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Research Article

Improving the Stability of the Passenger Vehicle by Using an Active Stabilizer Bar Controlled by the Fuzzy Method

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The rollover phenomenon is a particularly dangerous problem. This phenomenon occurs when the driver travels at high speed and suddenly steers. Under the influence of centrifugal force, the body vehicle will be tilted and cause the wheels to lift off the road. To solve this problem, the method of using an active stabilizer bar has been proposed. The active stabilizer bar is controlled automatically by a previously designed controller. The performance of the active stabilizer bar depends on the selected control method. Previous research often only used a half-car dynamics model combined with a linear single-track dynamics model to simulate the vehicle's oscillation. In addition, most of the research focuses only on the use of linear control methods for the active stabilizer bar. Therefore, the performance of the stabilizer bar is not guaranteed. This paper focuses on establishing the model of spatial dynamics combined with the nonlinear double-track dynamics model that fully describes the vehicle's oscillation most accurately. Besides, the fuzzy control method is proposed to control the operation of the hydraulic stabilizer bar. This is a completely novel model, and it is suitable for the actual traveling conditions of the vehicle. Also, simulations are done based on different scenarios. The results of the paper showed that the values of the roll angle, the difference in the vertical force at the wheels, and the displacement of the unsprung mass were significantly reduced when the vehicle used the active stabilizer bar, which is controlled by an intelligent control method. Therefore, the stability and safety of the vehicle have been guaranteed. This result will be the basis for performing other more complex research in the future.

1. Introduction

Today, the automotive industry is developing very strongly. Compared to before, the quantity and quality of the vehicles have also been greatly improved. When the vehicle is on the road, many unexpected situations happen. These situations can cause instability and be unsafe for the driver. In particular, the phenomenon of rollover is extremely dangerous, and it can directly affect life and cargo.

Usually, there are two types of accidents that are mentioned, including turnover and rollover. A turnover occurs when the vehicle travels at a high speed and impacts an obstacle. At that time, the roof of the vehicle will be in contact with the road surface, and the wheels will point up. Actually, the turnover phenomenon rarely occurs in practice. In contrast, the phenomenon of rollover often occurs with great frequency. The vehicle is rolled over, which means that the side of the vehicle will be in contact with the road, the wheels facing the same direction. When the driver is traveling at high speed and suddenly steers, the rollover phenomenon may occur [1]. The main cause of this phenomenon is the appearance of the centrifugal force, which is proportional to the square of the vehicle's velocity. In addition, vehicles that are bulky in size are also very easy to fall into this situation when traveling at high speeds. Besides, some other factors, such as lateral wind, and external force, can also affect this problem [2].

Because of the danger of the rollover phenomenon, automotive manufacturers are always trying to improve this situation. There are several solutions to limit this phenomenon, such as using the active suspension system, integrated air suspension system, and hydraulic antiroll system, in which the method of using the stabilizer bar is more appropriate [3].

The passive stabilizer bar is used on most vehicles today. Its structure is quite simple, with high durability and low cost [4]. Therefore, its performance is not good. In many dangerous situations, the passive stabilizer bar is not able to meet the requirements set in terms of vehicle stability and safety. To improve this problem, the active stabilizer bar is equipped to replace the conventional passive stabilizer bar. The active stabilizer bar is controlled automatically by a predesigned controller based on signals received from sensors around the body of the vehicle. The structure of the active stabilizer bar (using a hydraulic motor) is introduced as shown in Figure 1. The two arms of the bar (1) are connected to the wheel hub. The back of the bar can be rotated flexibly based on two rubber bearings (2) that are attached to the chassis (3). When the body of the vehicle is tilted and other parameters of the vehicle also change, the sensor will record this change and send it to the ECU. Then, the ECU will send a current signal to the actuator of the stabilizer bar to control the opening and closing of the internal valves. At this time, high-pressure hydraulic oil is supplied to the actuator (5) through the pipeline (4). The actuator of the stabilizer bar is a hydraulic motor, the operation of which is based on the opening and closing of the valves, which are controlled by an electric current signal. When the hydraulic motor rotates, the impact force F_S is generated at the ends of the lever arm, which is attached to the wheel hub. These two force values are opposite and act on the unsprung mass to bring the vehicle body back to a stable position. The performance of the active stabilizer bar depends entirely on the controller, which is designed. Therefore, the choice of the control method for the active stabilizer bar is extremely important.

Recently, there has been quite a bit of research on the problem of rollover and methods to improve this situation. According to [5], the rollover phenomenon occurs when the vehicle changes direction suddenly at high speed. The centrifugal force that is produced by the lateral acceleration will cause the vehicle body to tilt [6]. At this time, the wheels tend to separate from the road surface [7]. If the vertical force at the wheel reaction approaches zero, the vehicle can roll over at any time. This has been demonstrated by Anh in his research [8]. To evaluate the rollover phenomenon, Phanomchoeng and Rajamani introduced the concept of the rollover index R [9]. However, this is only true when both wheels on the same side are lifted off the road, which means that the value of the roll index R will approach "1." In general, rollover problems are extremely complex and dangerous.

In order to improve this situation, the option of using an active suspension system was proposed by Yim et al. [10]. Besides, Nguyen and Hoang also proposed equipping the stabilizer bar on the vehicle in their research [11]. The stabilizer bar can be fitted on other family cars, buses, or large vehicles. Its effect is positive [12]. In addition, the stabilizer bar can be combined with several other mechatronic systems to support the vehicle's stability. This idea was proposed by Yim et al. [13]. Passive stabilizer bars are widely used on many vehicles. It is made up of an elastic steel bar with a circular, hollow inside [14]. In [15], Nguyen proposed a new method for calculating the impact force of the passive stabilizer bar on the unsprung mass. However, the effect of the passive stabilizer bar is still not very good. In some dangerous cases, the vehicle can still roll over. Therefore, the active stabilizer bar is recommended for use.

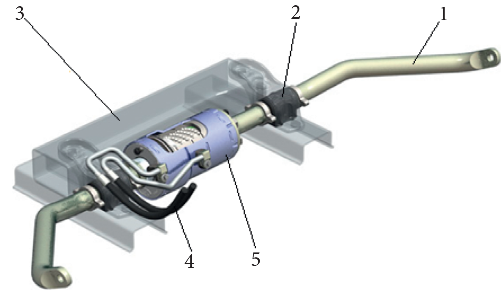


Figure 1: Hydraulic active stabilizer bar.

It is very important to control the operation of the active stabilizer bar. Zulkarnain et al. designed a linear controller LQG for the stabilizer bar [16]. In this paper, Zulkarnain only uses the half-car dynamics model combined with the conventional linear single-track dynamics model. Similarly, Vu et al. also introduced the linear parameter varying control law (LPV) for their stable bar model [17]. However, the actuator that is described in [17] is the hydraulic piston. Besides, the active stabilizer bar that uses a hydraulic motor-shaped actuator has also been researched and introduced. In [18], Varga et al. presented the LQ control law for this model. Based on Varga's ideas, Tan also published research on active stabilizer bars that use hydraulic motors with linear control methods [19]. In which, the influence of the parameters has been described more clearly. Recently, Nguyen introduced the PID control rules for hydraulic stabilizer bars [20]. For this paper, Nguyen used the model of spatial dynamics, which fully describes the effect of oscillations when a vehicle is traveling on the road. Because the vehicle is a very complex mechanical system, if conventional linear control methods are used, the performance of the stabilizer bar is not guaranteed. Therefore, more complex control methods are needed to control the operation of the hydraulic stabilizer bar.

Instead of just using the conventional linear control method, Zulkarnain et al. proposed the use of the composite nonlinear feedback control method (CNF) to improve the efficiency of the system. According to [21], the CNF controller is designed to increase the damping ratio when the system output approaches the desired threshold and avoids overshoot. Based on the idea of minimizing the error of the control signal to the lowest threshold, Konieczny et al. proposed a sliding mode control algorithm (SMC) to control the operation of the antiroll system [22]. The hydraulic actuator of the system is located on both sides of the wheel. Its efficiency is very high. Besides, intelligent control methods also bring stable effects to the vehicle's antiroll system. In [23], Muniandy et al. introduced the fuzzy PI-PD law for the controller. The parameters of the PID controller will be self-tuned by the fuzzy controller based on the error of the output signals. The results of the study have been proven through the simulation and experiment process, which is mentioned in the paper. Again, this content is expressed more specifically through [24]. Obviously, the vehicle's stability can be further improved by using intelligent control methods. Based on this idea, Marzbanrad et al. also proposed a fuzzy control algorithm using two modes [25]. However, this research has not addressed the effect of actuators on overall system performance. Recognizing the shortcomings of previous studies, Dawei et al. published a paper with the content towards controlling the operation of hydraulic actuators [26]. In this paper, the characteristics of the active stabilizer bar are shown. Besides, experiments were also conducted to prove the results of the paper. Recently, some research on controlling the active antiroll system has also been carried out and published [27, 28]. Overall, the results of these researches are very positive.

Intending to improve the stability and safety of the vehicle when steering, this paper focuses on calculating and simulating the oscillation of a vehicle equipped with a hydraulic stabilizer bar in dangerous situations. This is different from previous studies that used only half-car dynamics models integrated with the linear single-track dynamics model. This paper establishes a spatial dynamics model that describes the vehicle oscillations fully. Besides, the tire influence is also shown through the Pacejka tire model. Then, the control model of the actuator is given with fuzzy control algorithms. The contents of the paper are presented below.

2. Materials and Methods

2.1. Dynamic Model. To evaluate the vehicle body oscillation accurately, a spatial dynamics model that includes 7 degrees of freedom was used in this research (Figure 2). Separating the sprung mass (m) and the unsprung mass (m_{ij}), the equations describing the oscillation of each part are shown below:

$$\begin{aligned} mz &= \sum_{i,j=1}^{2} F_{Cij} + F_{Kij}, \\ \left(J_x + mh_\varphi^2\right)\ddot{\varphi} &= \sum_{i,j=1}^{2} \left[(-1)^{j-1}\left(F_{Cij} + F_{Kij}\right)t_{wi}\right] + \left(g \sin\varphi + a_y \cos\varphi\right)mh_\varphi, \\ \left(J_y + mh_\theta^2\right)\ddot{\theta} &= \sum_{i,j=1}^{2} (-1)^{i-1}\left(F_{Cij} + F_{Kij}\right)l_i, \\ m_{ij}\ddot{e}_{ij} &= F_{KTij} - F_{Cij} - F_{Kij} + (-1)^j F_{Si}, \quad i, j = \overline{1,2}, \end{aligned} \tag{1}$$

where a_y is the lateral acceleration in global coordinates, $a_y = \dot{v}_y + (\dot{\alpha} + \dot{\psi})v_x$.

Lateral acceleration (a_y) is what causes the vehicle's body to be tilt, which leads to a rollover phenomenon. This value is determined based on a nonlinear double-track dynamics model as shown in Figure 3.

$$\begin{aligned} M\left[\dot{v}_x - (\dot{\beta} + \dot{\psi})v_y\right] &= \sum_{i,j=1}^{2} \left(F_{xij}\cos\delta_{ij} - F_{yij}\sin\delta_{ij}\right) - F_1, \\ M\left[\dot{v}_y + (\dot{\beta} + \dot{\psi})v_x\right] &= \sum_{i,j=1}^{2} \left(F_{xij}\sin\delta_{ij} + F_{yij}\cos\delta_{ij}\right) - F_2, \\ J_z\ddot{\psi} &= \sum_{i,j=1}^{2} \left[(-1)^j\left(F_{xij}\cos\delta_{ij} - F_{yij}\sin\delta_{ij}\right)t_{wi} + (-1)^{i+1}\left(F_{xij}\sin\delta_{ij} + F_{yij}\cos\delta_{ij}\right)l_i + F_i c_i - M_{zij}\right], \end{aligned} \tag{2}$$

where $v_x = v\cos\beta$ and $v_y = v\sin\beta$.

In this research, the influence of external force (wind force) is not considered. Therefore, the value of F_i can be ignored.

To be able to determine equations (1) and (2), the forces and moments at the wheel need to be calculated specifically. There are many methods used to calculate longitudinal force (F_x), lateral force (F_y), and moment (M_z). This paper uses the

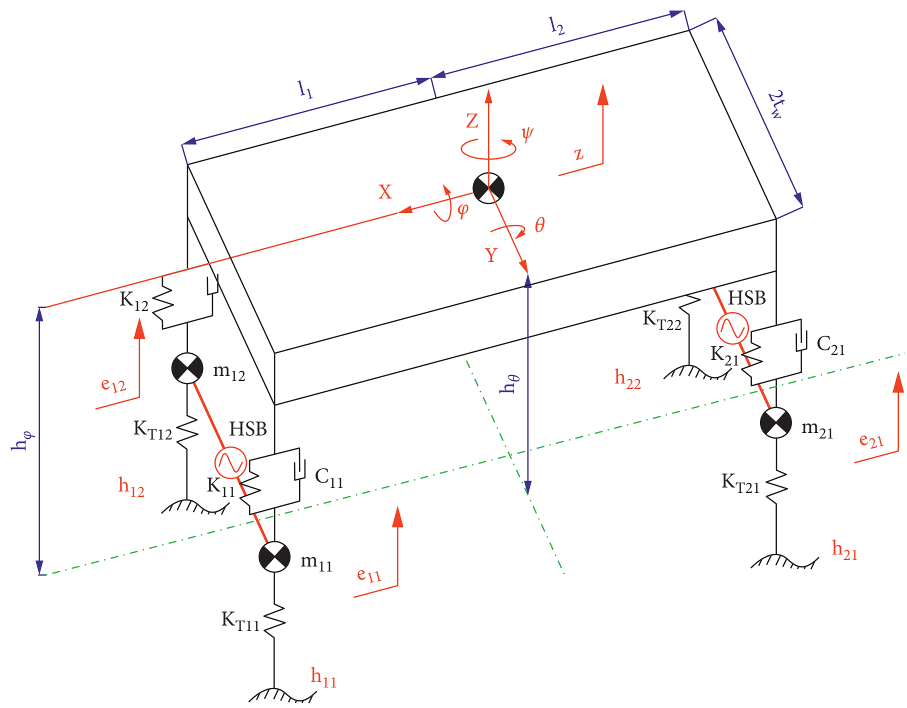


FIGURE 2: Spatial dynamics vehicle model 7 DOF.

Pacejka tire model to calculate the above values [29]. The parameters of slip ratio and slip angle at the wheel are shown in the following equations:

$$\begin{aligned}F_x &= D_x \sin\left(C_x \arctan\left[B_x\left(1-E_x\right)\left(s_x+S_{hx}\right)+E_x \arctan\left\{B_x\left(s_x+S_{hx}\right)\right\}\right]\right)+S_{vx},\\ F_y &= D_y \sin\left(C_y \arctan\left[B_y\left(1-E_y\right)\left(\alpha+S_{hy}\right)+E_y \arctan\left\{B_y\left(\alpha+S_{hy}\right)\right\}\right]\right)+S_{vy},\\ M_z &= D_z \sin\left(C_z \arctan\left[B_z\left(1-E_z\right)\left(\alpha+S_{hz}\right)+E_z \arctan\left\{B_z\left(\alpha+S_{hz}\right)\right\}\right]\right)+S_{vz},\end{aligned} \tag{3}$$

where $D_{x,y,z}$; $C_{x,y,z}$; $B_{x,y,z}$; $E_{x,y,z}$; $S_{vx,vy,vz}$; and $S_{hx,hy,hz}$ are coefficients of the tire model. Also, s_x is the slip ratio, and α is the slip angle of the tire.

2.2. Hydraulic Motor Model. The schematic diagram of the hydraulic actuator is shown in Figure 4. The hydraulic actuator consists of a rotating motor that is controlled by valves, which are widely used in engineering fields. In particular, the changes in the volume of liquid inside the motor are not constant; they vary in a discontinuous depending on the rotational speed of the motor shaft.

The hydraulic actuator is controlled by the current signal $i(t)$ [30]. When current is supplied, the valve inside the hydraulic motor will travel. The displacement of the servo valve X_v depends on current signal $i(t)$. It is shown in the following equation:

$$\dot{X}_v \tau + X_v - K_v i(t) = 0. \tag{4}$$

The linearization equation describes the flow of fluid ΔQ through the valve as follows:

$$\Delta Q = K_{qi} X_v - K_c \Delta P. \tag{5}$$

Assuming that $V_1 = V_2 = 0.5V_t$ and $\Delta P = P_1 - P_2$, the flow of the liquid can be calculated as follows:

$$\Delta Q = D_m \dot{\theta}_m + C_m \Delta P + \frac{V_t}{4\beta_e} \Delta \dot{P}, \tag{6}$$

where θ_m is the rotation angle of the motor shaft.

From equations (5) and (6),

$$K_{qi} X_v = D_m \dot{\theta}_m + K_{ce} \Delta P + \frac{V_t}{4\beta_e} \Delta \dot{P}, \tag{7}$$

where $K_{ce} = K_c + C_m$.

The torque balance equation at the output of the motor shaft is as follows:

$$D_m \Delta P = J_a \ddot{\theta}_m + B_m \dot{\theta}_m + T_r. \tag{8}$$

After the actuator's dynamics model has been established, the controller needs to be designed to be able to control the actuator's operation appropriately.

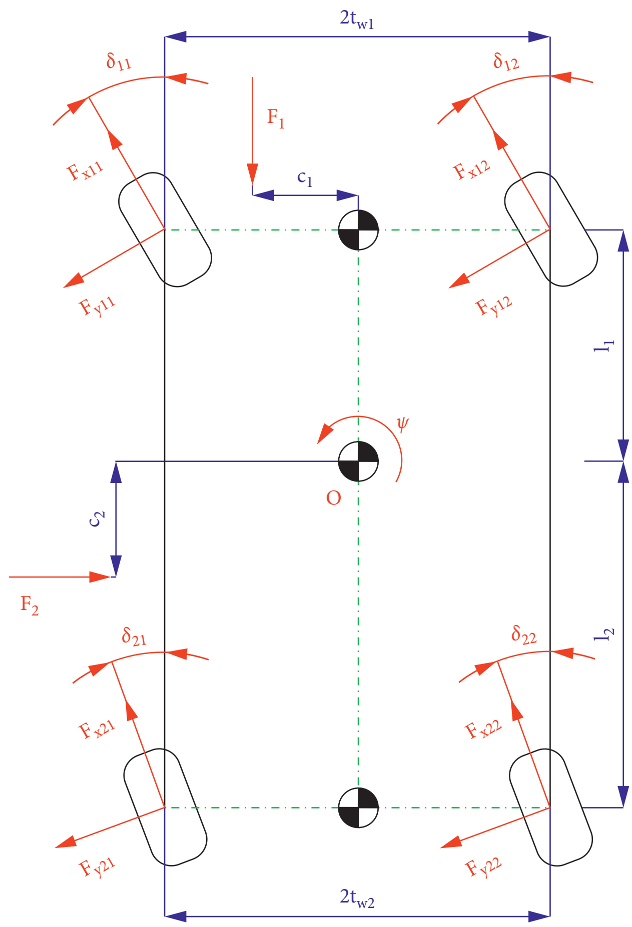


Figure 3: Nonlinear double-track dynamics model (3 DOF).

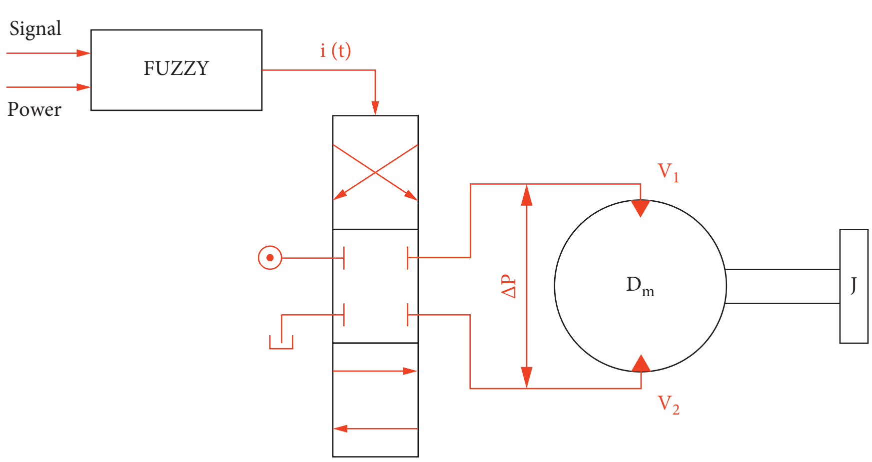


Figure 4: Hydraulic motor.

2.3. Designing the Controller. As discussed in the previous section, there are many methods used to control the operation of the hydraulic motor. Each method has its advantages and disadvantages. In this paper, the intelligent control method with fuzzy law is used to control the process of generating force F_S for the hydraulic actuator.

Unlike traditional linear or nonlinear control methods, the fuzzy control law is an intelligent control method. This method makes it possible to achieve a high level of system performance. In particular, complex systems like in this research can use the fuzzy control method. Besides, the fuzzy control method helps ensure the efficiency of the system, while the uncertainty factors change continuously.

The fuzzy controller consists of three main stages: fuzzification, rules, and defuzzification. The fuzzification stage helps convert the explicit values (the signal received from the sensor) into the linguistic values that are used in the fuzzy controller.

$$A' = \text{fuzz}(x\prime). \tag{9}$$

The second stage of the controller is the rules. At this time, the linguistic value of the fuzzy set will be converted to an explicit value based on the defined basis set. The algorithm diagram for this stage is given as shown in Figure 5. According to this figure, positions from A_1 to A_9 are the limits of the controller's input parameter.

The third stage of the controller is the defuzzification process. This process is performed based on the control rules that have been established for the controller. For each limit position, the output parameter will be set specifically. Therefore, if the input value is not at the limit position, this controller can easily interpolate its position. From there, the corresponding output value can be reasonably calculated.

3. Results and Discussion

3.1. Simulation Conditions. The process of simulating the vehicle's oscillations when traveling is performed based on two types of dangerous steering, including J-turn steering and fishhook steering (Figure 6) [20]. For each type of steering, the vehicle's speed will be shown in two cases, including case 1: $v_1 = 60$ km/h and case 2: $v_2 = 90$ km/h. In these situations, the vehicle can use the passive stabilizer bar (Pa), the active stabilizer bar controlled by the linear controller (PC), the active stabilizer bar controlled by the fuzzy controller (FC), or not use the stabilizer bar (none).

The technical parameters of the vehicle and the hydraulic actuator are given in Table 1 [31].

3.2. Simulation Results

3.2.1. J-Turn Steering. With the J-turn steering type, the results of two simulation processes are given below.

(1) Case 1: $v_1 = 60$ km/h. In this case, the car travels at a steady speed: $v_1 = 60$ km/h. This is the actual speed value, suitable for most traveling conditions on the road. Figure 7 shows the change in the roll angle of the sprung mass over time. The maximum value of the roll angle can reach $\varphi_{\max} = 5.77^\circ$ when the vehicle does not use the stabilizer bar. If the passive stabilizer bar is used, this value will decrease slightly, reaching 5.35°. This change is not much. However, when the hydraulic stabilizer bar is equipped instead of the passive stabilizer bar, the change is significant. The maximum value of the roll angle of the sprung mass decreased sharply, to only 4.90° and 4.59°, respectively, in the situation of using the PID controller and the fuzzy controller for the stabilizer bar. This change is dependent on the impact force, which is generated by the stabilizer bar.

The graph in Figure 8 shows the impact force values of the stabilizer bar. According to this result, the value of the impact force of the stabilizer bar at the front axle will be larger than that of the rear axle. However, this difference is small. The value of the impact force when the vehicle uses the hydraulic stabilizer bar controlled by the fuzzy controller is the largest. It can be up to 2,569.7 N and 2,486.9 N, respectively, for each axle. For the situation of a vehicle using the hydraulic stabilizer bar controlled by the conventional linear controller, the value of the impact force will be smaller. It only reaches 2,036.9 N and 1,951.7 N, respectively. The value of the impact force in the other case (mechanical stabilizer bar) is the smallest. It is only about 30% more than in the first case. Obviously, if the value of the impact force F_S is too small, the vehicle's stability support effect is not guaranteed. Therefore, the use of the fuzzy controller in this research has brought very positive results.

The peak of the impact force value appears at different times corresponding to the situations. Firstly, the delay of the control process can cause this situation. Secondly, because the actuator's inertia force is large, it will continue to operate for a short time afterward. However, this does not affect the safety of the vehicle.

The vertical force at the wheel is a value that characterizes the stability of the vehicle. If this value is too small or it even tends to approach zero, the rollover phenomenon may occur. Figure 9 shows the change in the value of the vertical force in this simulation case. According to the results obtained from these graphs, the value of the vertical force of the wheel at position (21) tends to decrease the most. This is completely consistent with reality. Their minimum value is 3,486.9 N, 3,044.3 N, 2,508.2 N, and 1,715.0 N, respectively. When the vehicle uses the hydraulic stabilizer bar, the difference in the vertical reaction of the two wheels at each ΔF_{zi} bridge will be controlled stably. For the vehicle using the hydraulic stabilizer bar that is controlled by a PID controller, the difference value of ΔF_{zi} will be smaller than that of the conventional passive stabilizer bar. If the fuzzy controller is used to replace the PID controller, this change occurs much more strongly. Because the force generated by the actuator is quite large, the difference between ΔF_{zi} will tend to be opposite to that of the other three cases.

The change in the vertical force's value at the wheel ΔF_{zij} has a direct effect on the displacement of the unsprung mass e_{ij} (Figure 10). When the value of F_{zij} approaches zero, that is, the value of e_{ij} approaches the maximum limit, lateral instability will occur. This can even lead to the rollover phenomenon. If the vehicle uses the hydraulic stabilizer bar controlled by a fuzzy controller, the maximum value of displacement of the unsprung mass is only 3.45 mm. Conversely, if the vehicle is not equipped with the stabilizer bar, this value can go up to 14.01 mm. Obviously, the hydraulic stabilizer bar that is controlled by the fuzzy controller works very well compared to other cases.

In this case, the vehicle only moves at an average speed of $v_1 = 60$ km/h. Therefore, the instability of the vehicle is still not much. In order to demonstrate the performance of the fuzzy controller that controls the hydraulic stabilizer bar's operation, it is necessary to perform simulations in a more hazardous case.

(2) Case 2: $v_2 = 90$ km/h. In this case, the vehicle moves at very high speeds and may fall into dangerous situations. Similar to Case 1, the outputs include the roll angle, the impact force, the vertical force at the wheel, and the displacement of the unsprung mass.

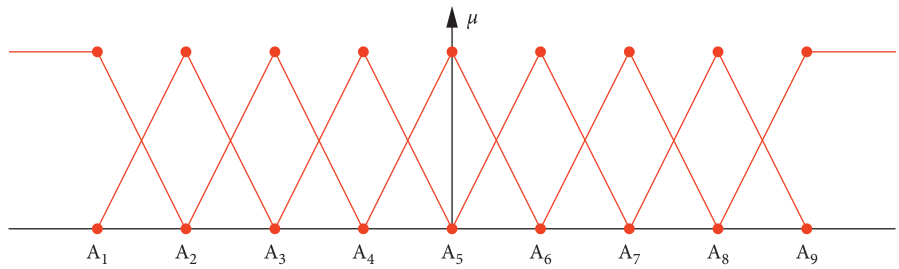


FIGURE 5: Rules of the fuzzy controller.

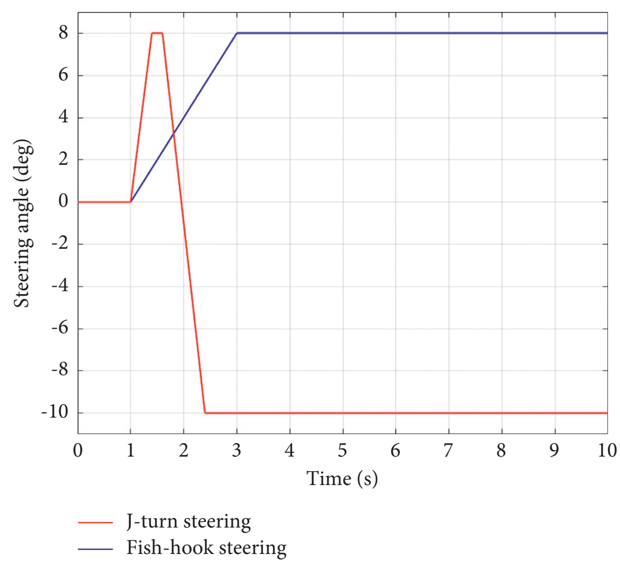


FIGURE 6: Steering angle.

TABLE 1: The technical parameters.

Symbol	Description	Value	Unit
M	Sprung mass	1,655	kg
m_{ij}	Unsprung mass	48	kg
M	Total mass	1847	kg
h_φ	Distance from center of gravity to roll axis	0.51	m
h_θ	Distance from center of gravity to pitch axis	0.44	m
t_{wi}	Half of the track width front/rear axle	0.725/0.720	m
l_i	Distance from center of gravity to front/rear axle	1.25/1.65	m
J_x	Moment of inertia of the x-axis	665	kg m^2
J_y	Moment of inertia of the y-axis	2,480	kg m^2
J_z	Moment of inertia of the z-axis	2,610	kg m^2
T	Time constant	0.005	S
K_v	Servo valve gain	0.025	m/A
K_{qi}	Valve flow gain coefficient	0.02	m^2/s
K_{ce}	Total flow pressure coefficient	4×10^{-11}	m^5/Ns
V_t	Total volume of trapped oil	1×10^{-3}	m^3
β_e	Effective bulk modulus of the oil	6×10^{6}	N/m^2
D_m	Flow per revolution	1.6×10^{-5}	m^3/rad
B_m	Viscous friction coefficient	12.5	Nms/rad
J_a	Moment of inertia of the hydraulic motor	2.5	kg m^2

In Figure 11, the change in the roll angle over time is shown. Obviously, these values have increased a lot compared to Case 1. If the vehicle does not use the stabilizer bar, the maximum roll angle can reach $\varphi_{\max} = 8.36^\circ$. When the passive stabilizer bar is used, this value drops slightly, to only 7.78°. If the hydraulic stabilizer bar that is controlled by the designed controller is fitted, the value of the roll angle of the vehicle body will be greatly reduced. At this moment, it is

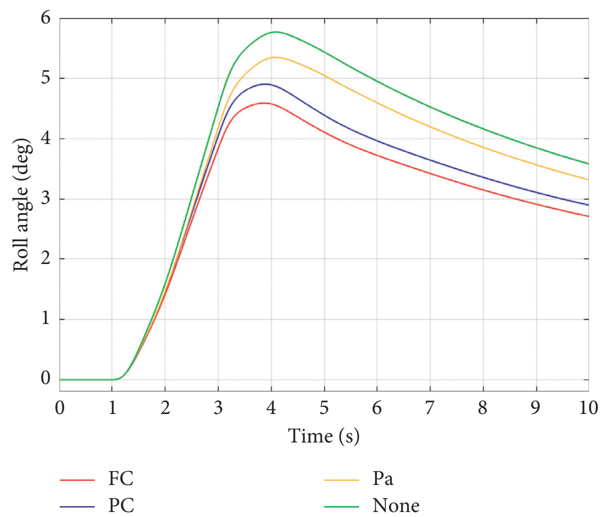


FIGURE 7: Roll angle (J-turn, v_1).

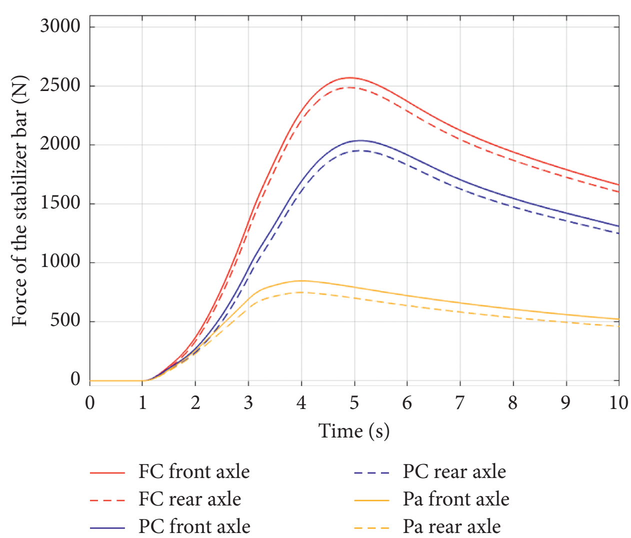


FIGURE 8: Impact force of the stabilizer bar (J-turn, v_1).

only about 7.17° and 6.73°, respectively, with the PID controller and fuzzy controller. The difference in the roll angle between using the fuzzy controller and not using the stabilizer bar is quite large, $\varphi_{\max} = 1.63°$.

The value of the impact force of the stabilizer bar is shown in the graph in Figure 12. In fact, the value of the force generated from the hydraulic stabilizer bar that is controlled by the fuzzy controller is larger than the other two cases. Its maximum value can be up to 3,715.2 N and 3,597.8 N, respectively, for each axle. Meanwhile, the stabilizer bar controlled by the linear controller can only produce lower forces, 2,945.5 N and 2,824.8 N, respectively. At the same time, the value of the passive stabilizer bar is very low, only about 1,232.5 N and 1,089.1 N If the F_S impact force is too low, the vehicle's stability will not be improved.

The values for the vertical force at the wheel F_{zij} are extremely important. It is an indication of possible lateral instability. In the case of the vehicle traveling at a high speed, $v_2 = 90$ km/h, this change is very large. If the vehicle does not have a stabilizer bar, the minimum value of the vertical force at the wheel is only about 649.0 N (Figure 13). This is a very small value, and it is close to the dangerous threshold. If the vehicle uses the passive stabilizer bar, this value has increased. It can reach 1,790.4 N. Besides, the active stabilizer bar can improve this problem more optimally. When the vehicle is equipped with the hydraulic stabilizer bar controlled by the linear controller, this value increases to 2,586.4 N. In addition, if the PID controller is replaced by a fuzzy controller, the minimum value of the vertical force at the wheel can reach 3,205.9 N, which is the safety threshold of the vehicle when moving.

The hydraulic stabilizer bar generates a large impact force F_S to ensure that the change in the vertical force at the wheel is minimal. Therefore, the displacement of the unsprung mass e_{ij} will also be improved. Figure 14 shows that the maximum value of the displacement of the unsprung

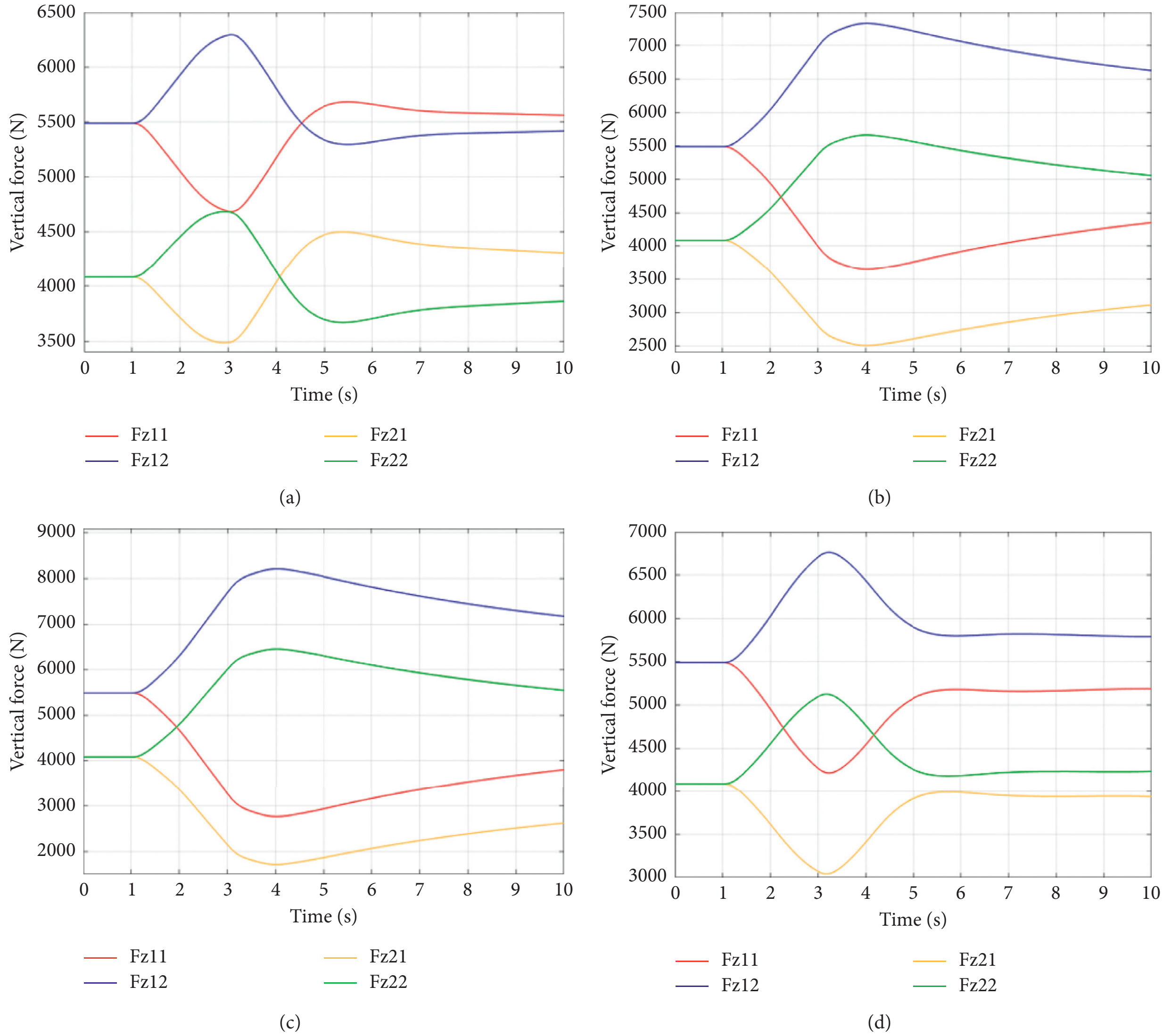


FIGURE 9: Vertical force: (a) FC, (b) PC, (c) Pa, and (d) none (J-turn, v_1).

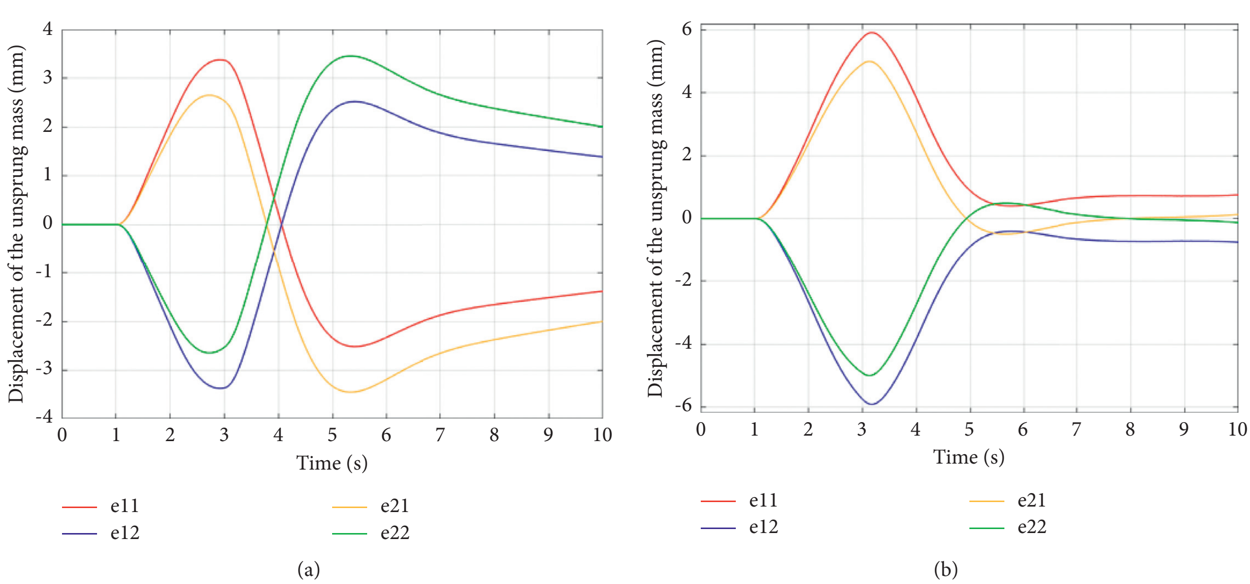


FIGURE 10: Continued.

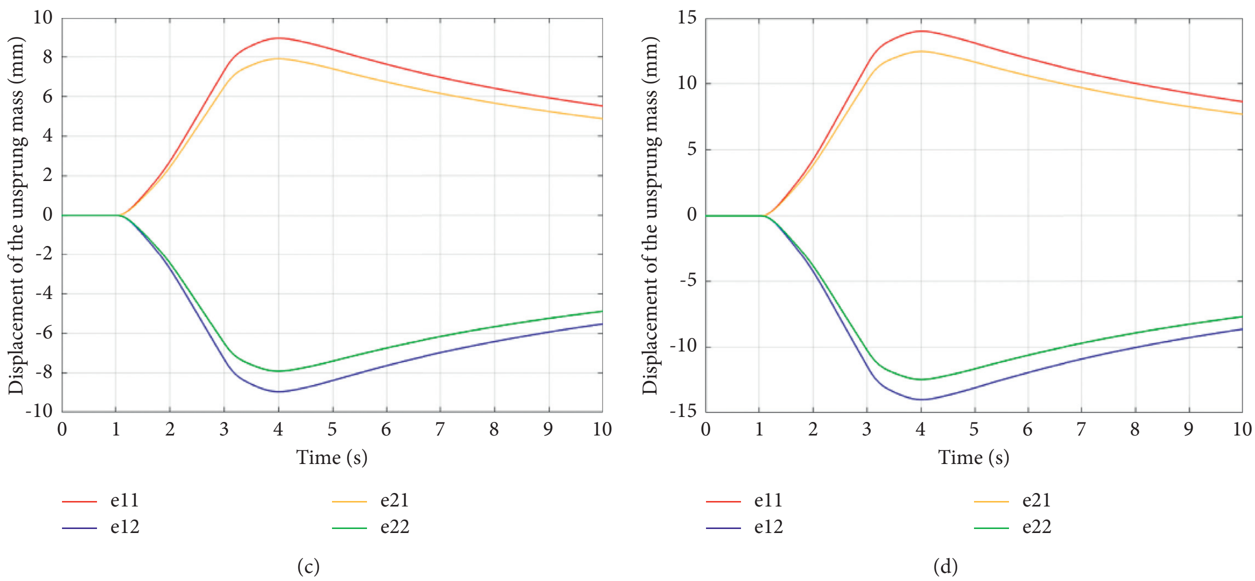


Figure 10: Displacement of the unsprung mass: (a) FC, (b) PC, (c) Pa, and (d) none (J-turn, v_1).

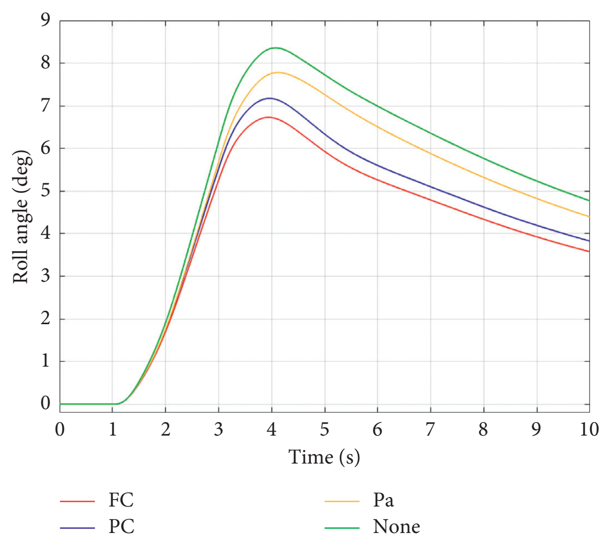


Figure 11: Roll angle (J-turn, v_2).

mass is only 5.20 mm. To achieve this value, the vehicle needs to be equipped with a hydraulic stabilizer bar that is controlled by the fuzzy controller. If the linear controller is used for the hydraulic stabilizer bar, this value will be larger, about 8.52 mm. In the case of vehicles using only the conventional mechanical stabilizer bar, this value is quite large, up to 13.03 mm. Conversely, if the vehicle does not use the stabilizer bar, the maximum value of the displacement of the unsprung mass can be as high as 20.31 mm; lateral instability can occur at any time.

3.2.2. Fishhook Steering. Compared to the normal J-turn steering type, the fishhook steering can cause many dangerous situations for the vehicle. In this section, two travel conditions will also be performed to simulate and evaluate the results of the research.

Vehicles often use the fishhook steering type when drivers want to change lanes or avoid obstacles. The value of the steering acceleration when the vehicle uses this type of steering is very high. Therefore, the results obtained from this situation will be very different from the first situation.

(1) Case 1: $v_1 = 60$ km/h. When the vehicle travels at a speed of $v_1 = 60$ km/h and performs fishhook steering, the roll angle of the sprung mass will change as shown in Figure 15. In the first phase, the value of the roll angle is not large because the steering angle is quite small. In contrast, this value in the second phase is quite large. The maximum value of the roll angle φ_{max} can reach 7.07° if the vehicle does not use the stabilizer bar. Besides, if the hydraulic stabilizer

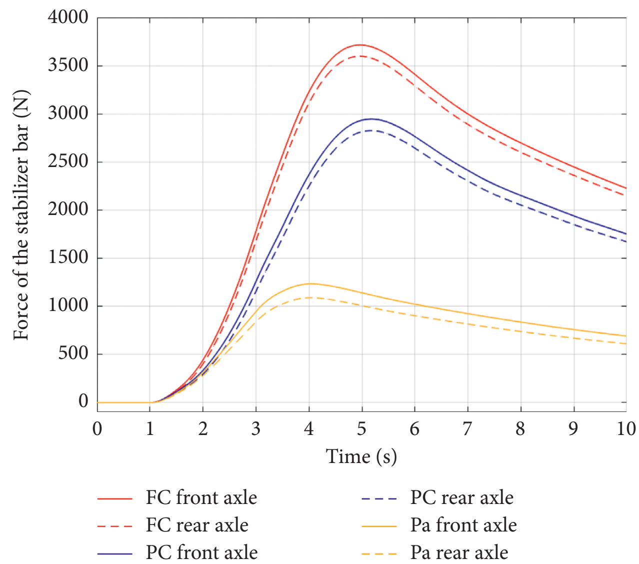


FIGURE 12: Impact force of the stabilizer bar (J-turn, v_2).

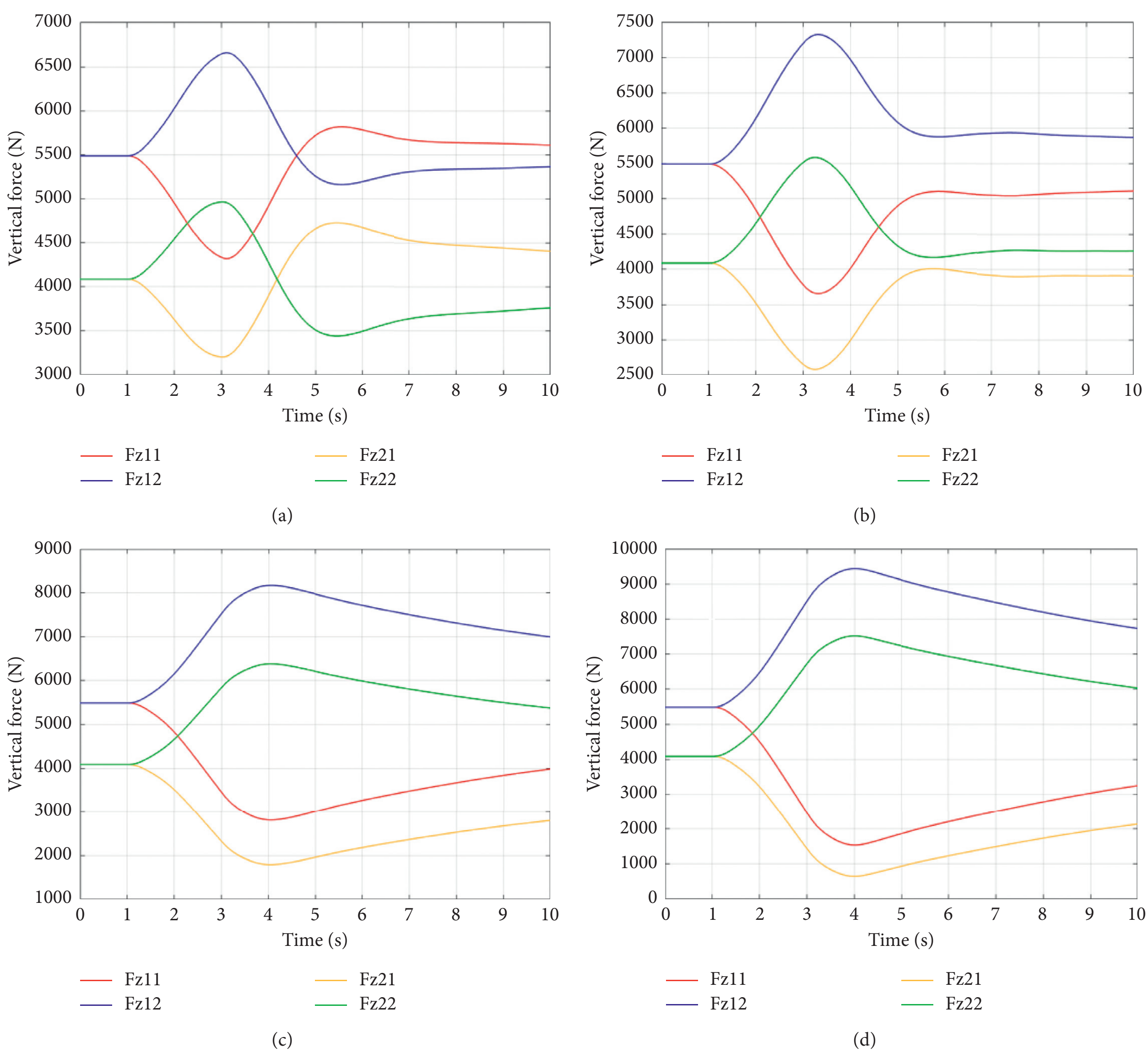


FIGURE 13: Vertical force: (a) FC, (b) PC, (c) Pa, and (d) none (J-turn, v_2).

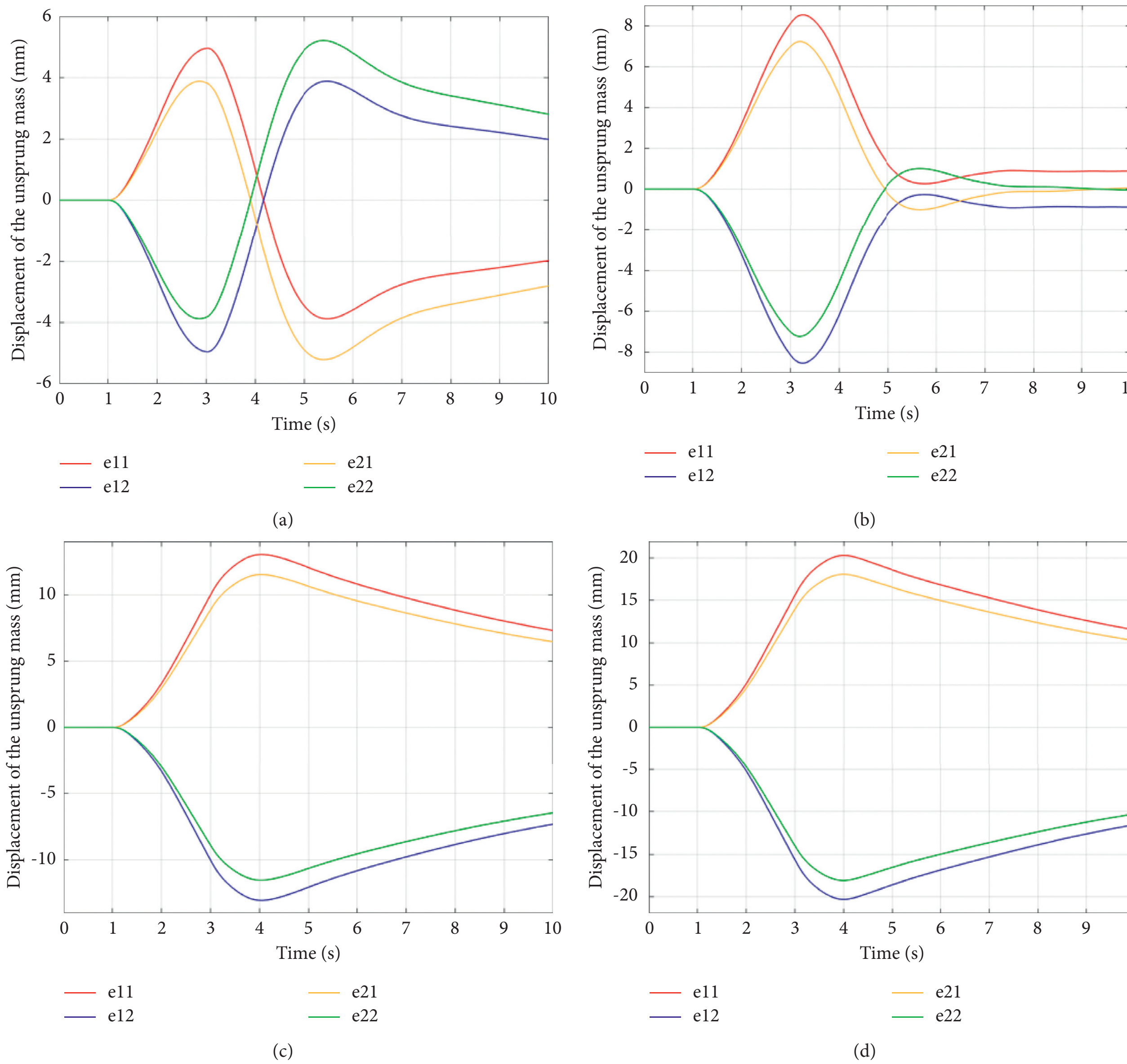


Figure 14: Displacement of the unsprung mass: (a) FC, (b) PC, (c) Pa, and (d) none (J-turn, v_2).

bar that is controlled by the fuzzy controller is equipped, this value is only 6.01°. Compared to the above situation, this value has been considerably reduced.

The value of the impact force that is produced by the stabilizer bar F_S also changes continuously. The fuzzy controller that was established in this research enables the hydraulic stabilizer bar to generate a larger torque than the other two cases. Based on the results from Figure 16, this can reach 3,145.1 N and 3,056.0 N for each axle, respectively. Meanwhile, the conventional linear controller can only produce smaller torque, only about 2,637.0 N and 2,543.7 N.

The value of the vertical force at the wheel changes following the law of the fishhook steering angle. Figure 17 shows this change when the vehicle uses the hydraulic stabilizer bar, which is controlled by the fuzzy controller. In the first phase, when the impact force of the actuator is small, the change in the value of the vertical force is completely compatible with the roll angle of the sprung mass. In the second phase, when the impact force generated by the actuator is large, the change in the value of the vertical force at the wheel has the opposite direction to the roll angle of the vehicle body. The minimum value of the vertical force is 2,435.8 N, which is obtained at the wheel position (22). Besides, the minimum value of the vertical force at the wheel when the vehicle uses the PID controller for the stabilizer bar is 2,036.3 N. In the case of a vehicle equipped with only the passive stabilizer bar, the value of the force F_{zij} is only about 2,151.1 N This value is slightly larger than the vehicle using the active stabilizer bar controlled by a linear controller. Obviously, in many special cases, the linear controller cannot be used to its full potential. If the vehicle does not have a stabilizer bar, the value of the vertical force at the wheel drops sharply. It is only 1,178.0 N. A dangerous situation can occur.

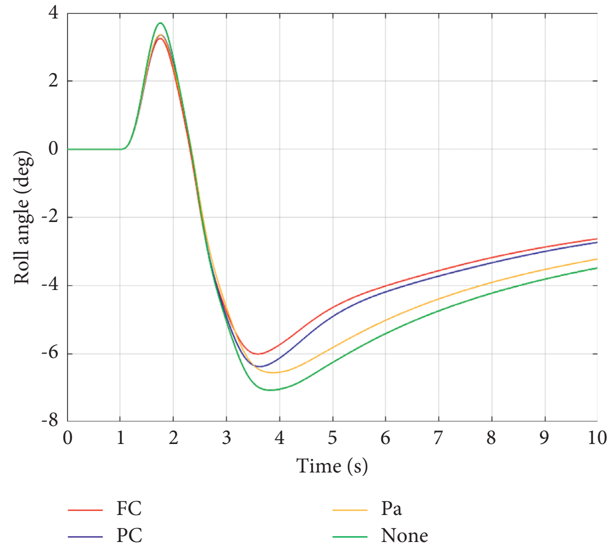


FIGURE 15: Roll angle (fishhook, v_1).

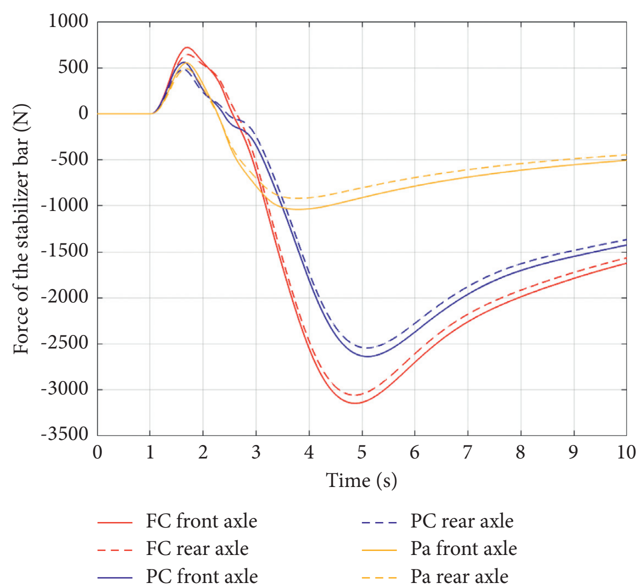


FIGURE 16: Force of the stabilizer bar (fishhook, v_1).

The displacement of the unsprung mass e_{ij} varies depending on the value of the vertical force at the wheel F_{zij}. This value is too large, which also means that the vehicle falls into a state of lateral instability. Even so, the rollover phenomenon can happen. In the case of the vehicle using the fuzzy controller to control the hydraulic stabilizer bar operation, the displacement of the unsprung mass was significantly reduced. Its maximum value is only 9.37 mm (Figure 18). Meanwhile, if the vehicle uses the active stabilizer bar, which is controlled by a PID controller, or uses the passive stabilizer bar, their values are equivalent, 11.71 mm and 10.98 mm. Therefore, the linear controller is completely ineffective in this case. In particular, the displacement of the unsprung mass is very large, reaching 17.18 mm if the vehicle does not have a stabilizer bar.

(2) Case 2: $v_2 = 90$ km/h. Fishhook steering at the high speed of $v_2 = 90$ km/h is extremely dangerous. Then, the problem of rollover can happen at any time. The change of roll angle over time is shown in Figure 19. Indeed, the maximum roll angle φ_{max} is very large. If the vehicle did not have the stabilizer bar, the vehicle would have rolled over at time $t = 3.38$ s. At the time the vehicle rollover phenomenon occurs, the maximum roll angle of the vehicle only reached $\varphi_{max} = 7.54°$. If the vehicle uses the stabilizer bar, this phenomenon can be improved. Besides, the threshold for the roll angle of the vehicle may be increased.

In the case of danger, the value of the impact force generated by the stabilizer bar is also much larger than in other cases. According to Figure 20, the maximum value of the impact force generated by the fuzzy controller can be up

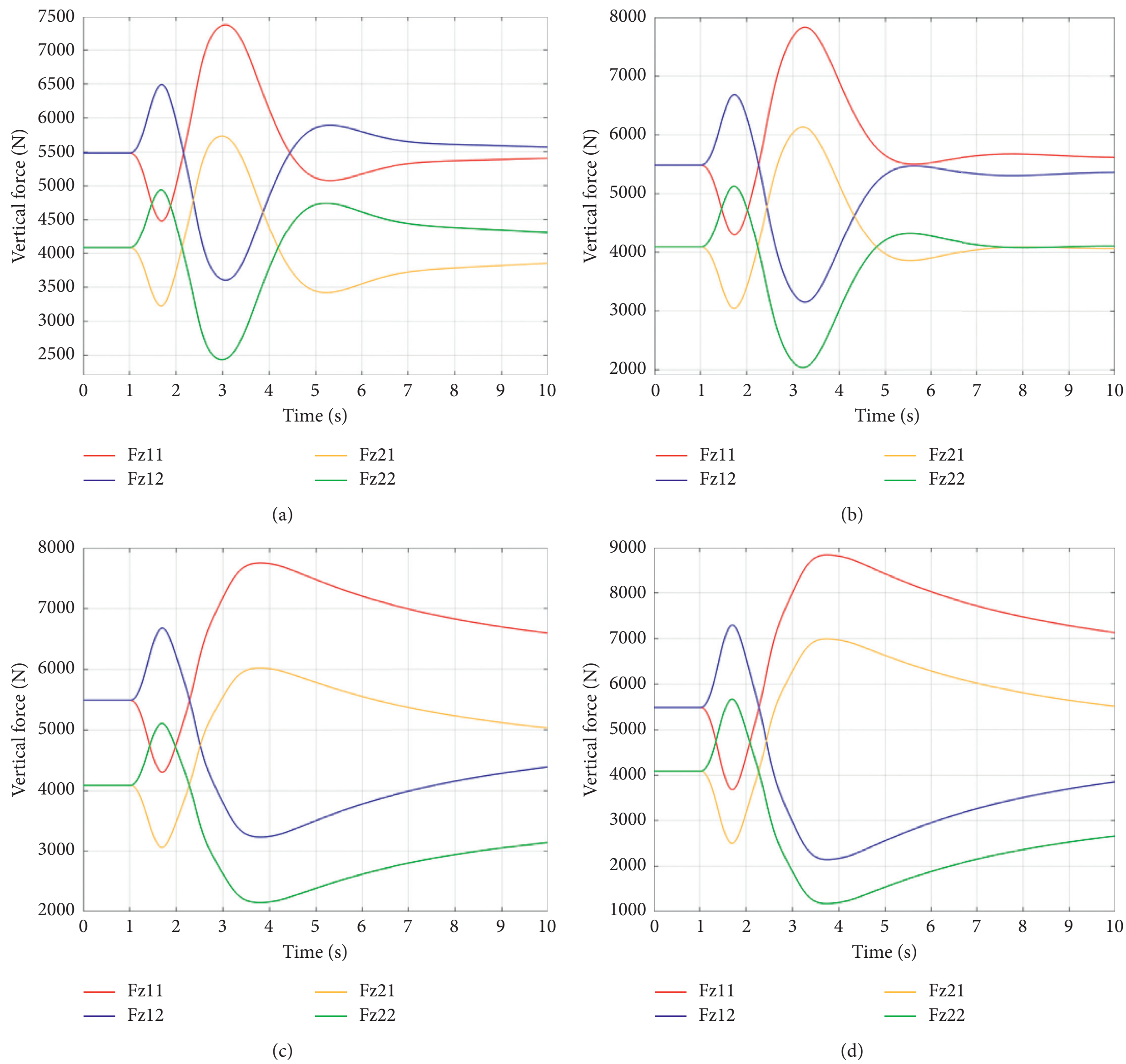


Figure 17: Vertical force: (a) FC, (b) PC, (c) Pa, and (d) none (fishhook, v_1).

to 4,414.5 N and 4,289.8 N. Meanwhile, the linear controller can only produce less force, reaching 3,630.9 N and 3,501.5 N.

When the vehicle is traveling at high speed and using this type of steering, the wheels tend to separate from the road surface. At time $t = 3.38$ s, the value of F_{z22} will equal zero if the vehicle has no stabilizer bar. If the vehicle only uses the conventional mechanical stabilizer bar, this value is only about 1,162.6 N (Figure 21). When the vehicle is equipped with the hydraulic stabilizer bar controlled by a PID controller, this value decreases slightly, reaching 947.3 N. In fact, the PID controller cannot solve complex problems like this. The efficiency of the stabilizer bar that uses a linear controller is only comparable to that of the passive stabilizer. However, the fuzzy controller that has been designed in this paper can assist the vehicle in this dangerous situation. The minimum value of the vertical force at the wheel can be reached at 1,693.8 N. Thus, the rollover phenomenon can be greatly improved.

Similar to the cases simulated above, the displacement of the unsprung mass will depend on the value of the vertical force at each wheel. Through the graphs in Figure 22, their maximum values are 13.58 mm, 17.93 mm, and 16.60 mm, respectively, corresponding to specific technical conditions.

With the above results obtained, the stabilizer bar has a noticeable effect when equipped on the vehicle. In addition, the fuzzy controller, which controls the hydraulic stabilizer bar operation, can help improve the stability and safety of the vehicle when traveling at high speed.

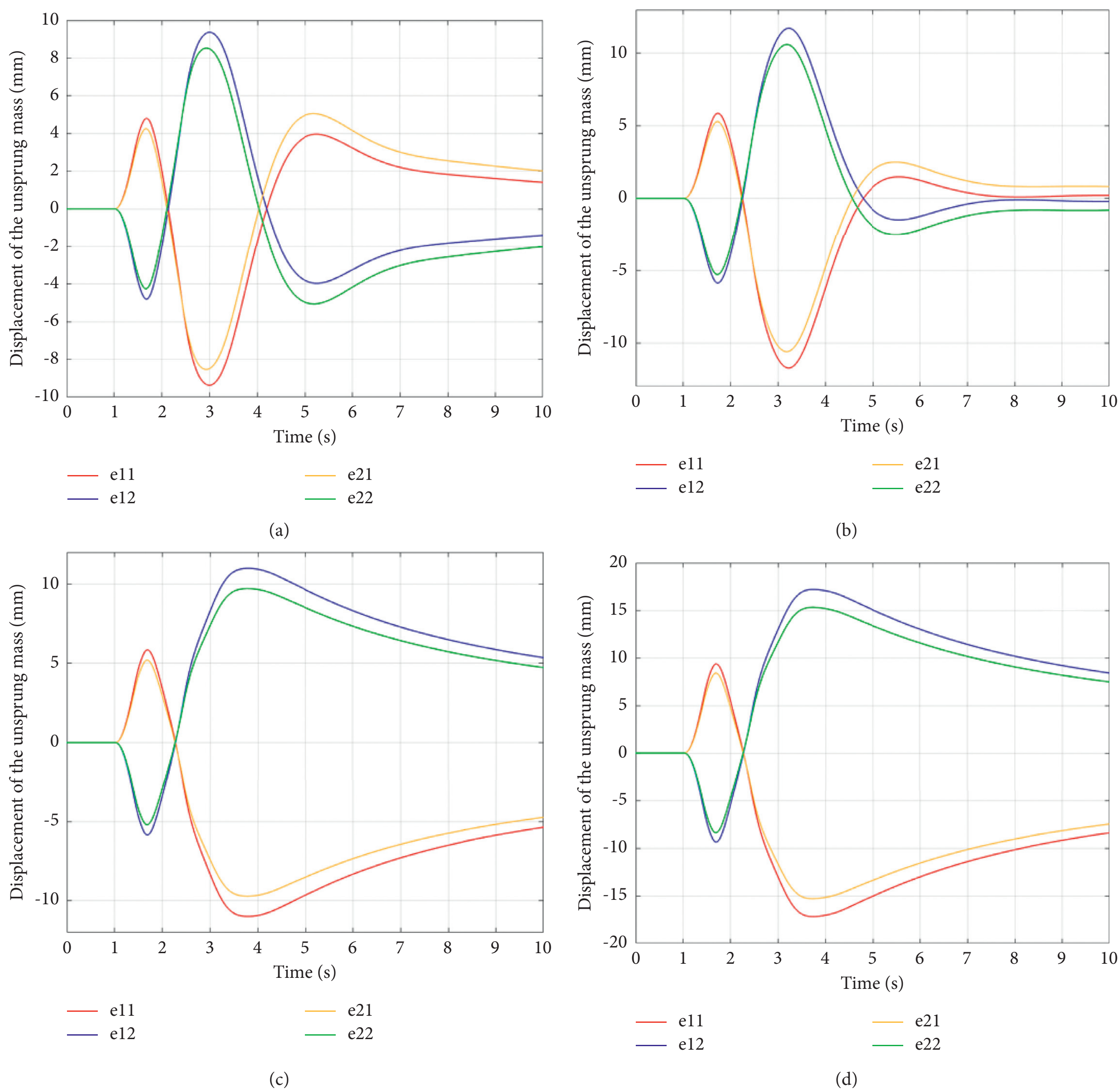


Figure 18: Displacement of the unsprung mass: (a) FC, (b) PC, (c) Pa, and (d) none (fishhook, v_1).

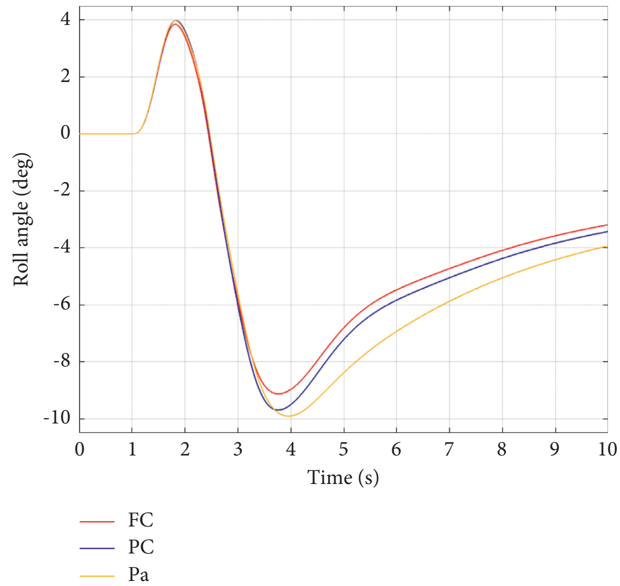


Figure 19: Roll angle (fishhook, v_2).

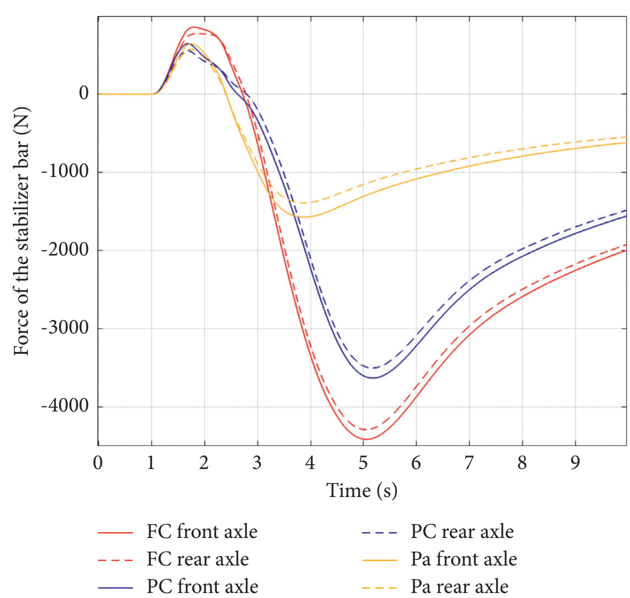
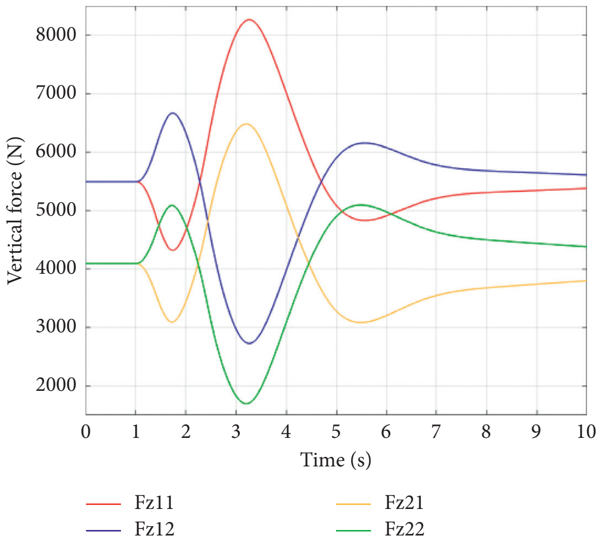
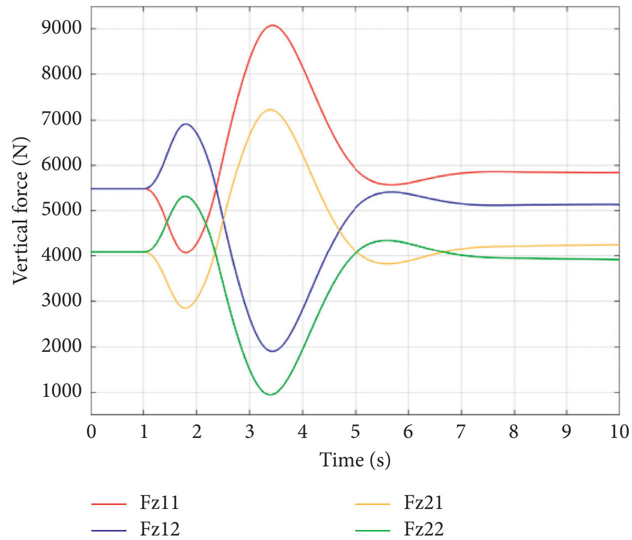


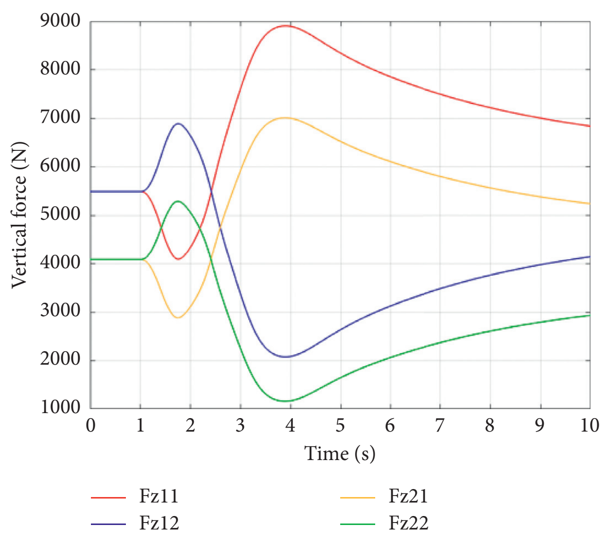
Figure 20: Force of the stabilizer bar (fishhook, v_2).



(a)

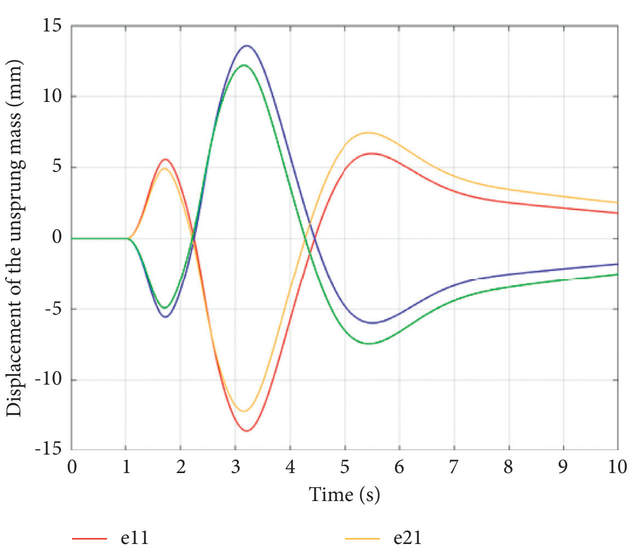


(b)

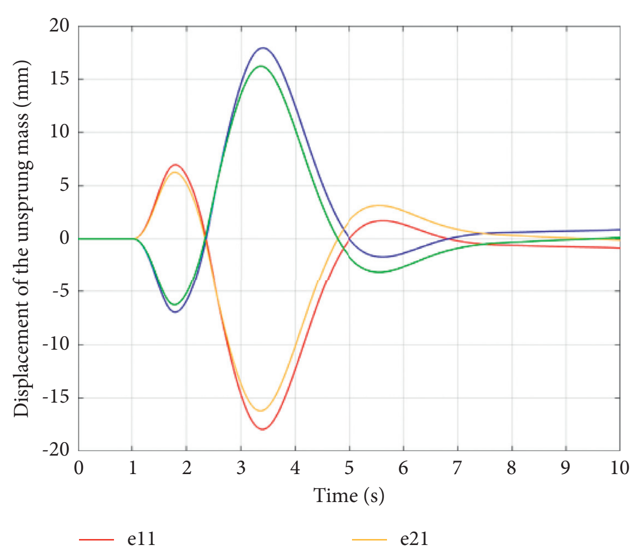


(c)

FIGURE 21: Vertical force: (a) FC, (b) PC, and (c) Pa (fishhook, v_2).

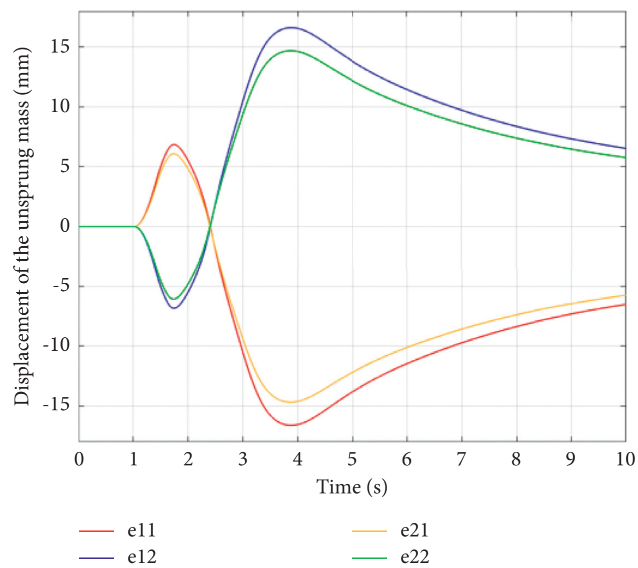


(a)



(b)

FIGURE 22: Continued.



(c)

Figure 22: Displacement of the unsprung mass: (a) FC, (b) PC, and (c) Pa (fishhook, v_2).

4. Conclusions

The rollover phenomenon is a very dangerous problem. It can happen at any time in many unexpected situations. To overcome this, the active stabilizer bar is recommended for use in vehicles. The active stabilizer bar that is simulated in this paper is a hydraulic actuator, which is controlled automatically through the previously established controller. The performance of the stabilizer bar depends on the control method that has been designed and selected.

This paper proceeds to establish the model of the spatial dynamics of the vehicle combined with the nonlinear double-track dynamics model that fully describes the factors affecting the vehicle's oscillation. Besides, the dynamics model of the hydraulic actuator is also clearly shown. The fuzzy control law is proposed to control the operation of the hydraulic actuator. The control law is selected based on actual experience, through previous researches.

The results of the paper show that when the vehicle uses the active stabilizer bar, parameters such as the roll angle, the difference in vertical force at the wheel, the displacement of the unsprung mass, and so on are all reduced significantly. According to the results of the simulation process, if the active stabilizer bar is controlled by the fuzzy controller, the maximum roll angle of the vehicle can be reduced by more than 10% compared to using the passive stabilizer bar. Also, the value of impact force that is generated by the active stabilizer bar is 2.8 times that of the corresponding mechanical stabilizer bar. Besides, the fuzzy control law also shows outstanding advantages compared with other conventional control methods. As a result, the vehicle's stability and safety have been improved. Compared with other control algorithms, the fuzzy control algorithm is quite complex. The control process relies heavily on fuzzy rules, which are used for the controller. It is very difficult to establish the fuzzy rule correctly. This depends on the experience of the expert as well as some experiments and simulations that have been done before. In some special cases, the fuzzy controller's performance is still not good. In the future, fuzzy control algorithms can be combined with some complex algorithms such as SMC, ANN, and so on to improve the efficiency of the system.

Nomenclature

φ: Roll angle (°)
θ: Pitch angle (°)
ψ: Yaw angle (°)
α: Slip angle (°)
β: Heading angle (°)
δ: Steering angle (°)
a_y: Lateral acceleration (m/s^2)
e_{ij}: Displacement of the unsprung mass (m)
F_{Cij}: Force of the damper (N)
F_{Kij}: Force of the spring (N)
F_{KTij}: Force of the tire (N)
F_{Si}: Impact force (N)
F_{xij}: Longitudinal force (N)
F_{yij}: Lateral force (N)
F_i: External force (N)
c_i: Distance from the center of gravity to external force (m)
g: Gravitational acceleration (m/s^2)
h_φ: Distance from the center of gravity to roll axis (m)
h_θ: Distance from the center of gravity to pitch axis (m)
J_x: Moment of inertia of the x-axis (kg m^2)
J_y: Moment of inertia of the y-axis (kg m^2)
J_z: Moment of inertia of the z-axis (kg m^2)
l_i: Distance from the center of gravity to front/rear axle (m)
m: Sprung mass (kg)
M: Total mass (kg)

m_{ij}: Unsprung mass (kg)
t_{wi}: Half of the track width front/rear axle (m)
v_x: Longitudinal velocity (m/s)
v_y: Lateral velocity (m/s)
z: Displacement of the sprung mass (m)
CNF: Composite nonlinear feedback
ECU: Electronic control unit
HSB: Hydraulic stabilizer bar
LQR: Linear quadratic regulator
LQG: Linear quadratic Gaussian
LQ: Linear quadratic
LPV: Linear parameter varying
PID: Proportional integral derivative
SMC: Sliding mode control.

Data Availability

The data used to support this research are included within this paper.

Conflicts of Interest

The author declares that there are no conflicts of interest regarding the publication of this paper.

References

[1] B. Li and S. Bei, "Research method of vehicle rollover mechanism under critical instability condition," *Advances in Mechanical Engineering*, vol. 11, no. 1, pp. 1–11, 2019.
[2] K. Parczewski and H. Wnęk, "The influence of vehicle body roll angle on the motion stability and maneuverability of the vehicle," *Combustion Engines*, vol. 168, no. 1, pp. 133–139, 2017.
[3] A. N. Tuan and B. H. Thang, "Research on dynamic vehicle model equipped active stabilizer bar," *Advances in Science, Technology and Engineering Systems Journal*, vol. 4, no. 5, pp. 271–275, 2019.
[4] N. T. Anh, H. T. Binh, and T. T. Tran, "Optimization of the stabilizer bar by using total scores method," *Advances in Science, Technology and Engineering Systems Journal*, vol. 5, no. 1, pp. 431–435, 2020.
[5] A. N. Tuan and B. H. Thang, "Research on determining the limited roll angle of vehicle," in *Proceedings of the International Conference of Advances in Engineering Research and Application*, pp. 613–619, Thai Nguyen, Vietnam, 1 December 2020.
[6] R. Rajamani, D. Piyabongkarn, V. Tsourapas, and J. Y. Lew, "Parameter and state estimation in vehicle roll dynamics," *IEEE Transactions on Intelligent Transportation Systems*, vol. 12, no. 4, pp. 1558–1567, 2011.
[7] N. T. Anh and H. T. Binh, "Determining the vertical force when steering," *Advances in Systems Science and Applications*, vol. 20, no. 4, pp. 27–35, 2020.
[8] N. T. Anh, "Predict the rollover phenomenon of the vehicle when steering," *International Journal of Mechanical & Mechatronics Engineering*, vol. 20, no. 4, pp. 31–40, 2020.
[9] A. H. Kazemian, M. Fooladi, and H. Darijani, "Rollover index for the diagnosis of tripped and untripped rollovers," *Latin American Journal of Solids and Structures*, vol. 14, no. 11, pp. 1979–1999, 2017.
[10] S. Yim, Y. Park, and K. Yi, "Design of active suspension and electronic stability program for rollover prevention," *International Journal of Automotive Technology*, vol. 11, no. 2, pp. 147–153, 2010.
[11] T. A. Nguyen and T. B. Hoang, "Review on the stabilizer bar equipped with the vehicle," *Journal of Mechanical Engineering Research and Developments*, vol. 44, no. 6, pp. 156–161, 2021.
[12] H. Y. Hwang, T. S. Lan, and J. S. Chen, "Developing a strategy to improve handling behaviors of a medium-size electric bus using active anti-roll bar," *Symmetry*, vol. 12, no. 8, Article ID 1334, 2020.
[13] S. Yim, K. Jeon, and K. Yi, "An investigation into vehicle rollover prevention by coordinated control of active anti-roll bar and electronic stability program," *International Journal of Control, Automation and Systems*, vol. 10, no. 2, pp. 275–287, 2012.
[14] S. S. Kelkar, P. Gautam, S. Sahai, P. Sanjay Agrawal, and R. Manoharan, "A detailed study on design, fabrication, analysis, and testing of the anti-roll bar system for formula student cars," *SN Applied Sciences*, vol. 3, Article ID 302, 2021.
[15] A. T. Nguyen, "New methods for calculating the impact force of the mechanical stabilizer bar on a vehicle," *International Journal on Engineering Applications (IREA)*, vol. 9, no. 5, pp. 251–257, 2021.
[16] N. Zulkarnain, H. Zamzuri, H. Zamzuri, and S. A. Mazlan, "Ride and handling analysis for an active anti-roll bar: case study on composite nonlinear control strategy," *International Journal of Automotive and Mechanical Engineering*, vol. 10, pp. 2122–2242, 2014.
[17] V. T. Vu, O. Sename, L. Dugard, and P. Gaspar, "H_∞/LPV controller design for an active anti-roll bar system of heavy vehicle using parameter dependent weighting functions," *Heliyon*, vol. 5, no. 6, Article ID E01827, 2019.
[18] B. Varga, B. Nemeth, and P. Gaspar, "Control design of anti-roll bar actuator based on constrained LQ method," in *Proceedings of the 14th IEEE International Symposium on Computational Intelligent and Informatics*, pp. 31–36, IEEE, Budapest, Hungary, 19 November 2013.
[19] V. V. Tan, "Preventing rollover phenomenon with an active anti-roll bar system using electro-hydraulic actuators: a full car model," *Journal of Applied Engineering Science*, vol. 19, no. 1, pp. 217–229, 2021.
[20] T. A. Nguyen, "Control the hydraulic stabilizer bar to improve the stability of the vehicle when steering," *Mathematical Modelling of Engineering Problems*, vol. 8, no. 2, pp. 199–206, 2021.
[21] N. Zulkarnain, H. Zamzuri, Y. M. Sam, S. A. Mazlan, and S. M. H. F. Zainal, "Improving vehicle ride and handling using LQG CNF fusion control strategy for an active antiroll bar system," *Abstract and Applied Analysis*, vol. 2014, Article ID 698195, 14 pages, 2014.
[22] J. Konieczny, M. Sibielak, and W. Raczka, "Active vehicle suspension with anti-roll system based on advance sliding mode controller," *Energies*, vol. 13, no. 21, Article ID 5560, 2020.
[23] V. Muniandy, P. M. Samin, H. Jamaluddin, R. Abdul Rahman, and S. A. Abu Bakar, "Double anti-roll bar hardware-in-loop experiment for active anti-roll control system," *Journal of Vibroengineering*, vol. 19, no. 4, pp. 2886–2909, 2017.
[24] V. Muniandy, P. M. Samin, and H. Jamaluddin, "Application of a self-tuning fuzzy PI-pd controller in an active anti-roll bar system for a passenger car," *Vehicle System Dynamics*, vol. 53, no. 11, pp. 1641–1666, 2015.

[25] J. Marzbanrad, G. Soleimani, M. Mahmoodi-k, and A. H. Rabiee, "Development of fuzzy anti-roll bar controller for improving vehicle stability," *Journal of Vibroengineering*, vol. 17, no. 7, pp. 3856–3864, 2015.

[26] P. Dawei, K. Zhenxing, W. Xianhui, W. Hongliang, and C. Shan, "Design and experimental validation of control algorithm for vehicle hydraulic active stabilizer bar system," *Proceedings of the Institution of Mechanical Engineers - Part D: Journal of Automobile Engineering*, vol. 233, no. 5, pp. 1280–1295, 2019.

[27] M. M. Khalil, M. R. A. Atia, and M. H. Mabrouk, "Improving vehicle rollover resistance using fuzzy PID controller of active anti-roll bar system," *SAE International Journal of Passenger Cars – Mechanical Systems*, vol. 12, no. 1, 2019.

[28] X. Zhang, C. Song, S. Song et al., "Stability control for vehicle dynamic management with multi-objective fuzzy continuous damping control," *Applied Sciences*, vol. 10, no. 21, Article ID 7923, 2020.

[29] T. A. Nguyen, "Establishing the dynamics model of the vehicle using the 4-wheels steering systems," *Mathematical Modelling of Engineering Problems*, vol. 7, no. 3, pp. 436–440, 2020.

[30] K. Sinthipsomboon, I. Hunsacharoonroj, J. Khedari, W. Pongaen, and P. Pratumsuw, "A hybrid of fuzzy and fuzzy self-tuning PID controller for servo electro-hydraulic system," in *Proceedings of the 6th IEEE Conference on Industrial Electronics and Applications*, pp. 220–225, IEEE, Beijing, China, 21 June 2011.

[31] T. A. Nguyen, "Preventing the rollover phenomenon of the vehicle by using the hydraulic stabilizer bar controlled by a two-input fuzzy controller," *IEEE Access*, vol. 9, pp. 129168–129177, 2021.