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COB-2023-0824 SUSPENSION SYSTEM STUDY ON THE PUSH ROD DOUBLE WISHBONE MODEL WITH VARIABLE STIFFNESS ACTUATOR

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Abstract. *One of the most famous tests to verify the performance of vehicles is the double lane change, where the vehicle avoids an unexpected obstacle without hitting cones in the designed lane. Generally, several vehicles fail that test because of the loss of control that can cause the vehicle to hit cones or even tip over. However, approved vehicles often have suspensions that have more control over the vehicle's behavior, and a functional traction control system. A suspension system used in the vehicles approved is the double wishbone Push Rod has prompted interest in using it to improve the performance of disapproved vehicles in test on the double lane change. This paper aims to present a suspension system study on the Push Rod double wishbone model with variable stiffness actuator, proposed to improve the performance of vehicles in the double lane change test. A model describing the lateral dynamics during the handling spin step in the normalized test and an actuator for improving vehicle performance were developed through the Davies method and a systematic VSA development. The paper presents an implementation of VSA on vehicle dynamic model to verify the performance enhancement, dealing successfully with the test stage where several vehicles typically fail.*

Keywords: *Performance, Variable stiffness actuator, Suspension system, Lateral vehicle dynamic model, Davies method .*

1. INTRODUCTION

The normalized double lane change is one required test that vehicles are put to prove. The driver must perform a maneuver that simulates the avoidance of an obstacle that suddenly appears in their path at a speed of 70 *km/h* (19.4 *m/s*) (ISO3888-2, E).

According to Cornelius and Chucholowski (2019), several commuting cars are usually rejected in this test due to poor behavior during the course. The reasons that lead to rejection are linked to the vehicle's speed and the execution of the abrupt lane change maneuver, both mandatory aspects of the test. However, another factor observed is that the failed vehicles have McPherson suspension system, which doesn't allow for precise vehicle control, unlike suspension models such as the double wishbone Push Rod.

The double wishbone Push Rod is characterized by occupying less space in the vehicle's design and being designed using a set of lighter and more compact elements. It also provides the driver with greater control over the longitudinal and lateral behavior of the vehicle (Ferreira, 2019).

However, to using this system in commuting car can increase the stiffness of the suspension components and, consequently, their natural frequency. With increase, vehicle users are affected if exposed to certain uncomfortable frequency ranges for more than 2 hours. Therefore, the need arises for the utilization of a variable stiffness actuator in the suspension system (Bernardi, 2021).

Presented by Vanderborght *et al.* (2013), the variable stiffness actuator (VSA) is able of varying its stiffness and assuming more than one configuration, including a rigid and a flexible. This device uses mechanisms in its operating structure to change its rigidity, which can involve mechanical means or means that influence the properties of the springs present in its structure.

Therefore, the application of the VSA in suspension system Push Rod, allows it to change the behavior between increased rigidity of these components when performing maneuvers and curves to have greater control, or more flexibility of these components when traveling in a straight line, thus not exposing the vehicle occupants to frequency ranges that could harm them.

This work deals with applying a VSA to Push Rod double wishbone suspension system for adapted use in commuting car and allow approval on the normalized double lane change test, analyzing the efficiency of the suspension system, using the Davies Method to calculate the static and kinematic behavior of the vehicle, along with a systematic approach to develop variable stiffness actuators for designing the VSA.

2. BIBLIOGRAPHY REVIEW

This section presents concepts about the Push Rod suspension system and the lever length VSA subcategory.

2.1 Push Rod double wishbone suspesion system

According to Ferreira (2019), the Push Rod double wishbone suspension system offers better vehicle control responses, occupies less volume in the design, and is built with lighter and more compact elements.

It allows for better weight transfer, with excellent absorption of road irregularities by the tires. Its components with this tribute the load more efficiently, contributing to better wheel traction.

The system consists of a rod that pushes a rocker, with a load exerted by the vehicle. This transmits the force to the spring/damper assembly at the same time that it exerts a reaction force coming from the rod (Ferreira, 2019).

2.2 Double lane change test

The popularly known "*moose test*" assesses a vehicle's ability to perform a maneuver, at a speed of 70 *km/h* or higher, by avoiding an obstacle that suddenly appeared in he path (ISO3888-2, E). The course for such the is test presented in Figure 1.

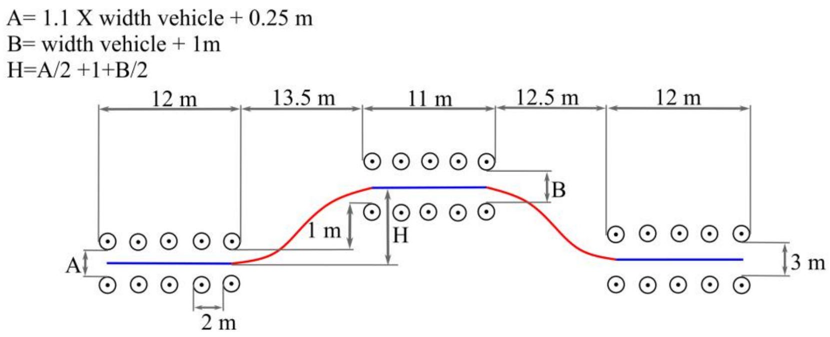

Figure 1. The course of normalized double lane change test.

In order to approve for the test, the driver must not collide with the obstacles on the track, leave the track, or tip over during its execution. In addition, a device is usually used to induce steering wheel rotation during the test, limiting the driver's reflex interruptions. Its intervention is only necessary in extreme cases when the vehicle is at risk of tipping over or hitting the obstacles (Katzourakis *et al.*, 2012).

2.3 Lever Length of VSAs

VSA are actuators able to change their stiffness and adopt several configurations, including rigid and complaint, through changes in their structure or design. A classification of the operating principles of VSA, with categories and subcategories, including the lever length, belonging to the changing transmission category. The stiffness alteration in this subcategory can occur by changing the position of three components in its structure: the springs, the pivot point of the lever arm, and the point of force application on the arm (Vanderborght *et al.*, 2013).

3. METHOD REVIEW

This section presents the main methods used to conduct the study, which are the Davies Method and the systematic stiffness actuator development.

3.1 Davies method

Based on Kirchhoff's laws for nodes and circuits, Davies developed a method that utilizes graph theory and screw to analyze the statics and kinematics of mechanisms. In this method, the circuit law is used to perform the analysis of the kinematics of mechanisms by using the motion graph with the circuit matrix and twists to obtain the network's motion matrix, which allows for the calculation of the joint velocities of the mechanism. On the other hand, the node law is used to perform the static analysis of mechanisms by using the action graph with the cut matrix and wrenches to obtain the network action matrix, which allows for the calculation of the forces and torques of the mechanism (Weihmann, 2011).

3.2 Systematic variable stiffness actuator development

Based on Murai (2013) methodology, Rosa (2018) presented a systematic approach to assist designers in producing a new VSA. The systematic allows the designer to develop a VSA without technical knowledge of this structure.

The systematics consist of developing a VSA for any application. With it, the designer can analyze several characteristics of other VSAs that already belong to category or subcategory, according to the classification of Vanderborght *et al.* (2013), in orde to develop a new device based on what has been analyzed or studied.

This new device will have functional characteristics and elements, such as the number of springs and motors, similar to those of the analyzed VSAs. However, the links and joints that compose it are different, presenting a distinct device that is potentially patentable.

4. SIMULATION SETTING

This section presents the simulation configurations, including the simulation steps and the vehicle behavior model.

4.1 Simulation steps

With the theory and methods presented in the previous sections, the following flowchart, as shown in Figure 2, outlines the steps that were followed to develop the desired vehicle simulation proposal in this paper. In the flowchart R_{ar} is the real turning radius, VW the vehicle width, F_{Ni}, F_{No}, F_{Li} and F_{Lo} are normal and lateral forces of inside and outside tires, F_{di} and F_{do} are inside and outside damper forces, α_i and α_o the slip angle, V_{ai} and V_{ao} dampers speed, V_{LPi} and V_{LPo} inside and outside lateral speed of tires, ω_{Pi} and ω_{Po} inside and outside angular speed, δ_c steering angle of tire center, CG gravity center, m_s sprung mass, K_s stiffness spring and finally, PT stands for percentage of load transfer (Bernardi, 2021).

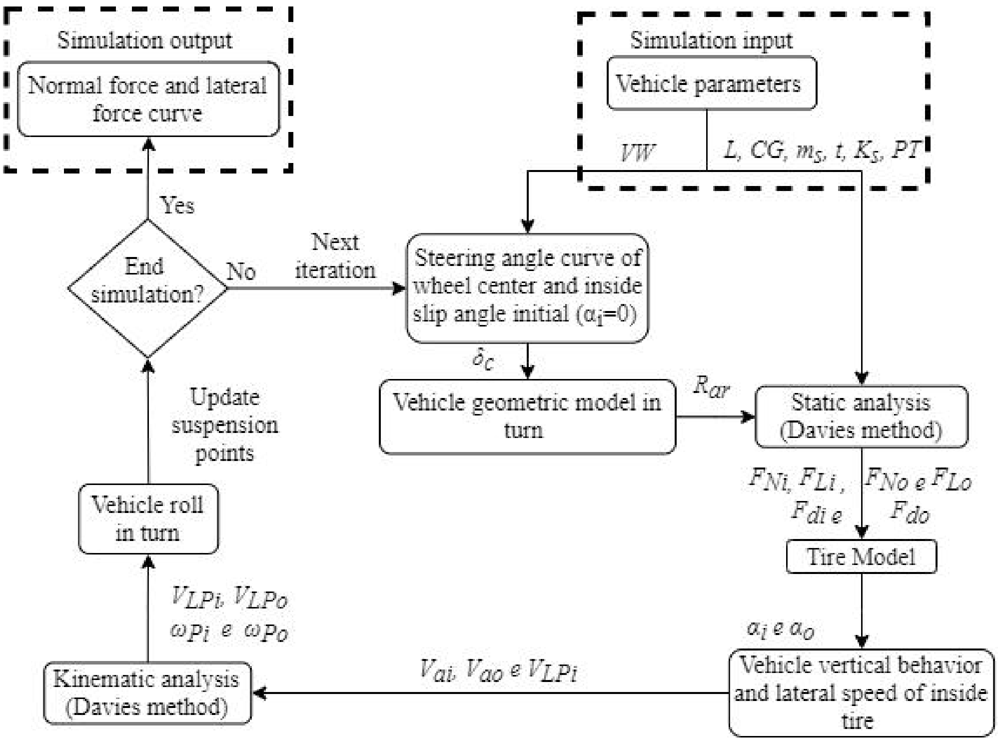

Figure 2. Flowchart representing the simulation steps.

4.1 Vehicle parameters

The chosen vehicle to perform the simulation steps described in Figure 2 was a Peugeot 307, with the specifications are presented in Table 1, according Doumiati *et al.* (2011). Originally, it is equipped with a McPherson suspension system, but in this work, was applied a Push Rod System.

Table 1. Peugeot 307 parameters.

Parameters	Value	Unit
Wheel base (L)	2.600	*m*
CG position	Y= 0.8; Z=0.3	*m*
Track	1.400	*m*
Sprung mass	1400	*kg*
Stiffness spring	40000	*N/m*
PT	60%	
Vehicle width	1.762	*m*

Furthermore, the chosen vehicle for the analyses doesn't have braking assistance systems, traction control, or any other system that assist in vehicle behavior control in its structure. Therefore, the analyses were conducted with the purpose to testing the efficiency of the suspension system.

4.1.2 Steering angle curve

The curve of the inside tire steering angle, which is used for the simulations and describes the path required for the tire to perform the double lane change test, is presented in Figure 3. It was obtained by analyzing the measurements of the curve present in the test.

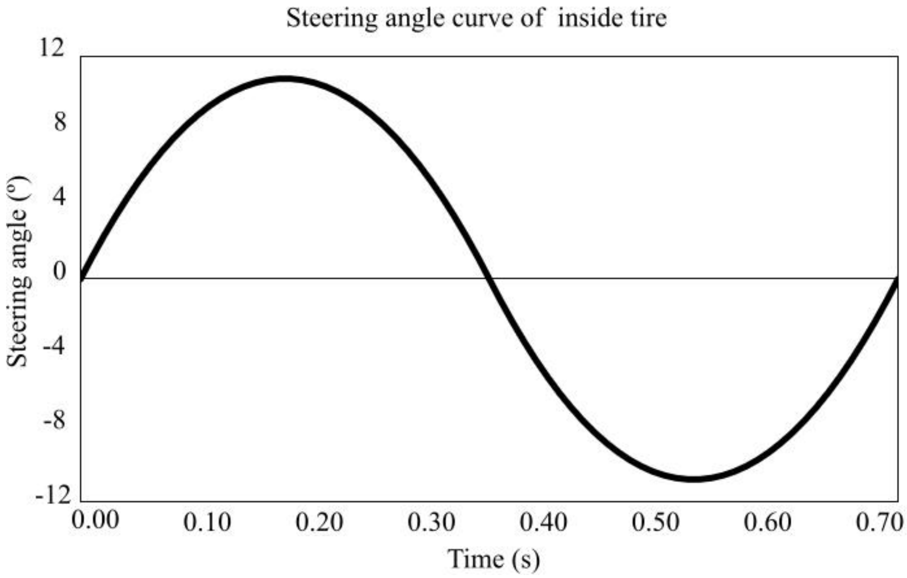

Figure 3. Steering angle curve of inside tire.

Following the curve in Figure 3, it can be observed that to above the first steering wheel rotation, the inside tire reaches a steering angle of $10.8°$ in 0.175 *s*. After this step, the driver returns the steering wheel to the initial state starting a second rotation, completing the first curve in 0.35 *s*. Once the first curve is completed, the driver initiates the third steering wheel rotation to begin the second turn, which is completed after 0.525 *s* with a steering angle of $-10.8°$. The fourth steering wheel rotation is performed after this point, returning the steering wheel to the initial state again, and concluding the lane departure in the normalized test in 0.7 *s*. The same maneuver is repeated during the lane return.

4.1.3 Vehicle geometric model in turn

In the stage, using the model present in the work of Viera *et al.* (2012), with only the front of vehicle with slip angle and in the simulation, the rear was disregarded.

4.1.4 Kinematic vehicle model

In order to describe all the components of the suspension system and the vehicle, the kinematic model of the system is presented in Figure 4. The model describes the tires by a revolute joint that allows for tire camber during a turn, a prismatic joint that describes tire slip, and a spring that represents the lateral force of the tire.

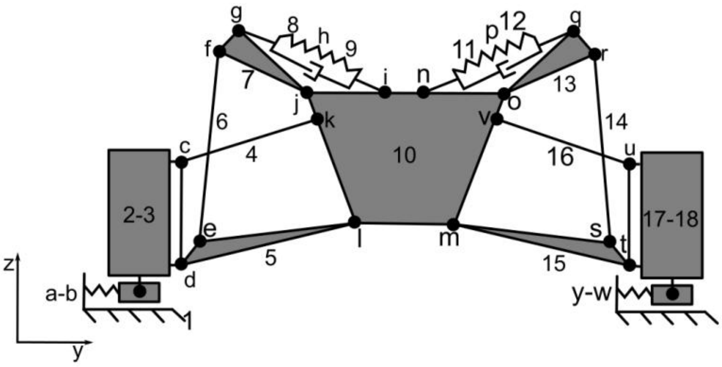

Figure 4. Kinematic model of vehicle with Push Rod suspension system.

4.1.5 Static analysis

In order to obtain tire forces, the static analysis of the suspension system is performed using the Davies Method with the law of cuts that uses the matrix of cuts and wrenches to calculate the network action matrix this allows the calculation to obtain the forces *(*F_{Ni}, F_{No}, F_{Li} and F_{Lo}). As follows, it becomes possible to perform the next steps that were presented in Figure 2, the definition of each tire slip angles using model. In addition to these forces, the damper forces are also obtained, which are necessary to calculate the vertical behavior of the vehicle.

4.1.6 Tire model

The model that describes the tire behavior is the "*Pacejka Formula*", used to calculate the lateral tire forces (Without considering tire pressure variation). It is presented in more detail in the dissertation by Bernardi (2021).

4.1.7 Vehicle vertical behavior

With the spring forces and the inner tire's slip angle of the turn, the speed of the dampers and its lateral speed of the are calculated. These are the necessary data to perform the kinematic analysis using the Davies Method.

In order to obtain the damper speed ($\dot{x}$), Equation (1), is used, where *Ca* is the damping coefficient, ΔF_s is the variation in spring forces, K_p and Δ_{xp} are the stiffness and deformation of the tires, which are obtained through the simulation of the load transfer model.

$$\dot{x} = \left| \frac{\left(K_p \Delta_{xp} - \Delta F_s\right)}{C_a} \right| \tag{1}$$

The equation (2) presented by Pacejka (2012) is used to calculate the lateral speed of the inside tire (V_{yi}). In this equation, the longitudinal speed of the tire (V_{wxi}) is multiplied by the tangent of the slip angle of the inside tire in turn (α_i). According to the mentioned author, the longitudinal speed of the tire is proportional to the longitudinal speed of the vehicle (V_{xv}), that is, $V_{wxi} = V_{xv}$.

$$V_{yi} = V_{xi} tan(\alpha_i) \tag{2}$$

4.1.8 Kinematic analysis

The study of performance and kinematic analysis of the vehicle the Davies Method was used. The steps of the kinematics in the Davies Method are obtained, providing the velocities of all suspension points as a result. However, only the lateral speed of the tires is required to update the points and initiate a new iteration until the vehicle completes the normalized double lane change test. The upgrade of vehicle roll can be performed using trigonometric relationships between the suspension points.

5. VARIABLE STIFFNESS ACTUATOR DESIGN

In this section, the developed VSA is presented, including its features, final appearance and its application in the suspension system.

5.1 VSA developed

The chosen subcategory of VSA for the drive is the variation of the lever length. The reasons for choosing this subcategory are:

- Having mobility 3, with the to move springs for variation of stiffness VSA, change de position of end effector and action end effector with locked motor;
- Is that the VSA has low energy cost to perform stiffness variation, making it more efficient;
- It has high efficiency in transferring forces from the springs to the system;
- It has balanced control between the movements of the effector and the stiffness variation of the springs.

After selecting the subcategory, the next steps of Rosa (2018) systematic approach are followed and a prototype of the VSA structure is developed in CAD environment, which is presented in the Figure 5 where, 1 is the fixed part connected to the vehicle chassis as shown in Figure 6, 2 is the linear motor responsible for changing the position of the springs, 3 are the springs, 4 is the element that supports motor 2, 5 is the motor responsible for changing the position of the end effector (EF), 6 is the element that compresses the springs, 7 is the end effector and 8 is the component where are the spring.

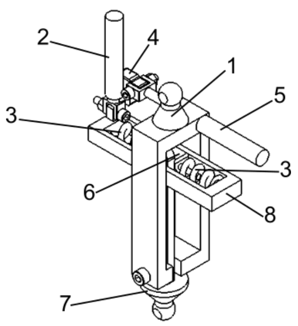

Figure 5. Final appearance of the VSA to be applied in the suspension system.

Now, with the final concept of the VSA defined, it is applied to the Push Rod suspension system to conduct simulations and assess the device's effectiveness in improving the vehicle's performance gain in the normalized double lane test.

5.2 VSA on suspension system

The chosen position for the developed VSA is parallel to the spring/damper assembly in both suspension systems, as shown in Figure 6. Thus, the equivalent stiffness of the system is formed by the sum of the VSA stiffness and the spring stiffness that are in parallel, while the tire stiffness is considered in series.

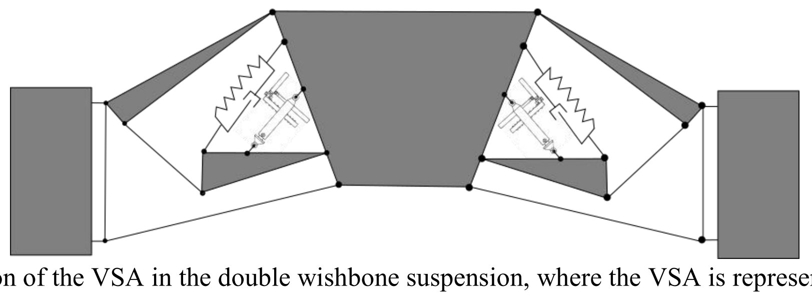

Figure 6. Position of the VSA in the double wishbone suspension, where the VSA is represented by a spring.

With the VSA positioned parallel to the spring/damper assembly, the equivalent stiffness of the system can be calculated using Equation (3), where K_{eq} represents the equivalent stiffness of the system, where K_s is the spring stiffness, K_p is the tire stiffness, and K_{VSA} is the VSA stiffness.

$$K_{eq} = \frac{K_p(K_s + K_{VSA})}{K_p + K_s + K_{VSA}} \tag{3}$$

In addition to its influence on the equivalent stiffness, the VSA also affects the natural frequency of the suspension. By altering the natural frequency range of the system, it will also impact the damping coefficient. With an increase in the damping coefficient, the damper speed decreases, decreasing the vehicle's roll speed as well. As the vehicle's roll decreases, its inclination reduces, and it takes more time for the tire to lose contact with the road surface.

Following this logic, it is necessary to apply the calculated K_{eq} from Equation (3) to determine the natural frequency using Equation (4).

$$W_n = \frac{1}{2\pi}\sqrt{\frac{K_p(K_s + K_{arv})}{K_p + K_s + K_{arv}\ m_s\ PT}} \tag{4}$$

In addition to changing the system's frequency, the VSA will modify the vertical behavior of the suspension. This modification involves an active internal force applied in Equation (1). That can be rewritten as Equation (5). Where C_c is damper coefficient.

$$\dot{x} = \left|\frac{\left(K_p\Delta_{xp} - \Delta F_s - F_{ARV}\right)}{C_c}\right| \tag{5}$$

Since the VSA exerts an active force on the suspension system, it is necessary to add two wrenches in the calculated static analysis in section 4.1.5.

In this way, the VSA will apply an internal torque to the system that originates from the rocker's arm and goes towards the fixed links. As a result, the system will have better control of the vehicle during cornering, as there will be a reduction in loads transfer from one tire to the other. This is done, because the side with higher stiffness will pull the load towards itself, contributing to an improved behavior.

6. RESULTS

In order to verify the vehicle's behavior in turn, it is necessary to analyze the curves of the normal and lateral forces of the tires and their slip angles. Figure 7 shows, the graph of the normal force on the inside tire of the turn, where the dashed line represents the system with VSA and the solid line represents the system without VSA.

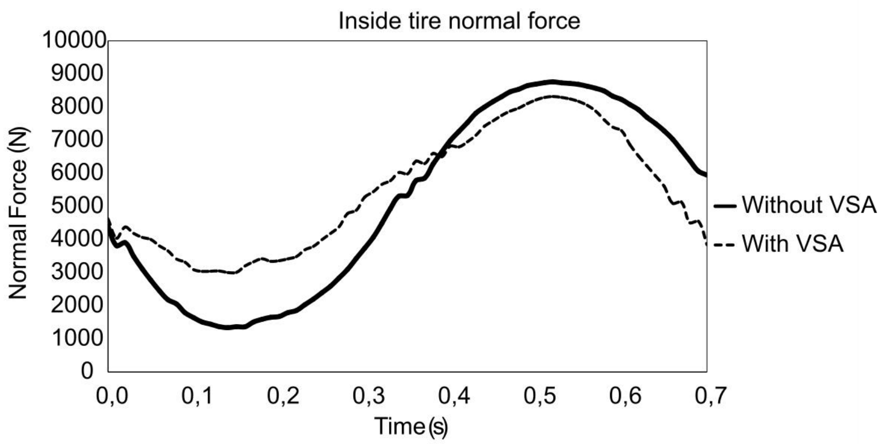

Figure 7. Normal force on the inside tire in the turn to the Push Rod system with and without VSA.

The graph in Figure 7 shows that with the VSA, higher values are obtained compared to the system without VSA, because the VSA applied more force in the system, allowing more control of the load transfer between two tires.

Now we analyze the graph of the lateral force on the inside tire of the turn in Figure 8, where it can be seen that there is an increase in the lateral force due to the increase in the normal force caused by the presence of VSA torque in the system.

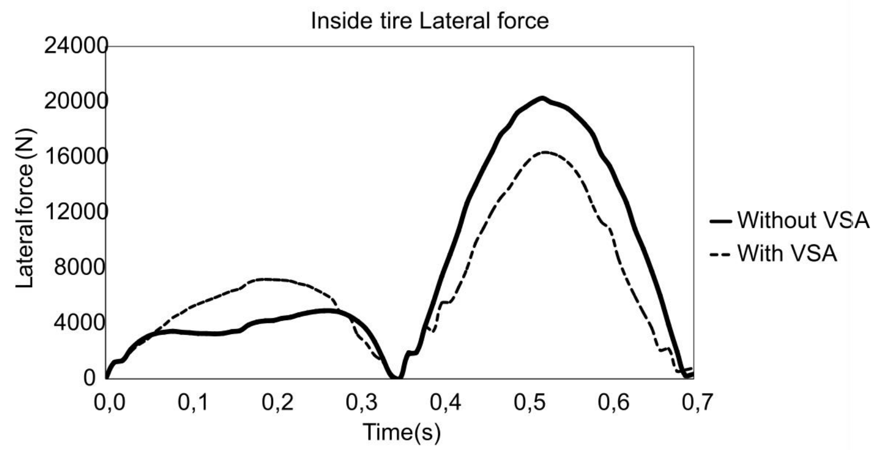

Figure 8. Lateral force on inside tire in the turn to Push Rod system with and without VSA.

With the two forces presented, we obtain the slip angle in the inside tire of the turn, as shown in Figure 9. It can be observed that the values obtained are higher because the tire remains in contact with the ground for a longer time and therefore experiences more slip.

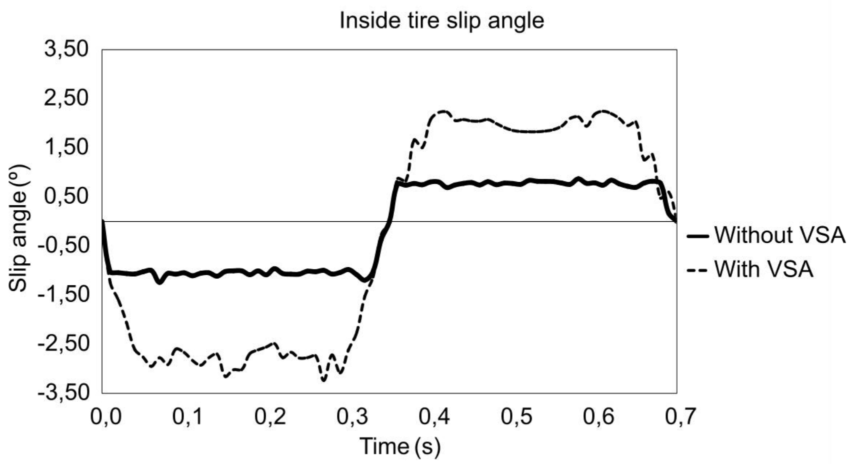

Figure 9. Slip angle on inside tire in the turn to Push Rod system with and without VSA.

After analyzing the inside tire behavior results, the outside tire performance responses are presented. The next graph shown in Figure 10 corresponds to the normal force on the outside tire. It exhibits an opposite behavior compared to the inside tire due to the weight transfer of the vehicle acting from the inside to the outside tire.

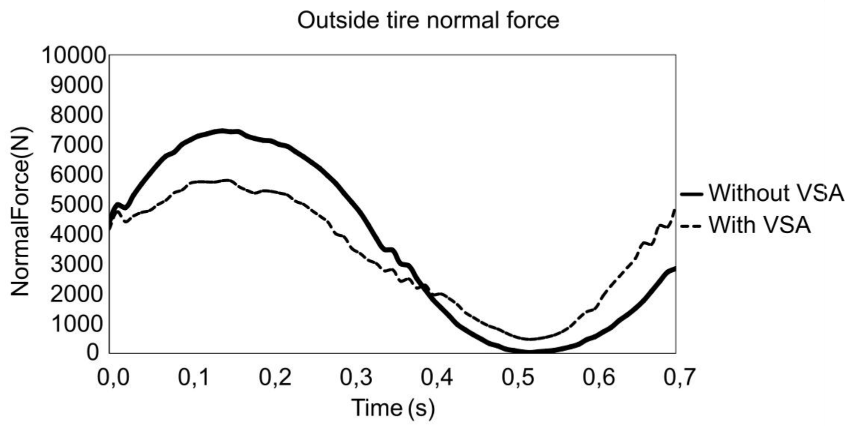

Figure 10. Normal force on the outside tire in the turn to Push Rod system with and without VSA.

As the VSA allowed for higher forces values on the inside tire, the values obtained for this tire are lower for the first curve and higher for the second curve, due to reduced weight transfer on the axle and changes in stiffness from one side to the other.

The next graph shown in Figure 11 corresponds to the lateral force of the outside tire at the turn. Based on the previously explained arguments, such a figure exhibits behavior opposite to the inside tire, as expected.

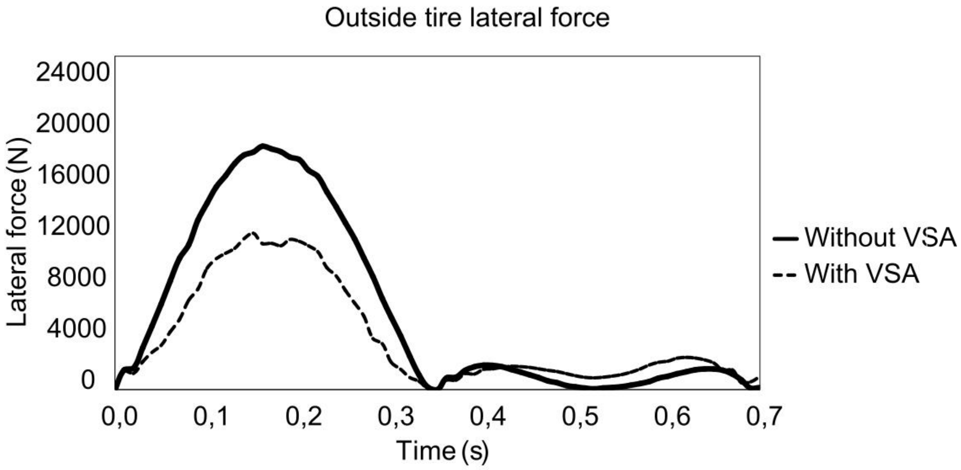

Figure 11. Lateral force on the outside in the turn to Push Rod system with and without VSA.

Eventually, Figure 12 presents the behavior of the slip curve obtained for the outside tire. It can be observed that the values obtained are lower than those of the inside tire, since the outside experiences less slip compared to the inside tire. This occurs because the higher load presence stabilizes the outside tire more effectively.

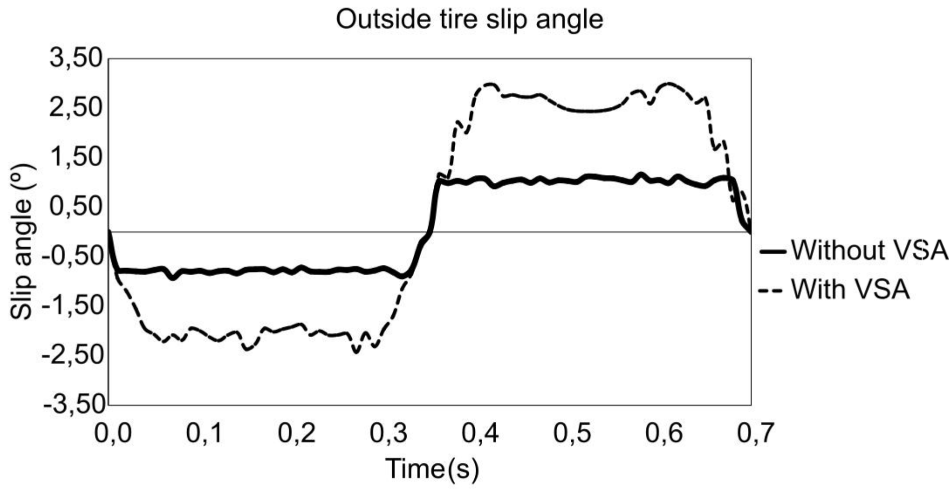

Figure 12. Slip angle on the outside tire in the turn to Push Rod system.

The non-linearity of the slip angle values found in Figure 9 and Figure 12, is justified using an optimization algorithm in these values. Since the algorithm analyzes ‘N’ possible angles in the “Magic Formula” randomly, to compare the tire’s lateral force with the forces obtained from the Davies Method calculation, the optimized values are random and do not follow a linear pattern.

With the presented graphs, it is possible to analyze the moment when the vehicle loses contact with the asphalt during the execution of the “moose test”. Without VSA the tire loses contact with the asphalt at 0.52 *s*, as shown Figure 10, and consequently is unable to perform the maneuver. However, according to the same graph, it can be observed that the vehicle equipped with VSA maintains contact with the asphalt at the 0.52 *s* mark and is able to complete the maneuver. In addition to this graph, it is possible to analyze in the slip angle graphs were higher with VSA, as presented in Figure 9 and Figure 12, indicating that the vehicle remained in contact with the asphalt for a longer period during the maneuver.

7. CONCLUSION

The simulation conducted in this study presents a kinematic and static analysis of all the points comprising the double wishbone Push Rod suspension system using the Davies Method, performing the normalized double lane change test. This approach offers a new perspective for analyzing the vehicle's behavior, allowing for a more detailed study of all the suspension joint positions.

The developed VSA contributed to the vehicle's ability to perform the normalized double lane change test successfully. As demonstrated, the vehicle equipped with VSA executed the test without losing tire contact with the asphalt or colliding with the track's obstacles, which are the requirements to pass successful.

8. ACKNOWLEDGEMENTS

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