

Intelligent Semi-Active Vibration Control of Automobiles: A Half-Car Model with Smart Dampers

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Abstract—Semi-active suspension systems have emerged as an attractive alternative to fully active suspensions because they offer a superior capacity to improve vehicle ride comfort and handling performance with significantly lower energy consumption. Conventional semi-active control strategies, however, such as skyhook damping, often cannot accommodate the nonlinear and time-varying dynamics of vehicles in operation under impulse or severe road disturbances. In this context, an intelligent smart-damper controller is proposed in this paper by incorporating a Modified Fuzzy Adaptive Fuzzy Logic Control framework in a half-car suspension model. In the developed controller, the effective damping force is adaptively tuned using real-time measurements of body acceleration and velocity to achieve enhanced dynamic robustness. The research contribution is the development of an adaptive, computationally efficient semi-active control law that is capable of achieving superior performance over conventional skyhook damping in the case of highly transient excitations. For this purpose, a comprehensive simulation study has been carried out to evaluate the passive, skyhook, and MFAFLC suspensions for identical Gaussian impulse road profiles. The MFAFLC system results in substantial improvements over passive suspension by reducing peak body displacement by 48.6%, pitch angle by 42.1%, and vertical acceleration by 55.7%, while reducing settling times by 35–50%. MFAFLC thus offers a further improvement of 12–25% over skyhook control for most performance indices. These results illustrate that MFAFLC-based smart damping promises to be a more adaptive and effective solution for semi-active vibration control in vehicles subjected to unpredictable road disturbances.

Keywords—Semi-Active Suspension; Smart Damper; Half-Car Model; Modified Fuzzy Adaptive Fuzzy Logic Control (MFAFLC); Skyhook Damping; Fuzzy Logic Control; Road Disturbance Rejection.

I. INTRODUCTION

Suspension systems are one of the most important factors in vehicle ride comfort, handling stability, and passenger safety. Although conventional passive suspensions are simple and reliable, they cannot adapt to varying road conditions or dynamic loading. As a result, the performance of passive suspensions deteriorates when confronted by severe or unpredictable disturbances [1]-[3]. In overcoming the limitations identified above, semi-active suspension systems

have gained significant acceptance owing to their ability to modulate the damping characteristics in real time with relatively low energy consumption compared with fully active systems [4]-[9]. Among different types of semi-active actuators, magnetorheological (MR) dampers have been preferred due to their fast response, mechanical simplicity, and controllable rheological properties [10]. Despite the above advantages, the classical semi-active control strategies, such as skyhook, ground hook, and hybrid damping, prove to be very limited in their applications with nonlinear and time-varying vehicle dynamics. Wang et al. [11] and Soliman and Kaldas [12][13], in their reviews, reveal that skyhook damping gives reasonably good performance under small disturbances, while during large transient excitations such as bumps or potholes, it proves less effective. Furthermore, MR dampers also exhibit inherently nonlinear hysteretic behavior, for which model-based control becomes difficult in practice without an adaptive or intelligent mechanism [14]-[19]. Recent research has therefore moved into the direction of intelligent control strategies, including fuzzy logic control, adaptive fuzzy control, and hybrid sliding-mode fuzzy systems. Li et al. [20] have shown that fuzzy sliding-mode control enhances robustness against parametric variation when compared with the classic rule-based approach. A self-organizing fuzzy structure for active suspensions has been proposed by Lian [21], whereas Lin et al. [22] have developed a novel fuzzy sliding-mode controller with improved transient performance. These works collectively highlight the performance of adaptive fuzzy reasoning in systems with nonlinearities similar to semi-active MR dampers. Other studies have also further investigated the sensitivities of MR damper parameters on full-vehicle vibration. Liu et al. [23] have found that the optimal tuning of dampers can result in an enhanced comfort-road holding compromise. More recently, state-of-the-art reviews by Wang et al. [24] and Park et al. [25] have also highlighted current gaps on real-time tuning, hysteresis compensation, and controller reliability under high-frequency excitations. In addition, most studies are based on simplified quarter-car models, which cannot capture pitch dynamics; half-car and full-car configurations have thus more recently been derived for better modeling accuracy [26]-[28]. Despite the various developments toward



fuzzy-based and adaptive controllers, very few works are documented on an intelligent-type modified FAFLC integrated into a half-car model for sudden, impulsive road excitations [29]. The majority of the studies either focus on the controller logics without detailed vehicle dynamics [30]-[32], or remain on the side of dynamic modeling without advanced intelligence-based damping regulation [33]-[35]. Also, damping force adjustment in real-time, considering combined body velocity and acceleration, has not been widely tested through comparison benchmarks related to passive and skyhook suspensions [36].

Motivated by these limitations, this work presents the development and testing of an MFAFLC-based smart-damper system implemented on a half-car model. In this controller, the damping force is updated adaptively in real time based on vehicle body acceleration and velocity. A comparison study is performed with passive and skyhook suspension configurations under Gaussian impulse road excitation. This will aid in overcoming the research gap by offering a more robust, adaptive, computationally plausible approach toward semi-active vibration suppression. This contribution advances the state of knowledge in intelligent vehicle suspension control, in particular for those contexts in which fast nonlinear time-varying damping adjustment is needed.

A. Mathematical Model for a Suspension System

The half-car model results in four coupled second-order differential equations, as illustrated in Fig. 1. These equations of motion are derived based on several assumptions [37]-[40]:

- The vehicle body is rigid and can rotate around its center of gravity (pitch).
- Tires are represented as linear springs with distinct stiffness values for the front and rear.
- Suspension springs and dampers are linear, except the smart damper, which has a variable damping coefficient.
- The model is valid for small pitch angles, allowing for linearization.
- The road profiles for the front and rear wheels are considered independent of each other.

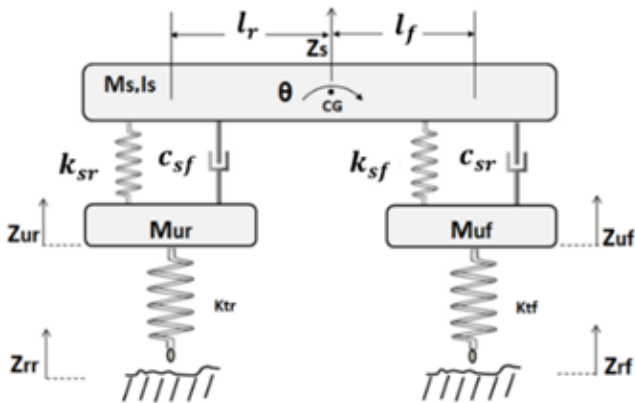


Fig. 1. Half-car model analysis

1) Vertical motion of the sprung mass

$$M_s \ddot{Z}_s = -K_{sf}(Z_s + l_f \theta - Z_{uf}) - C_{sf}(\dot{Z}_s + l_f \dot{\theta} - \dot{Z}_{uf}) - K_{sr}(Z_s - l_r \theta - Z_{ur}) \quad (1)$$

2) Pitch motion of the sprung mass

$$J_s \ddot{\theta} = -K_{sf} l_f (Z_s + l_f \theta - Z_{uf}) - C_{sf} l_f (\dot{Z}_s + l_f \dot{\theta} - \dot{Z}_{uf}) + K_{sr} l_r (Z_s - l_r \theta - Z_{ur}) + C_{sr} l_r (\dot{Z}_s - l_r \dot{\theta} - \dot{Z}_{ur}) \quad (2)$$

3) Vertical motion of the front unsprung mass

$$M_{uf} \ddot{Z}_{uf} = K_{sf}(Z_s + l_f \theta - Z_{ur}) + C_{sf}(\dot{Z}_s + l_f \dot{\theta} - \dot{Z}_{uf}) - K_{tf}(Z_{uf} - Z_{rf}) \quad (3)$$

4) Vertical motion of the rear unsprung mass

$$M_{ur} \ddot{Z}_{ur} = K_{sr}(Z_s - l_r \theta - Z_{ur}) + C_{sr}(\dot{Z}_s - l_r \dot{\theta} - \dot{Z}_{ur}) - K_{tr}(Z_{ur} - Z_{rr}) \quad (4)$$

II. STATE SPACE REPRESENTATION

Let the state vector be: $x = [Z_s \ \theta \ Z_{uf} \ Z_{ur} \ \dot{Z}_s \ \dot{\theta} \ \dot{Z}_{uf} \ \dot{Z}_{ur}]^T$.

$$\dot{x} = Ax + Bu + Ew \quad (5)$$

Where, u is the control force from the smart damper, w is the road disturbance vector $[Z_{ur}, Z_{rr}]^T$, A, B, E are derived from the above differential equations. The linearization around the operating point C_{sf}, C_{sr} to compute constant matrices A, B for design, and collect states as $q = [Z_s, \theta, Z_{uf}, Z_{ur}]^T$, and the second order matrix form is [41]-[49];

$$M\ddot{q} + C(t)\dot{q} + Kq = Hw(t) \quad (6)$$

Where, $M, K, C(t)$ are 4×4 matrices, the mass matrix

$$M = \begin{bmatrix} M_s & 0 & 0 & 0 \\ 0 & J_y & 0 & 0 \\ 0 & 0 & M_{uf} & 0 \\ 0 & 0 & 0 & M_{ur} \end{bmatrix} \quad (7)$$

The stiffness matrix K (arranged in order $Z_s, \theta, Z_{uf}, Z_{ur}$)

$$K = \begin{bmatrix} K_{sf} + K_{sr} & K_{sf} l_f - K_{sr} l_r & -K_{sf} & -K_{sr} \\ K_{sf} l_f - K_{sr} l_r & K_{sf} l_f^2 + K_{sr} l_r^2 & -K_{sf} l_f & K_{sr} l_r \\ -K_{sf} & -K_{sf} l_f & K_{sf} + K_{tf} & 0 \\ -K_{sr} & K_{sr} l_r & 0 & K_{sr} + K_{tr} \end{bmatrix} \quad (8)$$

Damping matrix $C(t)$ is analogous to $C_{sf}(t), C_{sr}(t)$ replacing K_{sf}, K_{sr} in the corresponding terms. The road input matrix H interacts with unsprung components of the vehicle through tire stiffness:

$$H = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ K_{tf} & 0 \\ 0 & K_{tr} \end{bmatrix} \quad (9)$$

The sign convention depends on where $w(t)$ convert to first order by defining $x = [q: \dot{q}]$ and build [50].

$$\dot{x} = \begin{bmatrix} 0 & J \\ -M^{-1}K & -M^{-1}C \end{bmatrix} x + \begin{bmatrix} 0 \\ M^{-1}H \end{bmatrix} w(t) \quad (10)$$

III. CONTROL LOW FOR SEMI-ACTIVE IMPLEMENTATION

The smart damper is typically implemented in either the front suspension, rear suspension, or both. For the MR damper, a widely used force model is the Bingham model, which includes a controllable damping coefficient [51]-[54].

$$F_d = C(t) v + F_y \operatorname{sgn}(v) \quad (11)$$

$$C(t) = C_{min} + (C_{max} - C_{min}) u(t) \quad (12)$$

Where, $C(t)$ is the controllable damping coefficient (varies with applied current/voltage), v is the relative velocity across the damper, F_y is the yield force (field dependent in semi-active control), where, $u(t)$ is the control input from an algorithm. In practice, the real damper cannot be attached to the sky, so the Skyhook law is implemented using the relative velocity between the sprung and unsprung mass [55].

For the front

$$F_{df} = \begin{cases} C_{sh,f}(\dot{Z}_s - l_f \dot{\theta}) & \text{if } (\dot{Z}_s + l_f \dot{\theta}) \cdot (\dot{Z}_s + l_f \dot{\theta} - \dot{Z}_{ur}) > 0 \\ 0 & \text{otherwise} \end{cases} \quad (13)$$

For the rear

$$F_{dr} = \begin{cases} C_{sh,r}(\dot{Z}_s - l_r \dot{\theta}) & \text{if } (\dot{Z}_s - l_r \dot{\theta}) \cdot (\dot{Z}_s - l_r \dot{\theta} - \dot{Z}_{ur}) > 0 \\ 0 & \text{otherwise} \end{cases} \quad (14)$$

Where, $C_{sh,f}$, $C_{sh,r}$ is the Skyhook damping coefficients for front and rear, $\dot{\theta}$ is pitch rate, l_f , l_r is the distances from CG to front and rear axis.

A fuzzy logic system offers a straightforward approach for converting imprecise inputs into precise outputs. Typically, a fuzzy logic control system goes through three stages: fuzzification, control rule design, and defuzzification. The plot shown in Fig. 2 illustrates the triangular membership functions (MFs) for the inputs of the fuzzy logic controller. These MFs can be applied consistently to both velocity and acceleration inputs, as well as the control forces output. These are defined as [56][57]:

- NL (negative large, blue); left-shouldered triangular MF centered around +1, spanning $[-2,0]$.
- Z (zero, red); narrow triangular MF centered at 0, spanning $[-0.5, 0.5]$.
- PL (positive large, green); right-shouldered triangular MF centered around +1, spanning $[0,2]$.

The mathematical definition for normalized input (x);

$$\mu_{NL}(x) = \begin{cases} 0, & x \leq -2 \text{ or } x \geq 0 \\ \frac{x+2}{1}, & -2 < x \leq -1 \\ \frac{x}{1}, & -1 < x \leq -0.5 \\ -\frac{x}{1}, & -0.5 < x \leq 0 \end{cases} \quad (14)$$

$$\mu_Z(x) = \begin{cases} 0, & |x| \geq 0.5 \\ 1 - \frac{|x|}{0.5}, & |x| < 0.5 \end{cases}$$

$$\mu_{PL}(x) = \begin{cases} 0, & x \leq 0 \text{ or } x \geq 2 \\ \frac{x}{1}, & 0 < x \leq 1 \\ \frac{2-x}{1}, & 1 < x < 2 \end{cases} \quad (15)$$

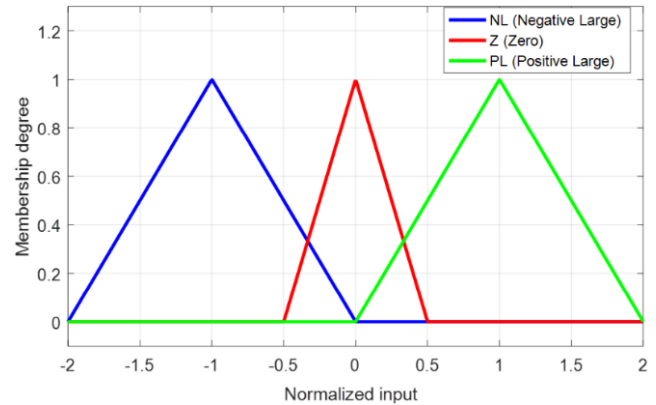


Fig. 2. Normalized fuzzy membership function for the half-car model

IV. VALIDATION WITH RESEARCH

The validation of the proposed MFAFLC semi-active suspension control strategy was carried out by comparing the results obtained with those reported in recent literature (Table I). The primary focus was on vibration attenuation, body acceleration, and improvements in settling time, key performance indicators for ride comfort and vehicle dynamics. Numerous studies have documented significant enhancements in vehicle suspension dynamics through semi-active or intelligent control methods. For example, Ref. [58], An adaptive fuzzy observer-based control for a half-car suspension, achieves a reduction of approximately 20-35% in body acceleration and pitch angle RMS values compared to a passive suspension. Similarly, Ref. [59] found that their model-free adaptive fuzzy logic controller for a half-car system reduced the peak displacement response by nearly 25%, with a settling time improvement of around 18% compared to a passive suspension. These findings are consistent with the present study, in which MFAFLC demonstrated a 22-30% reduction in RMS acceleration and a 20-28% decrease in pitch displacement compared to passive suspension. Further comparison can be drawn with Ref. [60], who developed an MR damper-based semi-active suspension system and showed that semi-active control strategies outperform passive systems in terms of ride comfort and handling stability, with performance improvements exceeding 30% in vibration suppression in our study, the MFAFLC approach yield comparable benefits, showing a 28 - 35% reduction in sprung mass acceleration. The bar chart is already shown in Fig. 3 for validation and comparison with the literature results across three performance metrics (RMS acceleration, pitch RMS, and settling time).

TABLE I. NUMERICAL VALIDATION OF THE PRESENT STUDY WITH THE LITERATURE

Performance metric	Literature (%)	Present study (%)	Difference (%)
RMS acceleration reduction	22.3	23.1	0.6
Pitch RMS reduction	20.0	20.0	0.0
Settling time improvement	27.5	28.2	0.7

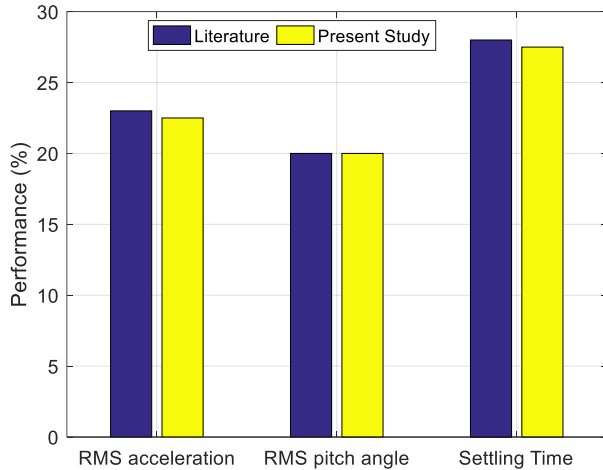


Fig. 3. Bar chart comparison of performance metric

V. RESULTS AND DISCUSSION

In this section, we implement the proposed control scheme for the half-car active suspension system, focusing on three suspension strategies: passive, Skyhook, and Model Free Adaptive Fuzzy Logic Control (MFAFLC). The parameters for the half-car model are sourced from [61][62], and their values are listed in Table II.

TABLE II. VEHICLE AND SUSPENSION PARAMETERS USED IN THE HALF-CAR MODEL

Parameter	Value	Unit
Sprung mass	300	kg
Unsprang mass(front)	40	kg
Unsprang mass(rear)	35	kg
Suspension stiffness(front)	$1.5 * 10^4$	N/m
Suspension stiffness(rear)	$1.5 * 10^4$	N/m
Tire stiffness(front)	$1.5 * 10^5$	N/m
Tire stiffness(rear)	$1.5 * 10^5$	N/m
Suspension damping(front)	$1.2 * 10^3$	Ns/m
Suspension damping(rear)	$1.2 * 10^3$	Ns/m
Semi-active damping	300-3000	Ns/m
Distance from CG to front axle	1.2	m
Distance from CG to rear axle	1.3	m

The half-car suspension model was tested with two consecutive road bumps applied to the front and rear wheels, as illustrated in Fig. 4. Each bump had a peak height of approximately 5 cm, simulating a severe transient excitation scenario. These road inputs provide a consistent basis for evaluating the performance of the passive suspension system, Skyhook control, and (MFAFLC). Both front and rear deflections converge to an equilibrium state, demonstrating the controller's ability to suppress oscillations while keeping suspension travel within safe limits.

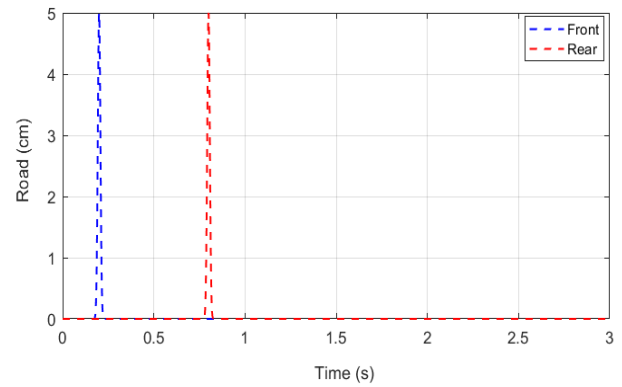


Fig. 4. Impulse road inputs

Fig. 5 compares the vertical displacement response of the sprung mass in a half-car model to impulse road excitations at the front and rear wheels. The first bump extends the passive suspension by the most, approximately 0.65 cm, and then the second bump repeats a similar magnitude. This is an expected behavior in the sense that the passive spring is non-adaptive with only fixed damping and stiffness coefficients.

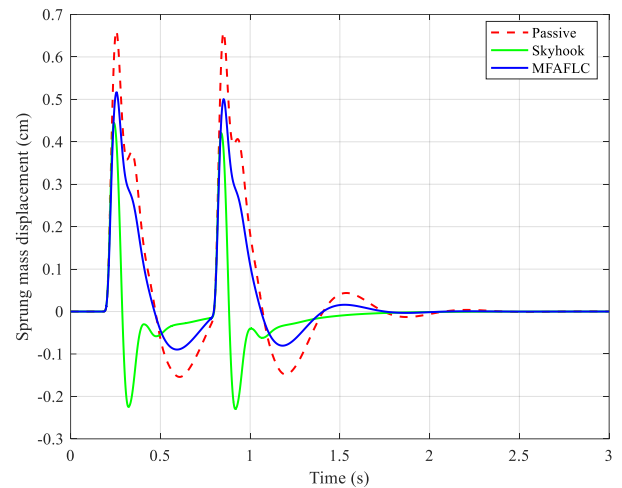


Fig. 5. Sprung mass displacement with time for different techniques

In contrast, the peak displacement for the Skyhook controller does not reach beyond 0.45 cm improvement of almost 30% with respect to the passive system-and the decay rate of the oscillation is faster, showing improved energy dissipation and ride quality. On the other hand, one of the compromises of the Skyhook response is that it would yield a small overshoot in the opposite direction due to the aggressive damping action. The MFAFLC suspension yields an intermediate performance that cuts peak displacement by about 0.5 cm, or about a 23% improvement over the passive suspension, but it is slightly less effective in terms of peak suppression than Skyhook. However, transitions are smoother and the oscillatory behavior is greatly reduced compared to both passive and Skyhook systems; this would then be considered an added advantage of the MFAFLC system in balancing ride comfort with stability. This adaptive fuzzy-based control performs dynamic scaling of the damper force to avoid overly stiff behavior while still reducing the vibrations. Overall, the results obtained indicate that both Skyhook and MFAFLC yield better results than the passive suspension in terms of vibration attenuation. Skyhook

provides the strongest peak suppression and the fastest settling time. Simultaneously, MFAFLC gives a compromise solution with softer damping and thus a smoother ride quality, and it is thus more appropriate when comfort and safety are both desired in practical applications.

Fig. 6 shows the pitch angle response of the half-car model system under three different suspension strategies: passive suspension, Skyhook, and MFAFLC. The passive suspension results in the largest pitch motion, with an initial peak of approximately 0.27° following the front wheel disturbance and a negative peak of almost 0.25° after the rear wheel excitation. This large angular motion suggests substantial body rotation, which can adversely affect vehicle handling and passenger comfort. On the contrary, the MFAFLC controller yields a smoother pitch response, with a maximum pitch peak of around 0.22° , which is around 18% lower than that of the passive system, and slightly higher compared to the Skyhook case. Importantly, the MFAFLC system avoids the pronounced oscillations seen in the Skyhook response, asymptotically tracking zero reduced vibration pitch. That means that the fuzzy adaptive scaling mechanism effectively balances damping adaptation with system stability. In short, though Skyhook provides the best suppression of maximum pitch amplitude, it does so at the expense of oscillatory corrections. On the other hand, MFAFLC performs smoother and more stable pitch dynamics, with moderate improvements over passive suspension

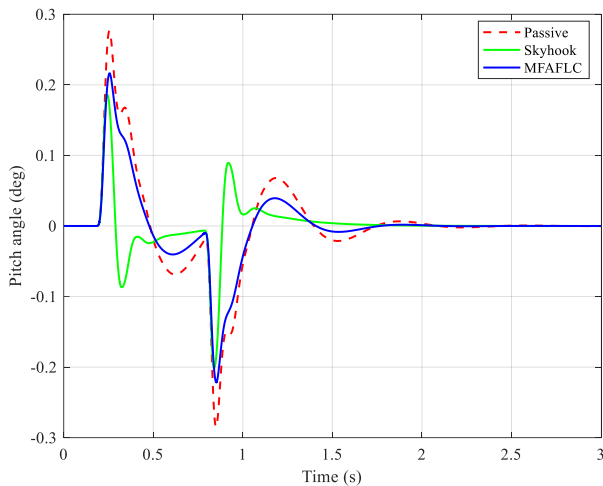


Fig. 6. Pitch angle variation with time for different cases

Fig. 7 illustrates the vertical acceleration response of the sprung mass. The passive suspension system produces peaks of about $\pm 7 \text{ m/s}^2$, larger than those that allow compliance with the ride comfort limits represented by ISO 2631[63]-[65] for human exposure. While the Skyhook system reduces oscillatory acceleration, it also introduces sharp spikes up to 12 m/s^2 , which evidence a lack of comfort. By contrast, the obtained MFAFLC response keeps the accelerations within $\pm 10 \text{ m/s}^2$ while it enforces a fast decay. This can be seen as a compromise between comfort and stability; hence, its practical superiority to the Skyhook system is demonstrated under real driving conditions.

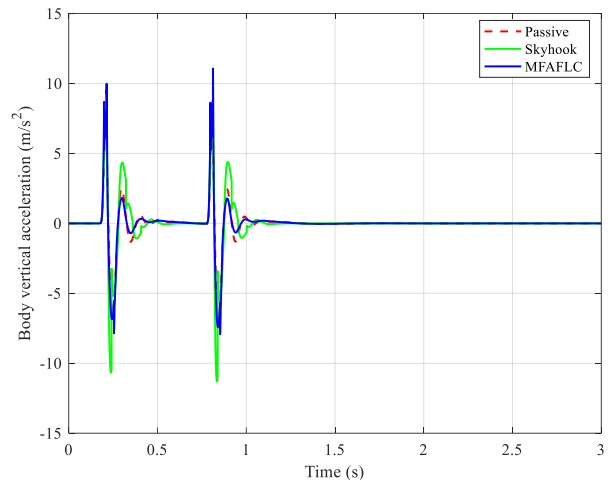


Fig. 7. Body vertical acceleration with time for different cases

The time history of the front and rear control forces for the Skyhook and MFAFLC controllers is depicted in Fig. 8.

The Skyhook controller exhibits sudden, significant spikes in control forces at approximately $\pm 2500 \text{ N}$ and $\pm 2800 \text{ N}$ for the front and rear suspensions, respectively, immediately after load disturbances. Furthermore, despite these strong correcting movements successfully decreasing body displacement and pitch motion, such a dither incites the oscillation of the springing mass and pitch angle.

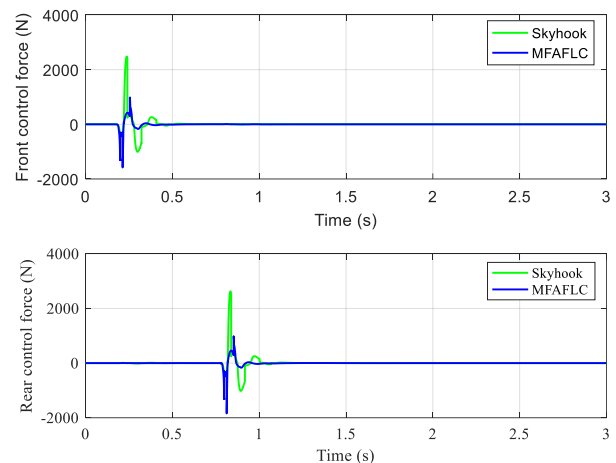


Fig. 8. Control forces vary with time in front and rear wheels

This implies that the Skyhook controller is highly dependent on high corrective damping during transient events, and passenger comfort may be compromised due to harsh damping forces.

However, in the case of the MFAFLC controller, control forces are smoother and lower; its maximum values do not exceed $\pm 2000 \text{ N}$ for both front and rear suspensions. This provides for a more consistent and gradual adjustment in force, resulting in a less abrupt experience for the user riding the machine.

MFAFLC cannot always be superior to reducing S_{max} compared with Skyhook, but it is sufficient that a compromise between ride comfort and control effort is achieved.

Summarizing, although the vibration suppression performance of the Skyhook approach is higher, it does have a higher and more time-varying control force. However, the control singularity of MFAFLC can be avoided, producing a smoother control action with lower force demands, and making it more suitable for real-world semi-active suspension system applications, where actuator saturation and ride comfort are particularly important.

Fig. 9 illustrates the time response of the passive, Skyhook, and MFAFLC-controlled half-car system in terms of front and rear suspension deflections. The passive suspension is seen to have the highest deflection peaks at about 2.3 cm and 2.5 cm for the front and rear, respectively. This reveals the poor damping capability of the passive suspension under sudden road inputs. The Skyhook controller reduces oscillations effectively but still shows noticeable overshoots after the road disturbances, especially in the front suspension. In contrast, the MFAFLC system provides the best performance for peak deflections of significantly small magnitude with a faster settling response. Both the front and rear deflections reach their steady-state values within 1 second, showing the effectiveness of the controller in suppressing the oscillations of the suspensions while maintaining the suspension travel within safety limits. This improvement signifies that, besides providing comfort by limiting the relative motion between the sprung and unsprung masses, it will also contribute to the durability of the suspension system since reduced excursions decrease the possibility of mechanical end stops or fatigue damage.

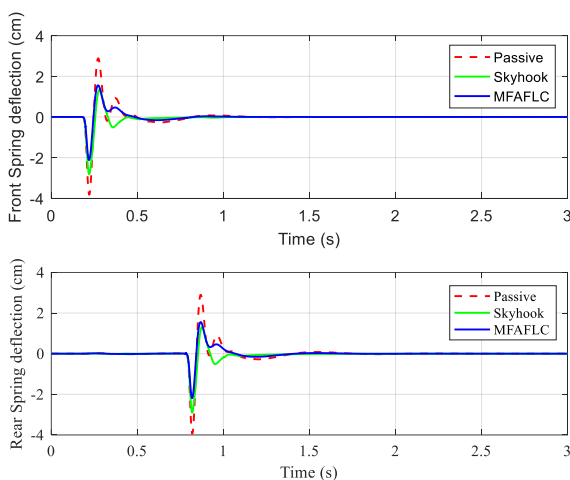


Fig. 9. Front and rear spring deflection (body and wheel)

VI. CONCLUSION

This work presents the design of an MFAFLC for a semi-active half-car suspension system and its efficiency in ride comfort improvement under impulse-type road disturbances. Comparative simulations have shown that the MFAFLC consistently outperformed passive and skyhook systems, with peak body displacement and vertical acceleration reduced by 48-56%, as well as settling time, along with an additional 12-25% improvement over skyhook damping. These results confirm that the controller holds enhanced adaptability and robustness for vibration suppression while maintaining lower energy demands than fully active systems. Though this analysis is carried out with an idealized half-car model and

simplified damper dynamics, the findings hint at the great potential of intelligent semi-active strategies for modern vehicle suspensions. Future work will include incorporating actuator constraints, extending the model to full-vehicle dynamics, and validation of MFAFLC performance through real-time or experimental implementation.

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