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Fuzzy Logic Controller for Half Vehicle Active Suspension System: An Assessment on Ride Comfort and Road Holding

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Research Article

Abstract

Vehicle suspension system results in a trade-off between the crucial requirements of road holding and ride comfort. To increase the suspension performance and overcome this difficulty, vehicle dynamic systems must be equipped with new technologies and intelligent materials. This paper proposes a fuzzy logic control strategy to enhance ride comfort and road holding dynamics for a half-vehicle active suspension system. The fuzzy logic controller offers a control approach that enables systems with uncertainty and complexity to be managed effectively. Two fuzzy logic controllers for each tire are executed and arranged for the characteristics of the unsprung and sprung masses. The controllers' inputs are suspension deflection and sprung mass acceleration for each tire. Moreover, actuator force is generated as the controllers' outputs. Bump road disturbances are applied to each tire for performance evaluation of the controllers. The performance criteria for the suspension system are selected as acceleration and displacement of sprung mass, suspension deflection, and dynamic tire load. These parameters are compared to the passive suspension system and evaluated on ride comfort and road holding. Ride comfort is enhanced by 34% with the active suspension system, including a fuzzy logic controller. Furthermore, road holding is improved by about 13% regarding suspension deflection and dynamic tire load. In conclusion of the simulation, the proposed control approach enhances ride comfort and road-holding dynamics concurrently.

Keywords: Active suspension system; Fuzzy logic; Half-vehicle model; Ride comfort; Road holding.

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1. Introduction

Controllable suspension systems such as semi-active and active ones improve Ride Comfort (RC) and Road Holding (RH) concurrently with an appropriate controller. While RC is supplied by reducing vibrations from the road input disturbances to the vehicle body, RH is achieved by establishing contact between the wheels and the road [1]. Semi-Active Suspension Systems (SASS) and Active Suspension Systems (ASS) enhance RC and RH by controlling parameters such as body acceleration, suspension deflection, and dynamic tire load with various control algorithms [2-4]. While the passive suspension system has no controllable part to achieve RC or RH, the SASS has a regulable damper to control the damping ratio. In contrast to a passive damper, an actuator force can both add and dissipate energy from the system through the spring and damper [5]. However, an ASS is more effective for enhancing better control response thanks to the force actuator [6, 7]. To achieve the required performance, actuators that can apply linear force vertically are added to the suspension systems [8].

These controllable suspension systems require a controller. The controller should be chosen appropriately and arranged perfectly to optimize RC and RH together because RC and RH have trade-off characteristics. Furthermore, the suspension system has nonlinear dynamics because of many nonlinear elements. A fuzzy logic controller (FLC) is a frequently employed control approach in addressing complex nonlinear systems using fuzzy sets to define the values linguistically [9]. Fuzzy rules include the nonlinear dynamics of the suspension system in the control process [1].

It has been extensively studied in recent years, focusing on modeling and controlling half-vehicle suspension systems using FLC. To ensure RC, Demir et al. [10] developed a hybrid FLC consisting of fuzzy logic and PID controllers on an analytical nonlinear half-vehicle model. Ozbek et al. [11] proposed a model-based adaptive control law integrated with a Mamdani FLC for ASS. RC was improved without compromising RH performance and suspension working space. Gandhi et al. [12] suggested that the ANFIS-based controller demonstrated superior performance in terms of parameters amplitude of road disturbances and settling time when compared to the alternative controllers, PID, LQR, and Fuzzy. Nagarkar et al. [13] presented a method for designing a passive suspension system that closely approached the performance of an ASS. The optimized passive suspension system aimed to match the characteristics of the ASS. The least-square technique was applied to optimize passive suspension parameters. Results showed that the FLC system significantly reduced RMS acceleration by 46%, providing a more comfortable ride compared to the initial passive system. In conclusion, the optimized passive suspension system closely emulated the initial FLC system, demonstrating superior health criterion-based results compared to other suspension systems. Yoshimura et al. [14] presented FLC for a half-car suspension model and evaluated vertical and rotary motion parameters, such as accelerations, velocities, and displacements. They obtained the results as the mean squares of the time responses and spectral density. The road input was not a special chassis input. Arslan et al. presented a FLC for a quarter car active suspension system [15]. They selected the vertical displacement and velocity of the system for fuzzy inputs and evaluated the results on body displacement, velocity, acceleration, and control forces. They showed that especially for control force, FLC is more appropriate in terms of energy saving, actuator dimensions, and cost.

Some studies are on improving FLC performance and model properties of the suspension systems. Jibril et al. [16] introduced a fuzzy model predictive control for a nonlinear quarter car active suspension system, considering the complex dynamics of the spring and damper. They could effectively stabilize the suspension displacement under various road profiles. Ozbek et al. [17] presented a novel fuzzy robust-adaptive controller to enhance the robustness of active vibration systems in the face of parametric uncertainties, unmodeled dynamic effects, and external disturbances. They applied to a full car active suspension system. The proposed approach was validated through Lyapunov theory and comparative analysis with an FLC, demonstrating superior performance in eliminating disturbances and improving RC. Hsiao and Wang [18] developed a self-tuning fuzzy sliding mode controller for an ASS and evaluated RC based on ISO-2631-1. Although they improved RC almost by 60%, they did not consider RH parameters. So this is not a realistic approach, as increasing ride quality decreases road holding. Rao and Prahlad [19] presented a fuzzy ASS for a quarter car and random road input was applied to the system. The system performance was evaluated only with visual results. There are not any numeric results for a comparison. Robert et al. [20] carried out active suspension quarter car suspension system control with FLC with the same road profiles. Compared to passive suspension, fuzzy controlled active suspension offered greater control over reducing suspension travel to increase a car's ability to maintain its road-holding. Barr and Ray [21] compared the LQG and passive suspension to the fuzzy logic controlled suspension and showed that FLC substantially decreased sprung mass acceleration. They used two road inputs, random linear PSD input and random Gaussian input. The fuzzy suspension improved ride comfort by 27% over both the passive and LQG suspensions but RH was not evaluated.

There are ongoing advancements in enhancing the performance of FLCs of the suspension systems by adapting and integrating PID, optimizing with well-adapted optimization techniques such as particle swarm optimization (PSO) and genetic algorithm (GA) [22, 23]. Talib et al. [24] presented FLC with a PSO Discoverer for a semi-active suspension system. They used a sinusoidal road profile and a round-top hump road profile for the simulation. They obtained better performance on RC when taking into account body mass acceleration and displacement but they did not analyze RH. Chao et al. [25] developed a fuzzy adaptive cuckoo search algorithm aimed at enhancing the performance of the cuckoo search algorithm, particularly in addressing multi-objective optimization challenges prone to local optima. The proposed algorithm incorporated a Fuzzy PID controller design for an active suspension system, contributing to improvements in both driving comfort and road handling. Chiou and Liu [26] developed a GA-assisted FLC for a quarter-car suspension model and with this controller, they obtained lower suspension deflection, a reduced sprung mass acceleration, and a lower bouncing distance between the tire and the ground. They improved suspension deflection by 24.7%, body mass acceleration by 58.24%, and beating in the distance of the tire and ground by 16.42%. Pekgökgöz et al. [27] proposed FLC for the active suspension on a quarter-car model and optimized the membership functions with GA. They evaluated the results on body deflection and control forces for the PID controller and optimized FLC. They showed that FLC had better performance than PID.

Overall, a controller plays a vital role in a half-vehicle ASS. FLCs, in particular, should improve RC by reducing vibration and RH by increasing the contact between the road and the tire.

One of the most important problems with suspension systems is improving RC and RH together. The main contribution and novelty presented in this paper is to supply the improvement of these two important parameters together with a fundamental controller for a half-vehicle suspension system. The assessments have been done numerically on basic performance parameters criteria for the half-vehicle model. Especially for the front side of the vehicle both of the parameters have been improved. For the rear side, RC has been optimized without decreasing the RH characteristics.

This study proposes an FLC for a half-vehicle ASS. The suggested method was evaluated on these performances by applying a bump road profile with dynamic tire loads and suspension deflections for the front and rear (F&R) vehicle sides, and body acceleration. These parameter values were obtained as continuous-time, cumulative root mean square (RMS), and integral of time-weighted absolute error (ITAE). Accelerations assessed the RC, while dynamic tire loads and suspension de-flections for the F&R vehicle sides assessed the RH. The improvement in road comfort was obtained by almost 34% with RMS value. The motivation of this study is to improve RC and RH performance. Consequently, except for the rear dynamic tire load for the rear tire, all of the parameters were enhanced.

2. Problem Formulation

2.1. Half-vehicle Model

The half-vehicle model refers to a mathematical model of one side of a vehicle. This type of modeling is generally more straightforward and computationally faster as it does not include the complexity of the whole vehicle and all details of suspension systems. An ASS was implemented using a four degrees of freedom (4-DOF) half-vehicle model. While numerous significant characteristics of pitch and vertical motions were considered, the motion along the y-axis was ignored.

The model, shown in Figure 1, comprises half of the sprung mass, two suspension structures, and two tires, while the suspension structure involves a spring, a damper, and an actuator. The elements are all assumed to be linear. Moreover, the tire is represented as an unsprung mass and a spring in the model. The force actuator in the ASS is located between the unsprung and sprung masses. The parameters for the half-vehicle system can be found in Table 1, and the system model is formulated, wherein the motion equations are deduced from Newton's Law of Motion [28, 29]. The equations for the sprung and unsprung masses are given in Eq. (1) below.

$$\begin{aligned} &M\ddot{z}_c + F_{df} + F_{dr} + F_{sf} + F_{sr} = u_1 + u_2 \\ &I\ddot{\phi} + a\left(F_{df} + F_{sf}\right) - b(F_{dr} + F_{sr}) = a.u_1 - b.u_2 \\ &m_f\ddot{z}_1 - F_{sf} - F_{df} + F_{tf} = -u_1 \\ &m_r\ddot{z}_2 - F_{sr} - F_{dr} + F_{tr} = -u_2 \end{aligned} \tag{1}$$

where F_{df} and F_{dr} are the forces produced by the F&R suspension dampers. Similarly, F_{sf} and F_{sr} are the F&R suspension spring forces. Finally, F_{tf} and F_{tr} are the forces produced by the F&R tires, respectively. The values of each force are calculated as follows:

$$\begin{aligned} &F_{df} = b_{e_1}\left(\dot{z}_c - z_1 + a.\dot{\phi}\right) \\ &F_{dr} = b_e\left(\dot{z}_c - \dot{z}_2 - b.\dot{\phi}\right) \\ &F_{sf} = k_1(z_c - z_1 + a.\phi) \\ &F_{sr} = k_{r_1}(z_c - z_2 - b.\phi) \\ &F_{tf} = k_{f_2}(z_1 - z_{01}) \\ &F_{tr} = k_{r2}\left(z_2 - z_{o_2}\right) \end{aligned} \tag{2}$$

The half-vehicle ASS was modeled as a state space model, which is given in Eq. (3):

$$\begin{aligned} &\dot{x} = Ax + Bu + Ed \\ &y = Cx + Du \end{aligned} \tag{3}$$

The coefficient matrices are A and C for the state variables, B and D for the control inputs, and E for the road inputs, respectively. x is the state vector, u is the control inputs vector, d is the external disturbance vector, and they are given as follows:

$$\begin{aligned} &x = [x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8]^T,\ u = [u_1, u_2]^T \\ &d = [z_{o1}, z_{o2}]^T \end{aligned} \tag{4}$$

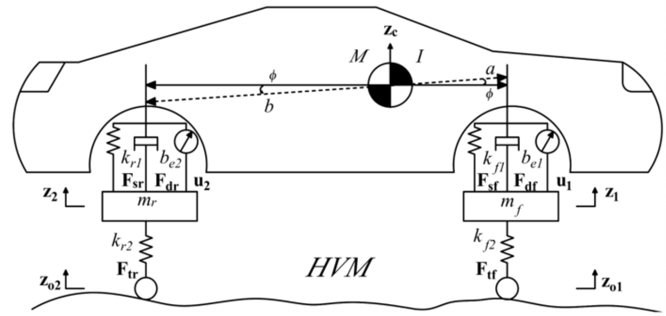

Fig. 1. Half-vehicle active suspension model

Table 1. Half-vehicle ASS parameters

Symbol	Unit	Value	Definition
M	kg	1200	Vehicle body mass (sprung mass)
m_f	kg	100	Front unsprung mass
m_r	kg	100	Rear unsprung mass
I	kgm2	600	Mass moment of inertia for pitch motion
k_{f1}	N/m	1500	Front suspension spring constant
k_{r1}	N/m	1500	Rear suspension spring constant
k_{f2}	N/m	200000	Front tire spring constant
k_{r2}	N/m	150000	Rear tire spring constant
b_{e1}	Ns/m	1500	Front suspension damping constant
b_{e2}	Ns/m	1500	Rear suspension damping constant
a	m	1.2	Distance between the front suspension structure and the center of body mass
b	m	1.5	Distance between the rear suspension structure and the center of body mass
z_c	m		Vertical displacement for sprung mass
z_1	m		Vertical displacement for front unsprung mass
z_2	m		Vertical displacement for rear unsprung mass
z_{01}	m		Road excitation for the front tire
z_{02}	m		Road excitation for the rear tire
ϕ	radian		Pitch angle for the sprung mass
u_1	N		Control input for the front suspension system
u_2	N		Control input for the rear suspension system

The initial stage in state space modeling involves identifying the state variables, which consist of the displacement and pitch motion angle of the sprung mass, displacements of the F&R unsprung masses, and the velocities associated with these variables. While the control inputs are actuator forces, road inputs are external disturbances for each tire. This model yields four outputs: body position (displacement), pitch angle of the vehicle body, and positions of the F&R unsprung masses. The state variables and their derivatives are as follows:

$$
\begin{aligned}
&x_1 = z_c && \dot{x}_1 = \dot{z}_c = x_2 \\
&x_2 = \dot{z}_c && \dot{x}_2 = \ddot{z}_c = \frac{1}{M}\begin{pmatrix} -F_{df} - F_{dr} - F_{sf} - F_{sr} \\ +u_1 + u_2 \end{pmatrix} \\
&x_3 = \dot{\phi} && \dot{x}_3 = \dot{\phi} = x_4 \\
&x_4 = \dot{\phi} && \dot{x}_4 = \ddot{\phi} = \frac{1}{I}\begin{pmatrix} -a\left(F_{df}+F_{sf}\right) \\ +b(F_{dr} + F_{sr}) \\ +a.u_1 - b.u_2 \end{pmatrix} \\
&x_5 = z_1 && \dot{x}_5 = \dot{z}_1 = x_6 \\
&x_6 = \dot{z}_1 && \dot{x}_6 = \ddot{z}_1 = \frac{1}{m_f}\left(F_{sf} + F_{df} - F_{tf} - u_1\right) \\
&x_7 = z_2 && \dot{x}_7 = \dot{z}_2 = x_8 \\
&x_8 = \dot{z}_2 && \dot{x}_8 = \ddot{z}_2 = \frac{1}{mr}\left(F_{sr} + F_{dr} - F_{tr} - u_2\right)
\end{aligned}
\tag{5}
$$

$\dot{x}_2$, $\dot{x}_4, \dot{x}_6, \dot{x}_8$ are rewritten by substituting F_{df}, F_{dr}, F_{sf}, F_{sr}, F_{tf}, and F_{tr} in Eq. (6), Eq. (7), Eq. (8), and Eq. (9):

$$
\dot{x}_2 = \ddot{z}_c = \frac{1}{M}\begin{pmatrix} -b_{e1}\left(\dot{z}_c - \dot{z}_1 + a\dot{\phi}\right) \\ -b_{e_2}\left(\dot{z}_c - \dot{z}_2 - b\dot{\phi}\right) \\ -k_{f1}(z_c - z_1 + a\phi) \\ -k_{r_1}(z_c - z_2 - b\phi) \\ +u_1 + u_2 \end{pmatrix} \tag{6}
$$

$$
\dot{x}_4 = \ddot{\phi} = \frac{1}{I}\begin{pmatrix} -ab_{e1}\left(\dot{z}_c - \dot{z}_1 + a\dot{\phi}\right) \\ -k_{f1}(z_c - z_1 + a\phi) \\ +bb_{e2}\left(\dot{z}_c - \dot{z}_2 - b\dot{\phi}\right) \\ +bk_{r1}(z_c - z_2 - b\phi) \\ +au_1 - bu_2 \end{pmatrix} \tag{7}
$$

$$
\dot{x}_6 = \ddot{z}_1 = \frac{k_{f1}}{m_f}\begin{pmatrix} (z_c - z_1 + a\phi) \\ +\dfrac{b_{e1}}{m_f}(\dot{z}_c - \dot{z}_1 + a\dot{\phi}) \\ -\dfrac{k_{f2}}{m_f}(z_1 - z_{01}) - \dfrac{1}{m_f}u_1 \end{pmatrix} \tag{8}
$$

$$
\dot{x}_8 = \ddot{z}_2 = \frac{1}{m_r}\begin{pmatrix} k_{r1}(z_c - z_2 - b.\phi) \\ +be_2\left(\dot{z}_c - \dot{z}_2 - b\dot{\phi}\right) \\ -k_{r2}(z_2 - z_{o2}) - u_2 \end{pmatrix} \tag{9}
$$

The coefficient matrices $A, B, C, D,$ and E are obtained from the above equations and they are presented in Appendix Section.

2.2. Controller Design

The controller for the ASS provides good RC by reducing vibration from the road to the vehicle and good RH by increasing the contact between the road and the tire surface. The significant conflict characteristics in designing suspension systems result in a trade-off between RC and RH [30]. While improving RC parameters, RH parameters worsen and vice versa. RC is known as the general well-being and comfort of the vehicle's occupants while they are traveling. RC is the overall outcome of intricate interactions among various wheel control, springing, and damping elements, as well as the chassis and its adjacent components.

The primary cause of vibrations and oscillations is the tire's interaction with the road. On the other hand, this interaction is a necessity for the RH. The effective force transfer between the tires and the road surface is important for road-holding and this has been supplied with an actuator force [31].

The controller's purpose for the ASS is to enhance the optimum point. In the literature, various control structures have been proposed to enhance the performance of ASSs. Among these, FLC represents a problem-solving approach, which involves the utilization of an algorithm that converts expert knowledge-based linguistic control strategies into automatic control strategies [32]. FLC exhibits an excellent dynamic response when dealing with complex, unclear, or uncertain interpretational situations [33]. The trade-off behavior of suspension system performance parameters leads to using FLC to overcome this challenge.

Figure 2 shows the proposed FLCs' structure for the half-vehicle ASS. As seen from the figure, two FLCs were used for the vehicle's F&R sides. Two actuator forces, u_1 and u_2, are the outputs of the FLCs. Body acceleration and suspension deflection (Eq. 10), for each tire, are the inputs of the FLCs.

$$
\begin{aligned}
sd_f &= z_c - z_1 \\
sd_r &= z_c - z_2
\end{aligned}
\tag{10}
$$

Five triangular membership functions (MFs) which are the most widely preferred in FLC design were used for the inputs as seen in Figure 3 (a) and (b) for the body acceleration and suspension deflection of the front vehicle and Figure 4 (a) and (b) for the rear vehicle respectively. The linguistic labels are Negative Big (NB), Negative Small (NS), Zero (ZE), Positive Small (PS), and Positive Big (PB). The nine MFs for the outputs of the FLCs were chosen as singular type and shown in Figure 5 (a) and (b).

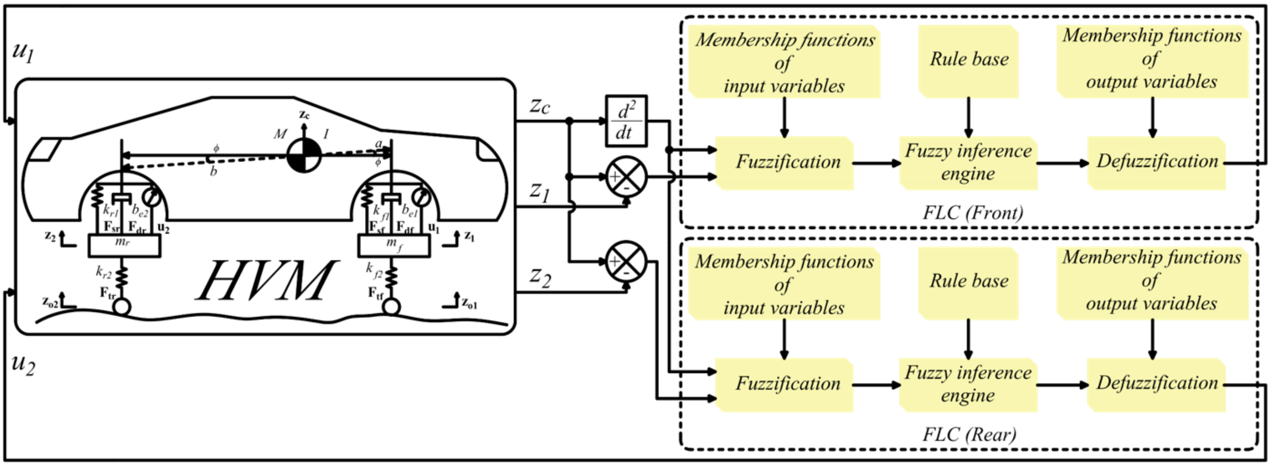

Fig. 2. Structure of FLCs for half vehicle active suspension system

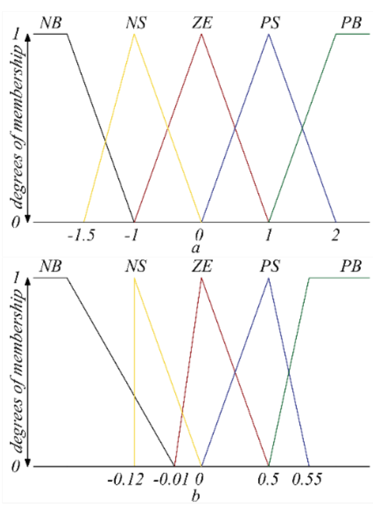

Fig. 3. FLC input MFs for front side a)MFs for body acceleration of the sprung m ass b)MFs for the front suspension deflection

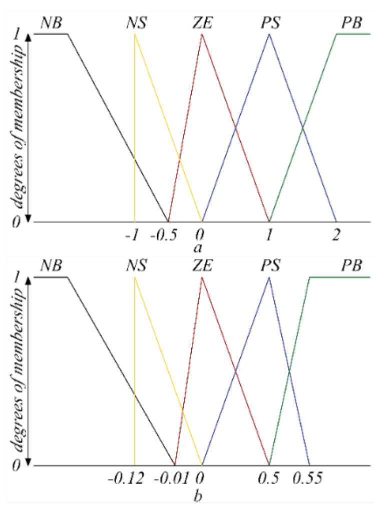

Fig. 4. FLC input MFs for rear side a) MFs for body acceleration of the sprung m ass b) MFs for the rear suspension deflection

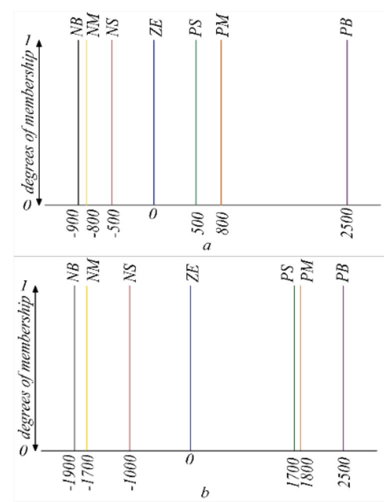

Fig. 5. FLC output MFs for a) Front side b) Rear side

The additive linguistic labels for the outputs are Negative Medium (NM) and Positive Medium (PM). The passive suspension system simulation determined the number of MFs and the function limits. Fine-tuning processes were done with repetitive simulation studies. The rule base utilized for the F&R vehicle sides fuzzy control is expanded from the expert and is given in Table 2. The max-min Method and Mamdani implication were used for the inference mechanism.

Table 2. Rule base for used front and rear FLCs

sd \ ba	NB	NS	ZE	PS	PB
NB	*PB*	*PB*	*PM*	*PS*	*ZE*
NS	PM	PM	PS	ZE	ZE
ZE	PS	ZE	ZE	ZE	NS
PS	PS	ZE	ZE	NS	NS
PB	ZE	NS	NS	NM	NB

3. Results and Discussion

The proposed FLCs' performances have been assessed through a comparative analysis of the outcomes obtained from the passive suspension system and the ASS with FLC. RC and RH were analyzed on a bump road input. This road profile is shown in Figure 6 (a). The system response to the road disturbance was assessed by observing of the sprung displacement and acceleration regarding RC. It can be seen from Figure 6 that the displacement and the acceleration are significantly enhanced with the ASS. The reduction in suspension deflection results in an improvement in RH. The suspension deflections for the F&R vehicle sides are shown in Figure 7 (a) and (b), respectively. While a noticeable improvement is obtained for the front vehicle side, a slight improvement is achieved for the rear vehicle side. In addition to the suspension deflection, the dynamic tire load parameter is related to the RH. The dynamic tire loads for the F&R vehicle sides are shown in Figure 8 (a) and (b), respectively.

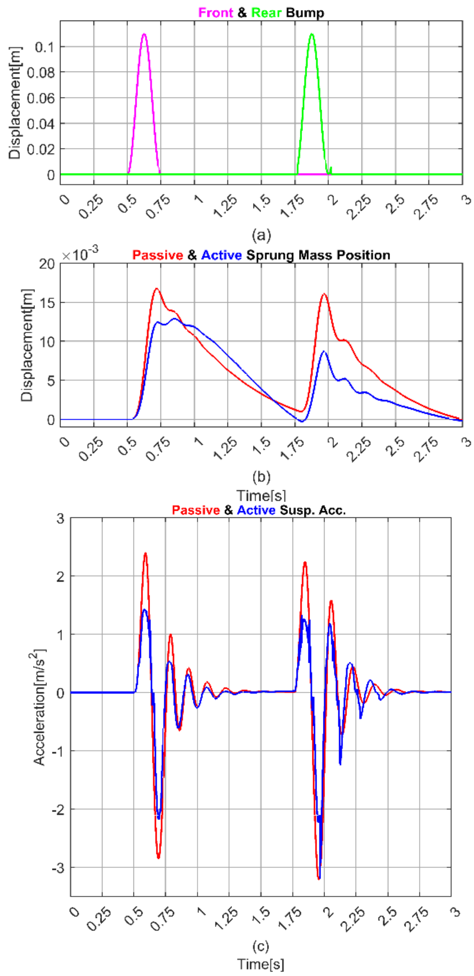

Fig. 6. Performance parameters for RC a) Road input in the shape of bump b) Body (sprung mass) displacement for passive and ASS c) Body (sprung mass) acceleration for passive and ASS

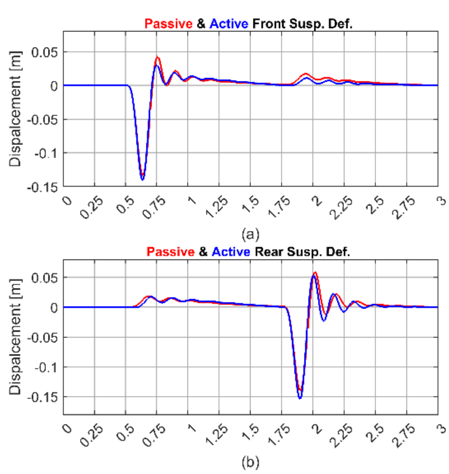

Fig. 7. Suspension deflections for a) Front vehicle side b) Rear vehicle side

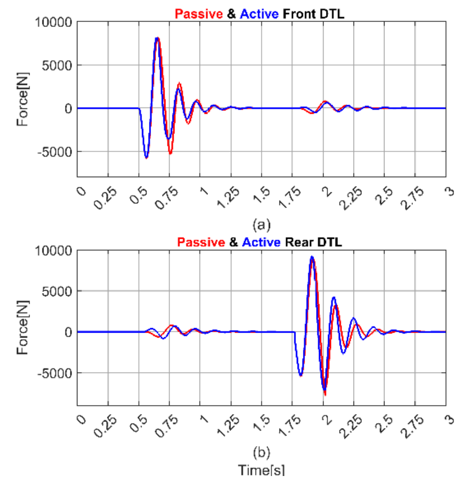

Fig. 8. Dynamic tire loads a) Front vehicle side b) Rear vehicle side

This parameter is improved for the front vehicle side. However, the ASS with FLC exhibits a slightly inferior performance compared to the passive suspension system for the rear side of the vehicle.

The performance results of the controller are given in Table 3. This table includes ITAE and RMS values of the body (sprung mass) acceleration, suspension deflection, and dynamic tire load for each vehicle side, F&R. The change percentage according to the passive suspension system is also presented in the table. Improvement of the parameter is shown with an up arrow and deterioration with a down arrow. According to the results, body acceleration is improved by 34% in RMS and 24% in ITAE. A significant enhancement in RC can be achieved because body acceleration is related to RC.

Additionally, front suspension deflection and front dynamic tire load are improved by 21% and 12% in ITAE, almost by 13% in RMS. Suspension deflection and dynamic tire load are related to RH. These results show that the front RH capability is enhanced as these values decrease. Furthermore, this results from an appropriate design of an FLC for the front vehicle side. For the rear vehicle side controller, rear suspension deflection is improved by 1.2% in ITAE. The rear dynamic tire load is deteriorated by 6.7% in ITAE and 5.4% in RMS. These results show that FLC for the rear vehicle has slightly worsened the RH. This situation is an expected outcome. RC and RH have contrasting behaviors by the nature of the suspension system. Conventional controllers for the suspension system can ideally improve one parameter while worsening or leaving the other parameter the same. This trade-off can be overcome with advanced controller techniques and optimized parameters for FLC.

Table 3. Results of ASS with FLC and passive suspension system

		Passive	Active	Improvement
Body Acceleration	RMS	1.315	0.861	↑34%
	ITAE	2.409	1.810	↑24%
Front Suspension Deflection	RMS	0.019	0.016	↑13%
	ITAE	0.031	0.024	↑21%
Rear Suspension Deflection	RMS	0.029	0.029	↓0.6%
	ITAE	0.052	0.051	↑1.2%
Front Dynamic Tire Load	RMS	1027	876	↑14%
	ITAE	1388	1212	↑12%
Rear Dynamic Tire Load	RMS	1997	2016	↓5.4%
	ITAE	3602	3844	↓6.7%

4. Conclusion

This paper presents a fuzzy logic-controlled half-vehicle ASS to improve RC and RH. A half-vehicle suspension system has been modeled as state space. Body acceleration, suspension deflection, and dynamic tire load have been selected as performance criteria. The road disturbance is a bump input based on 20 m/s velocity. The proposed controller outcomes have been compared to the passive suspension system. The foundational impetus behind this study is to improve RH and RC simultaneously. Sprung mass displacement and acceleration curves for the passive and ASSs have been drawn and obtained analytically. The simulation results show that the RC is enhanced by 34% with the ASS. Furthermore, the RH for the front vehicle side is improved by about 13% in terms of suspension deflection and dynamic tire load. While the rear vehicle RH characteristics are nearly the same as the passive suspension system in terms of suspension deflection, dynamic tire load is worse. This condition means that the rear FLC requires robust optimization. The solution may be an optimization method or another controller structure. Consequently, FLC is generally an adequate controller for ASSs, especially RC and RH. A future issue to work on is optimizing the MFs' limits and rule base with an optimization method.

Nomenclature

RC	:	Ride Comfort
RH	:	Road Holding
SASS	:	Semi-Active Suspension Systems
ASS	:	Active Suspension Systems
FLC	:	Fuzzy Logic Controller
PID	:	Proportional-Integral-Derivative
F&R	:	Front and Rear
RMS	:	Root Mean Square
ITAE	:	Integral of Time-Weighted Absolute Error
DOF	:	Degrees of Freedom

Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRediT Author Statement

Meral Ozarslan Yatak: Conceptualization, Formal analysis, Investigation, Software, Writing - original draft.
Çagdas Hisar: Conceptualization, Formal analysis, Investigation, Software, Validation, Visualization, Writing - review & editing.
Fatih Sahin: Conceptualization Data curation, Formal analysis, Supervision.

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Appendix

$$A=\begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{1}{M}(-k_{f1}-k_{r1}) & \frac{1}{M}(-b_{e1}-b_{e2}) & \frac{1}{M}(-a.k_{f1}+b.k_{r1}) & \frac{1}{M}(-a.b_{e1}+b.b_{e2}) & \frac{k_{f1}}{M} & \frac{b_{e1}}{M} & \frac{k_{r1}}{M} & \frac{b_{e2}}{M} \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ \frac{1}{I}(-a.k_{f1}+b.k_{r1}) & \frac{1}{I}(-a.b_{e1}+b.b_{e2}) & \frac{1}{I}(-a^2.k_{f1}-b^2.k_{r1}) & \frac{1}{I}(-a^2.b_{e1}-b^2.b_{e2}) & \frac{a.k_{f1}}{I} & \frac{a.b_{e1}}{I} & \frac{-b.k_{r1}}{I} & \frac{-b.b_{e2}}{I} \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_{f1}}{m_f} & \frac{b_{e1}}{m_f} & \frac{a.k_{f1}}{m_f} & \frac{a.b_{e1}}{m_f} & \frac{-k_{f1}-k_{f2}}{m_f} & \frac{-b_{e1}}{m_f} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ \frac{k_{r1}}{m_r} & \frac{b_{e2}}{m_r} & \frac{-b.k_{r1}}{m_r} & \frac{-b.b_{e2}}{m_r} & 0 & 0 & \frac{-k_{r1}-k_{r2}}{m_r} & \frac{-b_{e2}}{m_r} \end{bmatrix}$$

$$B=\begin{bmatrix} 0 & 0 \\ \frac{1}{M} & \frac{1}{M} \\ 0 & 0 \\ \frac{a}{I} & -\frac{b}{I} \\ 0 & 0 \\ -\frac{1}{m_f} & 0 \\ 0 & 0 \\ 0 & -\frac{1}{m_r} \end{bmatrix} \qquad C=\begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \end{bmatrix} \qquad D=0 \qquad E=\begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ \frac{k_{f2}}{m_f} & 0 \\ 0 & 0 \\ 0 & \frac{k_{r2}}{m_r} \end{bmatrix}$$