controller showed the best overall performance. Swethamarai and Lakshmi [19] applied adaptive fuzzy fractional-order PID controller to study the vibration reduction performance control of a three-degree-of-freedom quarter active suspension and achieves good control results. The optimal control parameters obtained by these improved PID controllers and meta-heuristic algorithms can improve the control accuracy, but they increase the complexity of the control calculation and the solution time, and obtain local optimal solutions.

Automobile suspension control requires fast real-time response. Although some complex control methods can achieve high accuracy, their computational burden limits real-time performance. Fuzzy control uses expert knowledge for rapid inference, while PID control has a simple and practical structure. Therefore, combining fuzzy control with PID control can improve the real-time vibration reduction performance of semi-active suspension. The proposed fuzzy-PID scheme is compared with passive suspension and conventional PID control under sinusoidal excitations of 2 cm, 6 cm, and 10 cm, as well as ISO 8608 random road inputs. Body acceleration, suspension travel, and tire deflection are analyzed to evaluate its performance.

2. RESEARCH METHOD

2.1. Semi-active suspension model

The 2-degree-of-freedom quarter-car semi-active suspension system with vertical vibration includes sprung mass and displacement (m_s, z_s), unsprung mass and displacement (m_u, z_u), actuator damping force (f_a), damping coefficient (b_s), spring stiffness (k_s), and tire stiffness and displacement (k_t, z_r), as shown in Figure 1. The sprung mass represents the vehicle body, whereas the unsprung mass mainly represents the wheel and associated suspension components. The interaction among the suspension spring, controllable damping force, and tire stiffness determines the vertical dynamic response of the system to road disturbances.

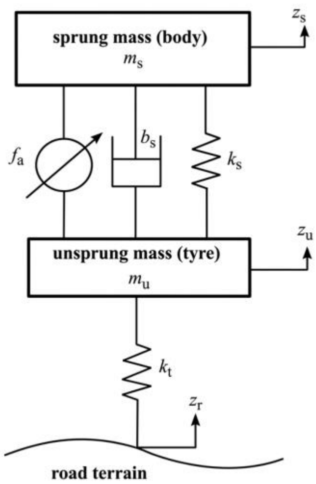

Figure 1. Semi-active suspension [20]

According to Newton's second law, the quarter-car suspension model is formulated under the assumptions that the suspension spring and tire stiffness remain linear within the operating range. And that the tire maintains contact with the road surface. Taking the static equilibrium position as the reference, the dynamic equations of the sprung and unsprung masses can be expressed as (1) and (2) [20].

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - b_s(\dot{z}_s - \dot{z}_u) - f_a \quad (1)$$

$$m_u\ddot{z}_u = k_s(z_s - z_u) + b_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) + f_a \quad (2)$$

2.2. Magnetorheological damper model

There are two types of MR damping modeling: parametric model and non-parametric model. The parametric model is more accurate in describing the dynamic characteristics of the MR damper and convenient for analyzing and understanding the working principles of the damper. Spencer's modified Bouc-Wen model, as shown in Figure 2, is adopted to describe the hysteretic behavior of MR damper [21]. This model provides a good fit to experimental data and effectively captures the nonlinear force-velocity response. The governing equations are given in (3) to (9).

$$F = c_1 \times \dot{y} + k_1 \times (x - x_0) \quad (3)$$

$$\dot{z} = -\gamma \times |\dot{x} - \dot{y}| \times z \times |z|^{n-1} - \beta(\dot{x} - \dot{y}) \times |z|^n + A(\dot{x} - \dot{y}) \quad (4)$$

$$\dot{y} = \frac{1}{c_0 + c_1}\left(\alpha z + c_0 \times \dot{x} + k_0(x - y)\right) \quad (5)$$

$$\alpha = a_a + a_b \times u \quad (6)$$

$$c_0 = c_{0a} + c_{0b} \times u \quad (7)$$

$$c_1 = c_{1a} + c_{1b} \times u \quad (8)$$

$$\dot{u} = -\eta(u - v) \quad (9)$$

Where x and $\dot{x}$ denote the relative displacement and velocity of the MR damper, respectively; k_1 is the accumulator stiffness; c_0 represents the viscous damping coefficient at high velocity; c_1 denotes the damping coefficient of the low-velocity viscous element; k_0 and x_0 are the linear spring stiffness and initial displacement, respectively; y and $\dot{y}$ are the displacement and velocity of the internal viscous element; z is hysteretic restoring variable; u is the output of the first-order filter; v is the input voltage; η is the time constant. and A, β, γ, a_a, a_b, c_{0b}, c_{1a}, c_{1b}, and n are empirical constants related to the MR fluid properties and damper structure. Table 1 lists the MR damper parameters, which are the parameters fitted and verified by Gao *et al.* [22] through experiments to accurately describe the mechanical properties of the MR damper.

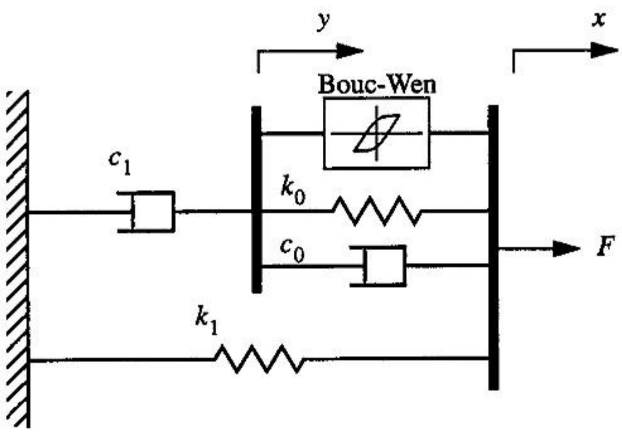

Figure 2. Spencer's modified Bouc–Wen model [21]

Table 1. MR damper parameters

Parameter	Value	Unit	Parameter	Value	Unit
a_a	1921.1	N/m	k_1	1.75	N/m
a_b	5882.5	N/(m V)	γ	36332.1	m^{-2}
c_{0a}	651.5	N s/m	β	36332.1	m^{-2}
c_{0b}	1043.76	N s/m V	A	155.32	-
c_{1a}	2089.3	N s/m	X_0	0.248	m
c_{1b}	14384.9	N s/m V	n	2	-
k_0	1940.4	N/m	η	60	s^{-1}

2.3. Fuzzy-proportional-integral-derivative controller

Fuzzy-PID control is formed by adding fuzzy control to PID controller. The fuzzy control takes the body displacement error and the body displacement error change rate as input variables. After fuzzification, fuzzy reasoning, and defuzzification, the PID control gains $\triangle K_p$, $\triangle K_i$, and $\triangle K_d$ are obtained to achieve real-time adjustment of PID parameters and increase control efficiency and robustness. Mathematical model of fuzzy-PID control is as in (10).

$$U_{PID} = (K_P + \Delta K_P)e(t) + (K_i + \Delta K_i)\int e\,(t).dt + (K_d + \Delta K_d)\frac{d}{dt}e(t) \quad (10)$$

Where U_{PID} is the PID output control variable, and K_p, K_i, and K_d are the PID control parameters. Figure 3 shows the simulation model of passive, PID controller, and fuzzy PID controller applied to the vehicle semi-active suspension system in MATLAB/Simulink. The passive suspension has no control and only relies on its own fixing mechanism to reduce vibration. The PID controller is a feedback controller. After multiple simulations, the control parameters K_p = 3, K_i = 0.15, and K_d = 0.1 are determined. The fuzzy PID adjusts K_p = 3, K_i = 0.15, and K_d = 0.1 through fuzzy control to achieve nonlinear control. The fuzzy controller is set with two inputs and three outputs. Its input membership functions are determined from the body displacement error and its rate of change under the 10 cm sinusoidal road profile. Gain of error (GE) and gain of change in error (GCE) are set to map the input variables of other road profiles to the membership ranges, and parameter K is added to adjust the control gain parameter of the fuzzy logic output.

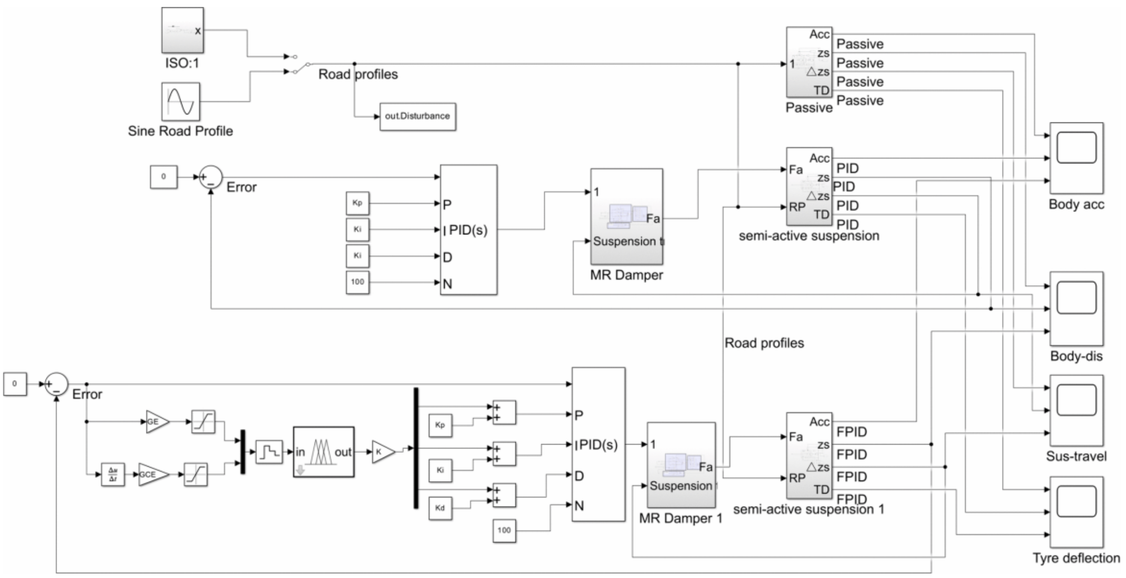

Figure 3. Semi-active suspension control system

The fuzzy rule base can be constructed using different linguistic sets, such as seven levels (NB, NM, NS, Z, PS, PM, and PB), five levels (NB, NS, Z, PS, and PB), or three levels (H, M, and L) [1], [23]. Although a larger number of linguistic levels may improve control accuracy, it also increases computational complexity. Considering the real-time requirement of vehicle vibration control, this study adopts the three-level set (H, M, and L), representing high, medium, and low, respectively. The corresponding fuzzy rules are established based on expert knowledge, as shown in Table 2.

Table 2. Fuzzy control rule base with three linguistic levels

e \ de	High	Medium	Low
High	HHH	HMH	HLH
Medium	MHH	MMM	MML
Low	LHH	LML	LLL

Figure 4 shows the structure of the fuzzy controller. The triangular membership function (trimf) is used, with Mamdani inference and centroid defuzzification. The fuzzy controller outputs the gain increments ($\triangle K_p$, $\triangle K_i$, and $\triangle K_d$) for the PID controller. The fuzzy inference process can be expressed as follows: if e = L and de = M, then $\triangle K_p$ = L, $\triangle K_i$ = M, and $\triangle K_d$ = L. The corresponding rules are shown in Table 2. If the input error and the error change rate are L and M, then K_p, K_i, and K_d are L, M, and L, respectively. Use "centroid" to obtain the exact values of the corresponding membership interval. Superimpose them on the PID control parameters to achieve control of nonlinear systems.

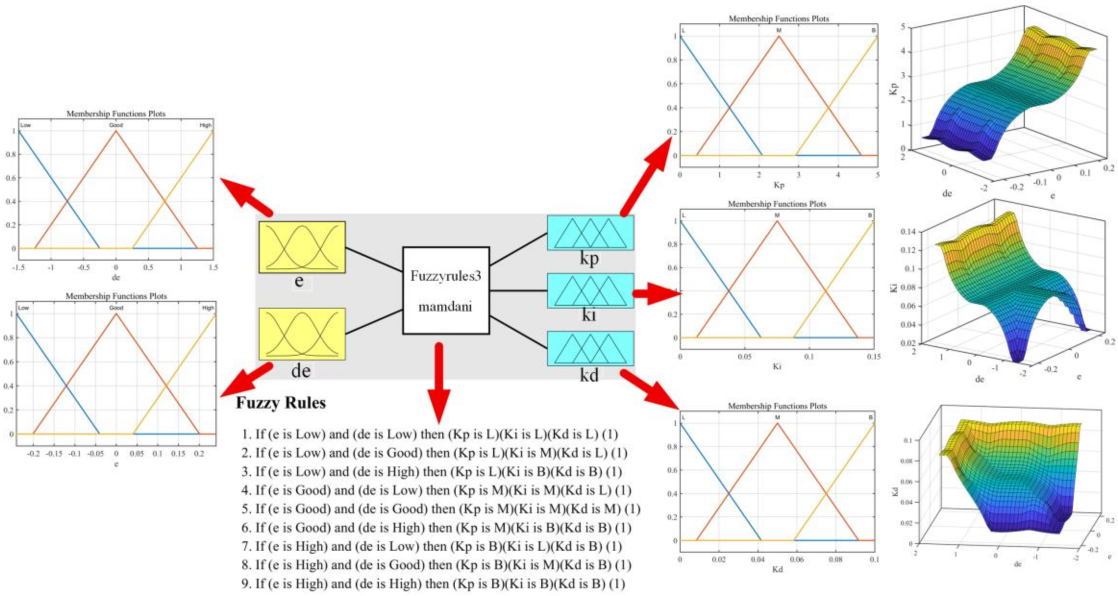

Figure 4. Fuzzy control controller

3. RESULTS AND DISCUSSION

Time-domain and frequency-domain simulations were conducted to compare the passive suspension, conventional PID controller, and proposed fuzzy-PID controller. The sinusoidal road inputs were set with amplitudes of 2 cm, 6 cm, and 10 cm at a frequency of 1 Hz. The random road input was generated according to the ISO 8608 class B profile using white noise [24], with the vehicle speed set to 20 m/s. Suspension performance is evaluated by the root mean square (RMS) and steady-state peak values of the vehicle body acceleration, suspension travel, and tire deflection. The semi-active suspension parameter settings are: m_s = 290 kg, m_u = 40 kg, K_s = 18,000 N/m, K_t = 210,000 N/m, and b_s = 1,000 N·s/m.

3.1. Sinusoidal road profiles simulation

Figures 5 to 7 show the time responses and peak values of the evaluation indices under sinusoidal road inputs with different amplitudes. Overall, the fuzzy-PID controller achieves the best performance. This is because passive suspension has fixed damping, while the conventional PID controller uses fixed gains and has limited adaptability. In contrast, the fuzzy-PID controller adjusts the PID gains through fuzzy inference, enabling better adaptation to nonlinear suspension dynamics and different road excitations.

For the 2 cm sinusoidal road profile, the fuzzy-PID controller produces the smallest evaluation indices (Figure 5). The peak body acceleration is reduced to 0.93 m/s^2, which is 0.25 m/s^2 and 0.97 m/s^2 lower than those of the PID controller and passive suspension, respectively. Suspension travel and tire deflection are also substantially reduced, indicating that the proposed controller can improve ride comfort while maintaining better suspension stability under small-amplitude excitation. This benefit is also supported by Li *et al*. [12], who showed that self-tuning control reduced body acceleration, suspension deflection, and tire displacement in an MR semi-active suspension.

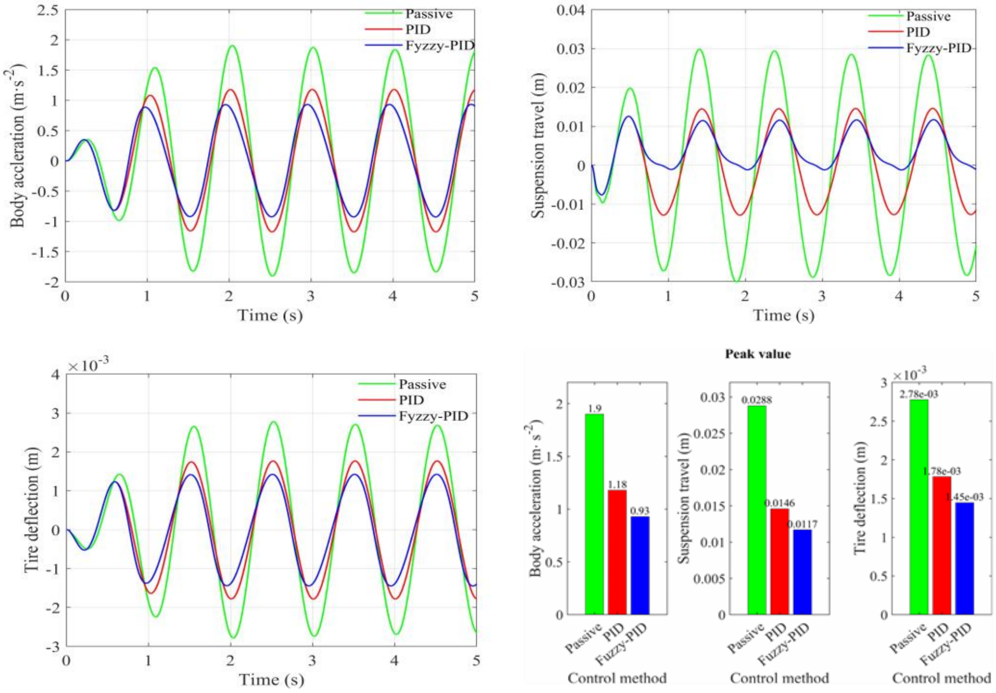

Figure 5. Time response and peak values of the evaluation indices on 2 cm amplitude sinusoidal road profile

For the 6 cm sinusoidal road profile, the superiority of the fuzzy-PID controller remains evident (Figure 6). The peak body acceleration is reduced to 3.01 m/s^2, which is 0.98 m/s^2 and 2.70 m/s^2 lower than those of the PID controller and passive suspension, respectively. The adaptive gain adjustment provided by fuzzy logic becomes increasingly beneficial as road excitation becomes stronger. The reductions in suspension travel and tire deflection further indicate that the proposed controller can suppress excessive suspension motion without causing obvious deterioration in tire-road contact performance.

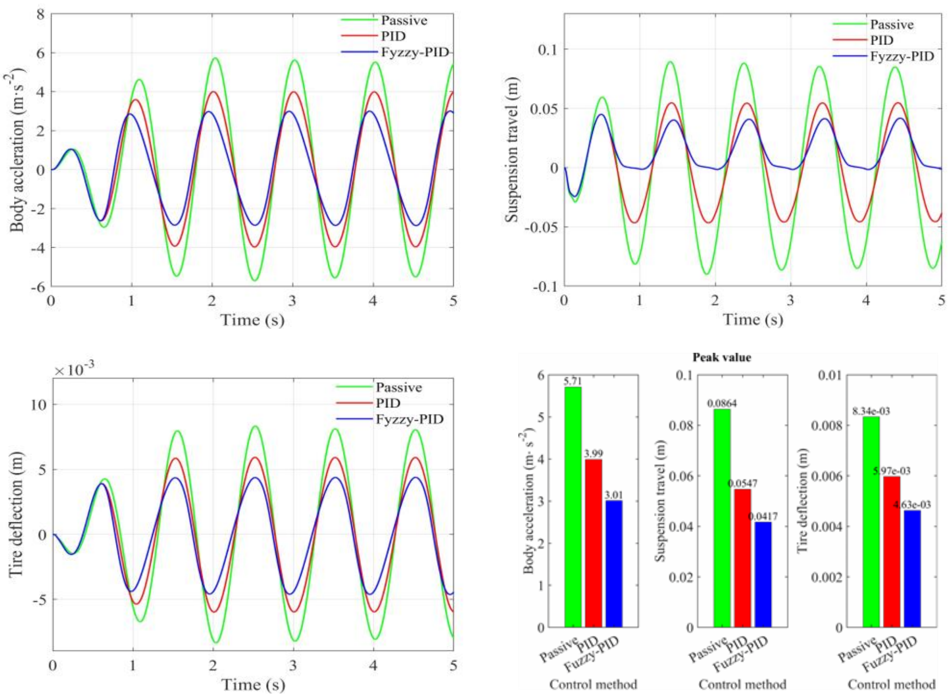

Figure 6. Time response and peak values of the evaluation indices on 6 cm amplitude sinusoidal road profile

For the 10 cm sinusoidal road profile, the fuzzy-PID controller still maintains superior vibration suppression (Figure 7). The peak body acceleration is reduced to 5.25 m/s^2, which is 1.47 m/s^2 and 2.70 m/s^2 lower than those of the PID controller and passive suspension, respectively. Although the vibration intensity increases with road amplitude, the proposed controller still effectively limits the growth of evaluation indices. This suggests that the fuzzy-PID controller has stronger robustness under large-amplitude road disturbances.

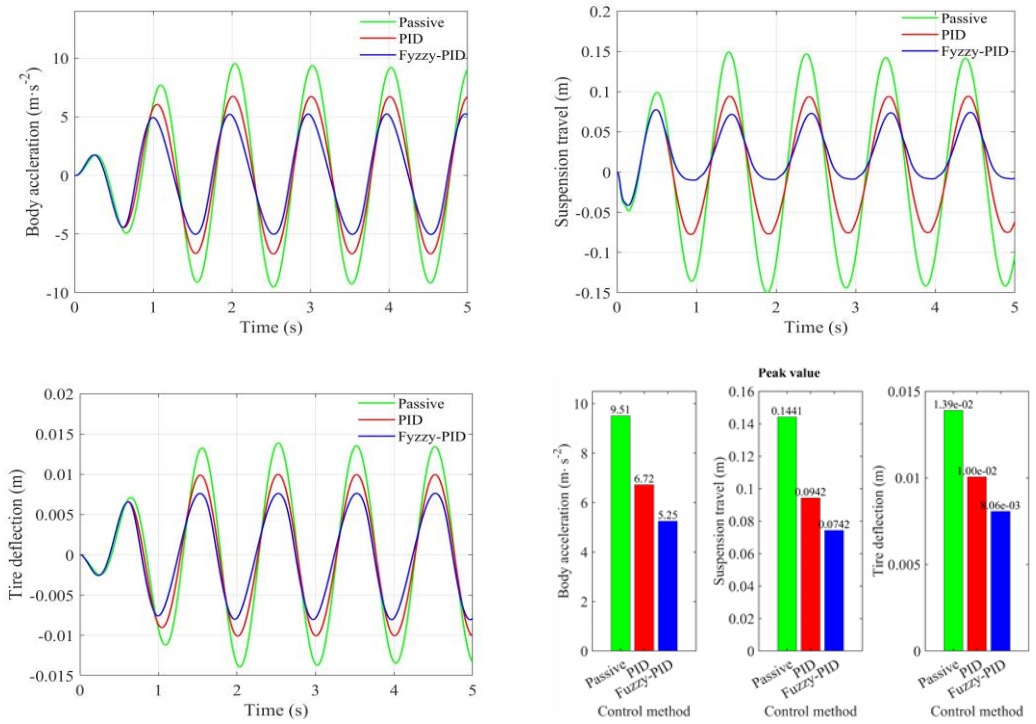

Figure 7. Time response and peak values of the evaluation indices on 10 cm amplitude sinusoidal road profile

Table 3 quantitatively confirms the superior performance of the proposed fuzzy-PID method. Compared with passive suspension, it reduces body acceleration by 46.97%, 44.56%, and 42.35%, suspension travel by 68.36%, 63.24%, and 60.71%, and tire deflection by 44.00%, 41.76%, and 39.70% under the 2 cm, 6 cm, and 10 cm sinusoidal road profiles, respectively. It also achieves lower RMS values than the PID controller for all evaluation indices, demonstrating that fuzzy gain adjustment enhances the adaptability and stability of the semi-active suspension system.

Table 3. Suspension performance of sinusoidal road profiles

Evaluation index	Road profiles (cm)	Passive		PID		Fuzzy-PID
		RMS	Fuzzy-PID with respect to passive (%)	RMS	Fuzzy- PID with respect to PID (%)	RMS
Body acceleration (m·s^{-2})	2	1.19	46.97	0.769	17.91	0.632
	6	3.57	44.56	2.60	23.91	1.98
	10	5.96	42.35	4.40	21.92	3.43
Suspension travel (m)	2	1.95E-02	68.36	9.46E-03	34.72	6.17E-03
	6	5.85E-02	63.24	3.49E-02	38.28	2.15E-02
	10	9.75E-02	60.71	5.90E-02	35.07	3.83E-02
Tire deflection (m)	2	1.74E-03	44.00	1.16E-03	15.96	9.76E-04
	6	5.23E-03	41.76	3.89E-03	21.76	3.04E-03
	10	8.71E-03	39.70	6.57E-03	19.99	5.25E-03

3.2. Random road profiles simulation

Figure 8 shows the time responses of the evaluation indices under the ISO 8608 random road profile. Compared with the sinusoidal road profiles, the difference between the fuzzy-PID controller and the

PID controller is smaller under random excitation. However, both controlled suspension systems perform better than the passive suspension.

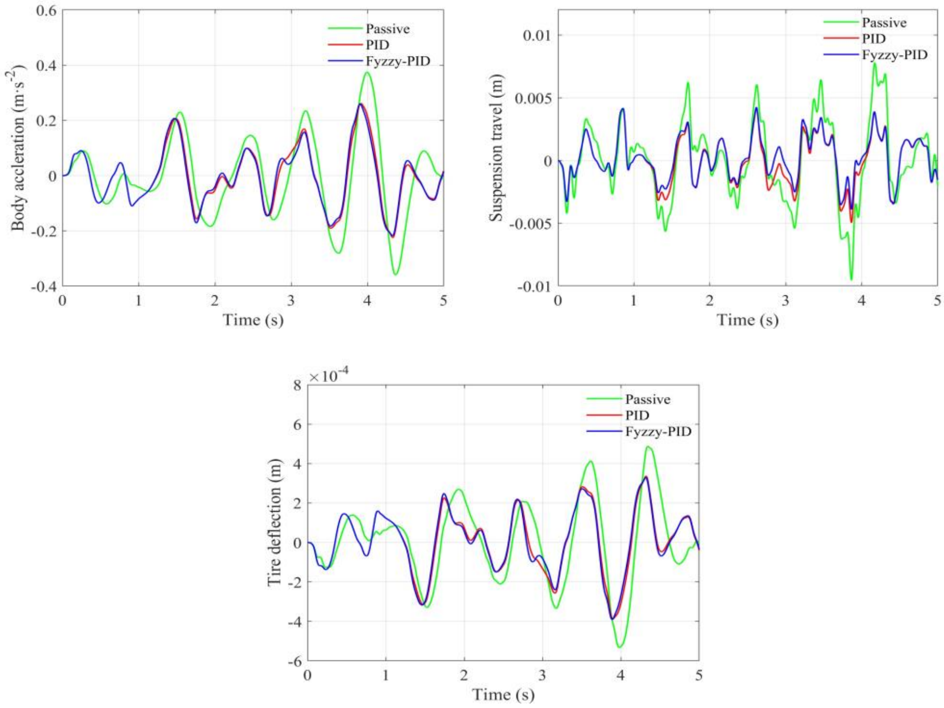

Figure 8. Time response of the evaluation indices on random road profile

As shown in Table 4, compared with the passive suspension, the fuzzy-PID controller reduces the RMS values of body acceleration, suspension travel, and tire deflection by 30.09%, 44.41%, and 25.97%, respectively, while the reductions relative to the PID controller are 1.77%, 6.32%, and 1.14%, respectively. These results indicate that the fuzzy-PID controller still achieves the best vibration reduction performance under random road excitation, although its improvement over the conventional PID controller is relatively limited. This is mainly because passive suspension is generally designed to provide acceptable performance on random road surfaces, and random excitation contains more complex frequency components, which may weaken the advantage of fuzzy gain adjustment. Nevertheless, the lowest RMS values obtained by fuzzy-PID controller confirm its effectiveness and stability under random road conditions. Min and Wei [25] similarly demonstrated that road-dependent adaptation of MR damper control parameters can improve semi-active suspension performance by enabling the controller to respond to variations in road excitation.

Table 4. Suspension performance of random road profiles

Evaluation index	Passive		PID		Fuzzy-PID
	RMS	Fuzzy-PID with respect to passive (%)	RMS	Fuzzy-PID with respect to PID (%)	RMS
Body acceleration (m·s^{-2})	0.149	30.09	0.106	1.77	0.104
Suspension travel (m)	3.12E-03	44.41	1.85E-03	6.32	1.74E-03
Tire deflection (m)	2.11E-04	25.97	1.58E-04	1.14	1.56E-04

3.3. Frequency domain analysis

Frequency-domain analysis is conducted to further examine the vibration suppression performance of the three suspension systems. According to ISO 2631-5:2018, body acceleration is an important indicator for evaluating ride comfort. Therefore, the power spectral density of body acceleration was analyzed under different road profiles. Figure 9 shows that the frequency-domain results are consistent with the time-domain analysis. The passive suspension exhibits the highest spectral response, indicating that road excitation is more directly transmitted to the vehicle body. The PID controller reduces the spectral amplitude to some extent, whereas the fuzzy-PID controller produces the lowest spectral response under the same road conditions. Figures 9(a) to 9(c) show the responses under 2, 6, and 10 cm sinusoidal road excitations, respectively, while Figure 9(d) presents the response under a random road profile. This improvement is mainly attributed to the adaptive adjustment of PID gains by fuzzy logic, which enables the controller to provide more suitable damping control according to the displacement error and its change rate. Xiong *et al*. [26] likewise reported that adaptive control of MR semi-active suspension can effectively reduce vehicle-body vibration under varying road excitations, supporting the benefit of adaptive damping regulation for vibration suppression. Therefore, the proposed fuzzy-PID controller can more effectively reduce the vibration energy transmitted to the sprung mass, thereby improving ride comfort and further verifying its vibration reduction capability.

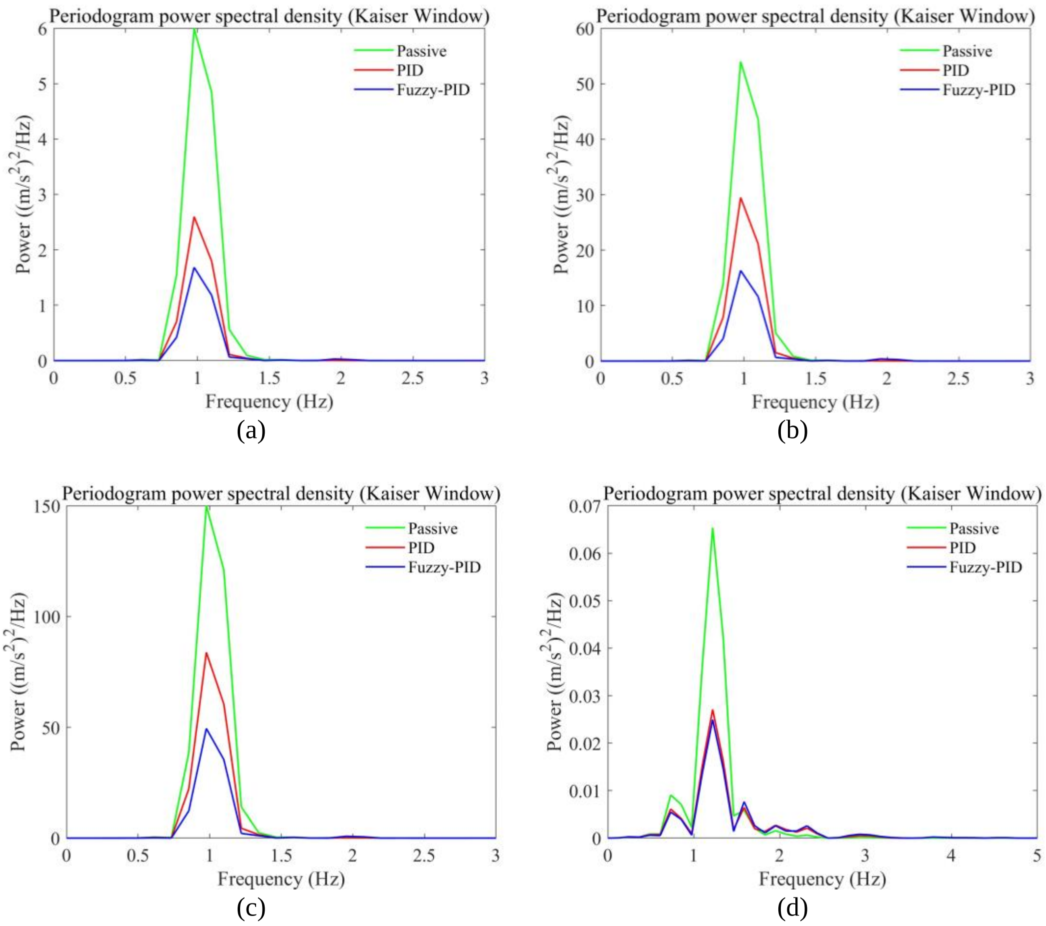

Figure 9. Frequency responses of body acceleration on different road profiles for fuzzy-PID, PID, and passive suspension systems: (a) 2 cm amplitude sinusoidal, (b) 6 cm amplitude sinusoidal, (c) 10 cm amplitude sinusoidal, and (d) random

4. CONCLUSION

A fuzzy-PID control strategy is proposed for a semi-active suspension system with an MR damper. By combining the robustness of PID control with the adaptive adjustment capability of fuzzy logic, the proposed controller improves vibration reduction performance under different road excitations. MATLAB/Simulink simulations were conducted under sinusoidal road inputs and an ISO 8608 random road profile. Compared with passive suspension, the fuzzy-PID controller reduces the RMS value of body acceleration by 46.97%, 44.56%, 42.35%, and 30.09% under the four road conditions, respectively. Compared with the conventional PID controller, the corresponding reductions are 17.91%, 23.91%, 21.92%, and 1.77%, respectively. These results confirm that the proposed method can effectively reduce vehicle body vibration and improve ride comfort. Future work may further investigate experimental validation and real-time implementation of the proposed control strategy. This method can also be applied to other areas of fast motion control such as robot motion and unmanned aerial vehicle (UAV) flight.

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AUTHOR CONTRIBUTIONS STATEMENT

This journal uses the Contributor Roles Taxonomy (CRediT) to recognize individual author contributions, reduce authorship disputes, and facilitate collaboration.

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C : **C**onceptualization
M : **M**ethodology
So : **So**ftware
Va : **Va**lidation
Fo : **Fo**rmal analysis
I : **I**nvestigation
R : **R**esources
D : **D**ata Curation
O : Writing - **O**riginal Draft
E : Writing - Review & **E**diting
Vi : **Vi**sualization
Su : **Su**pervision
P : **P**roject administration
Fu : **Fu**nding acquisition

CONFLICT OF INTEREST STATEMENT

Authors state no conflict of interest.

DATA AVAILABILITY

The authors confirm that the data supporting the findings of this study are available within the article and its supplementary materials.

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