CONEM 2016
CONGRESSO NACIONAL DE ENGENHARIA MECÂNICA
21 - 25 AGOSTO DE 2016 FORTALEZA - CEARÁ

MODELING AND ANALYSIS OF SOLID AXLE SUSPENSION AND ITS IMPACT ON THE HEAVY VEHICLES STABILITY

Gonzalo Moreno[1,2], e-mail: gmoren@hotmail.com
Lauro Nicolazzi[1], e-mail: lauro@grante.ufsc.br
Rodrigo S Vieira[1], e-mail: rvieira@grante.ufsc.br
Daniel Martins[1], e-mail: daniel.martins@ufsc.br

[1]Federal University of Santa Catarina. 88040-900 Florianopolis, SC - Brazil.
[2]University of Pamplona. 543050 Pamplona - Colombia.

Abstrac: *The leaf spring is the oldest type of automotive suspension, and continues to be a popular choice for solid axles in heavy vehicles. This type of suspension, though simple in appearance, causes many problems in modelling. This paper presents the modelling of a leaf spring using mechanism theory and subsequent kinematic analysis using the Davies method. The proposed model is planar, with two degrees-of-freedom and allows relative motion between the sprung and unsprung masses of the vehicle. This model also allows a better representation of this type of suspension in researches on vibration control and vehicle lateral stability study.*

Key words: *Leaf spring, Suspension system, Kinematics, Davies method*

1. INTRODUCTION

The stability of heavy vehicles has been the focus of research efforts in the last decades. The vehicle lateral stability can be evaluated through the static stability factor (*SSF*). This factor represents the maximum lateral acceleration - a_y (expressed in terms of gravity acceleration - g) in a quasi-static situation before one or more of the tires lose contact with the ground, (Winkler, 1987; Gillespie, 1992; Hac, 2002).

According to the classic vehicle stability analysis, the *SSF* factor is a function of the vehicle track width (t) and the height of the CG (h) (Eq. 1) (Gillespie, 1992).

$$SSF = \frac{a_y}{g} = \frac{t}{2h} \qquad (1)$$

However, when a heavy vehicle is making a turn or an evasive maneuver, many characteristics of the vehicle, such as the suspension, tires, and chassis allow the inclination of the vehicle body, changing the position of the center of gravity (CG) and modifying the *SSF* factor.

Important researches (Gillespie, 1992; Winkler, 2000; Chang, 2001, and Miegè and Cebon, 2005) have reported that the suspension system is one of the main factors influencing vehicle stability.

Several models have been developed to represent the associated mechanism and its functioning at the rollover limit; however, such models do not take into account the change in the roll center height (CR) of the vehicle, and the change in the lateral separation between the springs. Reimpell *et al.* (2001) defined the roll center as the point in the center of the axle, around which the body begins to roll when a lateral force acts.

This paper focuses on a modeling and kinematic analysis of the suspension system under the action of a lateral inertial force. Mechanism theory is applied to model the tires and suspension systems, while the static analysis of the mechanism can be carried out with the use of three concepts: screw theory, graph theory and the Davies method (Tsai, 1999; Davies, 2000; Erthal, 2010; Moreno *et al.,* 2015) in order to describe the movement of the center of gravity (CG) and its roll center (CR).

The rest of the paper is organized as follows: in section 2, a mechanism that represents a vehicle as a two-dimension model is developed; section 3, the static analysis of the vehicle stability using the proposed method is presented. The results presented in Section 3 are analyzed and discussed with a case study in section 4. Finally, conclusions are drawn in section 5.

2. VEHICLE MODEL FOR LATERAL STABILITY

There are several types of suspension systems, but the most commonly used by vehicles is the leaf spring (Rill *et al.,* 2003), as shown in Fig. 1. For this analysis, only the rigid suspension system is taken into account, then, a two-dimensional vehicle model that represent the last unit of a heavy vehicle (trailer) is developed below.

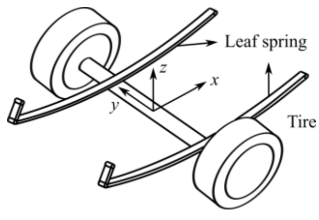

Figure 1 Solid axle with leaf spring (Rill *et al.,* 2003).

2.1. Suspension system

The suspension system comprises the linkage between the sprung and unsprung masses of a vehicle, which reduces the movement of the sprung mass, allowing the tires to maintain contact with the ground, and filtering disturbances imposed by the ground (Ledesma and Shih, 1999).

The leaf spring is a mechanism that allows three motions of the vehicle body under the action of lateral forces, this is, displacement in the *z*- and *y*-direction and a roll rotation about the *x*-axis (Rempel, 2001; Jazar, 2014), as shown in Fig. 2(a).

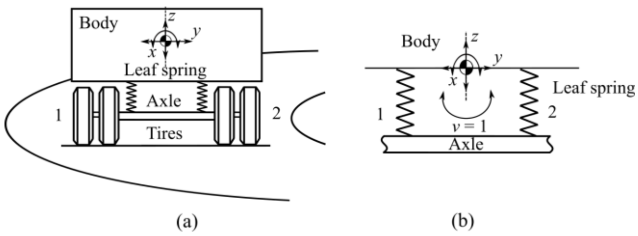

Figure 2: (a) Body motion. (b) Suspension system.

2.1.1. Kinematic chain of the suspension system

Mechanical systems can be represented by kinematic chains composed by links and joints, which facilities their modeling and analysis, (Kutzbach, 1929; Crossley, 1964; Tsai, 2001). To model the kinematic chain Eq. (2) was used together with to Eq. (3).

$$M = \lambda \times (n - j - 1) + j \tag{2}$$

$$\nu = j - n - 1 \tag{3}$$

where M is the degrees of freedom (DoF) or mobility of a mechanism, λ is the degrees of freedom of the space in which the mechanism is intended to function, n is the number of mechanism link, including the fixed link, j is the number of mechanism joints and v is the number of independent loops in the mechanism.

The kinematic chain of the suspension system in Fig. 2(b), has 3-DoF ($M= 3$), the work space is planar ($\lambda = 3$) and the number of independent loops is one ($v = 1$). Based on Eqs. (2) and (3) the kinematic chain of the suspension system should be composed by six links ($n = 6$) and six joints ($j = 6$).

To model this system the following consideration is taken into account: leaf springs are assumed as compression spring and can be represented by prismatic joints “P" supported in revolute joints “R" (Lee, 2001; Erthal, 2010). Applying these concepts to the kinematic chain of the suspension system, a model with the configuration shown in Fig. 3(b) is proposed.

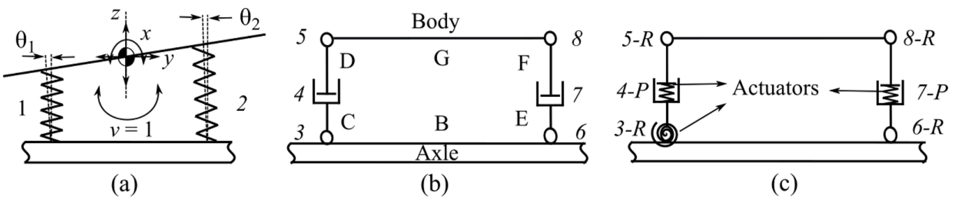

Figure 3: (a) Movement of the suspension system. (b) Kinematic chain of the suspension system. (c) Suspension system including actuators.

The kinematic chain of the suspension system is composed of six links identified by the letters B (vehicle axle), C and D (spring 1), E and F (spring 2) and G (the vehicle body); and six joints identified by numbers as follow: four revolute joints "R" (3,5, 6 and 8) and two prismatic joints "P" that represent the leaf springs of the system (4 and 7), as shown in Fig. 3(b and c).

The mechanism of Fig. 3(b) has 3-DoF, and requires three actuators to control its movement. The mechanism has one passive actuator in each prismatic joint (suspension system), and one passive actuator in the joint 3 (torsion spring), which controls the movement along the *x*- *y*- and *z*- axes, as shown in Fig. 3(c).

2.1.2. Kinematics of the suspension system

Given the pose (position and orientation) of the tires system, the kinematic problem consists of finding the corresponding rotation angle or displacement of all joints (active and passive) to achieve this position (Mejia *et al.*, 2013).

The movement of the tires is orientated by the forces acting on the mechanism (vehicle weight (W) and the inertial force (ma_y)). These forces affect the passive actuators of the mechanism, as shown in Fig. 4.

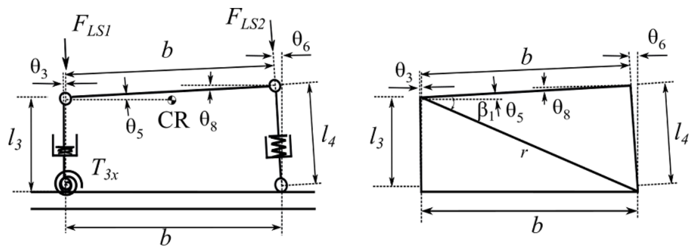

Figure 4: Movement of the suspension system.

The kinematics of the suspension system is defined by Eqs. (4) to (10).

$$\theta_3 = T_{3x} / k_{ts} \tag{4}$$

$$l_{3,4} = \delta_s + l_s = \frac{3Fl^3}{8ENBT^3} + l_s \approx \frac{-F_{LS1,2} + (W/2)}{k_s} + l_s \tag{5}$$

$$r = \sqrt{l_3^2 + b^2 - 2lb\cos\left(\tfrac{\pi}{2} + \theta_3\right)} \tag{6}$$

$$\beta_1 = \arccos\left(\left(b^2 + r^2 - l_4^2\right)/(2br)\right) \tag{7}$$

$$\theta_5 = \beta_1 + a\sin\left(\frac{b}{r}\sin\left(\tfrac{\pi}{2} + \theta_3\right)\right) - \frac{\pi}{2} \tag{8}$$

$$\theta_6 = \theta_3 + a\sin\left(\frac{b}{r}\sin\left(\tfrac{\pi}{2} + \theta_3\right)\right) - a\sin\left(\frac{b}{l_4}\sin(\beta_1)\right) \tag{9}$$

$$\theta_8 = \frac{\pi}{2} - \beta_1 - a\sin\left(\frac{b}{l_4}\sin(\beta_1)\right) \tag{10}$$

where T_{3x} is the moment around the x-axis on the joint 3, k_{ts} is the spring's torsion coefficient, δ_S is the vertical deformation of the leaf spring (Dhoshi *et al*., (2011)), F is the static force on the leaf spring, l is the length of the leaf spring, N is the number of leaf, B is the width of the leaf, T is the thickness of the leaf, E is the modulus of elasticity of a multiplate leaf, $F_{LS1;2}$ are the normal forces of the spring, W is the vehicle weight, l_S is the initial suspension height, k_S is the equivalent vertical spring stiffness of the leaf spring, $l_{3;4}$ are the instantaneous height of the spring, b is the lateral separation between the springs; and $\theta_{3,5,6,8}$ are the rotation angles of the revolute joints 3, 5, 6 and 8 respectively.

2.2. Vehicle model

Taking into account the model developed in our previous research (Moreno *et al.,* 2015), we developed a two-dimensional vehicle model, which integrates the vehicle body, the suspension system and the tires, as shown in Fig. 5(a). The kinematic chain of the vehicle model is composed of seven links identified by the letters A (road), B (vehicle axle and tires), C and D (spring 1), E and F (spring 2) and G (the vehicle body); and eight joints identified by numbers as follow: one revolute joint "R" (1 – tire-road contact - outer tire in the turn - Fig. 2(a)), four revolute joints "R" (3, 5, 6 and 8), two prismatic joints "P" that represent the leaf springs of the system (4 and 7), and one prismatic joint "P" (2 – tire-road contact - the lateral slide of inner tire in the turn 2 - Fig. 2(a)), the work space is planar (λ = 3); the proposed model has the parameters shown in the Figs. 5(b and c).

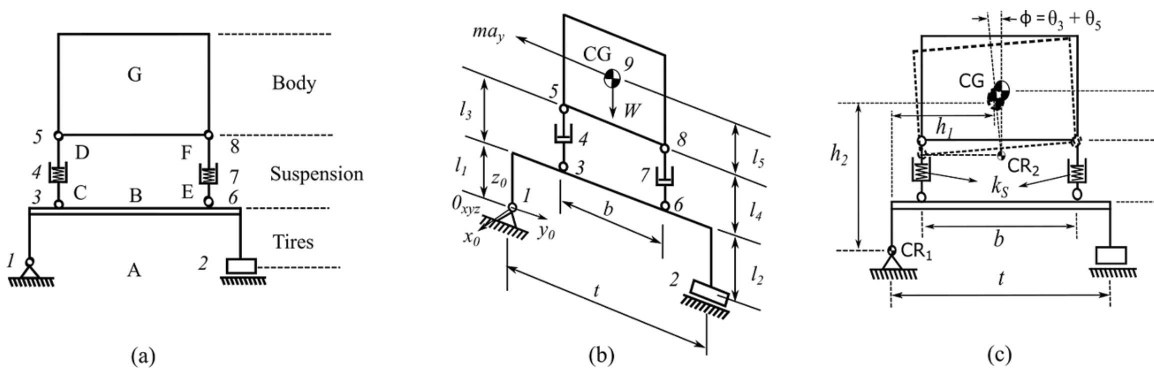

Figure 5: Vehicle model.

where t is the vehicle track width, b is the lateral separation between the springs, l_1 and l_2 are the dynamic rolling radii of the tires, l_5 is the CG height above the chassis, h_1 is the instantaneous lateral distance between the zero-reference frame and the CG, and h_2 is the instantaneous CG height, and l_S is the initial suspension height.

3. STATIC ANALYSIS OF MECHANISM

There are several methodologies which allow us to obtain a complete static analysis of the mechanism; however, in this paper the formalism described in Davies (1983 a) is used as the primary mathematical tool to analyze the mechanisms statically. The Davies method appears in many publications in the literature and further details regarding its use can be found in (Davies, 1983 a, b; Davies, 2000; Tsai, 1999; Erthal, 2010, and Moreno *et al.,* 2015). The Davies method was selected since it allows the static model for the mechanism to be obtained is a straight forward manner and it is also easily adaptable by using this approach.

Davies method is a mathematical tool, which uses screw theory and graph theory together with Kirchhoff laws for build and to solve the static and kinematic analysis of any mechanism. The Davies method for static analysis can be described in a simplified way through the following steps:

1. Given a mechanism, draw its kinematic chain identifying all of its "n" links, "f" external forces, and "e" direct couplings.
2. Draw the direct coupling graph "GA" for the mechanism with the links of the mechanism as the vertices of the graph, and the joints and external forces of the mechanism as the edges of the graph. Assign positive directions to each edge with an arrow pointing from the minor to major vertex.
3. Write the incidence matrix of the direct coupling graph $[I]_{n,e+f}$.
4. Generate the cut-set matrix $[Q]_{k;e+f}$ from the $[I]_{n,e+f}$ using Gauss-Jordan elimination method. Where is defined the number of cuts ($k = n - 1$) (Identity matrix), and chords ($l = e + f - n + 1$) in the action graph and depict them.
5. Write the expanded cut-set matrix $[Q]_{k;c}$, where c is the number of constraints of the joints and external forces of the mechanism.
6. Write a wrench $\$_{J\lambda;c}$ for each constraint or external force of the mechanism, as follows:

$$\$_{J_{\lambda;c}} = \begin{bmatrix} -z \\ 1 \\ 0 \end{bmatrix} J_{F_y} + \begin{bmatrix} y \\ 0 \\ 1 \end{bmatrix} J_{F_z} + \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix} J_{M_x} \tag{11}$$

where λ is the degrees of freedom of the space in which the mechanism is intended to function.

7. Replace each wrench $\$_{J\lambda;c}$ in the expanded cut-set matrix $[Q]_{k;c}$ in order to obtain the generalized action matrix $[A_N]_{\lambda k;c}$.

8. Operate algebraically the generalized action matrix $[A_N]_{\lambda k;c}$ in order to statically solve the system.

The proposed model (Fig. 5) represent a vehicle making a planar curve, to simplify the model, the following considerations are taken into account:

- the analysis model is made in steady state in the x-direction;
- the model does not have into account disturbances imposed by the road, and the friction forces (tire-ground contact);
- the only external forces included were: the vehicle weight (W), the inertial force acting on the mechanism ($m\ a_y$), the normal forces of the springs ($F_{LS1;2}$), and T_{3x} is the moment around the x-axis on the joint 6.

Using this method the statics of the mechanism can be defined, as exemplified in Eq. (12):

$$\left\lfloor \hat{A}_n \right\rfloor_{18\times 20} [\Psi]^T_{20\times 1} = [0]_{18\times 1} \tag{12}$$

where $[\hat{A}n]$ is the network unit action matrix, and $[\Psi]$ is the action vector of the magnitudes of the mechanism.

It is necessary to identify the set of primary variables $[\Psi_p]$ (known variables), among the variables of Ψ, starting from the system (Eq. 12). Once identified, the system is divided into two sets, as shown by Eq. (13).

$$\left\lfloor \hat{A}_{ns} \right\rfloor_{18\times 18} [\Psi_s]^T_{18\times 1} + \left\lfloor \hat{A}_{np} \right\rfloor_{18\times 2} [\Psi_p]^T_{2\times 1} = [0]_{18\times 1} \tag{13}$$

where $[\Psi_p]$ is the primary variable vector, $[\Psi_s]$ is the secondary variable vector (unknown variables), $[\hat{A}_{np}]$ are the columns corresponding to the primary variables and $[\hat{A}_{ns}]$ are the columns corresponding to the secondary variables. In this case, the primary variable vector and the secondary variable vector are shown in Eq. (14).

$$\begin{aligned} [\Psi_p]^T_{2\times 1} &= [W \quad ma_y]^T \\ [\Psi_s]^T_{18\times 1} &= [F_{1y} \quad F_{3y} \quad F_{3z} \quad M_{4x} \quad F_{4n} \quad \cdots \quad T_{3x} \quad F_{LS1} \quad F_{LS2} \quad F_{1z} \quad F_{2z}]^T \end{aligned} \tag{14}$$

where F (forces) and M (moments) denotes the constraints of the mechanism joints (unknown variables), and W (vehicle weight) and ma_y (inertial force) are the known variables.

Solving the system in Eq. (13), using the Gauss-Jordan elimination method, all secondary variables are functions of the primary variables (vehicle weight (W) and the inertial force (ma_y)). For this analysis only is taken into account the forces acting on the suspension and the tires; the last five rows of solution system provide the next equations:

$$T_{3x} + c_1 \times W + c_2 \times ma_y = 0 \tag{15}$$

$$F_{LS1} + c_3 \times W + c_4 \times ma_y = 0 \tag{16}$$

$$F_{LS2} + c_5 \times W + c_6 \times ma_y = 0 \tag{17}$$

$$F_{1z} + ((h_1 - t)/t) \times W - (h_2 / t) \times ma_y = 0 \tag{18}$$

$$F_{2z} - (h_1 / t) \times W + (h_2 / t) \times ma_y = 0 \tag{19}$$

where c_i are constants of the equation system solution.

At the rollover limit condition, the normal load, F_{2z}, reaches zero and on applying this in the Eq. (19), the *SSF* factor can be calculated as:

$$SSF = \frac{a_y}{g} = \frac{h_1}{h_2} \tag{20}$$

Making a comparison between the Eq. (1) and Eq. (20); it is demonstrated that the *SSF* factor is dependent on the lateral and vertical location of the vehicle center of gravity (CG).

For calculating the *SSF* factor, the inertial force is increased until the lateral load transfer (*LLT*) in the axle is complete (all the load is transferred of the inner tire to the outer tire when the vehicle makes a turn) (Rill, (2011), Moreno *et al., (2015)*). From the Eqs. (15) to (19) the instantaneous forces acting on the mechanism can be obtained. These forces modifying the configuration of the suspension system Eqs. (4) to (10).

4. CASE STUDY

For this analysis, a two-dimensional vehicle model that represents the last axle of the trailer is proposed. In this model a suspension system with tandem axle is used and the suspension parameters are dependent on the construction materials. Harwood *et al.,* (2003) indicated that the range of values for the vertical spring stiffness per axle is k_s = 1500-2400 kN.m^{-1}. Other important parameter is the dynamic rolling radius or loaded radius l_i, the model proposed considers Michelin® XZA® (2013) radial tires with dynamic rolling radius l_i = 0:499 m. Figure 6 and Table 1 shown the heavy vehicle parameters used in this analysis (Ervin and Guy, 1986).

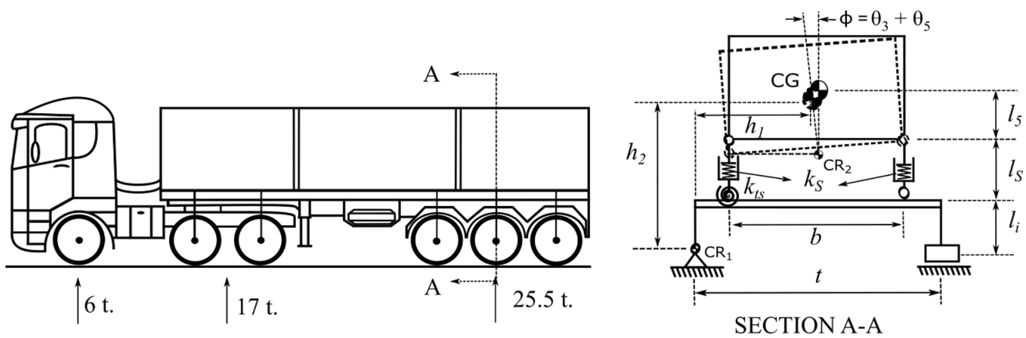

Figure 6: Parameters of heavy vehicle.

Table 1: Parameters of heavy vehicle.

Parameter	Value	Units
Vehicle weight - *W*	250	kN
Track width -t	1.86	m
Equivalent vertical spring stiffness – $k_{s.equ}$.	5400	kN/m
Spring's torsion coefficient – k_{ts}	k_{ts} >>> $k_{s.equ.}$	kN/m
Initial suspension height - l_s	0.205	m
Dynamic rolling radius - l_i	0.499	m
Lateral separation between the springs - *b*	0.95	m
Height of CG above the chassis - l_5	1.346	m

The simulation model was made in MATLAB®, for the *SSF* factor calculation (Eq. (19) and (20)), the inertial force is increased until the entire lateral load transfer (*LLT*) was transferred from the inner to the outer tires, along the turn (Rill, (2011)).

In the analysis of the proposed model, at the rollover limit, a reduction of around 10% in the *SSF* factor of the vehicle can be observed. This reduction results from the action of the suspension system, which allows a body roll angle of $\phi = \theta_3 + \theta_5 = 3.87^0$, as shown in Fig. 7.

Furthermore, this model allows the determination of the vertical (h_2 - the instantaneous CG height – shown in Fig. 8) and lateral (h_1 - the instantaneous lateral distance between the zero-reference frame and the center of gravity - shown in Fig. 9) movements of the vehicle center of gravity (CG) .

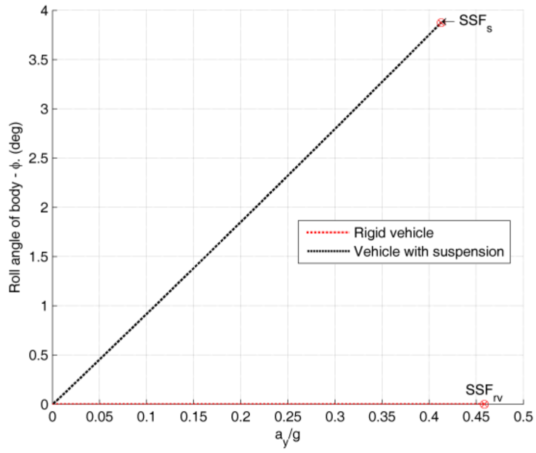

Figure 7: Roll angle of the vehicle model.

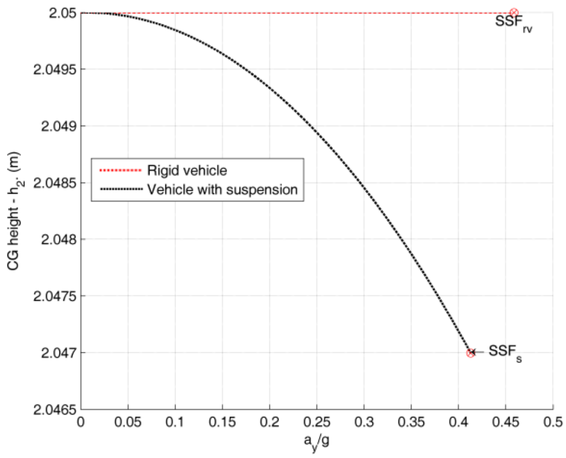

Figure 8: Instantaneous CG height (h_2).

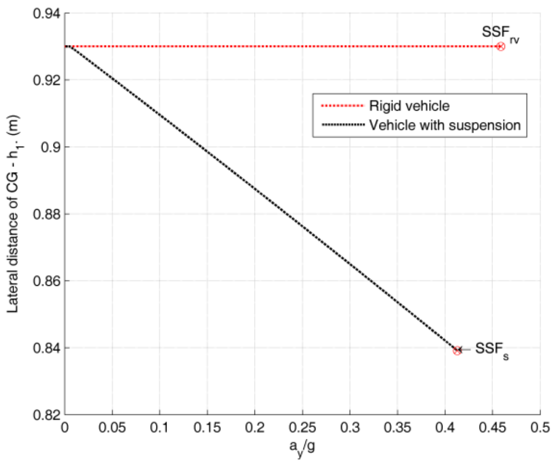

Figure 9: Instantaneous lateral distance between the zero-reference frame and the center of gravity (h_1).

In the Fig 10 can be observed the vertical movement of the roll center (CR) of the vehicle body (Fig. 6 – SECTION A-A); for the rigid vehicle model, all vehicle rotates around the joint 1 (CR_1: $h=0$); and for the vehicle model with suspension system, the CR is located in the instant center of rotation of the vehicle body (CR_2: $h = l_1 + l_3 \cos(\theta_3)$)

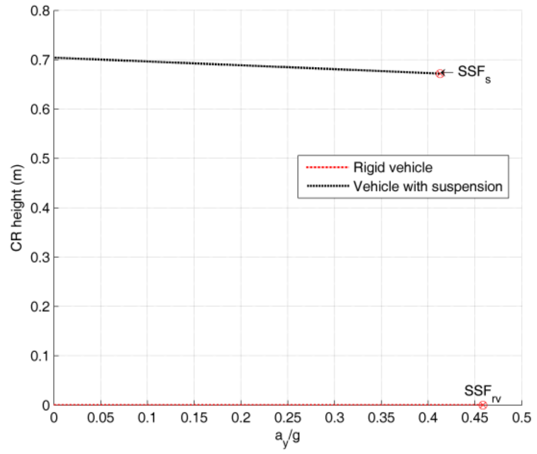

Figure 10: Instantaneous CR heights.

Finally, the proposed model shows how the change of *b* (lateral separation between the springs) influences the *SSF* factor. Some heavy vehicles with tanker semi-trailer have more lateral separation between the springs, this causes that decrease its roll angle, and therefore increase its *SSF* factor; one centimeter of lateral separation between the springs corresponds to the gain of stability around 0.001 g. as shown in Fig. 11.

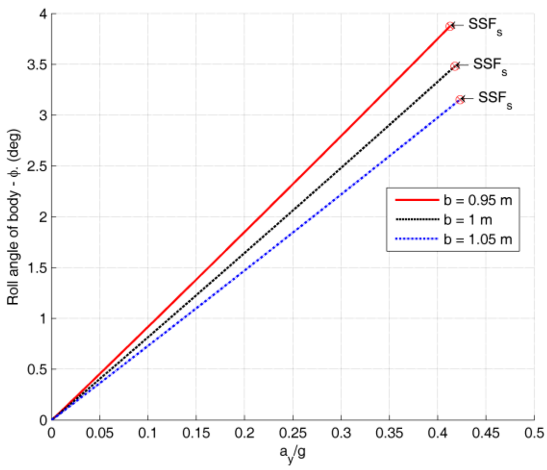

Figure 11: Change of b - *SSF* factor.

5. CONCLUSIONS

The proposed suspension system allows the determination of its influence on the calculation of the *SSF* factor of the vehicle and the changes in the position of the center of gravity (CG) and roll center (CR) of the vehicle.

This model also shows how the change in the lateral separation between the springs (*b*) is an important factor to consider in the design and construction of vehicles, as more lateral separation will increase the vehicle stability.

The decrease in the *SSF* factor is important since, it allows new road speed limits to determined, aimed at improving road safety and decreasing the occurrence of accidents related to vehicle stability.

6. FUTURE WORKS

As future works in this line of research, we propose the further development of the kinematic models of vehicles, in order to include others factors related to lateral stability, such as tire system, the fifth wheel system and longitudinal effects.

ACKNOWLEDGMENTS

This research was supported by the Brazilian governamental agencies Coordenação de Aperfeiçoamento de Pessoal de Nível Superior (CAPES) and Conselho Nacional de Desenvolvimento Cientifico e Tecnológico (CNPq).

REFERENCES

Chang, T. Effect of vehicles' suspension on highway horizontal curve design. Journal of Transportation Engineering (2001) 89-91.

Crossley, F.R.E. A contribution to Grübler's theory in number synthesis of plane mechanisms. ASME Journal of Engineering Industry (1964) 1-8.

Davies, T.H. Mechanical networks-III wrenches on circuit screws, Mechanism and Machine Theory 18 (2) (1983a) 107-112.

Davies, T.H. Mechanical networks-I passivity and redundancy, Mechanism and Machine Theory 18 (2) (1983b) 95-101.

Davies, T.H. The 1887 committee meets again. Subject: freedom and constraint, Ball 2000 Conference (2000) 56.

Dhoshi, N. P. and Ingole, N. K. and Gulhane, U. D. Analysis and modification of leaf spring of tractor trailer using analytical and finite element method, International Journal of Modern Engineering Research, (2011), v.1, n. 2, pp. 719-722.

Erthal, J. Modelo cinestático para analise de rolagem em veículos. Ph.D. Thesis, Federal University of Santa Catarina, Florianópolis, Brazil. (2010).

Ervin, R.D., Guy, Y. The influence of weights and dimensions on the stability and control of heavy duty trucks in Canada, Tech. rep., UMTRI – The University of Michigan Transportation Research Institute, Final Report UMTRI-86-35/III (1986).

Gillespie, T.D. Fundamentals of Vehicle Dynamics, 7th Edition, SAE International, ISBN 1560911999, 1992.

Hac, A. Rollover stability index including effects of suspension design. SAE 2002 World Congress (2002) 4-7.

Harwood, D.W., Torbic, D.j.,Richard, K.R. Review of Truck Characteristics as Factors in Roadway Desing, National Cooperative Highway Research Program, ISBN 0-309-08779-1, 2003.

Jazar, R. Vehicle Dynamics - Theory and Application. 2nd Edition, Springer, ISBN 978-0387742441, 2009.

Kutzbach, K. Mechanische leitungsverzweigung, Maschinenbau 8 (21) (1929) 710-716.

Ledesma, R., Shih, S. Heavy and medium duty vehicle suspension-related performance issues and effective analytical models for system design guide, SAE Technical Paper 1999-01-3781doi:10.4271/1999-01-3781.

Lee, U. (2001) ‘A Study on a Method for Predicting the Vehicle Controllability and Stability Using the Screw Axis Theory’, PhD. These. Hanyang University, Seoul, South Korea.

Mejia, L., Simas, H., Martins, D. Force capability maximization of a 3rrr symmetric parallel manipulator by topology optimization., 22nd International Congress of Mechanical Engineering (COBEM 2013).

Michelin, Michelin XZA tire (2013).

Miegè, A., Cebon, D. Active roll control of an experimental articulated vehicle. Journal of Automobile Engineering (2005) 791-806. DOI:10.1243/095440705X28385.

Moreno, G.G., Nicolazzi, L., Vieira, R., Martins, D. Three-dimensional analysis of the rollover risk of heavy vehicles using Davies method, 14th World Congress in Mechanical and Machine Science (IFToMM2015) DOI10.6567/IFToMM.14TH.WC.PS4.006.

Reimpell, J., Stoll, H., Betzler, J. The Automotive Chassis. 2nd Edition, Published by Butterworth-Heinemann, ISBN 0 7506 5054 0, 2001.

Rempel, M.R. Improving the dynamic performance of multiply-articulated vehicles., Master's thesis, The University of British Columbia, Vancouver, Canada. (2001).

Rill, G. Road Vehicle Dynamics: Fundamentals and Modeling. CRC Press, ISBN 978-1-4398-3898-3, 2011.

Rill, G., Kessing, N., Lange, O., J, M. Leaf spring modelling for real time applications. 18th IAVSD-Symposium in Atsugi-Japan. (2003) 1-22.

Tsai, L.W. Robot Analysis - The Mechanics of Serial and Parallel Manipulators. John Wiley and Sons, Ltd., ISBN 0-471-32593-7, 1999.

Tsai, L.W. Mechanism Design: Enumeration of Kinematic Structures According to Function. CRC Press., ISBN 0849309018, 2001.

Winkler, C. Experimental determination of the rollover threshold of four tractor-semitrailer combination vehicles. UMTRI Research Review. The University of Michigan Transportation, (1987).

Winkler, C. Rollover of heavy commercial vehicles, UMTRI Research Review, The University of Michigan Transportation Research Institute 31 (2000) 1-20.

7. COPYRIGHT RESPONSIBILITY