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RESEARCH PAPER

Lateral Safety Enhancement in a full Dynamic Vehicle Model Based on Series Active Variable-Geometry Suspension

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Abstract
Today, the importance of providing safety and stability while pay attention to the ride comfort and providing road holding is of paramount importance. This issue has become more important due to the many accidents related to vehicle rollover. In this article, an attempt has been made to reduce the risk of rollover prevention of the vehicle while pay attention to the needs of the occupant and the road. In this research, an attempt has been made to reduce the overall acceleration of the GT vehicle by using a series of active variable geometry suspensions and by using a variety of control strategies such as Fuzzy PID, LQR, Sliding mode. In previous works, PID and Skyhook controllers have been used. However, in this study, the choice of the controllers is based on attention to accuracy and optimization while pay attention to control aims. This study was performed in conditions of severe asymmetric roughness and cornering maneuvers. The examination of the results shows an improvement of more than 20% for the goal of vehicle stability while providing other suspension goals. This performance improvement occurs with the effect of suspending variable geometry along with the use of a suitable controller. It should also be noted that the improvement achieved by consuming energy is far less than other suspensions, which is the strength of the research.
Keywords: Series active variable-geometry suspension, SAVGS, Rollover prevention, LQR, Sliding mode, Fuzzy

Introduction

Given the high importance of vehicle overturning in recent years and the spread of concern in this regard, the issue of preventing lateral overturning of the vehicle has become a paramount concern [1]. Therefore, it is necessary and crucial to developing the vehicle rollover moot point and anti-rollover control system to improve the roll stability and avoid rollovers. This goal largely depends on the design and performance of the vehicle's suspension. On the other hand, today, given the importance of the energy issue, this point should be taken into account in designing the suspension system [2] [3]. Besides, the primary function of the suspension system should be considered in creating suitable ride comfort while the road holding [4, 5].

The use of an active suspension system that provides instantaneous control of the amount of stiffness and dumping of the system primarily covers the above objectives [1, 6]. The biggest problem of active suspensions is their high energy consumption. The series variable-geometry suspension system by changing the position and angle of the suspension stratum with the help of the added link and a servo motor covers the mentioned need [4, 7].

In previous works, many procedures were used for vehicle roll prevention, such as active differential braking system [8-10], active steering [11, 12], active and semi-active suspension [13-15], and together with active suspension and differential braking [3, 16, 17].

This article uses a series active variable geometry suspention to study a ground tour (GT) vehicle in certain conditions. In this study, a recovery control strategy has been used in the vehicle's dynamic response. Given the importance of the issue mentioned above and the high

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casualties of vehicle overturning, extensive research has been done in this field. However, the main differences and innovations of the study are as follows:

- Paying attention to the majority of the vehicle's dynamic parameters using the full vehicle model instead of the one-quarter model
- The use of a new type of suspension system that has not been studied extensively in the field of vehicle rolls
- Paying attention to performance improvement in several areas by choosing the type of the suspension and operator in addition to providing dynamic response purposes, reduces energy consumption
- Possibility to add driving maneuvers to the vehicle's mathematical model and use hard condition maneuvers (combining road conditions and environmental conditions) that will lead to the realization of the study.
- Implement a variety of modern and classic controllers that provide the conditions for multifunctional and more comprehensive studies

For this purpose, first, a complete model providing the dynamic characteristics of the vehicle and the conditions for performing the test on the designed model will be discussed is designed (section 2). Then, a complete description of the control methods used (section 3) to achieve the appropriate system response will be provided. In the last section, the results will be described, and the results of the work will be done.

Modeling of a Full Vevicle Model

The mathematical model is used to explain the dynamic states of the vehicle, and the suspension is the existing model of the Cheng reference paper [18]. This model of vehicle mode space has 10 inputs and 15 outputs given in equation 1.

The details of how to calculate the state space matrices with the exact value of the components along with the vibrational and force relations governing the base are explained in the reference [9].

$$u=[T_p,T_r,z_{r1},z_{r2},z_{r3},z_{r4},\dot{\theta}_{SL1},\dot{\theta}_{SL2},\dot{\theta}_{SL3},\dot{\theta}_{SL4}]^T$$
$$y=[\Delta l_{t1},\Delta l_{t2},\Delta l_{t3},\Delta l_{t4},\ddot{z}_{s1},\ddot{z}_{s2},\ddot{z}_{s3},\ddot{z}_{s4},\ddot{Z},\ddot{\theta},\ddot{\varphi},\Delta l_{s1},\Delta l_{s2},\Delta l_{s3},\Delta l_{s4}]^T \quad (1)$$

This model has rough road inputs and external torques on the vehicle, which the road roughness assumed in the research will be from a standard random road type C and E. These two types of road roughness are obtained by considering the car speed of 100 km/h and using the rules of Wong [19, 20] and are applied asymmetrically to both sides of the vehicle. Figure 1 shows the asymmetric unevenness diagram generated as a random standard. Where the left side of the vehicle passes with type C roughness and the right side with more severe type E roughness and the car is in an asymmetric roughness.

To simulate Tp/Tr (to excite the rotational movements) as 1 and 2 inputs, The spring travel difference between the front and rear suspension and the left and right of the vehicle is calculated. Then by calculating the suspension force resulting from the compression difference and considering the distance to the center of mass and installation ratio, the torque values are calculated [21].

$$T_{p(1,2)}=-\Delta l_{s1}(or\ \Delta l_{s2})\times K_{sf}\times wheelbase\times Weight\ distribution$$
$$T_{p(3,4)}=-\Delta l_{s3}(or\ \Delta l_{s4})\times K_{sr}\times wheelbase\times Weight\ distribution \quad (2)$$
$$T_p=T_{p1}+T_{p2}-T_{p3}-T_{p4}$$

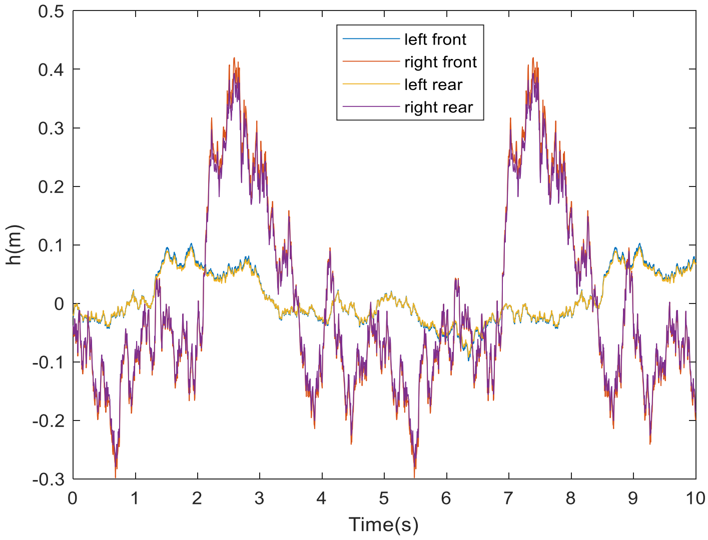


Figure 1. Road disturbance on type C and E ISO random road

The equation 2 shows the calculated pitching torque of each corner of the vehicle, and we can get the total Tp of the vehicle. Similarly, the amount of this torque for the vehicle roll can be obtained.

Table 1 lists the values of important and influential parameters in the design of the vehicle model, which are obtained by using the Ferrari F430 vehicle catalog.

Table 1. Parameters of the model

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*

One of the main differences between the existing model and the reference article is the possibility of adding different maneuvers (desired maneuver output information or simultaneous stimulation of the maneuver in dynamic analysis software) to achieve a more comprehensive study of road conditions and control objectives.

The crosswind maneuver consists of a path where the vehicle is affected by lateral winds while traveling at speeds of 100 km/h. The purpose of this work is to make the wind test have a high impact on the dynamics of the system and to face the most challenging conditions according to the type of rough road. The wind speed terms of time during the test diagram are given in figure 2, which shows that a high wind with variable wind conditions has been used.

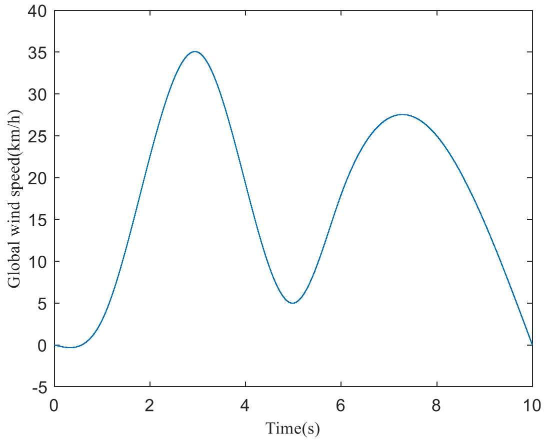


Figure 2. Global wind speed in the crosswind test

Validation of the model

As mentioned earlier, the model used is the Cheng reference model, which has already been verified. However, since the front torque and roll of the vehicle have been calculated separately and given to the model, it is necessary to verify again with the model.

As can be seen in figure 3, the amount of acceleration and tire deflection of the vehicle in the passive state is very close to its value in the reference article, and the reason for its small is more complicated due to the change in problem conditions for testing.

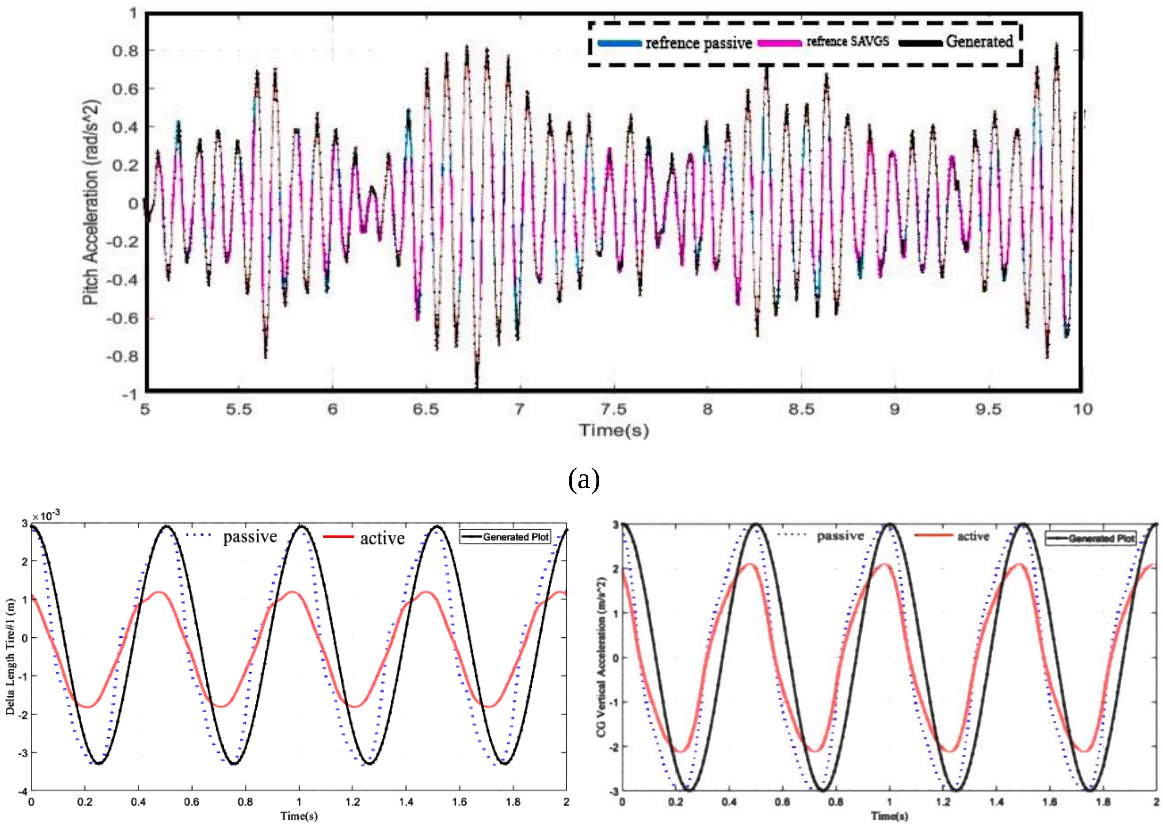


Figure 3. Vehicle responses of reference and designed model (a) Pitch angular acceleration (b) Tire deflection (c) Bounce acceleration

Table 2 shows the numerical values obtained from the results of the reference model in comparison with the model designed in passive mode under a road by type A at speed of 100 km/h.

Table 2. numerical values for validation

Parameter	Reference value	Generated value	Unit	Error
Max Pitch acc	0.83	0.85	*Rad/s*^{2}	**4%**
Min Pitch acc	-0.94	-0.99		
Max Roll acc	2.2	2.1	*Rad/s*^{2}	**4.5%**
Min Roll acc	-2	-2.1		
CG max acc	2.95	3	*m/s*^{2}	**1.3%**
CG Min acc	-2.98	-3.01		
Tire Max deflection	2.95	2.98	*cm*	**1.6%**

Controller Scheme

The primary purpose of this work is to design a control process based on the full vehicle dynamics to reduce the amount of roll to achieve the desired conditions to avoid the rollover in various profiles of the road under certain conditions. The general nonlinear control scheme for this work is shown in figure 4 [22, 23].

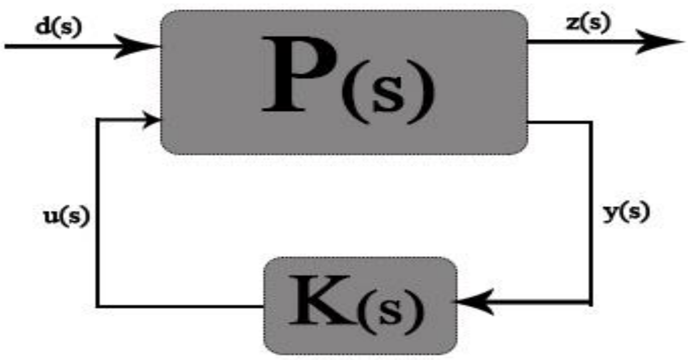


Figure 4. General schematic of the control strategy

The components of d(s) are the exogenous input disturbances, and the components of z(s) are the performance variables of interest to be minimized in this work. The measurement signals that are used by controller K(s) are denoted by y(s), and the controlled inputs generated by the controller are denoted by u(s) that are given in equation 3:

$$d(s)=[T_p,T_r,z_{r1},z_{r2},z_{r3},z_{r4}]^T$$
$$z(s)=[\Delta l_{t1},\Delta l_{t2},\Delta l_{t3},\Delta l_{t4},\ddot{z}_{s1},\ddot{z}_{s2},\ddot{z}_{s3},\ddot{z}_{s4},\ddot{Z},\ddot{\theta},\ddot{\varphi},\Delta l_{s1},\Delta l_{s2},\Delta l_{s3},\Delta l_{s4}] \tag{3}$$
$$y(s)=[\ddot{\varphi}]$$
$$u(s)=[\dot{\theta}_{SL1},\dot{\theta}_{SL2},\dot{\theta}_{SL3},\dot{\theta}_{SL4}]^T$$

The following is a brief description of how the controllers are designed and implemented.

Fuzzy PID Controller

The first controller used in this research is the use of Fuzzy PID controller. The controller has been used in previous research. In this paper, a fuzzy generator is used to determine the values of the PID controller coefficients, which are shown in figure 5 of the input and output memberships [24-26].

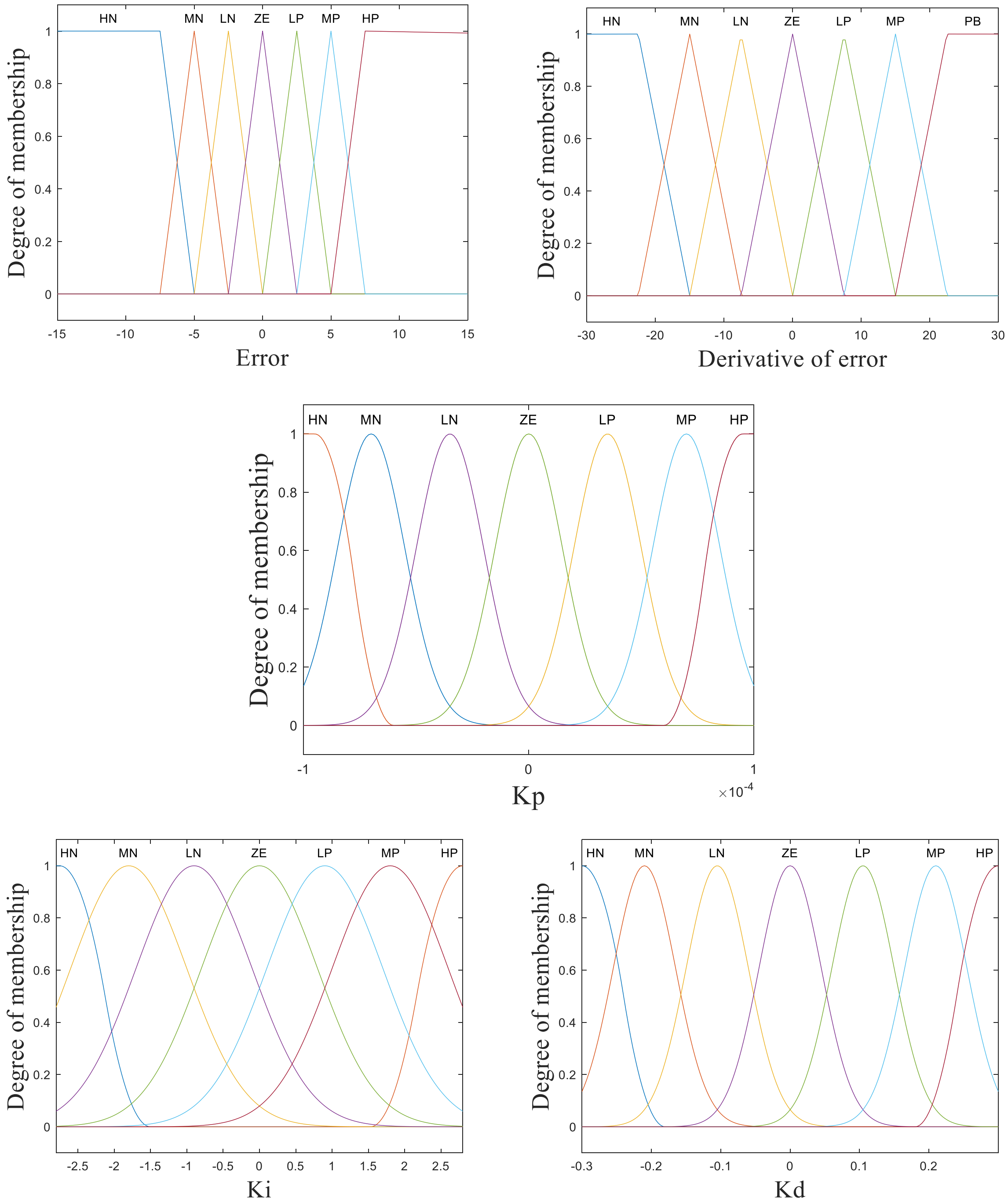


Figure 5. Memberships of inputs and outputs

The control rules between input and output are also given in another article is being published by the authors of this article, that the value range and forms of memberships and rules of this controller have been changed to appropriate values after trial and error and compliance with the references [27].

LQR Controller

Fuzzy PID controller is responsive to many conditions of the suspension due to its robustness property. However, a controller that is useful for state-space design. With the feature of being optimal is needed [28, 29]. To achieve this aim, an LQR controller is designed and used to compare with two controller strategies.

LQR controller design is for linear systems by generating system states and using them to create a controlling factor. Therefore, before designing the controller, one must design an observer [30].

$$\begin{aligned} \dot{x}(t) &= Ax(t) + Bu(t) \\ y(t) &= Cx(t) + Du(t) \\ u(t) &= -Ke(t) \end{aligned} \tag{4}$$

Due to the design of the observer, the values related to the road design, and also the ineffectiveness of the maneuvers performed, the mentioned values must be entered into the system as follows:

$$\begin{aligned} e(t) &= r(t) - x(t) \\ u(t) &= u_m(t) + d(t) \end{aligned} \tag{5}$$

The observer designed in the article is a combination of system inputs and outputs, taking into account the initial conditions and specific values of the system.

The K is evaluated concerning the system dynamics for the minimum value of the quadratic performance index j as specified in the equation 6 [31].

$$j = \int_0^{\infty} [x(t)^T Qx(t) + u(t)^T Ru(t)]dt \tag{6}$$

One of the categories that LQR controller design depends on is choosing the right values for Q and R matrices.

For this purpose, first of all, by referring to the articles that had worked in the field of vehicle rolls, the mentioned matrices were defined.

After that, considering the suggested values of the existing articles [32] and rules for selecting the range of high matrices, we reached the following values that are given in equation 7 , with several changes.

$$\begin{aligned} Q_{14\times14} &= 10^{-3} \times C' \times C \\ R_{10\times10} &= diag[1] \end{aligned} \tag{7}$$

Sliding Mode Controller

After implementing the two mentioned controllers, the slider mode controller will be designed to create an optimal and resistant mode for the problem. In this controller that the general view of which is shown in figure 6, will quantify the output of the controller in the optimal range by receiving the measured error resulting from the output value of the system and the reference value of zero and the time derivative of the error [33].

Before defining and implementing the basic rules for the slider mode controller mentioned, first, the changes in the parameters used in table 3 are given [24].

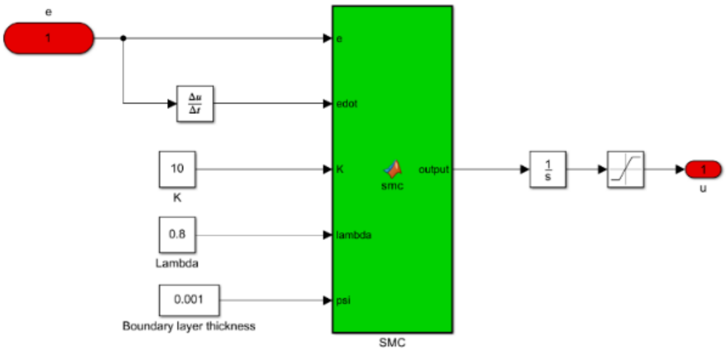


Figure 6. Simulink diagram from the sliding mode controller

Table 3. Sliding mode control coefficient design table

Equivalent Parameter	Parameter
0	y_r
$\ddot{\varphi}$	y_d
$e = 0 - \ddot{\varphi}$	$e = y_r - y_d$
$S = -(\lambda\dddot{\varphi} + \ddot{\varphi})$	$S = \lambda\dot{e} + e$
$u_{SMC} = SMC(e, \dot{e}, k_{\varphi}, \lambda, \phi)$	$u = u_{eq} - k\,\mathrm{sgn}(S)$

In the equation 8 and equation 9, the basic rules of controller design are given, which have been implemented with the authorities after verification.

$$s = \lambda e + \dot{e}$$
$$if \begin{cases} |s| \succ psi & \mathrm{sat}(s) = \mathrm{sgn}(s) \\ else & \mathrm{sat}(s) = s/psi \end{cases} \quad (8)$$

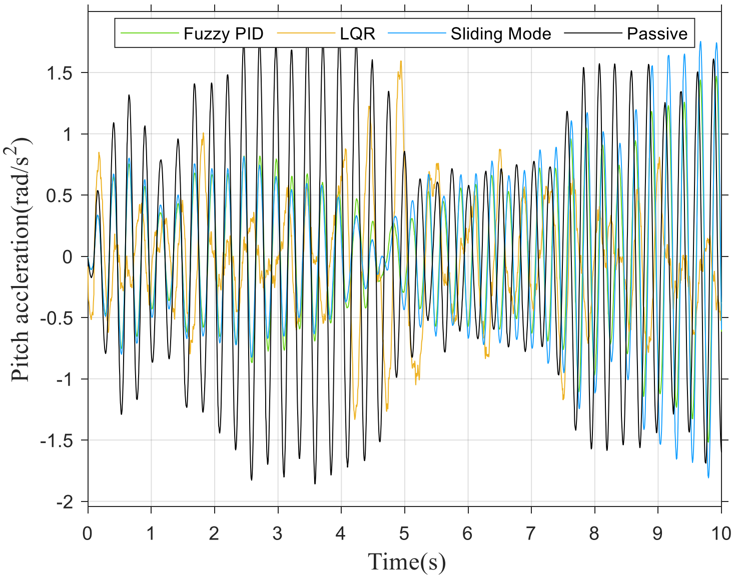
Values related to the design of the controller operation are obtained from the references [34] and after finding the relationship between the inputs and outputs of the controller and placing them in the governing equations [35].

$$output = SMC(e, \dot{e}, k, \lambda, psi) \quad (9)$$

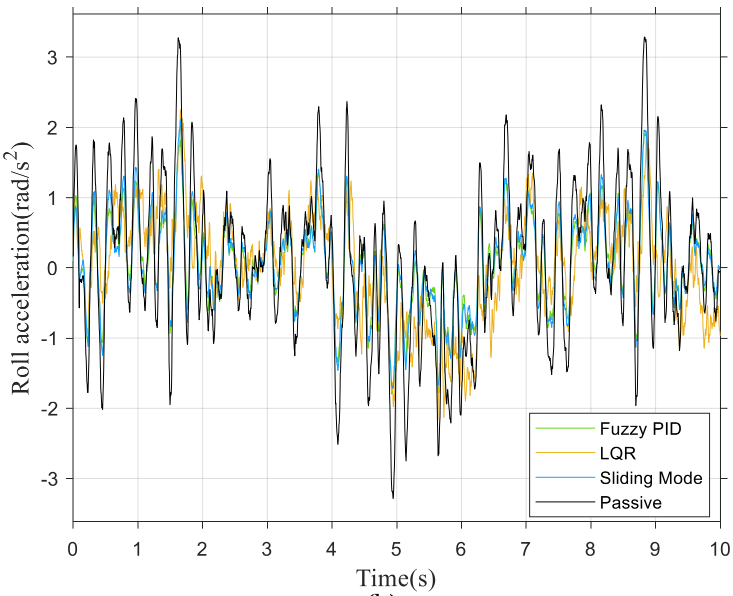
Results and Discussion

In this section, the results of the behavior of the vehicle model will be discussed. For this purpose, the vehicle traveled with the said conditions and mentioned in the maneuver and road conditions for 10 seconds, and the control methods discussed are applied to the model. The results of the vehicle oscillation response in each of the factors are compared with its value under the same conditions without the use of SAVGS.

Figure 7 shows the values obtained from the vehicle's angular acceleration response. The reason for the importance of these two values is to ensure the lateral security of the vehicle and prevent rollover.



(a)



(b)

Figure 7. Angular accelerations during the test via SAVGS model (a) Pitch angular acceleration (b) Roll angular acceleration

As shown in figure 7, the acceleration and roll of the vehicle in the face of lateral winds are significantly reduced. This reduction in pitching and rolling acceleration values will reduce the corresponding angle and create a safe margin for the lateral stability of the vehicle. To be more precise, the values of the critical angle of the vehicle swing have been improved by 25%, which is an acceptable value for the problem.

As mentioned at the beginning, the purpose of this study is to pay attention to the lateral stability of the vehicle while providing the desired driving conditions and fulfilling the main tasks of the suspension. For this purpose, along with examining the angular accelerations of the vehicle, the factors of ride comfort and optimal driving have also been examined in figure 8. The first important factor is to check the vertical acceleration of the vehicle. For this purpose, the two vertical accelerations of the vehicle mass center and the vertical accelerations of the sprung masses of each quadrant are given below. Figure 8 shows the vertical acceleration of the vehicle mass center (bounce acceleration) and the sprung mass acceleration of each corner of the vehicle.

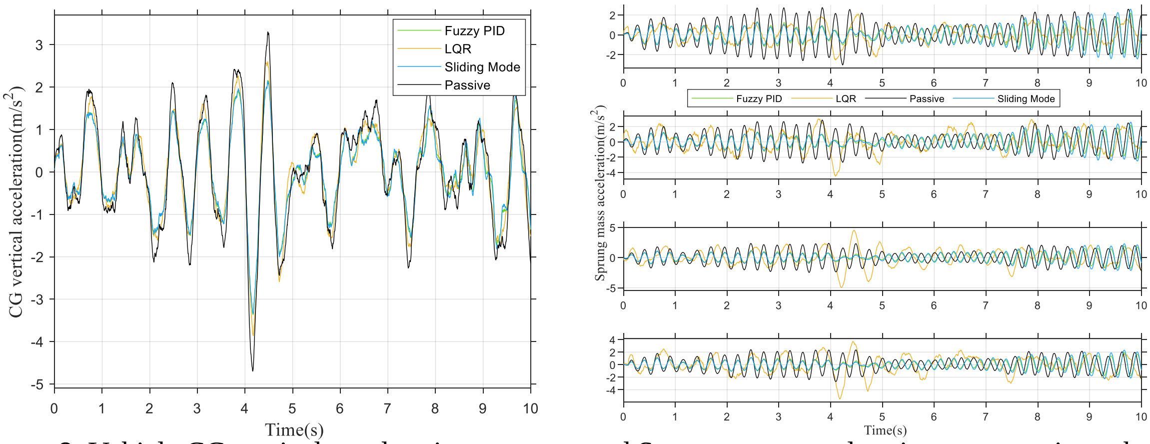


Figure 8. Vehicle CG vertical acceleration response and Sprung mass acceleration response in each corner of the vehicle

As can be seen, the parameters affecting the comfort of the vehicle have also been improved along with the focus on the angular acceleration of the vehicle. This achievement is of great value, which achieved several desirable goals by using a special suspension system and using the appropriate control method while reducing the energy consumption of the operator and the vehicle. This means that if a lateral disturbing factor such as strong winds that disrupts the stability of the vehicle is applied to the model and the model enters a very high and asymmetric roughness at a very high speed, preserve a significant improvement in lateral stability. More of them, the vehicle does not loss of ride comfort.

Now that the stability of the vehicle and the comfort of the passenger have been examined, the road holding and the favorable driving conditions are being considered. The reason for this is due to the nature of the suspension system and the coverage of the suspension targets in a different frequency range and instantaneously. The influential factor in the mentioned factor is the instantaneous compression of the vehicle tire. Figure 9 shows the instantaneous oscillation response of a vehicle tire to road roughness and vertical load application.

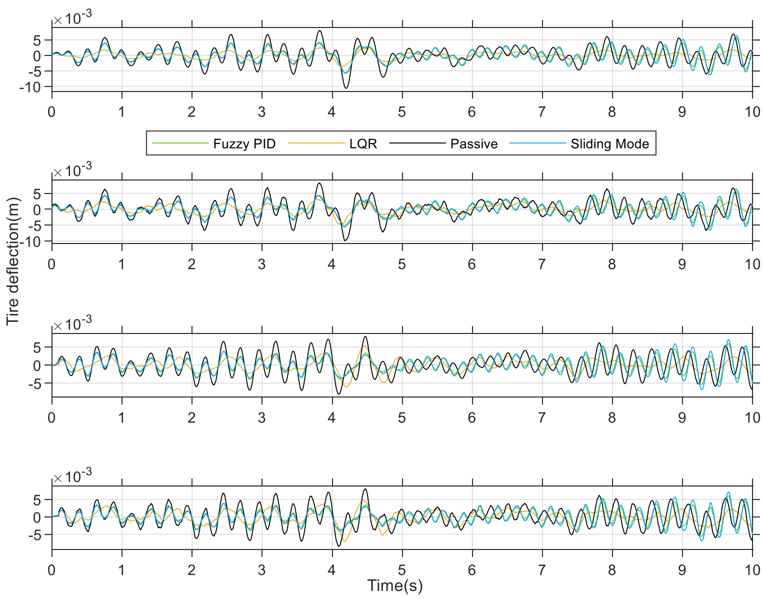


Figure 9. Deflection value per each vehicle tire

As can be seen, as in the previously studied variables, there is a slight improvement in the compression of the tire in each corner of the vehicle with the assumption of one-way wind and severe asymmetric unevenness, which indicates that the model used with the assumptions and designs mentioned all parameters improved the vehicle as a whole. The numerical and exact values of the vehicle response and the diagrams are shown in table 4.

Table 4. The numerical value of system responses

Parameter	Passive	Fuzzy PID		LQR		Sliding Mode		Unit
	Value	Value	Change	Value	Change	Value	Change	
Max Pitch acc	1.815	1.47	19.01%	1.595	12.12%	1.754	3.36%	***Rad/s²***
Min Pitch acc	-1.859	-1.518	18.34%	-1.332	28.35%	-1.809	2.69%	
Max Roll acc	3.286	1.911	41.84%	2.25	31.53%	2.108	35.85%	***Rad/s²***
Min Roll acc	-3.285	-1.798	45.27%	-2.169	33.97%	-1.719	47.67%	
Max CG acc	3.303	2.159	34.64%	2.601	21.25%	2.145	35.06%	***m/s²***
Min CG acc	-4.698	-3.363	28.42%	-3.863	17.77%	-3.34	28.91%	
Max Tire deflection	0.008	0.0058	27.5%	0.0054	32.5%	0.006	25%	***cm***
Min Tire deflection	-0.00009	-0.0057	36.7%	-0.0037	48.89%	-0.006	33.33%	
Max Sprung mass acc	2.785	2.195	21.18%	2.108	24.31%	2.582	7.29	***m/s²***
Min Sprung mass acc	**-3.045**	**-2.051**	**32.64%**	**-2.543**	**16.49**	**-2.436**	**20%**	

As shown in table 4, the rate of improvement of the variables in all vehicle output parameters is in the range of 10 to 50%, which varies according to the nature of the variable and the controller used. The noteworthy point is the reduction of high energy consumption and low manufacturing complexity while operating the improvement, which promises the development of this suspension system and conducting more accurate and comprehensive research.

Conclusions

This study presents the technology of a new suspension system with the right control strategy to the roll prevention of vehicle that effected in challenging conditions of maneuvers. This vehicle model includes the effects of the road profile and test conditions. These effects are made significant-high rollover risks. In summary, the work done in the three main sections is as follows:

- Suspension and vehicle modeling with its effects
- Define maneuvering and implementation on the complete vehicle model
- Definition of modern and classic control strategy and the possibility of the implementation for multi-parameter control

With this condition, a significant improvement of about 35% has been achieved to prevent overturning. This performance improvement has occurred because due to the use of the SAVGS system, the vehicle's energy consumption has been dramatically reduced compared to the active suspension. This improvement in overturning and roll performance is of great importance by reducing energy consumption and improving vertical acceleration parameters and occupant ride comfort and road holding.

In conclusion, the improvement of the vehicle's roll parameters is about 40%, and the comfort and adhesive parameters are about 25% while reducing the energy consumption of the suspension.

In future research, it is suggested that the existing model be linked to automotive analysis software and that the values of the variables be taken from the software simultaneously. By doing this, the used model can be implemented on all vehicles .Also, the possibility of performing laboratory tests and using laboratory operators to make the model more really is of paramount importance. In future work, it is possible to study different objectives of variety vehicles by using the existing model, and it is also possible to implement more stability maneuvers such as fishhook and double lane change, etc. in the model.

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