INTERNATIONAL JOURNAL OF AUTOMOTIVE SCIENCE AND TECHNOLOGY

2024, VOL. 8, NO:4, 439-450 www.ijastech.org e-ISSN: 2587-0963

Comparison of Control Methods for Half-Car Active Suspension System

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Research Article

Abstract

Suspension systems are of great importance in ensuring stable driving in vehicles and the appropriate reaction of vehicle sub-elements against disturbing inputs. Active suspension systems react quickly to road and driving conditions and positively affect vehicle dynamics and passenger comfort. The main factor in improving active suspension systems' performance is determining the suitable control method against the determined disturbing inputs. This study aims to contribute to the development of vehicle suspension technology by investigating the potential of optimizing the performance of active suspension systems with different control algorithms. In the study, a half-car model with an active suspension system was simulated on three different road profiles: bump-pit, sinusoidal, and ISO-8608. Fuzzy logic, PID, Fuzzy-PID, and MPC control methods are used to control the active suspension system, and their advantages over each other and the passive suspension system are investigated. The effectiveness of the control methods determined on each road profile has been analyzed in the evaluations made regarding vehicle dynamics and passenger comfort. As a result of the study, it is observed that the MPC control algorithm was able to control the active suspension system stably and quickly on all three road profiles with a high success rate. The best results are obtained by the MPC algorithm for bump pit, sinusoidal, and ISO8608 road profiles, and the RMSE values for each road profile are 1.466, 0.047, and 0.449, respectively. The suitable control method minimized vehicle body displacement and pitch angle and improved vehicle stability. In the passenger comfort evaluation, 33.49%, 47.79%, and 12.26% improvements were obtained using the MPC control method on bump-pit, sinusoidal, and ISO-8608 road profiles, respectively.

Keywords: Active Suspension System; Fuzzy Logic Control; Half-Car Model; Model Predictive Control; Vehicle Dynamics

History
Received 02.11.2024
Revised 28.11.2024
Accepted 16.12.2024

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To cite this paper: Çelik, İ., Arslan, T.A., Aysal, F.E., Bayrakçeken, H. and Oğuz, Y. Comparison of Control Methods for Half-Car Active Suspension S ystem. International Journal of Automotive Science and Technology. 2024; 8(4): 439-450. http://dx.doi.org/10.29228/ijastech..1578123

1. Introduction

Suspension systems have an important role in providing a stable ride in vehicles, minimizing the impact of road conditions on vehicle body elements, and improving driving comfort. Suspension systems are known as complex mechanical structures designed to optimize the driving dynamics of vehicles and improve road-holding [1,2]. These systems include a series of components that ensure the adaptation of the vehicle body structure and wheels to changing conditions on the road surface. From the past to the present, vehicle suspension systems have undergone various changes. The basis of these changes is MacPherson suspension systems, which are still widely used today [3].

Conventional vehicle suspension systems consist of two basic components. These are coil springs and shock absorbers. These systems contribute to vehicles in many important ways (high maneuverability, passenger comfort, driving safety, etc.). Suspension systems are classified as passive, semi-active, and active according to the type of control [4]. Passive suspension systems have fixed characteristics, and their ability to adapt to changing road conditions or vehicle speeds is limited. Therefore, they cannot respond adequately to sudden changes in the road profile. Although positive results have been obtained from the studies conducted with these systems, it has been observed that the vehicle dynamics improvements are insufficient because their responses to sudden changes cannot be improved [5-7].

Researchers have developed semi-active and active suspension systems to overcome the shortcomings of passive suspension systems. Semi-active suspensions are systems that can control the damping force of the shock absorber according to the data from the changing road surface. In these systems, many different approaches are used to control the damping force, such as orifice dampers, visco/elastic-plastic and controllable fluids [8,9]. Passive and semi-active systems are directly exposed to changes in the road surface. Despite these disadvantages, semi-active suspension systems have been able to provide significant performance improvements for vehicle ride [10-12]. However, regarding ride and passenger comfort, these improvements have not reached a sufficient level according to the "ISO-2631 Mechanical Vibration and Shock" standard [13]. For this reason, researchers have planned to develop active suspension systems that can be controlled more effectively than semi-active systems [14,15].

Active suspension systems monitor vehicle motion instantaneously through various sensors and continuously change the suspension settings based on this data. These systems can react quickly to road surface conditions and driver behavior, thus optimizing vehicle holding and stability while increasing passenger comfort [16,17]. In the literature, many studies have been conducted on increasing passenger comfort with an active suspension system based on the ISO-2631 standard [18-21]. In this study, it is aimed to improve vehicle dynamics by minimizing the pitch angle and body displacement of the vehicle. The pitch angle refers to the degree to which a vehicle's front and rear axles tilt up or down while driving. This angle is related to the vehicle's size, weight and suspension system design and is also a parameter that affects the dynamic driving characteristics. Improving vehicle dynamics is essential in terms of increasing passenger comfort, optimizing road holding, and increasing safety levels [22].

The design of active and semi-active suspension systems with the appropriate control structure and their integration into the vehicle is one of the main requirements for the efficient use of these systems. For this reason, manufacturers and researchers design active suspension systems with different control approaches. In these studies, various methods such as Proportional-Integral-Derivative (PID) [23], Linear Quadratic Regulator (LQR) [24], Linear Quadratic Integral (LQI) [25], Fuzzy Logic Control [26], Model Predictive Control (MPC) [27] are considered for the control of active suspension systems [28].

In this study, due to the complex structure of the active suspension system, several different control algorithms were used, and comparisons were made. These control algorithms are PID, Fuzzy logic, Fuzzy-PID, and MPC. The PID controller uses the error signal obtained from the difference between the current state of the system and the target value. Fuzzy logic is an artificial intelligence method for modeling and controlling uncertain systems [29,30]. The Fuzzy-PID controller combines the simplicity of the PID controller and the flexibility of Fuzzy logic. This method regulates the parameters of the PID controller with Fuzzy logic principles and thus improves the stability and performance of the system. MPC predicts the future system behavior to obtain an optimal solution to a control problem and uses these predictions to determine the optimal control action [31]. These control methods offer different approaches to optimize the suspension system's dynamics and minimize the vehicle's pitch angle and body displacement.

This study aims to support efforts to develop more effective and reliable suspension systems in future vehicle designs by evaluating the effectiveness of various algorithms to better control the dynamics of suspension systems. In this study, a half-car model with an active suspension structure is tested on three different road profiles: bump-pit, sinusoidal, and ISO-8608. The vehicle dynamics and passenger comfort parameters of this model with the developed Fuzzy logic, PID, Fuzzy-PID and MPC control methods are presented comparatively. Each control method is evaluated according to certain performance criteria, and the results are analyzed.

2. Materials and Methods

2.1. Half-car active suspension model

The half-car model has been widely used in vehicle dynamics and suspension system simulations thanks to its relatively simple and straightforward structure [32]. Half-car simulations, which include many critical features of the full-car, result in high accuracy [33,34]. The half-car model represents the longitudinal section of the full-car, which includes a front and rear suspension structure. The passive half-car model includes an unsprung mass, two springs, and a damper in the front and rear suspension structure. An actuator positioned between the vehicle body and the axle gives this system an active structure. These actuators control the suspension system by generating force in both directions vertically. The front and rear suspension structures are connected to the vehicle body, considered the sprung main mass. Active control minimizes the adverse effects of road and driving disturbances on the vehicle body and passengers [35,36]. In addition, the active control system positively affects vehicle dynamics and road holding [37]. Figure 1 shows a 4-degree-of-freedom active suspension half-car model in which the effects of different control algorithms on vehicle dynamics and passenger comfort are investigated.

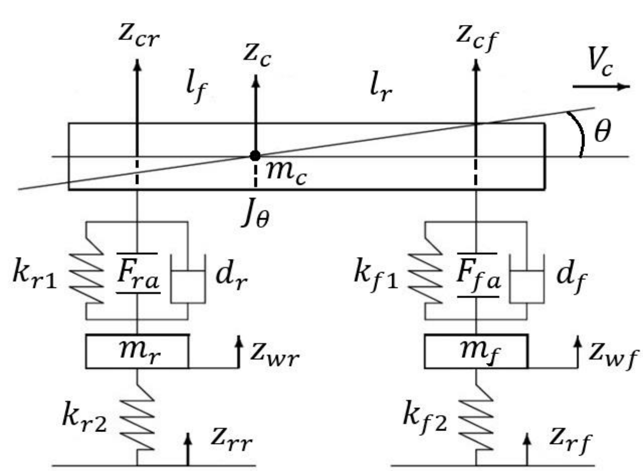

Fig. 1. Half-car active suspension model [38]

In the half-car model, the sprung vehicle's upper body mass is denoted by m_c, and the unsprung front and rear wheel masses are denoted by m_f and m_r. Here, z_c represents the displacement of the vehicle's center of gravity. The linear velocity of the vehicle is V_c. The active suspension dynamics on the front and rear axles are driven by dampers, springs and actuators positioned between the body and the wheels. The horizontal distance of the front axle to the vehicle center of gravity is defined by l_f, and the distance of the rear axle to the vehicle center of gravity is defined by l_r. The vehicle's upper body's pitching motion during its movement is expressed as the angular deflection of the center of gravity and is denoted by θ. The angular moment of inertia of the vehicle body due to the pitching motion is J_θ. In the half-car model, the spring constants, damping constants and actuator control forces of the front and rear active suspension systems are defined as k_{f1}, k_{r1}, d_f, d_r, and F_{fa}, F_{ra}, respectively. In addition, the wheel displacements at the front and rear axles are denoted by z_{wf} and z_{wr}, respectively. The spring constants of the vehicle wheels are k_{f2} and k_{r2} for the front and rear axles, respectively. Finally, the change in the road profile to which the front and rear wheels are subjected is represented by z_{rf} and z_{rr}.

The linear equations of motion for the sprung and unsprung masses of the system are determined using Newton's second law. Eq. (1) and Eq. (2) show the equations of motion for the unsprung front and rear wheel masses respectively. Eq. (3) presents the equation of motion for the sprung vehicle body.

$$m_f \cdot \ddot{z}_{wf} = k_{f2} \cdot (z_{rf} - z_{wf}) + F_{fa} - k_{f1} \cdot (z_{wf} - z_c - l_f \cdot \theta) - d_f \cdot (\dot{z}_{wf} - \dot{z}_c - l_f \cdot \dot{\theta}) \quad (1)$$

$$m_r \cdot \ddot{z}_{wr} = k_{r2} \cdot (z_{rr} - z_{wr}) + F_{ra} - k_{r1} \cdot (z_{wr} - z_c + l_r \cdot \theta) - d_r \cdot (\dot{z}_{wr} - \dot{z}_c + l_r \cdot \dot{\theta}) \quad (2)$$

$$m_c \cdot \ddot{z}_c = k_{f1} \cdot (z_{wf} - z_c - l_f \cdot \theta) + d_f \cdot (\dot{z}_{wf} - \dot{z}_c - l_f \cdot \dot{\theta}) - F_{fa} + k_{r1} \cdot (z_{wr} - z_c + l_r \cdot \theta) + d_r \cdot (\dot{z}_{wr} - \dot{z}_c + l_r \cdot \dot{\theta}) - F_{ra} \quad (3)$$

In order to complete the equations of motion for the half-car, the pitching motion of the vehicle's upper body must also be modeled. Eq. (4) shows the angular acceleration equation expressing the pitching motion of the vehicle body.

$$J_\theta \cdot \ddot{\theta} = k_{f1} \cdot (z_{wf} - z_c - l_f \cdot \theta) \cdot l_f + d_f \cdot (\dot{z}_{wf} - \dot{z}_c - l_f \cdot \dot{\theta}) \cdot l_f - F_{fa} \cdot l_f - k_{r1} \cdot (z_{wt} - z_c + l_r \cdot \theta) \cdot l_r - d_r \cdot (\dot{z}_{wr} - \dot{z}_c + l_r \cdot \dot{\theta}) \cdot l_r + F_{ra} \cdot l_r \quad (4)$$

2.2. Vehicle parameters and road conditions

Vehicle parameters are important factors that affect the vehicle's dynamic performance and driving characteristics [39,40]. These include vehicle weight, wheelbase, tire and suspension characteristics. In the simulations, the values of a passenger vehicle used in the city are taken as the basis for the dimensional and mass parameters. An approach is developed by taking into account the different values in the literature for the suspension system and tire values [41-43]. The constant vehicle parameters used in the half-car active suspension model are shown in Table 1.

Table 1. Vehicle parameters

Parameters	Symbols	Values	Units
Half-car weight	m_c	495.5	kg
Front and rear wheel weight	m_f-m_r	28.05	kg
Distance of the front axle from the center of gravity	l_f	1.1	m
Distance of the rear axle from the center of gravity	l_r	1.47	m
Front and rear suspension spring constants	k_{f1}-k_{r1}	15000	N/m
Front and rear suspension damping constants	d_f-d_r	1828	Ns/m
Front and rear tire spring constants	k_{f2}-k_{r2}	155900	N/m
Vehicle speed	V_c	8.65	m/s

In order to compare the control methods examined in this study, simulations were performed using three different road profiles. In the first stage of the simulations, the bump and pit conditions created using the unit step function were analyzed. The bump and pit values of the unit step function are 0.1 m and -0.1 m, respectively. A sinusoidal bump-pit structure is used for the second road profile considered. The peak-to-peak distance was set to 0.2 m and the period to 22 s for the sine signal. Finally, simulations were completed on a road profile created in accordance with the ISO-8608 standard in order to consider real road conditions. The ISO-8608 standard provides formulas to express the vertical roughness of the road as a signal in meters. The process of obtaining the road profile is mainly based on the Power Spectral Density (PSD). The ISO-8608 standard provides a random road profile signal for a total of seven different road qualities [44]. These road classes are determined by expressions deriving the roughness of the road surface in Eq. (5) and Eq. (6) [45].

$$F_d(\Omega) = F_d(\Omega_0) \cdot (\Omega/\Omega_0)^{-w} \quad (5)$$

$$F_d(\Omega_0) = 4^b \times 10^{-6} \quad (6)$$

Where F_d is the displacement PSD, Ω is the spatial frequency, Ω_0 is the reference spatial frequency, w is the waviness index, and b is the road class. According to the classification in "ISO-8608 Mechanical Vibration Road Surface Profiles" standard, D class road profile is considered in the study. In the literature, "Shaping Filter" and "Sinusoidal Approximation" methods are widely used to obtain road profiles in accordance with ISO-8608 standard [46,47]. In this study, the "Sinusoidal Approximation" method was used to create a road profile in accordance with the standard. The road profiles used in the study are shown in Figure 2.

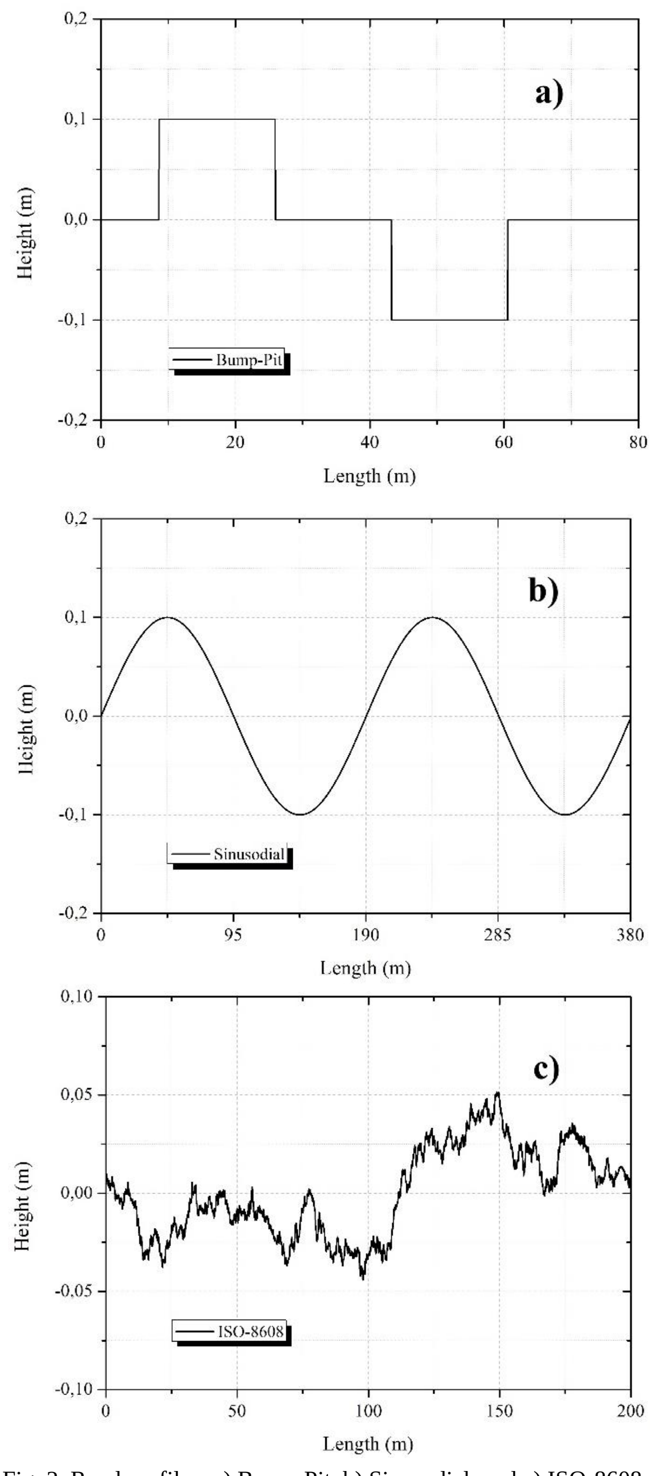

Fig. 2. Road profiles: a) Bump-Pit, b) Sinusodial, and c) ISO-8608

2.3. Control algorithms

The control algorithms control the force that the active suspension system will generate in response to the road profile. In the study, the maximum force that the suspension system can generate is determined as 2500 N. Studies for similar vehicle characteristics and road profiles have observed that the maximum actuator force is limited to the range of 2000-3000 N [48,49]. Within these limits, the duration and the moment of application of the reaction force of the suspension system to the road profile varies with the control algorithms. This paper analyzed the performances of PID, Fuzzy-PID, Fuzzy logic and MPC algorithms on a half-car active suspension system.

2.3.1. PID control

PID control is frequently used in the automotive industry for different purposes [50-53]. The error value between the reference value and the instantaneous actual value of the system is the input signal for the PID control algorithm. Eq. (7) shows the output signal of the PID as a function of the input signal.

$$F(t) = K_P e(t) + K_I \int e(t)dt + K_D \frac{de(t)}{dt} \tag{7}$$

K_P, K_I ve K_D are the proportional, integral and derivative coefficients, respectively. These control parameters are calculated using the Ziegler-Nichols reaction curve method. Table 2 shows the PID control parameters.

Table 2. PID controller parameters

Parameters	**Values**
K_P	26096
K_I	0.3198
K_D	26.6

The PID control algorithm generates an output signal to zero the input signal. The same control algorithm is used for the half-car model's front and rear wheels. Figure 3 shows the block diagram of the PID-controlled active suspension system.

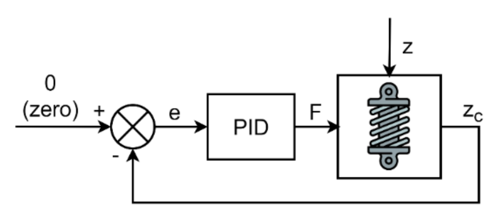

Fig. 3. Active suspension system with PID controller

2.3.2. Fuzzy logic control

In Fuzzy logic applications, the input-output parameters, membership functions and control rules must be set correctly by the expert [54,55]. In this study, the error value of the amount of vertical displacement on the front and rear axles of the vehicle and the time-dependent change of this error value are used as Fuzzy logic inputs. The controller's output signal is the force reference of the actuator of the active suspension system. The input and output membership functions consist of five trapezoidal functions. The boundary values of the error value in the membership functions are set between [-1, 1], and the boundary values of the error change are set between [-5, 5]. Figure 4 shows the block diagram of the active suspension system controlled by Fuzzy logic.

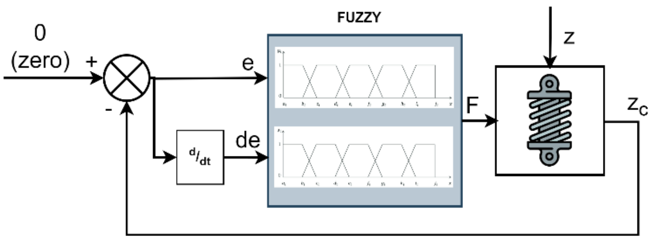

Fig. 4. Active suspension system with Fuzzy logic controller

The elaboration of Fuzzy logic rules is based on the general knowledge of the displacement of the vehicle body. Based on this knowledge, possible combinations of input data must be associated with an output value. The relationships between this input and output are shown in Table 3. The output signal is generated for each input signal by evaluating the relationships in the table. Fuzzy logic input and output parameters are denoted by PL Positive Large, PS Positive Small, Z Zero, NS Negative Small and NL Negative Large.

Table 3. Rule table for the Fuzzy logic controller

Rules		**Change of Error**				
		NL	**NS**	**Z**	**PS**	**PL**
Error	**NL**	NL	NL	NS	NS	Z
	NS	NL	NS	NS	Z	PS
	Z	NS	NS	Z	PS	PS
	PS	NS	Z	PS	PS	PB
	PL	Z	PS	PS	PB	PB

2.3.3. Fuzzy-PID control

The Fuzzy-PID method is an alternative approach to controlling the active suspension system. The control approach where PID coefficients are determined using Fuzzy logic is known as Fuzzy-PID [56-58]. In this method, K_P, K_I, and K_D parameters are determined at the output layer in response to the input signals of the Fuzzy logic controller. Figure 5 shows the block diagram of the active suspension system controlled by Fuzzy-PID.

Five triangular functions are used in the input layer of the controller, and seven triangular functions are used in its output layer. Table 4 shows the relationships between each output data and input data. The boundary values of the error value in the membership functions are set between [-0.5, 0.5], and the boundary values of the error change are set between [-0.5, 0.5]. The control parameters K_P, K_I and K_D coefficients take values in the range of [0, 90000], [0.0001, 0.006] and [0.1, 1.5], respectively. Fuzzy-PID input parameters are denoted by PL Positive Large, PS Positive Small, Z Zero, NS Negative Small and NL Negative Large. The output parameters are VL Very Large, L Large, ML Medium Large, M Medium, MS Medium Small, S Small, and VS Very Small.

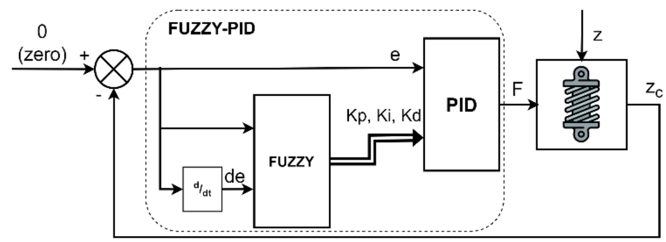

Fig. 5. Active suspension system with Fuzzy-PID controller

Table 4. Rule tables for the Fuzzy-PID controller

Rules for K_P		**Change of Error**				
		NL	**NS**	**Z**	**PS**	**PL**
Error	**NL**	VL	VL	VL	VL	VL
	NS	ML	ML	ML	ML	ML
	Z	VS	VS	S	MS	MS
	PS	ML	ML	ML	M	M
	PL	VL	VL	VL	VL	VL

Rules for K_I		**Change of Error**				
		NL	**NS**	**Z**	**PS**	**PL**
Error	**NL**	M	M	M	M	M
	NS	MS	MS	MS	MS	MS
	Z	S	S	VS	S	S
	PS	MS	MS	MS	MS	MS
	PL	M	M	M	M	M

Rules for K_D		**Change of Error**				
		NL	**NS**	**Z**	**PS**	**PL**
Error	**NL**	VL	VL	VL	VL	VL
	NS	MS	MS	MS	MS	MS
	Z	S	S	VS	S	S
	PS	MS	MS	MS	MS	MS
	PL	M	M	M	M	M

2.3.4. Model predictive control

Unlike other control algorithms, MPC uses a mathematical model of the system to control the system dynamics better. This method predicts a future control action and updates this prediction in real-time, taking into account the dynamics and limits of the system [31,59]. There are several important parameters for this control method. These are the prediction horizon, the control horizon, and the cost function. The objective of the control algorithm is to minimize the cost function calculated using the errors of the prediction and control horizon. Eq. (8) shows the cost function formula.

$$J = \sum_{i=1}^{n_y} (r_{k+i} - y_{k+i})^2 + \sum_{i=0}^{n_u-1} W(u_k - u_{ss})^2 + W_d \Delta u_k \qquad (8)$$

J is the cost function, r is the reference signal, y is the output signal, u is the input signal, and W is the weights. For this control algorithm, a mathematical model representing the dynamics in the system must first be created. This model predicts a future control action and selects the optimal control action according to the optimization problem. The optimal control motion is implemented in real-time in the last step. Figure 6 shows the MPC strategy for an active suspension system.

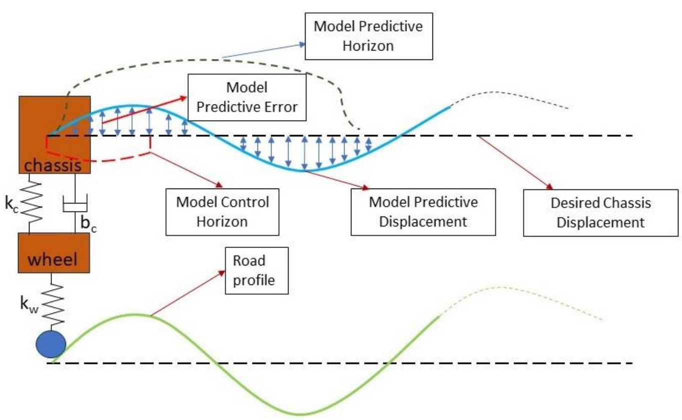

Fig. 6. MPC strategy for the active suspension system

3. Results and Discussions

In the present study, the control of the half-car active suspension system is achieved by using different control algorithms, and the system performances are comparatively analyzed in different road profiles. Four different controllers, PID, Fuzzy logic, Fuzzy-PID, and MPC, were used to control the active suspension system. In order to observe the active suspension characteristics and controller performances most effectively, simulations were carried out on three road profiles: bump-pit, sinusoidal, and ISO-8608. As a result of the research, the effects of controller performances on vehicle dynamics and passenger comfort were analyzed. In order to analyze the effects of control methods on vehicle dynamics, body displacement and pitch angle variation were considered. In evaluating passenger comfort, the Root Mean Square Error (RMSE) value for the vehicle body acceleration was calculated, and the percentage of improvement obtained with each control algorithm was determined. The passive suspension system was used as a reference to determine the improvement amounts. RMSE and percentage improvement values obtained from vehicle body acceleration for all road profiles and control algorithms are shown in Table 5.

Table 5. Vehicle body acceleration RMSE values and percentage improvements

Comfort Evaluation	Bump-Pit		Sinusodial		ISO-8608	
	RMSE	Imp. (%)	RMSE	Imp. (%)	RMSE	Imp. (%)
Passive	2.205	-	0.089	-	0.512	-
Fuzzy Logic	1.869	15.24	0.067	25.18	0.491	4.10
Fuzzy-PID	1.723	21.83	0.074	17.74	0.453	11.61
PID	1.529	30.63	0.076	14.97	0.473	7.61
MPC	1.466	33.49	0.047	47.79	0.449	12.26

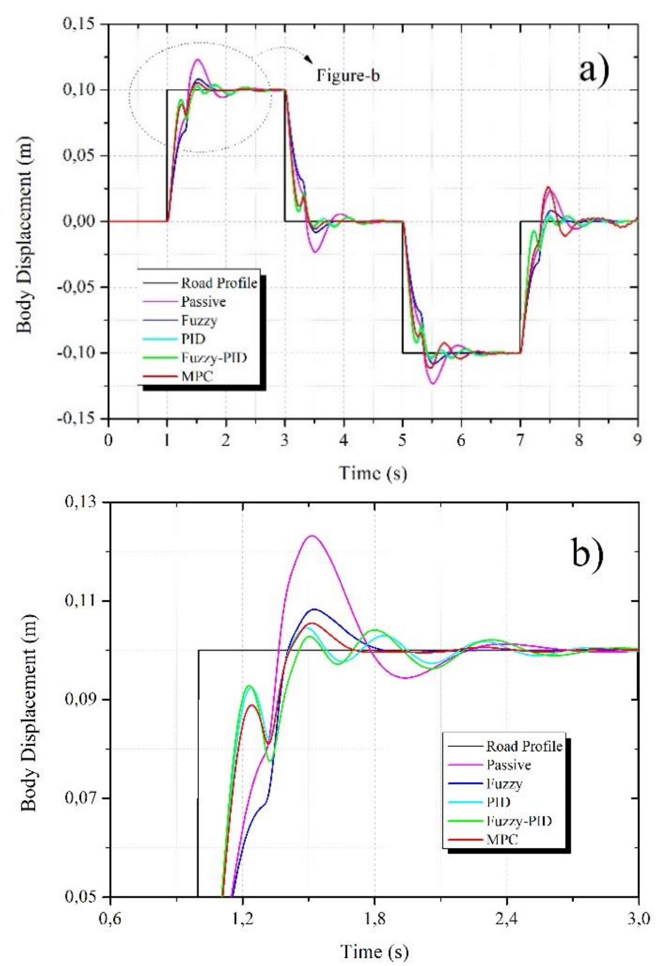

Fig. 7. Variation of vehicle body displacement on bump-pit road profile

The effects of different control algorithms on the body displacement and pitch angle change in the bump-pit road profile are shown in Figures 7 and 8, respectively. When the vehicle body displacement results are analyzed, it is clearly seen that all active suspension control approaches provide better results than the passive suspension system. The amount of overshoot, settling, and rise times are important in evaluating the control algorithm performance [60,61]. When the overshoot amounts of the body displacement are analyzed, it is seen that the active suspension overshoot amounts are less than half of the passive system. The lowest overshoot amount was found in the active suspension structure using the Fuzzy-PID controller. Considering the settling time, it is seen that the best result belongs to MPC. There was no significant difference between the rise times of the control algorithms. However, when the controller performance is evaluated, it is noteworthy that the settling time of the Fuzzy-PID control algorithm is quite long, even though the overshoot amount is small.

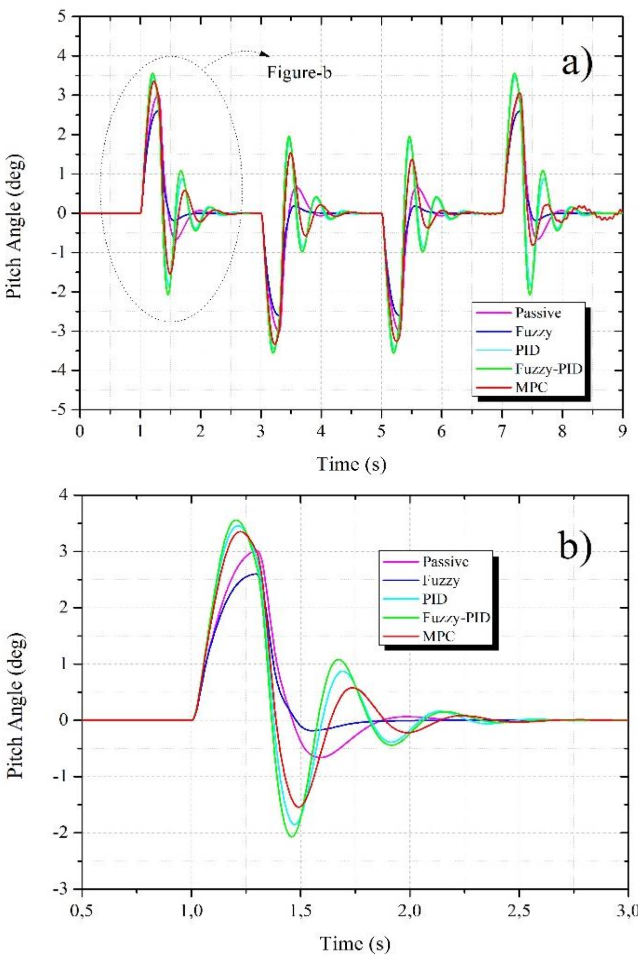

Fig. 8. Variation of pitch angle on bump-pit road profile

When the pitch angle variation is analyzed, it is seen that the Fuzzy logic control method works successfully on the front and rear wheels simultaneously and positively affects the vehicle dynamics. In other control algorithms, vehicle dynamics are adversely affected by pitch angle values that result in higher values than the passive suspension system. When the vehicle body acceleration RMSE is calculated for passenger comfort evaluation, the best results are obtained as 1.466 and 1.529 for MPC and PID, respectively. The RMSE value for passive suspension was calculated as 2.205. The MPC control algorithm was the best improving algorithm, with a rate of 33.49% in the bump-pit road profile.

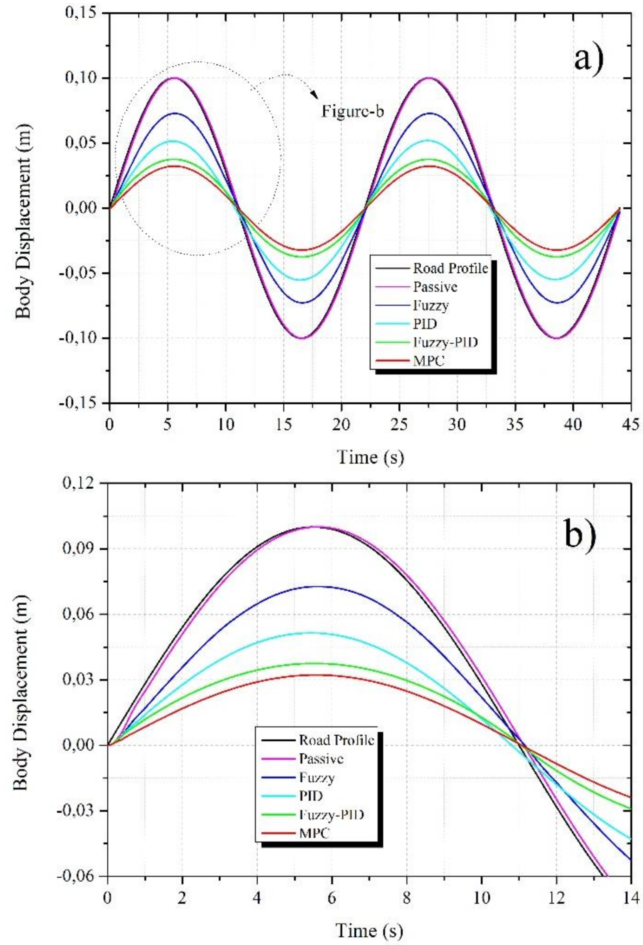

Fig. 9. Variation of vehicle body displacement on sinusodial road profile

The effects of the control algorithms on the body displacement and pitch angle change in the sinusoidal road profile are shown in Figures 9 and 10, respectively. In the second road profile, vehicle body displacement and pitch angle were minimized using the MPC control algorithm. When the changes in these parameters are analyzed, MPC is followed by Fuzzy-PID, PID and Fuzzy logic controllers, respectively. The MPC control algorithm stands out with its low overshoot and fast response capability. In the pitch angle variation, it is observed that the Fuzzy logic and Fuzzy-PID control algorithms have sudden angle changes with the peak. This is because the Fuzzy control structure's membership functions and rule tables are created for general purposes. With the MPC control method, the front and rear suspensions reached the appropriate operating conditions simultaneously and predictably in the sinusoidal road profile. When the vehicle body acceleration RMSE was calculated for passenger comfort assessment, the best results were obtained as 0.047 and 0.067 for MPC and Fuzzy logic, respectively. The RMSE for passive suspension was calculated as 0.089. The MPC control algorithm obtained a maximum improvement of 47.79% in the sinusoidal path profile.

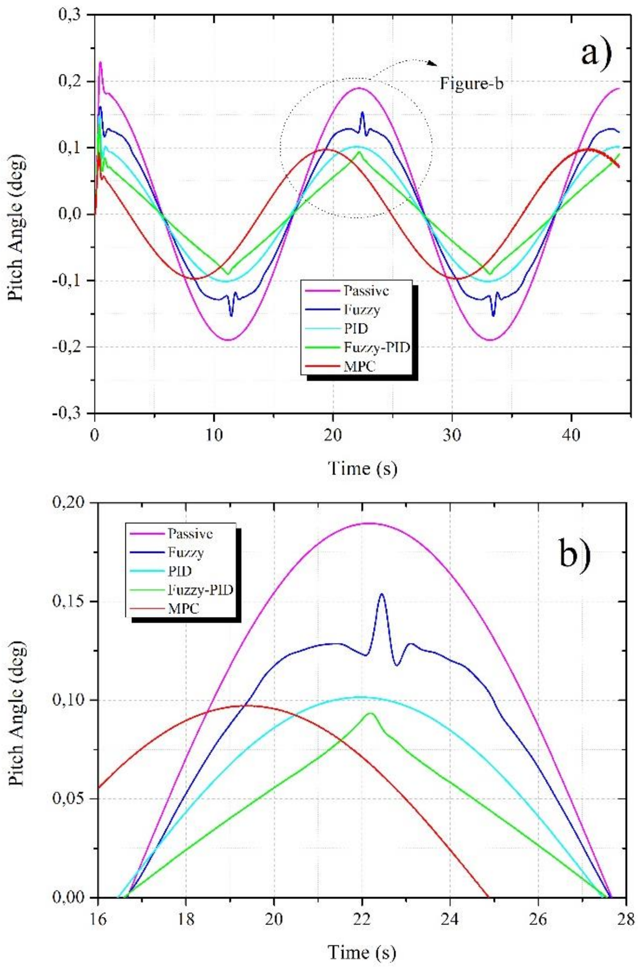

Fig. 10. Variation of pitch angle on sinusoidal road profile

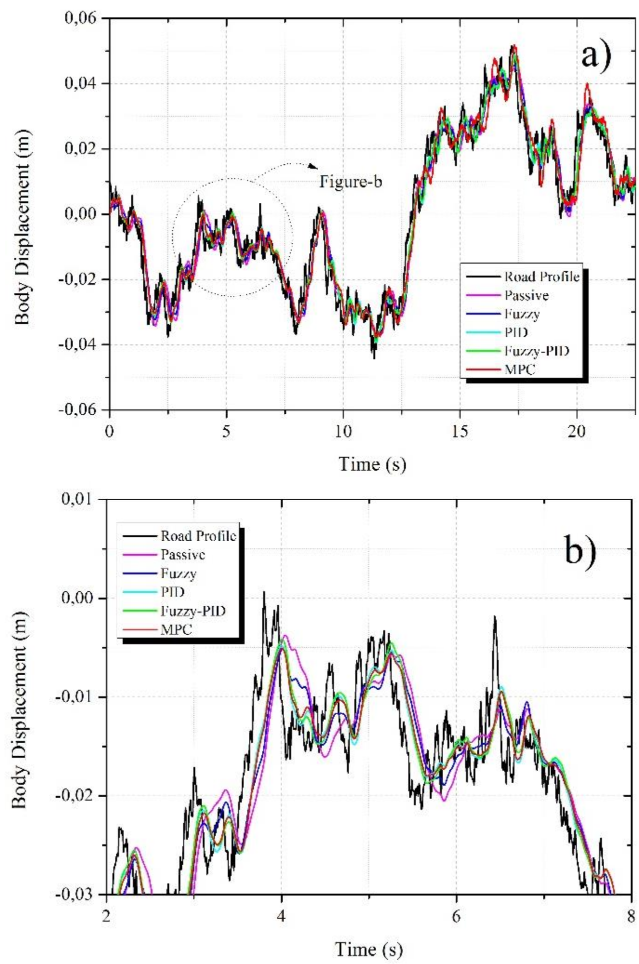

Fig. 11. Variation of vehicle body displacement on ISO-8608 road profile

The effects of the control methods on the body displacement and pitch angle changes on the ISO-8608 road profile are shown in Figures 11 and 12. Considering the overshoot of body displacement and pitch angle variation, all active suspension control methods positively affect vehicle dynamics. The lowest overshoot was obtained with the MPC control algorithm in the third road profile. Similarly, the body displacement was minimized using the MPC algorithm. There was no significant difference in the rise times as in the other road profiles. Fuzzy logic for body displacement and PID controllers for pitch angle variation did not perform as expected on the ISO-8608 road profile due to the long response time. When the RMSE of the vehicle body acceleration is calculated for passenger comfort, the best results are obtained as 0.449 and 0.453 for MPC and Fuzzy-PID, respectively. The RMSE for passive suspension was calculated as 0.512. The MPC control algorithm achieved the highest improvement value of 12.26% on the ISO-8608 road profile. The Fuzzy-PID control algorithm ranked second with a comfort improvement of 11.61%.

As a result, all active suspension control methods used in the study positively affect vehicle dynamics and passenger comfort at different rates. Due to the limited response capability of the passive suspension system to disturbance inputs, high amplitude and oscillation occur on the body displacement, which negatively affects these parameters. Based on vehicle dynamics and passenger comfort, optimum results in all road profiles were achieved with the active suspension system with the MPC controller. The MPC control algorithm stands out with its fast response time, adaptability to disturbance inputs, low overshoot and settling time.

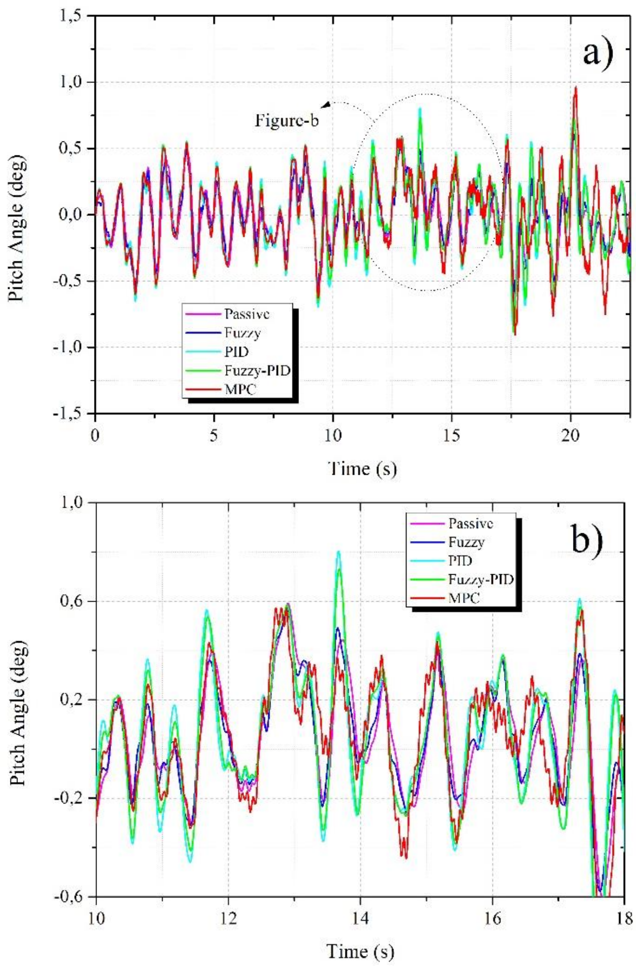

Fig. 12. Variation of pitch angle on ISO-8608 road profile

4. Conclusion

This study uses Fuzzy logic, PID, Fuzzy-PID and MPC control methods for active suspension control on a half-car model. Modeling, controlling and simulating a half-car active suspension system are carried out. The simulation results for three different road profiles, namely bump-pit, sinusoidal and ISO-8608, are analyzed and compared with the passive suspension system and each other. It is observed that all control algorithms stabilize the suspension system, provide significant improvements in passenger comfort, and positively affect vehicle dynamics. The study obtained very successful results with the MPC control algorithm in controlling the half-car active suspension system. On the other hand, the Fuzzy logic control algorithm was observed to improve vehicle dynamics and stability in the bump-pit road profile, which can be characterized as a pulse signal and sinusoidal road disturbances with high frequency. In the third road profile, where the standard defines random road disturbances, it is observed that the active suspension system using the Fuzzy-PID controller offers advantages in passenger comfort with results close to MPC. In the comfort evaluation calculated with the RMSE of the vehicle body acceleration, the MPC control method improved by 33.49% in the bump-pit road profile, 47.79% in the sinusoidal road profile and 12.26% in the ISO-8608 road profile. As a result of the research, it was determined that the MPC control algorithm, which is a new approach to active suspension control, will contribute to the development studies in this field with its fast and stable control capability. In the future, developing active suspension systems on multi-axle full-car models will be useful using different disturbance inputs and control methods.

Conflict of Interest Statement

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

CRediT Author Statement

İbrahim Çelik: Software, Methodology, Writing-original draft, Conceptualization,
Turan Alp Arslan: Methodology, Writing-original draft, Investigation, Visulation,
Faruk Emre Aysal: Methodology, Writing-original draft, Investigation, Writing - review & editing
Hüseyin Bayrakçeken: Methodology, Investigation, Supervision,
Yüksel Oğuz: Methodology, Investigation, Supervision.

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