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Implementation of an Asymmetric Actuator Distribution in a Series Active Variable Geometry Suspension

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A B S T R A C T

Today, a large part of a vehicle's performance depends on its suspension. These expectations are addressed in this paper, including ride comfort, road-holding, and lateral stability. Due to the high statistics of lateral overturning, preventing lateral overturning and providing lateral stability of the vehicle is one of the most important goals of this paper. In this paper, a new type of suspension based on the Series Active Variable-Geometry is used by designing a simple Sliding Mode Controller (SMC) to improve vehicle dynamics. On the contrary previous studies in this field, asymmetric distribution of control command has been used to increase the usefulness of suspension in standard road roughness and during longitudinal and transverse maneuvers. In this paper, by simulating crosswind and double lane change maneuvers, several ideas have been used to command the suspension links, and a 25% to 30% improvement in vehicle dynamic performance parameters has been achieved.

1. Introduction

Given the trend in the vehicle market in recent years, a turnover of up to about $85 billion can be considered for the vehicle suspension [1]. This turnover raises expectations to meet the needs of easy driving as well as the ride comfort for luxury vehicles. It is also important to have a sticky road holding and high acceleration in ordinary passenger and military vehicles [2].

On the other hand, vehicle safety and stability are some of the most important needs of customers [3]. This is all the more important given the high rate of overturning and injuries [4]. In recent years; extreme casualties have been reported due to the overturning of vehicles. These overturn are for various road, human, and vehicle operators [5]. One way to reduce the possibility of lateral overturning and provide lateral stability is study the quality of the vehicle's suspension [6]. In this study, an attempt has been made to control the safe lateral stability area of the vehicle and keep it at an acceptable level. This has been done based on the study of acceleration parameters and the rolling angle of the vehicle.

One of the most up-to-date and new systems in the field of active suspensions is a series active variable-geometry suspension (SAVGS). In this type of suspension, instead of using an actuator with very high energy consumption [7], the suspension geometry is designed to absorb vibrations by changing the position and angle of links, and suspension duties are provided. Most of the research in these systems is related to ride comfort and road holding [8].

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This suspension system, which belongs to the group of active systems, has been used in light passenger vehicles, small heavy vehicles (large heavy vehicles are prone to failure of suspension components due to their heavyweight), but it has not been studied widely in off-road vehicles and military vehicles as well as snowplows.

The SAVGS system has been required by a group of researchers based on the model presented by Arana [9], and in recent years studies have been conducted to measure the efficiency of this suspension model in different vehicles to meet expectations the vehicle. In these studies, the SAVGS system in uneven driving conditions at a constant speed with H infinity control has been studied to improve the ride comfort [10, 11]. In the present study, an attempt has been made to match the ride comfort of the mentioned authorities. These studies are high quality with road roughness. Automotive prototypes have also been used to enhance the dynamic model. Most researchers focus on modeling and try to make the operator used to perform better through experimental and simulation methods [12].

The general concept of the sliding mode controller is based on the design of the sliding surface to optimize the control cost function and forming an oscillating response of the output to the desired state [13]. This nonlinear controller has interesting features such as accuracy, robustness, and easy adjustment and implementation [14] . The main basis of this controller is based on the implementation and design of the sliding surface to be directed to the designed surface using the control roles [15]. Past studies indicate the use of a specific type of optimal sliding mode controller in active suspensions [16, 17]. Studies also show that a sliding mode controller is used in active suspension with cost function resulting from suspension displacement, vehicle tire deflection and vehicle sprung mass velocity [18]. There are also studies using a sliding surface technique to assess ride comfort in a linearized quarter-vehicle system [19].

In previous studies, an attempt was made to the issue of lateral stability of the vehicle and prevent its lateral overturning based on active differential braking system [20], To improve its structure and control method [21], active and semi-active suspension [22] by focusing on determining a beneficial control strategy [23] and together with active suspension and differential braking [24].

In most of these references, there is simulation limit of the vehicle model, and often simple models of a quarter car have been used. Also, the control strategy is based on limited methods and a comprehensive study in the field of actuator has not been done. As most studies have been done in conditions of poor roughness. In this study, an attempt has been made to cover the mentioned limitations as well as also utilized the study purposes mentioned in previous studies for this paper.

In this paper, a new useful sliding mode control method is used to achieve lateral stability of the vehicle. This control method was simply implemented in previous studies. In this study, an attempt has been made to increase the efficiency of the control method and also to investigate the implementation of the suspension system in the vehicle. For this purpose, an attempt has been made to use two types of lateral maneuvers to increase the challenge of the lateral overturning of the vehicle and double lane change to measure the efficiency of suspension on a rough road. In addition, all the parameters of passenger ride comfort and road holding of the vehicle have been considered and an attempt has been made to improve each one. In the previous study [25], it was shown that the use of this control method and suspension system is very beneficial. In this study, an attempt has been made to consider a challenge for the suspension actuator.

Our contributions are as follows:

- Design and implementation of fuzzy sliding mode controller in series active variable-geometry suspension
- Improving the vehicle dynamics by using a new type of suspension and control method in asymmetric roughness and indirect driving maneuvers
- Achieve a beneficial process in the variable geometry suspension actuator for passenger vehicles
- Prevent vehicle overturning and provide lateral stability while improving road-holding and ride comfort

For this purpose, first in section 2, a complete mathematical model of the SAVGS system mounted and the full vehicle model is presented. After validating the available model in this section, road roughness and driving maneuvers have also been modeled.

Then, in section 3, the control strategy of a simple fuzzy sliding mode controller will be discussed. The reasons for the asymmetric distribution of the control command and how to implement it are also described.

Then, examples of the results of the study will be presented and in section 4, the summary of the achievement of the article will be expressed in the last section.

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2. Modeling

To study the expected effects of road roughness and driving maneuvers in the intended passenger vehicle, a comprehensive dynamic model of the vehicle is modeled by taking into account all the coordinates of motion. In this modeling, a full vehicle model is used along with the SAVGS, and after extracting the equations of motion and vibration for one corner of the self and extending it to the whole model, a state-space model is reached. In the mentioned modeling that is shown in Figure 1, the dependence between the horizontal and vertical vehicle dynamics is ignored and also the assumption of permanent tire contact with the road surface, not considering the horizontal and transverse forces of the tires and also the turning torque of the tires, elimination of gearbox dynamics and vehicle transmission system, vehicle steering system and aerodynamics are also used to simplify the mathematical model [26].

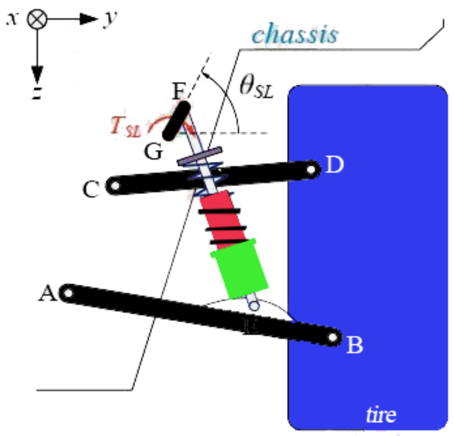

Figure 1: Schematic of the SAVGS suspension [26]

Finally, the vehicle model and suspension can be simulated with a state space with 10 inputs and 14 state variables, which in (1), input and output vectors and state variables are shown. Full details on how to extract state-space equations are given in the reference [26, 27].

(1)

$$\dot{x} = Ax + Bu$$
$$y = Cx + Du$$

$$u = [T_p, T_r, z_{r1}, z_{r2}, z_{r3}, z_{r4}, \dot{\theta}_{SL1}, \dot{\theta}_{SL2}, \dot{\theta}_{SL3}, \dot{\theta}_{SL4}]^T$$

$$y = [\Delta l_{t1}, \Delta l_{t2}, \Delta l_{t3}, \Delta l_{t4}, \ddot{z}_{s1}, \ddot{z}_{s2}, \ddot{z}_{s3}, \ddot{z}_{s4}, \ddot{Z}, \ddot{\theta}, \ddot{\varphi}, \Delta l_{s1}, \Delta l_{s2}, \Delta l_{s3}, \Delta l_{s4}]^T$$

$$x = [z_1, z_2, z_3, z_4, \theta, \varphi, z_{t1}, z_{t2}, z_{t3}, z_{t4}, \Delta l_{s1}, \Delta l_{s2}, \Delta l_{s3}, \Delta l_{s4}]^T$$

As can be seen, the full model of the present vehicle has 14 degrees of freedom, and by removing the dynamics of the actuator and the gearbox and its accessories, it has used the rotational speed of the suspension links as the system input. Assuming that the PMSM rotary actuators, which generate the T_{SAVGS} rotational torque at the end of the link added to the double-wishbone suspension, are Equivalent to a rotational speed and simulated (see Figure 1).

Dynamic system model inputs include vertical four-wheel-drive excitations (4 inputs) along with the vehicle's overall rolling and Swing torques (2 inputs) and the rotational speed of rotating links (4 inputs), each of which will be described by design. Also, the numerical values used to simulate the vehicle model are shown in Table 1.

Table 1: Parameters of the model [26]

Number	Parameter	Value	Unit
1	Wb	2.6	*m*
2	h_{CM}	450	*mm*
3	m / m_s	1325/1525	*kg*
4	W_{dis} (F/R)	57/43	%
5	k_s (F/R)	92/158	*N/mm*
6	k_t (F/R)	275	*N/mm*
7	T (F/R)	1.405/1.43	*m*
8	I_{xx} / I_{yy}	300/1500	*kg.m*^{2}
9	m_{us} (F/R)	47.5/52.5	*kg*
10	K_Φ / C_Φ	189506/6364	*N/rad* *N.s/rad*

Equation 2 has been used to generate signals related to vehicle torque [12].

(2)

$$T_{p(1,2)} = -\Delta l_{s1} (or\ \Delta l_{s2}) \times K_{sf} \times wheelbase \times Weight\ distribution$$
$$T_{p(3,4)} = -\Delta l_{s3} (or\ \Delta l_{s4}) \times K_{sr} \times wheelbase \times Weight\ distribution$$

It should be noted that indexes 1 and 3 are used for the front and rear wheels on the left and indexes 2 and 4 are used for the right side of the vehicle. Using (3), the pitch torque of the vehicle can be calculated at any time.

(3)

$$T_p = T_{p1} + T_{p2} - T_{p3} - T_{p4}$$

Similarly, for roll acceleration of the vehicle can calculate the amount of rolling torque. It is worth mentioning the details of the calculations of these torques are available in [25].

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To simulate the input of vertical road excitations, the standard model of the random road at constant speed ISO 8606: 1995 has been used [10, 28, 29]. In this paper, a very severe asymmetric roughness is used to evaluate the performance of the variable geometry suspension. In previous studies, all articles focused on the type A road profile. In this paper, in this article, type B road profile via 80 km/h speed of the vehicle is used for both sides of the vehicle, assuming a time delay on the right. This delay will cause severe asymmetric roughness. Figure 2 shows the diagram of vertical excitations on the left front tire of the vehicle.

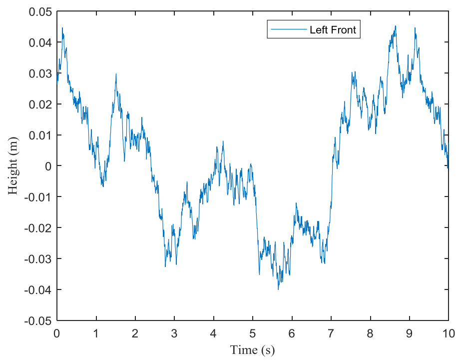


Figure 2: Road disturbance on type B ISO random road

Also, to study the vehicle dynamics and improve performance, double lane change and crosswind maneuvers have been used.

Full details of the design of driving maneuvers are available in [25] and the generalities of each are described below.

In the implementation of lateral crosswind maneuver, a wind with variable wind speed is used along with the driving distance. Depending on the type of vehicle, very uneven road conditions have a maximum wind speed of 35 km/h (which is a reasonable and repetitive amount in real city-wide driving) and it is also assumed that wind is applied to the vehicle from only one side.

In this paper, the double lane change maneuver is used on a road or friction coefficient 1, assuming strict command. For this purpose, a large field without roughness with a short path and high angle barrier arrangement has been used to design the maneuver. Also, according to the reference sources, a direct route maneuver with a speed of 80 km/h has been performed.

As mentioned in section 2, the dynamic model used to conduct studies in this article uses references [26, 30]. This initial model is based on laboratory modeling and practical tests and is the founder of subsequent studies in this field, some examples of which were mentioned in section 1.

This means that the prototype is self-validating and provides accurate and reliable answers. Since the system inputs (vehicle pitch and rolling torques) are calculated in a way other than that mentioned in the original references, a reasonable validation is needed to use the model.

Due to the use of the mentioned model in previous studies and the source of practical experiences of the model, the responses of the modeled system have been evaluated in comparison with the model of Cheng paper. In this section, the general results of the percentage difference in proportion to the model outputs are presented and the results of the tests in the previous study are given[25] .Based on the model validation diagrams and the difference between the output response values in Figure 3, it can be shown that the difference between the response acceleration response of the model and the reference model is less than 4%, which seems to be a reasonable value. In vehicle acceleration, the maximum difference is about 0.1 Rad/s^2 radians per square second and the minimum difference is about 0.2 Rad/s^2 .

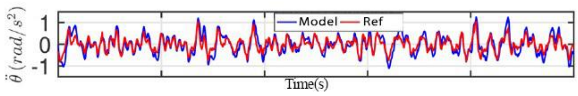
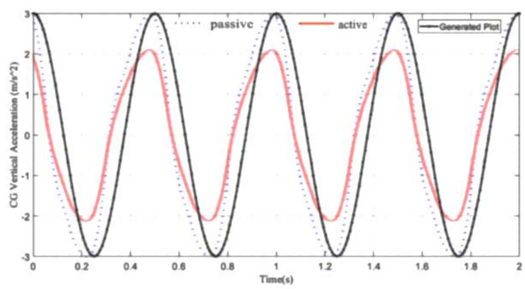


Figure 3: Model validation designed with the Cheng model

This means that the average difference between the two graphs is approximately less than 4.5%. According to the numerical values obtained from the analysis of the model response, there is a difference of about 1.3% in the acceleration of the vehicle mass center and a difference of about 1.6% in the tire deflection, which is due to changes in model inputs, that shown in Figure 4. It is stated that, of course, they are within a reasonable and acceptable range.

Figure 4 examines the parameters of bounce acceleration and tire deflection in the existing model and the model used in the main reference.



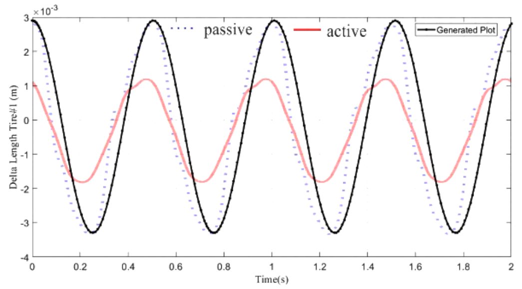


Figure 4: Model validation designed with the Min Yu model

Also in the studies related to Min Yu [31], the results of active variable geometry controller have been used to show the effects in terms of reference model with speed of vehicle and common road roughness.

3. Controller Scheme

The main purpose of this paper is to define and implement a robust and useful control strategy to increase the lateral stability of the vehicle and improve the longitudinal dynamic characteristics of the existing vehicle [32]. For this purpose, by defining the appropriate control parameters, can the expectation of suspension was met .The selection of the controller input parameter is determined based on the purpose of the study and the operation of the vehicle .In this paper, the Angular roll acceleration of the vehicle is used as the input of the control unit in order to improve the lateral stability of the vehicle and also improve the other mentioned cases [33].

Before proceeding with the implementation of the control unit, it is necessary to know that, in this paper, a challenge for the control command distribution actuator will be addressed. In previous studies and most studies of the series active variable-geometry suspension, the same distribution of control steering (angular velocity of links) in the four corners of the vehicle has been used. The usefulness of this type of distribution was shown in previous studies due to the adoption of control methods. In this paper, inspired by studies in the field of active suspension, the use of asymmetric distribution methods will be tested. In the studies of active suspension, two methods of symmetric distribution based on the weight distribution of the sprung mass have been used [34-37]. In this article, 3 types of identical distribution in all 4 corners of the vehicle, symmetrical distribution on the left and right of the vehicle, as well as distribution with weight ratio are distributed and used symmetrically.

Figure 5 shows a schematic of the modeled system and the control unit. In this figure, $d_{(s)}$ equal to the vector of the model inputs of the vehicle mode space and suspension, $Z_{(s)}$ represents the output vector of the dynamic model, $P_{(s)}$ indicates the linearized mathematical model of the vehicle and the effects of the series active variable-geometry suspension in the form of state space, $y_{(s)}$ equal to the inputs of the control unit, known as the control parameters, $u_{(s)}$ is equal to the output vector of the controller unit, which is the angular velocity of each of the suspension links, and $K_{(s)}$ indicates the Fuzzy sliding mode control unit of the system.

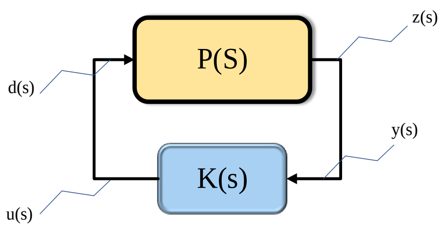


Figure 5: General schematic of the control strategy

3.1. Sliding Mode

As mentioned in section 1, the sliding mode controller has been used extensively in variable geometry suspensions, often for ride comfort [38], but has not been used in series variable geometry suspensions. Therefore, in this article, an attempt is made to use types of sliding mode controllers that have high resistance.

At first, the rolling angle acceleration of the vehicle model as the control base and its error relative to the reference value is considered as input [39]. Equation (4) is the formula of the input signal of the control unit.

$$e(t) = y_r - y_d \tag{4}$$

In this formula, y_r is the reference value of the control parameter to achieve the answer to it. Also y_d a measured signal of the parameter moment, which here is equal to the acceleration of the vehicle rolling resulting from the solution of the dynamic model. Thus, the amount of error to the controller will be equal to the value of (5).

$$e(t) = 0 - \ddot{\varphi} \tag{5}$$

According to the rules governing controller as well as the study of repetitive studies of the active suspension area, a sliding surface is used as follows for modeling.

$$S = \lambda\dot{e} + e \xrightarrow{e=0-\ddot{\varphi}} S = -(\lambda\dddot{\varphi} + \ddot{\varphi}) \tag{6}$$

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Which is that λ is positive constant coefficient is used to determine the overall shape of the sliding surface, which is determined by the type of problem. This coefficient will be effective in determining the form of the control law (to redirect to output responses) [38].

It is noteworthy that the mentioned slip occurs in the state of $\dot{S}=0$. The form of second-order problem-solving in the case of zero time derivative in which is mentioned in (7).

$$S\left(e,t\right)=\left(\frac{d}{dt}+\lambda\right)^{n-1}e=0 \tag{7}$$

Now, assuming the control equation mentioned to define the Instantaneous sliding surface, the control output rule of the controller for slip near the designed slip surface will be determined. This process is defined as the (8).

$$u_{SMC}=u_{eq}-k_{\varphi}\,\mathrm{sgn}(S) \tag{8}$$

Which u_{eq} is equal to the equivalent control input that $\dot{S}=0$ is assumed [38]. k_{φ} is equal to the effective coefficient of control (here for rolling acceleration). Also $\mathrm{sgn}_{(S)}$ is a function of the sliding surface symbol, which is used to determine the output response range and to determine the boundary layer and the control command [40]. This function creates a high-frequency oscillation layer in the control system.

As can be seen, the output response, after being pushed towards the sliding surface $S=0$, begins to slide thickly in 2ϕ one neighborhood and oscillates towards the desired shape.

The control condition for creating the mentioned conditions and limiting the response in the neighborhood of the oscillating layer is in the form of the (9).

$$if\begin{cases}|S|\succ 2\phi & \mathrm{sat}(S)=\mathrm{sgn}(S)\\ else & \mathrm{sat}(S)=S/2\phi\end{cases} \tag{9}$$

Now, using the controller rules, the control unit is implemented with the conditions mentioned in the Simulink environment. It should be noted that one of the main requirements for the design and implementation of this controller is to study its stability, which in this article uses the Lyapunov method. The condition $V(x)=(1/2)S^2$ has been utilized for this purpose. The following equation is used to implement this condition in the problem.

$$s\dot{s}\le -\eta|s| \tag{10}$$

In which η is a constant positive coefficient that is defined based on the type of problem and the design coefficient is related $k_{\varphi}\ge\eta$ to it. This coefficient is used to determine the amount of controller power to withstand perturbations and uncertainties. The expected performance of the controller designed in this paper is the most ideal case possible. The existence of external disturbances and internal uncertainties of the system will take the controller out of the specified mode and cause system instability. If $\dot{V}(x)\succ 0$ so, the mentioned instability will happen. Now, to stabilize the controller and limit the output responses to slip in the vicinity of the selected mode (to the thickness of the oscillating layer), a resistance beyond the limit must be considered for the controller. The mentioned limit will be determined by a η coefficient. In this paper, due to the non-consideration of perturbations in modeling (perturbations such as lateral wind in the maneuver and defined in the initial dynamic model), the η coefficient value is ignored.

3.1.2 Fuzzy Sliding Mode

One of the major problems on the sliding mode controller due to the determination of an oscillation layer and the determination of the lower and upper response limit is the high output oscillations in the sliding surface range. This phenomenon reduces the performance of the controller and also reduces its stability. This phenomenon in the physical dimension increases the cost of control and reduces the efficiency of the system. One of the main effective measures to reduce the mentioned fluctuations is to convert the oscillation layer with specific boundaries to the oscillation layer with multilayer boundaries. In which, instead of determining an identical oscillating thickness, this value is obtained from a fuzzy unit and can change over time and adapt to the conditions of the problem [41, 42].

To design this type of oscillation layer, a fuzzification with 2 inputs and 1 output of Mamdani type have been utilized. The fuzzification inputs are the controller error signal and its time derivative, the fuzzification output is the signal of the thickness of the oscillating layer that changes over time. In order

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to determine the fuzzification membership functions for scientific validation, the reference membership form [43, 44] has been used and combined with [40, 45, 46]. Then related values for the range of these membership functions are then set after performing several tests and observing the result. Table 2 provides information on designing membership functions from [44].

Table 2: membership function values

Parameter	Membership	Range	Type
Error	S	[-10,-4,0]	trimf
	M	[-5,0,5]	
	B	[0,4,10]	
Derivation of error	S	[-10,-0.4,0]	
	M	[-0.5,0,0.5]	
	B	[0,0.4,10]	
Layer thickness	MB	[-50,0.1,0.3]	
	NS	[0.1,0.3,0.5]	
	ZR	[0.3,0.5,0.7]	
	PS	[0.5,0.7,0.9]	
	BS	[0.7,0.9,50]	

These membership functions with the desired control rules result in a general answer form, which is shown in Figure 6 of the design surface of the fuzzification process result [47-49].

The philosophy of using membership and fuzzy rules is based on LQR control output as well as a benchmark from previous studies [50]. Based on this and according to the researches done in the field of fuzzy sliding mode controllers in different types of active controllers [45, 51], the type with triangular mode and the medium number has been used.

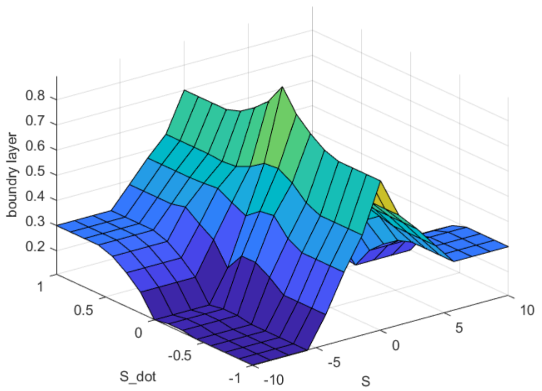

Figure 6: fuzzification surface

It is worth noting that in the results section, according to the design process, the results related to the same distribution of the control command as FSMC1 (Fuzzy Sliding Mode Controller Type 1), the results related to the symmetric distribution with the symbol FSMC2 and the results of the distribution based on the weight distribution with the symbol FSMC3 are displayed.

4. Results

In this section, an attempt is made to simulate the results of the mentioned vehicle model along with the control method based on fuzzy sliding mode in the 3 mentioned modes for the distribution of the angular velocity shown and a comprehensive comparison for the changes resulting from the use Simultaneously from the series active variable-geometry suspension, as well as the fuzzy sliding mode control method and the asymmetric distribution concerning the passive suspension mode should be investigated. The results are classified according to the expectations of the suspension and the factors of lateral stability, ride comfort and road holding are the factors under consideration.

The main factors for checking the lateral stability of a vehicle are its angular accelerations. The roll acceleration, which results in the vehicle overturning angle, the most important expectation is to reduce the range of oscillation while driving in different maneuvers.

Figure 7 Shows the oscillating response changes of the vehicle's roll angular acceleration during two modeled driving maneuvers. It should be noted that most diagrams use different time ranges for high resolution, while the test time was 10 seconds.

As shown in Figure 7, the oscillation amplitudes of roll acceleration of the vehicle have been highly variable relative to the idle state, which are fully described in previous studies.

For pitch acceleration of the vehicle (Figure 8), which leads to the longitudinal overturning of the vehicle, efforts are made to reach a suitable safety limit.

What needs to be described is the values obtained from the three types of angular velocity distributions between the suspension links, as can be seen, the series active variable-geometry suspension system is very useful and has a good oscillating response according to most distributions. It is also necessary to keep in mind that the reduction of the oscillation amplitude of the mentioned angular accelerations, which results in the reduction of the roll angle from the critical value (about 8 degrees) to below 4 degrees, must be such that it does not tend to negative angles.

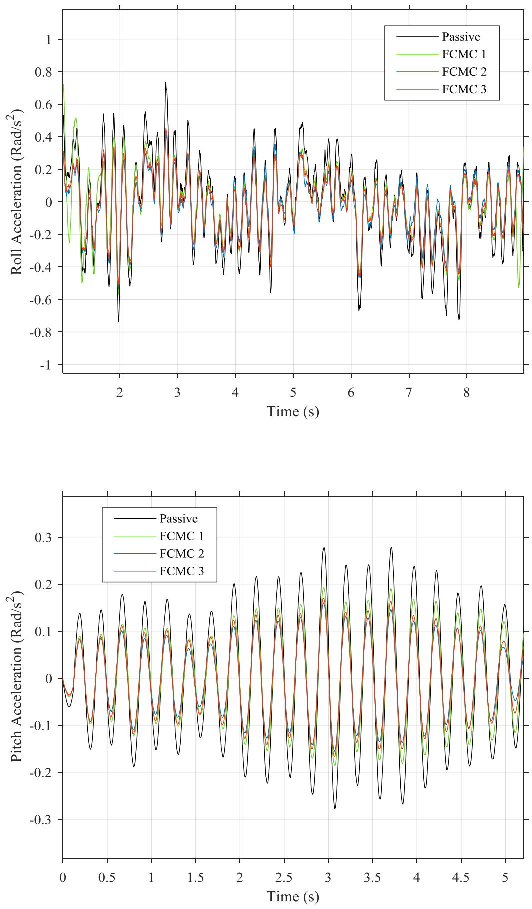

Figure 7: Vehicle roll acceleration, Top: Crosswind Maneuver, Bottom: Double Lane Change Maneuver

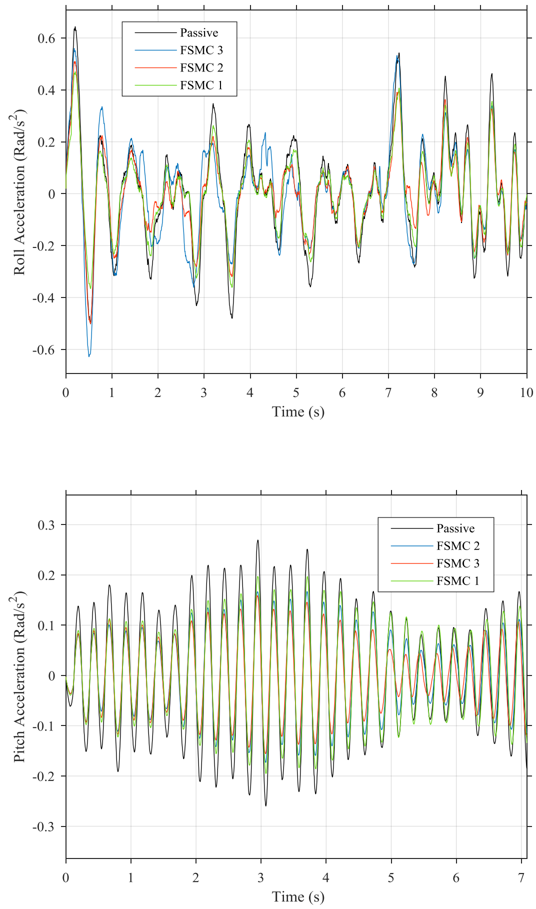

Figure 8: Vehicle pitch acceleration, Top: Crosswind Maneuver, Bottom: Double Lane Change Maneuver

This will lead to the establishment and maintenance of a sense of driving. In this way, if the vehicle tends to bend to one side due to external factors such as wind or the effects of driving and road, the passenger must also lean in the same direction. But in negative (and zero) angles, this is reversed and apart from the possible dangers, it will lead to the loss of the desired sense of driving. Table 3 is shown the numerical values of the mentioned parameters for comparison. In this table, which is for the parameters of the vehicle lateral stability, an attempt has been made to get a good view of the idea improvement of this article, considering the closeness of the diagrams of the three modes to each other.

As observed, the maximum and minimum values of the parameters affecting of the vehicle lateral stability have been significantly improved. It is noteworthy that to achieve a comprehensive and more accurate comparison of the 3 modes of actuator distribution, further information will be provided from the data review.

As it is like the active suspension, the controlled instantaneous response at different frequency intervals improves the performance of several factors simultaneously.

It was observed that the vehicle gave a favorable response to lateral stability and was very far from lateral overturning. In the following, two main factors of suspension that are important for customers and consumers are examined. The road-holding of the vehicle is the first major factor.

This factor, which is effective in forwarding the acceleration of the vehicle, is often measured by the vertical force of the tires since the compression of the tire is directly related to its force. In this article, this parameter is used as a criterion for examining the adhesive road. Figure 9 shows the deflection of the left front tire (for example) for inspection during crosswind maneuvers.

This diagram is also shown in Figure 10 for the double lane change maneuver and numerical values of results are given in Table 4.

It should be noted that the usefulness of the sliding mode controller has already been proven, and in this paper, the focus is more on its fuzzy mode and how to distribute the control command for greater efficiency.

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Table 3: numerical value of system angular acceleration response

Parameter	Maneuver	Factor	Passive	FSMC 1	FSMC 2	FSMC 3
Roll Acceleration (Rad/s^2)	Crosswind	max	1.078	0.7985	0.5521	0.5269
		min	-0.9511	-0.7384	-0.5355	-0.5093
	Double Lane Change	max	0.6436	0.47	0.5599	0.5094
		min	-0.5018	-0.3659	-0.6284	-0.5008
Pitch Acceleration (Rad/s^2)	Crosswind	max	0.5815	0.438	0.3577	0.4023
		min	-0.5867	-0.427	-0.3056	-0.4057
	Double Lane Change	max	0.6219	0.4378	0.3958	0.405
		min	-0.622	-0.446	-0.391	-0.402

Table 4: numerical value of tire deflections

Parameter	Maneuver	Factor	Passive	FSMC 1	FSMC 2	FSMC 3
Tire Deflection ($\times 10^{-2}$) (m)	Crosswind	max	1.88	1.46	1.21	1.33
		min	-1.93	-1.55	-1.32	-1.46
	Double Lane Change	max	1.97	1.44	1.31	1.38
		min	-2.03	-1.61	-1.408	-1.42

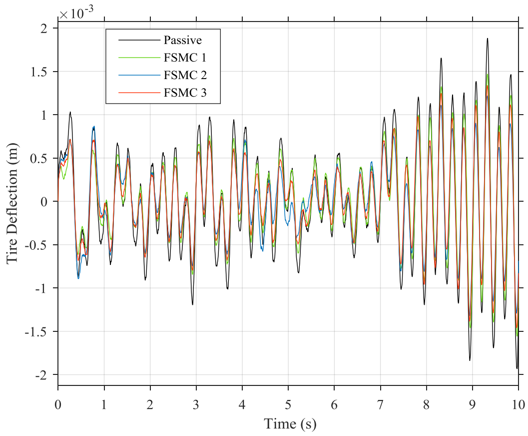

Figure 9: Tire Deflection in crosswind maneuver

Thus, reducing the vehicle tire deflection increases the road holding and vehicle acceleration performance [52, 53]. This is achieved with respect to the vertical tire force. In this study, due to the linear equation between the tire force and tire deflection, the criterion of changes in the height difference of the sprung mass with the road roughness surface has been used

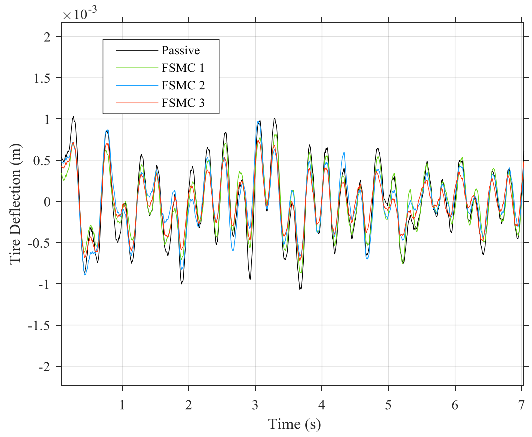

Figure 10: Tire Deflection in double lane-change maneuver

Another noteworthy point from the diagrams and tables provided by the vehicle oscillation response is the review criterion. As can be seen, the diagrams of the two distribution modes in the sliding mode controller may be very close to each other and the presented invoice may not be fully comparable in the table provided.

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Table 5: numerical value of bounce acceleration

Parameter	Maneuver	Factor	Passive	FSMC 1	FSMC 2	FSMC 3
Sprung Mass Acceleration (m/s^2)	Crosswind	max	0.8061	0.6052	0.4952	0.5493
		min	-0.8512	-0.655	-0.5043	-0.5885
	Double Lane Change	max	0.8553	0.601	0.5588	0.5693
		min	-0.9125	-0.6467	-0.5631	-0.5731
CG Acceleration (m/s^2)	Crosswind	max	1.078	0.5897	0.5221	0.5269
		min	-0.9511	-0.5788	-0.5355	-0.5093
	Double Lane Change	max	1.794	1.479	1.01	1.135
		min	-1.304	-0.9473	-0.751	-0.72

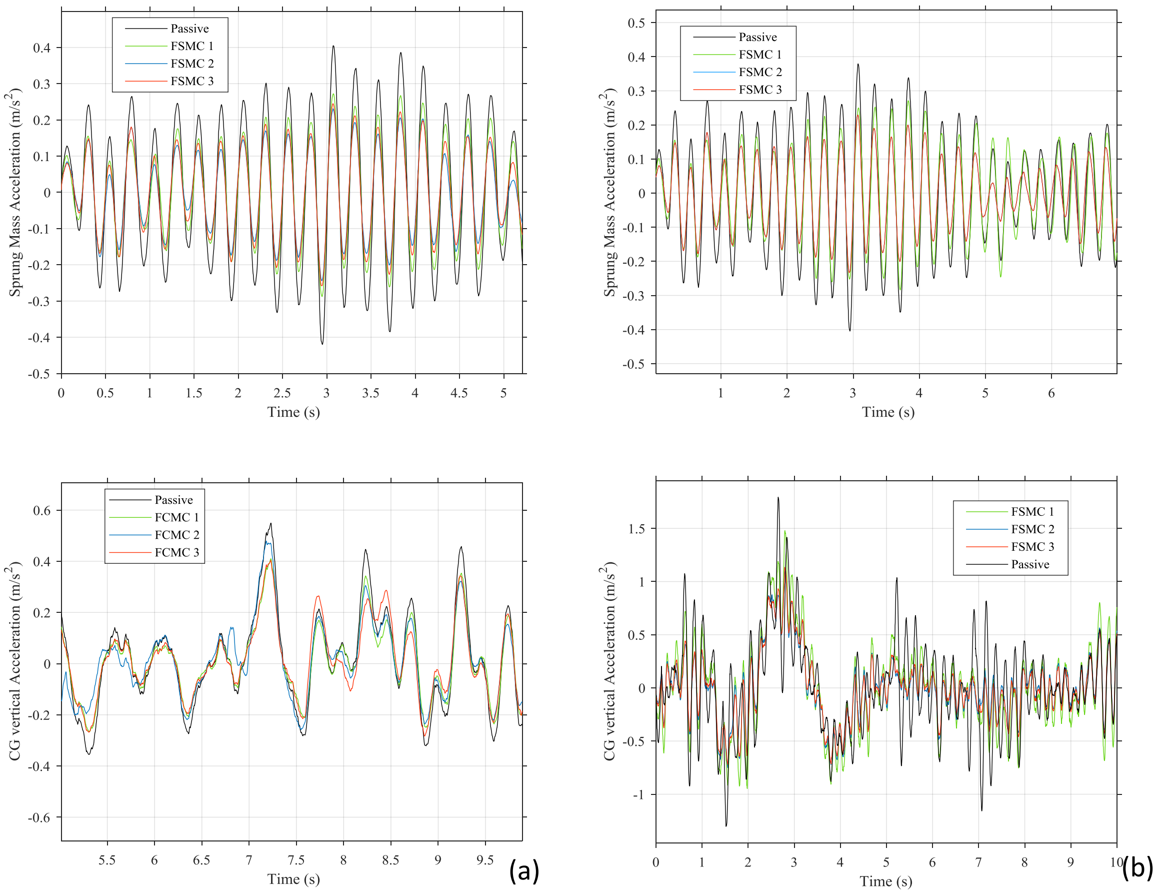

Figure 11: Bounce acceleration during maneuvers, Top: sprung mass acceleration; Bottom: center of gravity vertical acceleration a) crosswind maneuver b) double lane-change maneuver

For this purpose, at the end of this section, an attempt is made to pay RMS in addition to the mentioned items. The following diagrams and table of information related to the ride comfort factor, which includes all accelerations in the vertical direction of the vehicle are shown in Figure 11 and Table 5.

Carefully in the acceleration diagrams in the center of gravity of the vehicle and also in the left corner of the vehicle front (for example), it can be seen that by reducing the amplitude of vertical acceleration and reducing its amount, the vibration absorption of vertical excitations from road roughness is properly absorbed and not transferable to cabins and occupants. In some cases, a reduction of 0.3 m/s^2 relative to the inactive suspension mode is a very good performance for the suspension.

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Table 6: RMS of suspension performance

Maneuver	Parameter	Passive	FSMC 1	FSMC 2	FSMC 3
Crosswind	Roll Acc (×10^{-1}) (Rad/s^2)	2.043	1.51	1.64	1.87
	Pitch Acc (×10^{-1}) (Rad/s^2)	2.053	1.52	1.21	1.36
	CG Acc (×10^{-1}) (m/s^2)	2.89	2.65	1.83	1.89
	Sprung Mass acc (×10^{-1}) (m/s^2)	2.82	2.11	1.67	1.87
	Tire Deflection (×10^{-4}) (m)	6.29	4.99	4.45	4.62
Double Lane Change	Roll Acc (×10^{-1}) (Rad/s^2)	2.039	1.52	1.53	1.91
	Pitch Acc (×10^{-1}) (Rad/s^2)	2.101	1.48	1.31	1.34
	CG Acc (×10^{-1}) (m/s^2)	4.42	4.208	3.17	3.18
	Sprung Mass acc (×10^{-1}) (m/s^2)	2.88	2.039	1.81	1.85
	Tire Deflection (×10^{-4}) (m)	6.44	4.87	4.54	4.73

This reduction in the range of oscillations of the stimuli on the occupants will ensure the health and comfort of the occupants in the long run. This feature is very important in luxury vehicles as well as passenger vehicles for long distances, and also in military vehicles as well as special military vehicles that need low cabin vibration is very necessary and desirable.

Up to this point, an attempt has been made to express the improvements resulting from the use of the series active variable-geometry suspension system using a fuzzy sliding mode controller and to express the usefulness of the method presented in the paper compared to the passive mode. In the following, we will focus on the results of the control command distribution challenge and express the improvements obtained from the 3 modes and compare these methods. Also, as mentioned, due to the proximity of graphs and the lack of comprehensive factors of minimum and maximum values in the graphs, go to the criterion of Root Mean square, and all the results of the simulation will be examined from this perspective.

Table 6 shows the RMS values of the output parameters based on the three criteria of lateral stability, ride comfort, and road holding in two simulated maneuvers for the vehicle. The main advantage of this factor is the inclusion of output results throughout the test and provides a better view of the efficiency of the method adopted in the study.

It should be noted that the numerical data presented in the tables in this section are based on the minimum and maximum parameters, which due to the continuity of the maneuvers and to create a wide view of the charts in Table 6.

Based on this description, an attempt has been made to compare the data in Table 6 to examine the percentage of improvement or non-improvement of the method has been utilized. Because, as shown in the diagrams, reducing the maximum value of a parameter does not mean improving and reducing the amplitude over the entire test time.

As can be seen, better results are obtained by applying a symmetrical distribution on both sides of the vehicle. In this case, according to the control command, the existing angular velocity of the links enters the 2 sides of the vehicle in 2 different directions. In other words, the torque of the series active variable-geometry suspension links is entered in 2 opposite directions. The improvement mentioned in this case compared to previous studies and a uniform distribution is about 15% in lateral stability, about 20% in ride comfort, and about 8% in vehicle road holding. Assuming this is the most useful method of distributing the control command and torque of the links in this study, we will have a significant improvement using the suspension system and the control method mentioned and modifying the distribution method. The details of this improvement are given in Table 7, stating the

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improvement of the parameters compared to the passive mode. It should be noted that the improvement values are calculated according to the data in Table 6 and based on the RMS parameters in order to have a better coverage of the performance of SAVGS and adopted control method. It is also based on the comparative performance of the previous tasks mentioned in Section 2 and the passive suspension.

Table 7: improvement of model output parameters

Maneuver	Roll Acc	Pitch Acc	CG Acc	Sprung Mass Acc	Tire Deflectio
Crosswind (%)	19.7	41	36.6	40.7	29.2
Double Lane Change (%)	24.9	37.6	28.2	37.1	29.5

It should be noted that the rate of improvement of FSMC 2 compared to FSMC 3 is not very high and the 2 modes are very close to each other. It can be accurately stated that in the crosswind maneuver we see an improvement of 3% to 12% in the mentioned factors and in the double lane change maneuver we see ~~a~~ 4% to 11% improvement. But since it is more costly to implement the actuator based on the weight distribution of the work, using a simple symmetric actuator will be a better and more useful method for vehicles.

At the end of this section, after presenting all the graphical and numerical data of the test, it is necessary to pay attention to the fact that the data of this article is a new control method in SAVGS and apart from comparing with the author's previous claims, it should be compared with the original model references.

For this purpose, it should be noted that in the basic studies of this suspension system, which was based on the H-infinity controller in the smooth path of type A road at a constant speed, all previous studies based on the control method mentioned in the direct path have examined the ride comfort parameters [26, 54]. They were in this paper, by mentioning the modeling details of the previous section, the values of lateral stability, ride comfort, and road-holding were investigated.

5. Discussion

Reducing the vehicle's angular acceleration during maneuvers with high steering and lateral overturning factors reduces the vehicle's overturning angle and ensures its stability. This is important for any type of vehicle, especially military vehicles, with the degree of command and participation in operations. Reducing vertical acceleration and tire compression also provides a comfortable environment for vehicle occupants and the possibility of forwarding accelerating of the vehicle. In this study, by utilizing the beneficial control method and modern suspension system, all these cases are provided and in the general view, the longitudinal dynamics and stability of the vehicle are improved.

The results of the study provide the primary goal of lateral stability. This is achieved by reducing the amplitude of acceleration angles and reversal angles. In this regard, it is important to know that the reduction of the vehicle's rolling angle is towards zero but does not reach zero (and does not become negative) and this emir is necessary to maintain the driving sense. Driving sense means that if the vehicle overturns to one side, the occupant will also lean somewhat in the same direction to avoid both possible dangers (with the sense of overturning transmitted to the occupant) and inconsistent behavior with the vehicle's natural entrance.

Reducing the range of vertical acceleration oscillations reduces the displacement rate of the vehicle's sprung mass. This reduction creates a suitable bed for long-term driving without injury to the occupants and maintains health and avoids ergonomic complications. On the other hand, trying to make the road holding will increase the vehicle's accelerate. This happens by reducing energy consumption and also has lower control costs and less complexity. The sum of these achievements will increase the dynamic efficiency of the vehicle.

6. Conclusions

In this paper, after modeling a full-vehicle model with a series active variable-geometry suspension systems, crosswind and double lane change maneuvers redirection on a rough road at a relatively high speed were modeled. Also, after verifying the model, the fuzzy sliding mode controller was implemented based on asymmetric distribution and based on vehicle weight dispersion.

- Improvement of about 25% in the parameters of the vehicle lateral stability and improvement of the vehicle conditions to prevent proximity to the probability range

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- About 30% improvement in occupant comfort factors to increase the ability to drive properly over time and ensure occupant health, which can be used for vibration-sensitive load transfer modes in military applications.
- 29% improvement in acceleration factors and vehicle adhesive road supply, which is important in military and passenger use

All the effort in this study, which was in line with previous studies in this field, was to make the modeling conditions more realistic and to increase the accuracy of the output responses. This was tested by considering the different functions of the control command distribution operator in the suspension links. Rejection of further studies It is hoped to use a dynamic model based on vehicle variables to connect to appropriate simulation software. Efforts are also made to reduce simplistic assumptions and increase the accuracy and efficiency of the control method by using third-party optimization methods.

Declaration of Conflicting Interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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