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THE INFLUENCE OF TIRE CHARACTERISTICS ON SHIMMY STABILITY

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Abstract. *The shimmy phenomenon can be described as a self-excited oscillation of the front part of a motorcycle around its steering axis. It has been studied since the beginning of the twentieth century due to its effect in system stability and passengers' safety. Beside motorcycles, this phenomenon can occur in some carriages and aircraft landing gear. The system oscillation can be stable or unstable depending on the system characteristics and its work conditions. It is known that shimmy stability is affected by the fork and handlebars stiffness, the trail and the speed of the system, the wheel bearings clearance and the tire characteristics. Podgorski and other authors propose mathematical models known as front wheel system models that make possible the analysis of the shimmy behavior and the factors that affect it. In this paper, it is analyzed the tire stiffness and damping influence on shimmy stability using the mathematical model presented by Podgorski. First, the model was implemented in MATLAB. Then, the values of stiffness and damping coefficients were varied in a range of 30% to 200% of their respective original values. Finally, two graphics were plotted to show the shimmy stability curve behavior. According to these graphics, increasing the value of tire stiffness coefficient causes an increase at the stable region of the graphic regardless of the trail and the speed of the system. On the other hand, when the value of tire damping coefficient increases, the stable region of the graphic also increases for low speeds, but it decreases for high speeds. It shows the importance of tire characteristics on shimmy behavior and possible ways of avoid that the system works in unstable conditions.*

Keywords: *shimmy, tire, simulation*

1. INTRODUCTION

At the last centuries, many laboratories and research centers have focus on the study of the means of transport. Many research projects have been developed aiming at improving the performance and the safety of cars, aircrafts, vessels and others means of locomotion. Thus, it is important to study and understand the phenomena that affect these two factors, for example, the shimmy phenomenon.

Pacejka (2006) describes shimmy as a self-excited oscillation motion of the front wheel around its steering axis. According to Ran *et al*., 2014, it is an intense vibration that can occur in the front wheels of automobiles, motorcycles and aircrafts landing gears. Zhuravlev and Klimov (2009) claim that such self-oscillations harm the system performance and the safety of the users.

Ding (2009) states that the shimmy effect in front wheels can reduce the durability of mechanical components and cause fatalities. Therefore, its prevention is still a concern for designers and operators of vehicles and aircrafts.

According to Podgorski *et al.*,1975, the system oscillation decreases in stable conditions and increases in unstable conditions. Using mathematical models, it is possible to analyze the dependence of the stability of shimmy on system parameters.

In this paper, it is used the mathematical model known as front wheel model proposed by Podgorski *et al*., 1975 in order to evaluate the influence of some tire characteristics with respect to the shimmy phenomenon. It is studied the behavior of the shimmy stability by varying the values of the lateral stiffness and the lateral damping coefficient of the tire.

2. THE SHIMMY PHENOMENON

Zhuravlev and Klimov (2009) define shimmy as a strong angular self-oscillation phenomenon that occurs in wheels. Ding (2009) states that shimmy occurs when enough kinetic energy of the vehicle movement is transferred to the oscillating movement of its front wheel.

The studies of shimmy started in the early twentieth century and Broulhiet (1925) is responsible for one of the first known researches. Becker *et al.*, 1931, were the first to develop a theory for the shimmy motion of automobiles. As a result of their work, the gyroscopic coupling between the angular motions about the longitudinal axis and about the steering axis was appointed as the main factor causing shimmy.

Den Hartog (1940) and Rocard (1949) analyzed this gyroscopic shimmy in systems with live axles and Olley (1947) examined this phenomenon experimentally. By means of his experiments, Schrode (1955) concluded that a wheel system accelerating is slightly less stable than a system decelerating.

Pacejka (1966) observed another kind of shimmy to occur in aircraft landing gears and automobiles equipped with independent front wheel suspensions. This shimmy is closely related to tire and suspension lateral compliance.

The deformation of elastic tires was studied as an important factor in the analysis of shimmy by von Schlippe and Dietrich (1947), Moreland (1954) and Smiley (1957). Ho and Lai (1970) analyzed the shimmy effect on aircraft landing gears and found a relationship between the phenomenon occurrence and the lateral deflection of the landing gear support.

Besselink (2000) evaluated the performance of several linear tire models and the influence of parameters on the shimmy stability, showing that linear models and analytical techniques are efficient to analyze it.

Throughout the years, some variations of the mathematical model known as the front wheel model has been used to study the phenomenon. Zhuravlev and Klimov (2009) proposed a model based on dry friction theory to explain the shimmy phenomenon in rigid wheels. Ding (2009) proposed a mechanical model consisting of two front wheels models and some parts coupled up directly to the vehicle to understand the influence of the vehicle motion on the front wheel shimmy.

At his paper, Podgorski, *et al.*, 1975, presented a mathematical model that it is able to analyze the shimmy stability and which can be seen in Fig. 1. Although it is older than the models mentioned above, Podgorski’s model is presented in more details and it is the only one that enables the reproduction of the results obtained by the author and the implementation of new analyzes. For this reason, Podgorski’s model was chosen for the study performed at this paper.

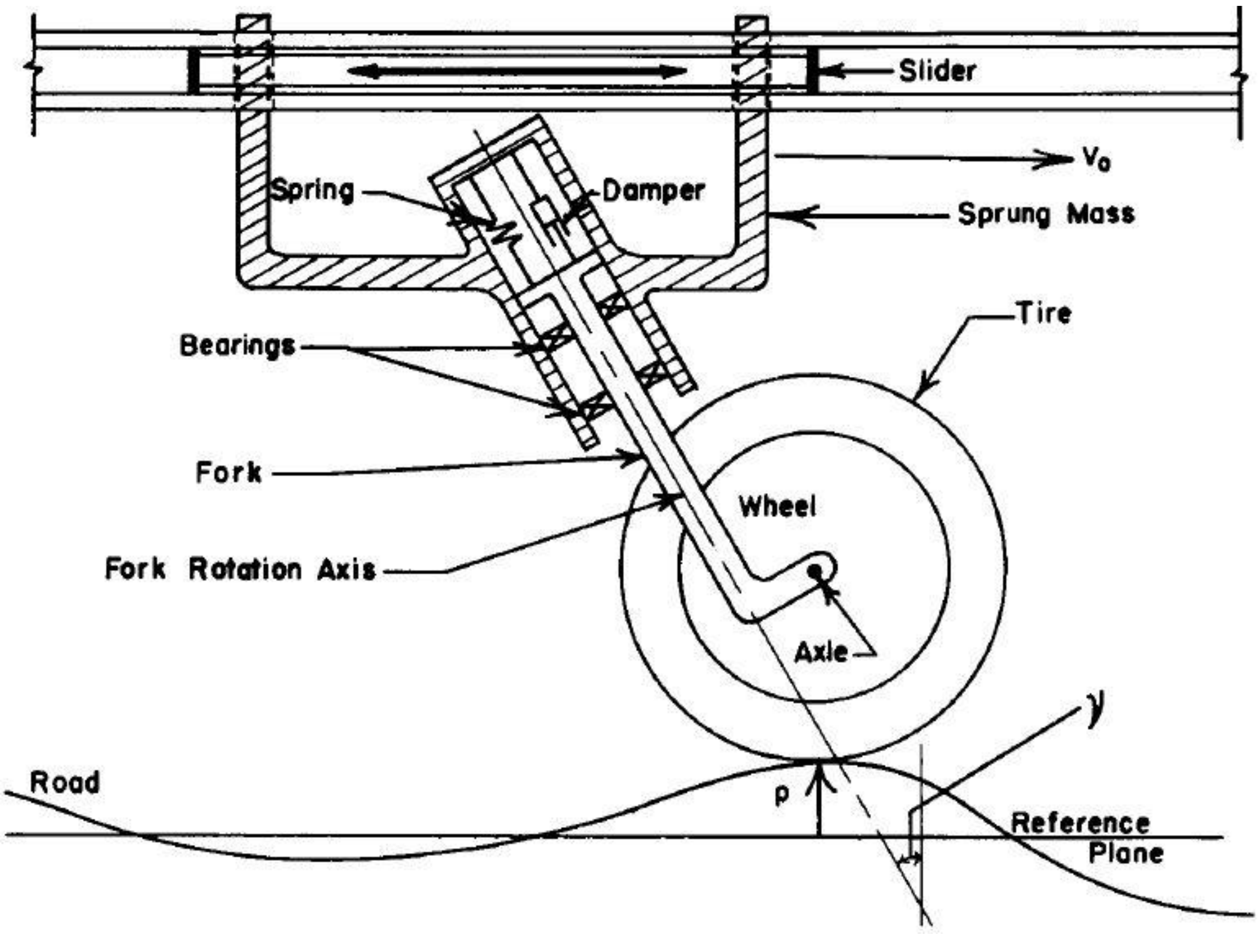

Figure 1. Front wheel model developed by Podgorski *et al.*, 1975.

The proposed model consists of a pneumatic tire mounted on the wheel which rotates about the axle fixed to the fork. On the other hand, the fork rests in bearings that are attached to the sprung mass, which translates horizontally at a speed *V*. The fork is free to rotate in the bearings and to translate along its rotation axis. The fork's translational motion is resisted by a spring and a damper. Besides this, the fork rotation axis makes an angle ε (known as caster angle) with the vertical.

There are three independent movements in the system developed by Podgorski *et al*., 1975: the wheel rotation angle θ, the fork rotation angle (shimmy angle) ψ and the suspension deflection η. Besides this, the tire contact point is allowed to deflect with respect to the wheel in. The deflection is defined by Δ and it is measured perpendicularly to the intersection of the ground plane and the wheel center plane.

The reactions from the road on the tire considered in the model are the normal load *N*, the side force F_t and aligning torque M_t. The tire's overturning moment, rolling resistance moment, and the braking-traction force are not considered.

Podgorski *et al*., 1975, adopted the contact point theory proposed by Moreland (1954) and Collins (1969) in order to characterize the side force and the aligning torque. Equation (1) corresponds to the side force equation:

$$F_t = K_I.\Delta + C_L.\dot{\Delta} \tag{1}$$

where K_I is the lateral tire stiffness and C_L is the lateral tire damping.

Equation (2) corresponds to the aligning torque based on Moreland's theory:

$$M_t = \mu_I.\lambda \tag{2}$$

where μ_I is the torsional stiffness constant of tire and λ is the sideslip angle.

In his analysis, Podgorski *et al*., 1975, admits that the normal load is constant and the same happens in this work.

The state variable formulation of the problem reduces to three uncoupled sets of differential equations when it is considered that the wheel is perfect (the wheel is true, balanced and perfectly round) and the road is flat.

The first set consists of a second order equation for the suspension deflection, as shown in Eq. (3):

$$\left(m_f + m_w\right)\frac{d\dot{\eta}}{dt} = -\left(m_f + m_w\right)gsen(\varepsilon) - C_s\dot{\eta} - K_s\eta + Ncos(\varepsilon) \tag{3}$$

where m_f is the fork mass, m_w is the wheel mass, g is the acceleration of gravity, K_s is the suspension spring rate and C_s is the suspension damping constant.

The second set contains an equation for the wheel rotation θ, which can be solved to yield:

$$\theta = \frac{-Vt}{r_e} \tag{4}$$

where r_e is the tire rolling radius and t is the time variable.

Finally, the last set consists of four first order differential equations which are linear in the state variables Δ, λ, ψ *e* $\dot{\psi}$. These equations can be written as shown in Eq. (5):

$$\frac{d\hat{Y}}{dt} = [B].\hat{Y} \tag{5}$$

where $\hat{Y}$ is a 4×1 state vector with components Δ, λ, ψ and $\dot{\psi}$, and [*B*] is a 4×4 matrix whose components depend on the tire physical characteristics, the caster angle (ε), the normal load (*N*), the system longitudinal speed (*V*) and the system trail (*a*), as shown in Eq. (6):

$$[B] = \begin{bmatrix} 0 & -V & -Vcos(\varepsilon) & -acos(\varepsilon) \\ \frac{CK_I}{C_I} & \frac{-(1+CC_LV)}{C_I} & \frac{-\left(CC_LVcos(\varepsilon)-C_2sen(\varepsilon)\right)}{C_I} & \frac{-\left(CC_Lacos(\varepsilon)\right)}{C_I} \\ 0 & 0 & 0 & 1 \\ \frac{\left(Nsen(\varepsilon)+K_Iacos(\varepsilon)\right)}{I_\psi} & \frac{\left(\mu_Icos(\varepsilon)-VC_Lacos(\varepsilon)\right)}{I_\psi} & \frac{\left(Nsen(\varepsilon)cos(\varepsilon)a-VC_L\,acos^2(\varepsilon)\right)}{I_\psi} & \frac{-\left(C_D+C_La^2\,cos^2(\varepsilon)\right)}{I_\psi} \end{bmatrix} \tag{6}$$

where C is the tire yaw coefficient, C_1 is the tire time constant, C_2 is the second order yaw stiffness coefficient and I_ψ is a composition of the wheel and the fork moments of inertia. Collins (1971) provides a detailed explanation of each tire parameters and clarifies the meaning of each one.

According to Podgorski *et al*., 1975, the trail is a physical parameter of great importance owing to its influence on the shimmy stability.

The trail is the distance measured at the ground plane between the front wheel contact point and the intersection point of the rotation steering axis and the ground (DONADIO, 2009).

Podgorski *et al*., 1975, built the shimmy stability curve as a function of longitudinal velocity and Trail, as shown in Fig. 2. The stability plot was obtained from the eigenvalues of matrix [B]. At each point in the plane, the eigenvalues of this matrix consist of two which are purely real and two which form a complex conjugate pair.

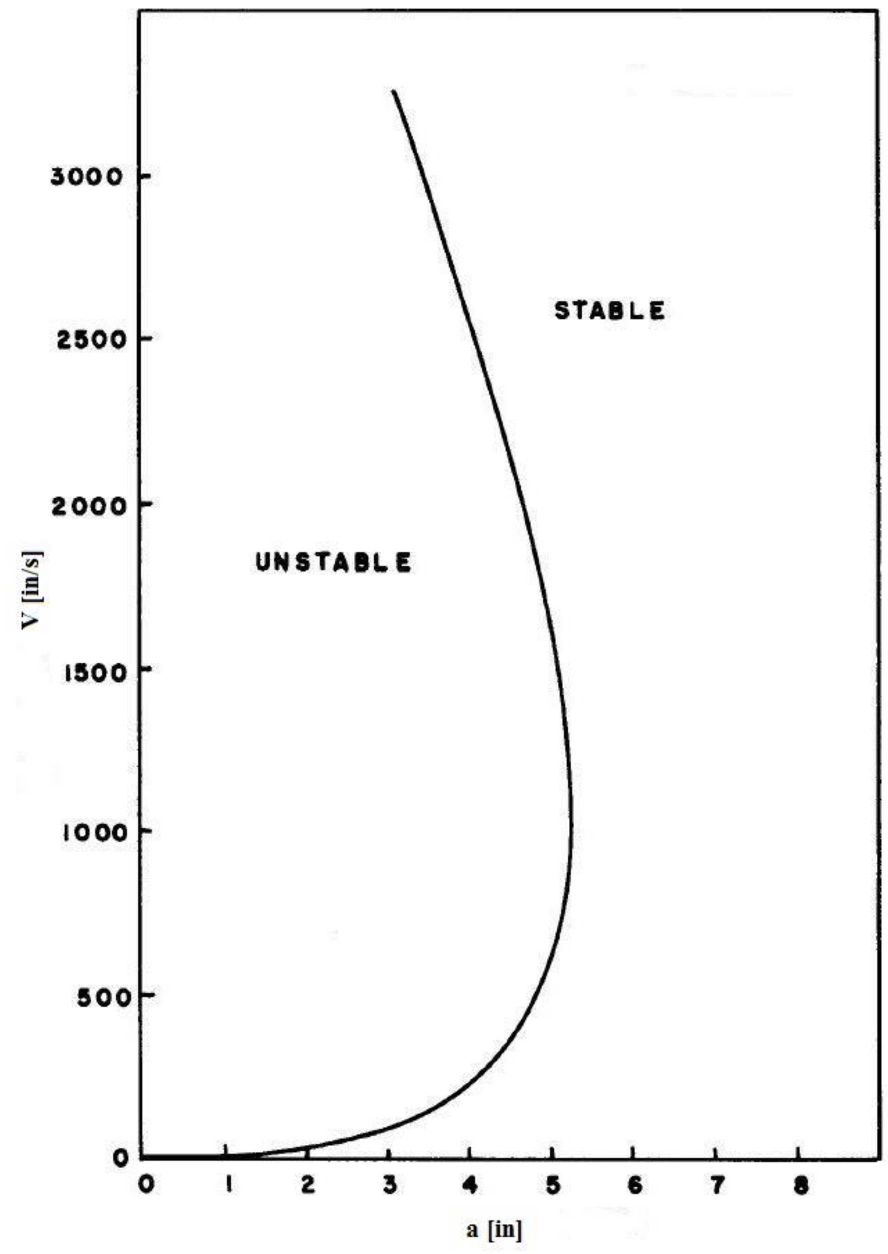

Figure 2. The stability plot obtained by Podgorski *et al*., 1975.

Since the two real eigenvalues are negative at all points in the plane, the shimmy oscillations are determined by the complex conjugate pair. The shimmy frequency (ω_s) is given by the magnitude of the imaginary parts and the exponential growth or decay rate of the shimmy oscillation (E_r) is given by the value of the real part. Both depend on V and a, and the value of ω_s increases monotonically as either V or a increase (PODGORSKI *et al*., 1975).

According to Podgorski *et al*., 1975, the shimmy motion is stable if E_r is negative and unstable if it is positive. When the point defined by the instantaneous velocity and the trail is located at the stable region of the figure above, the shimmy motion decays. On the other hand, the shimmy motion increases when this point is located at the unstable region.

2.1 Tire parameters effects on shimmy stability

The shimmy phenomenon can be avoided by designing a system where the resonance conditions occur at speeds that are outside the operating range. A possible alternative to minimize the effects of this phenomenon is to increase the inherent system stability, designing it to not operate in regions of instability.

It is possible to analyze the influence of the lateral tire stiffness and the lateral tire damping on shimmy stability using the eigenvalues of matrix $[B]$ from Podgorski's model. Therefore, in this paper his model was implemented in MATLAB. Podgorski described all the parameters he used to feed his model; thus his results can be reproduced and it is possible to rebuild the diagram shown in Fig. 2 and to check if the model built in MATLAB is working well. This reproduction can be seen in Fig. 3.

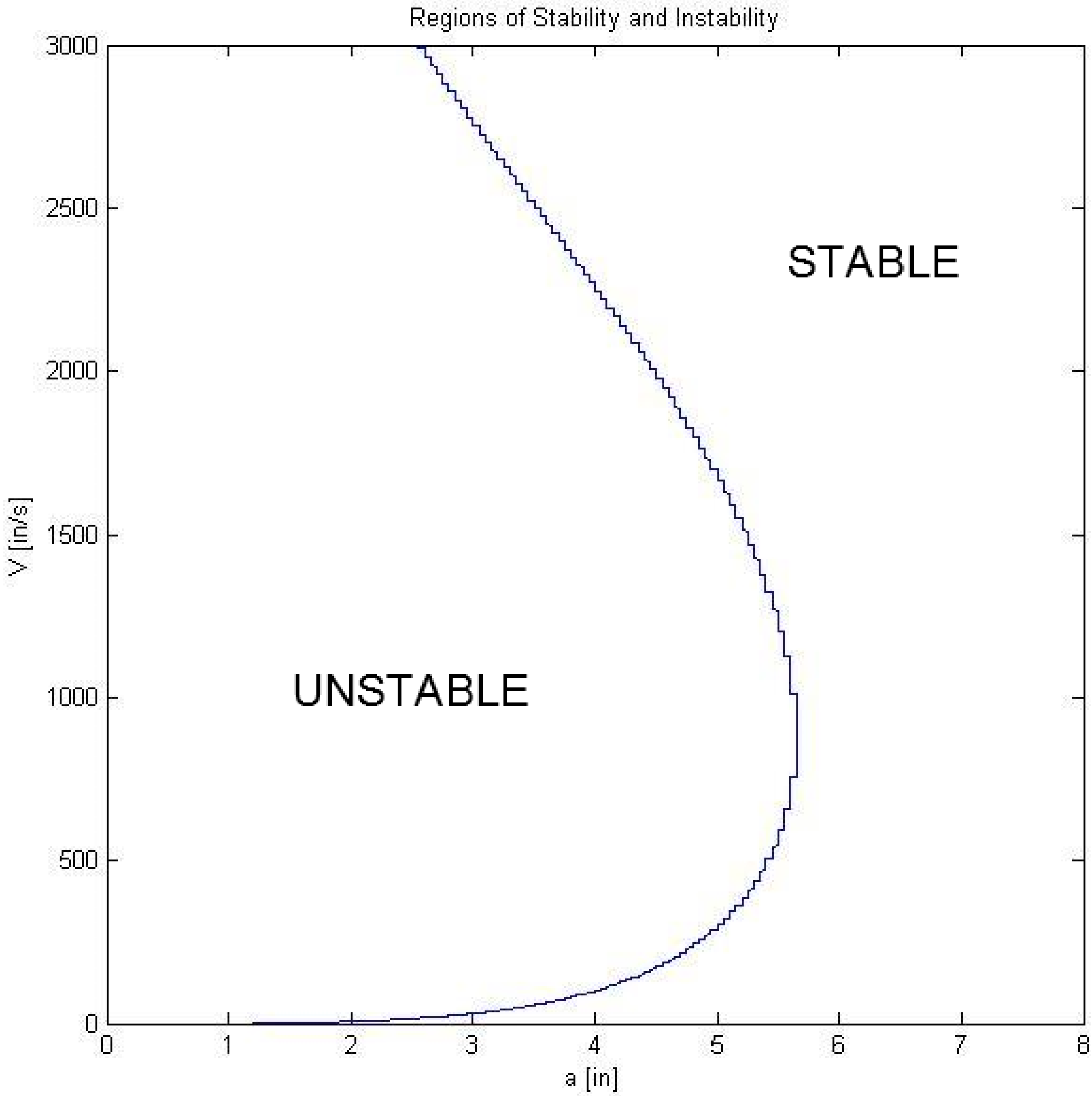

Figure 3. The stability plot obtained from MATLAB which is similar to the one obtained by Podgorski *et al.*, 1975.

It is possible to analyze the influence of the lateral tire stiffness and the lateral tire damping on shimmy stability varying the original values of K_1 and C_L. Thus, it was chosen to vary both coefficients in a range of 30% to 200% of their respective original values.

The plot shown in Fig. 4 was obtained by varying the value of the lateral tire stiffness:

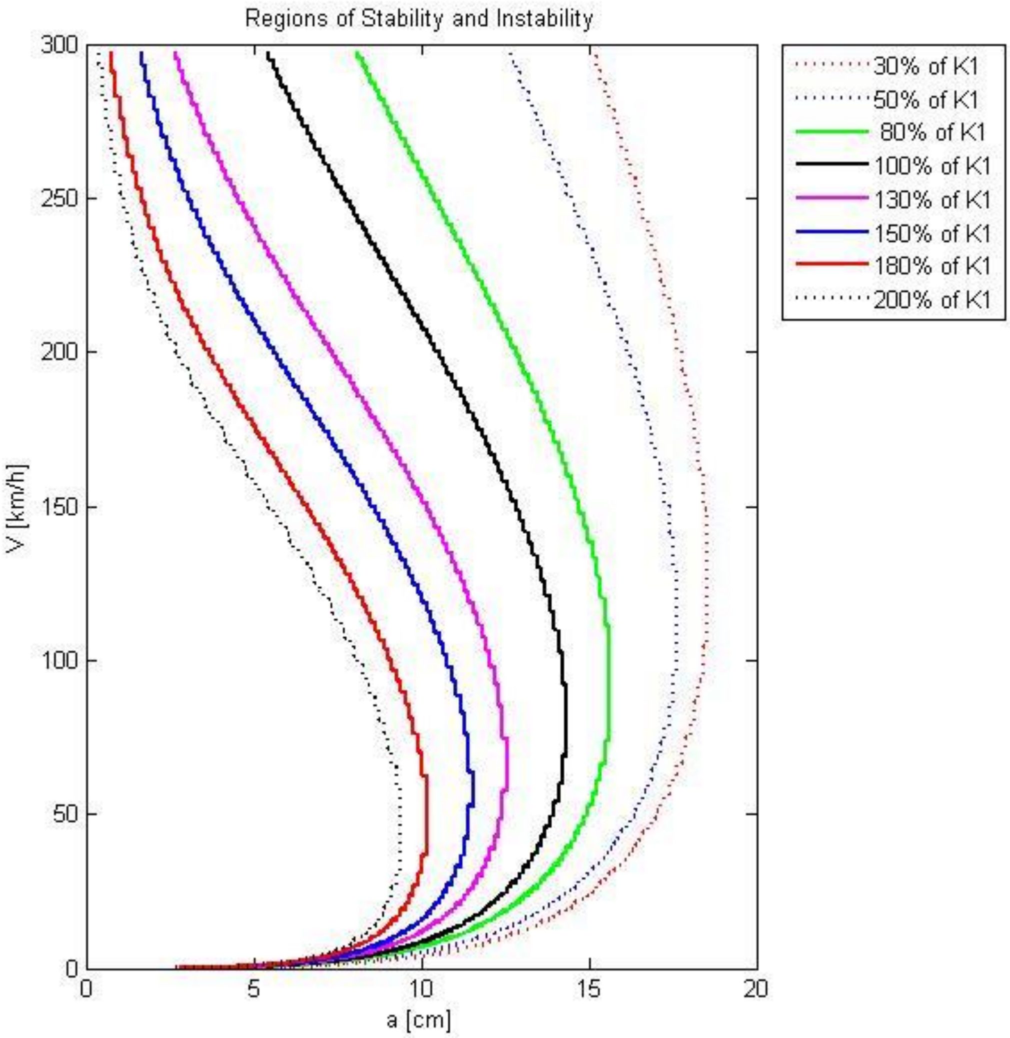

Figure 4. The shimmy stability curve behavior by varying the lateral tire stiffness.

Since the unstable region is located on the left side of the curve and the stable region on the right side, it can be noted that increasing the value of tire stiffness coefficient causes an increase at the stable region of the graphic regardless of the trail and the speed of the system. However, it is also found that the unstable region is even lower at high speeds.

On the other hand, the variation of the lateral tire damping coefficient resulted in the plot presented in Fig. 5:

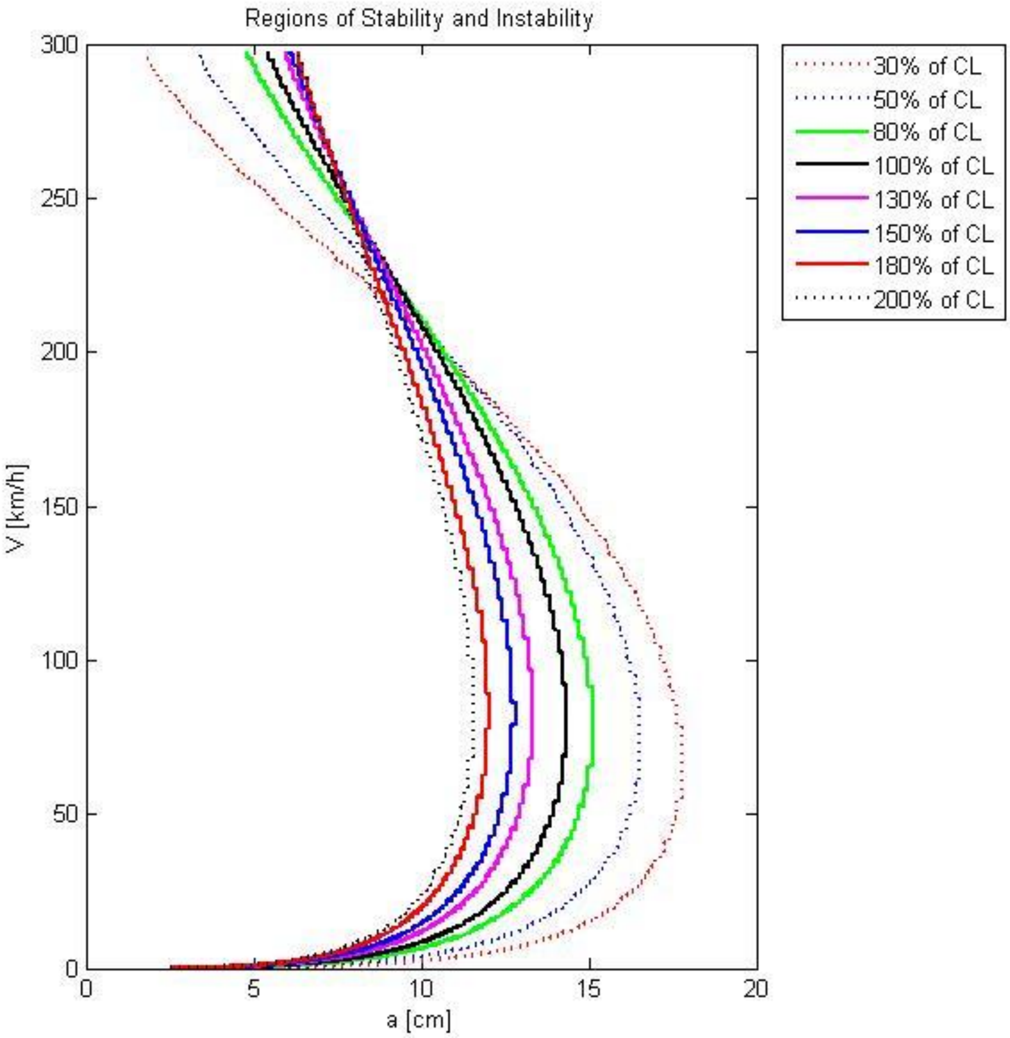

Figure 5. The shimmy stability curve behavior by varying the lateral tire damping.

The variation of the lateral tire damping resulted in a shimmy stability curve behavior different from the one observed when the lateral tire stiffness was varied. The stable region of the graphic increases for low speeds when the value of the lateral tire damping coefficient also increases. However, increasing the value of C_L at high speeds causes a decrease at the stable region.

Therefore, lower values of C_L may be interesting for systems operating at high speeds. On the other hand, high values of C_L ensure greater stability for systems working in low speed.

3. CONCLUSIONS

Shimmy is a phenomenon that can compromise the comfort and the safety of the users of many means of transport. Therefore, it is important to understand why it occurs and to discover some ways to avoid it.

The simulation resulted in a larger stable region as the value of the lateral tire stiffness increased. As stated in the literature, the energy required to cause the wheel oscillating movement comes from the vehicle's kinetic energy. Thus, it can be assumed that the most rigid systems need more energy to execute a high amplitude motion than the less rigid bodies.

At low speeds, the simulation showed that increasing the lateral tire damping coefficient expands the stable region. It is assumed that the amount of energy available for generating the front wheel oscillating motion is reduced since more damped systems absorb higher energy levels. On the other hand, by increasing the speed there is a certain point where the less damped systems become stable for a range of trail larger than the one of the most damped system. There is still no hypothesis to justify this behavior and the next steps of this research aim at trying to understand it.

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