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DYNAMIC ANALYSIS OF COMFORT AND HEALTH EFFECTS OF A BAJA SAE VEHICLE

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Abstract. *The suspension system plays a key role in a vehicle, since its behavior affects significantly the performance of the vehicle, and the comfort of the passengers. The design of a suspension is a challenging task, in view of the multiple control parameters that need to attend the requirements of vertical and lateral dynamics, for comfort and performance. In the present work, the behavior of the vertical dynamics of a Baja SAE is analyzed, in order to verify the comfort level and the limit of exposure time to the ground unevenness provided by the off-road track. The analysis is done through the analytical construction of a full car model of the vehicle suspension, containing 7 degrees of freedom. The system's mass, stiffness and damping matrices are included in the state space equations. Simulations are performed for three common track conditions: a sine road, cavities and elevations, and for two different speeds. The response obtained allow the evaluation of the suspension behavior and the influence of the main parameters in the accelerations transmitted to the driver. The results are compared with the limits supplied by the ISO standards, regarding the aspects related to the comfort and health of the driver, and so, the damping properties are determined in order to achieve a limit exposure longer than 4 hours.*

Keywords: *Vehicle Suspension, Vertical Dynamics, Comfort, Health Exposure, Baja SAE.*

1. INTRODUCTION

The Baja SAE is an off-road vehicle, fully designed and built by undergraduate engineering students, in order to participate in the SAE organized competitions. These competitions are exclusive to undergraduate students, with the purpose of exposing the participants to real engineering problems, involving several areas of knowledge, since the organization of a competition team, going through the design and manufacture of the vehicle until the performance of it in several conditions. UFSM participates of SAE competitions since 2003, represented by the Bombaja UFSM team, which currently competes with the BJ-14 prototype, as seen in Fig. 1.

Figure 1. BJ-14 prototype

Due to the rough off-road terrain, the suspension system in a Baja SAE vehicle requires a careful study, in order to guarantee the performance in the track, but also the comfort and the limit time of exposure to the health of the driver. The vertical dynamic analysis is encouraged by the studies of Griffin (1996), where it has shown that the vibrations in a vehicle have a significant effect on the comfort of the passengers. Herszenhaut (2013) highlight the importance in reducing the base movements translated to the chassis, to provide a more comfortable ride in the vehicle. The ISO-2631(1997) correlates the accelerations of the chassis movements, in the range of frequency, to the comfort level of the driver, and the maximum time of exposure that will not entail in health risks for the passengers. The accelerations felt by the driver in a Baja SAE vehicle were studied by Buarque et. al (2003), using a four degree of freedom model. Although this model provides good results for the bouncing movements, it is insufficient for a more careful analysis, since it neglects the pitching or the rolling excitations of the sprung mass, encouraging the construction of a full car model.

In a previous work, the spring stiffness was determined, in order to enable the vehicle to operate in the recommended ranges of frequency, which align performance with the driver comfort. The next step in the comfort analysis is the change of the damping value of the shock absorbers, with the adjustment buttons, in order to analyze the behavior of the vehicle in the same conditions. This analysis is correlated with Sun and Cui (2010), who showed a notable influence of the sprung damping in the behavior of a full car model, and indicates the high sensitiveness of this parameter in comparison to others, as spring stiffness and sprung mass inertia. Fernandes et. al (2015) applied asymmetrical damping properties to a quarter car model, aiming to increase the passengers comfort with the regulation in the shock absorber fluid, analyzing the accelerations and displacements felt by the passengers.

In this work, three linear damping values are compared, in order to analyze the displacements and accelerations translated to the sprung mass of a Baja SAE vehicle, along two very common endurance obstacles, and at two speeds of the vehicle. The values obtained will be compared with the parameters established in the ISO-2631(1997) standard, to classify the regulations in the levels of comfort, and to verify if any of them imply in a health risk to the Baja driver.

2. COMFORT AND HUMAN TOLERANCE

The transference of vibrations to the human body is not only related with the comfort of the occupants, but also implies a health risk, depending on the magnitude, frequency and exposure time. In the case of a Baja SAE, there is no daily exposure, but there is a constantly unevenness in the ground, besides a long duration of the tests in the competition. Therefore, it requires a close study in the vibrations translated to the driver, and if they imply any health risk.

To analyze the equivalent magnitude of vibration that is felt by the driver, the methodology proposed by the ISO 2631-1 (1997) standard is applied, since it considers several kinds of vibration, besides it establish limit values to discomfort, sickness and occupational risk. To quantify the vibration motion considering the magnitude and frequency, since both act on the human perception, the ISO 2631-1 (1997) suggest a parameter that represents an averaged acceleration over a period of time, called *weighted acceleration (a_w),* that is defined as:

$$a_w = \left\{\frac{1}{T}\int_0^T [a_w(t)]^2\, dt\right\}^{\frac{1}{2}} \tag{1}$$

where $a_w(t)$ is the instantaneous acceleration, T is the duration of measurement, and t is the time. To include the frequency in the acceleration value, the standard proposes frequency weighting curves, that will result in a multiplying factor for the weighted acceleration, resulting in the *frequency-weighted acceleration (a_{wf}),* defined as:

$$a_{wf} = \left(\sum_i \left(w_i\, a_{w,i}\right)^2\right)^{\frac{1}{2}} \tag{2}$$

where w_i is the weighting factor for the *i* th one-third octave band taken of the frequency weighting curves, and a_{wi} is the weighted acceleration for the *i* th one-third octave band. When the vibration consists in more than one period of exposure, with different magnitudes or durations, the value of equivalent vibration $a_{w,e}$ is evaluated as:

$$a_{w,e} = \left[\frac{\sum a_{wf,i}{}^4 T_i}{\sum T_i}\right]^{\frac{1}{4}} \tag{3}$$

This value is used to classify the level of risk to the health of the driver, according to the zones defined in the Fig. 2. Above the dashed area, the occupational risk is high, and health damages are very probable, requiring an immediate intervention. In the zone, there is a small probability of damages, but there is no emergency for an intervention. Below the zone, the effects are not clearly documented. The zone contains a vibration exposure from 4 to 8 hours.

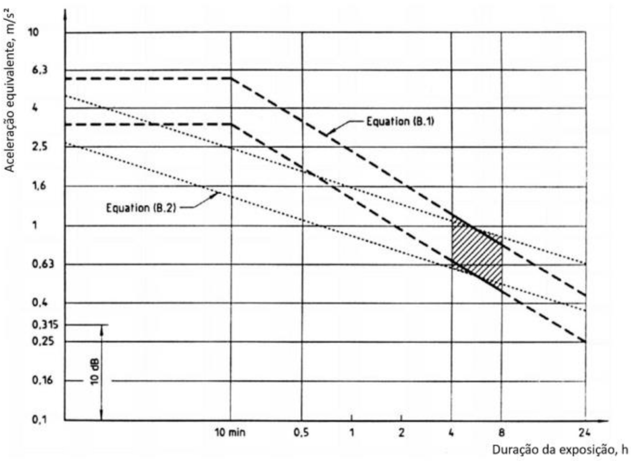

Figure 2 - Health guidance zones (ISO, 1997)

For the comfort analysis, the rotational motions are included in the equivalent magnitude of vibration. To combine vibrations in more than one direction, the total value is determined by vibration in orthogonal coordinates. This is performed with the coordinate system of a seated person, as seen in Fig. 3, and it is calculated by:

$$a_v = \left(k_z{}^2 a_{wz}{}^2 + k_{rx}{}^2 a_{wrx}{}^2 + k_{ry}{}^2 a_{wry}{}^2\right)^{\frac{1}{2}} \tag{4}$$

where a_v is the total value of vibration; a_{wz}, a_{wrx} and a_{wry} are the acceleration values for the orthogonal axes *x*, *y* and *z*; and k_z, k_{rx} and k_{ry} are multiplying factors, in which for a seated person, they assume the values of 1, 1,4 and 1,4, respectively. The acceleration components required by the standard are estimated using the seven degrees of freedom full car model simulation. This model is described in the next section.

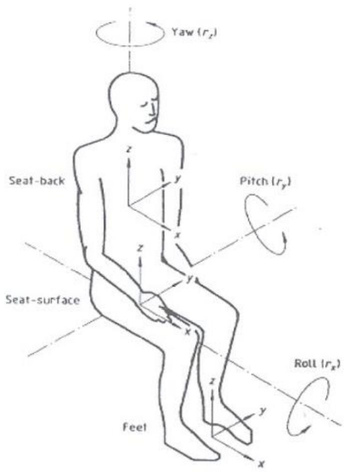

Figure 3 - Coordinate system for a seated person

3. MATHEMATICAL MODEL

The simulation procedure is done by using a full car mathematical model, where its matrices of mass, damping and stiffness are included in the state space equations, which are posteriorly solved by the Runge-Kutta method, returning the system's response to the obstacles of the track.

3.1 The Full Car Model

The model used in the analysis of this work is proposed by Nicolazzi, Machado and Edison (2012), describing the bounce, roll and pitch motions of the sprung mass, as well as the bounce motion of each wheel, for a vehicle with an independent suspension geometry, in both front and rear axle. A representation of the model is given in Fig. 4, where the seven degrees of freedom are shown, along with the road input on each wheel.

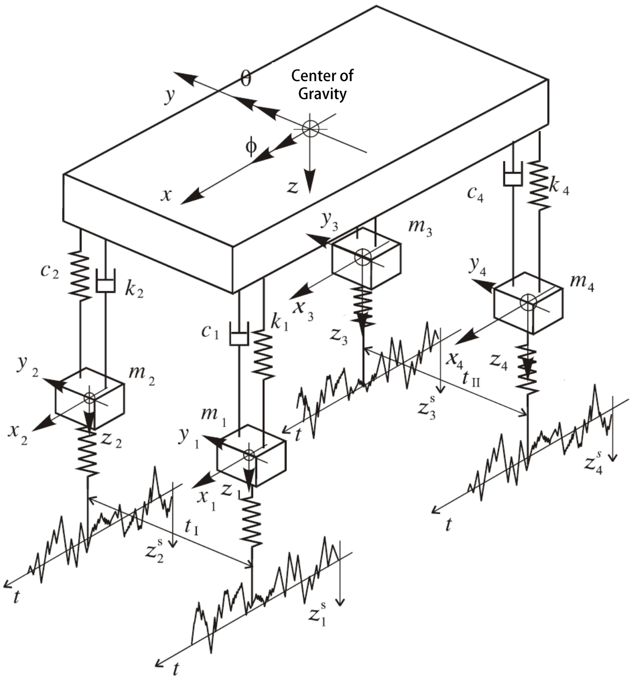

Figure 4 - Full Car Model (Adapted from Nicolazzi, Machado and Edison, 2012)

Starting from the dynamic balance equations, it is possible to determine the seven equations of the system, since the model contains seven degrees of freedom. With the motion equation to each degree of freedom, it is possible to arrange the system in a matricidal form, that is a very useful form to perform a numerical analysis, besides it will allow to enter the system matrices in the state space equations.

For the model of this work, the displacement vector *x(t)* is defined by:

$$\{x(t)\}=\{z(t) \quad \Phi(t) \quad \theta(t) \quad z_1(t) \quad z_2(t) \quad z_3(t) \quad z_4(t)\}^T \tag{5}$$

where $z(t)$, $\Phi(t)$ and $\theta(t)$ are the bounce, roll and pitch displacements of the sprung mass, respectively; $z_1(t)$ the bounce of the front left wheel, $z_2(t)$ the bounce of the front right wheel, $z_3(t)$ the bounce of the rear right wheel and $z_4(t)$ the bounce of the rear left wheel.

For the bounce movement of the sprung mass, the equation of motion becomes:

$$m\ddot{z}=(-k_1-k_2-k_3-k_4)z+(-c_1-c_2-c_3-c_4)\dot{z}+\left((k_1-k_2)\frac{t_I}{2}-(k_3-k_4)\frac{t_{II}}{2}\right)\Phi+\left((c_1-c_2)\frac{t_I}{2}-(c_3-c_4)\frac{t_{II}}{2}\right)\dot{\Phi}+\big((k_1+k_2)\alpha_I-(k_3+k_4)\alpha_{II}\big)\theta+\big((c_1+c_2)\alpha_I-(c_3+c_4)\alpha_{II}\big)\dot{\theta}+k_1z_1+k_2z_2+k_3z_3+k_4z_4+c_1\dot{z}_1+c_2\dot{z}_2+c_3\dot{z}_3+c_4\dot{z}_4 \tag{6}$$

where m is the sprung mass, k_i is the stiffness of each spring, c_i is the damping of each damper, t_I and t_{II} are the front gauge and rear gauge, respectively, and α_I and α_{II} are the distance of the center of mass to the front, and to the rear axle, respectively.

For the roll movement, the equation become:

$$I_x\ddot{\Phi}=\left((k_1-k_2)\frac{t_I}{2}-(k_3-k_4)\frac{t_{II}}{2}\right)z+\left((c_1-c_2)\frac{t_I}{2}-(c_3-c_4)\frac{t_{II}}{2}\right)\dot{z}+\left(-(k_1+k_2)\left(\frac{t_I}{2}\right)^2-(k_3+k_4)\left(\frac{t_{II}}{2}\right)^2\right)\Phi+\left(-(c_1+c_2)\left(\frac{t_I}{2}\right)^2-(c_3+c_4)\left(\frac{t_{II}}{2}\right)^2\right)\dot{\Phi}+\left(-(k_1-k_2)\frac{\alpha_It_I}{2}-(k_3-k_4)\frac{\alpha_{II}t_{II}}{2}\right)\theta+\left(-(c_1-c_2)\frac{\alpha_It_I}{2}-(c_3-c_4)\frac{\alpha_{II}t_{II}}{2}\right)\dot{\theta}-k_1\frac{t_I}{2}z_1+k_2\frac{t_I}{2}z_2+k_3\frac{t_{II}}{2}z_3-k_4\frac{t_{II}}{2}z_4-c_1\frac{t_I}{2}\dot{z}_1+c_2\frac{t_I}{2}\dot{z}_2+c_3\frac{t_{II}}{2}\dot{z}_3-c_4\frac{t_{II}}{2}\dot{z}_4 \tag{7}$$

where I_x is the inertia moment of mass along the x axis. For the pitch movement:

$$I_y\ddot{\theta}=((k_1+k_2)\alpha_I-(k_3+k_4)\alpha_{II})z+((c_1+c_2)\alpha_I-(c_3+c_4)\alpha_{II})\dot{z}+(-(k_1-k_2)\frac{\alpha_I t_I}{2}-(k_3-k_4)\frac{\alpha_{II}t_{II}}{2})\Phi+(-(c_1-c_2)\frac{\alpha_I t_I}{2}-(c_3-c_4)\frac{\alpha_{II}t_{II}}{2})\dot{\Phi}+(-(k_1+k_2)\alpha_I^2+(-(k_3+k_4)\alpha_{II}^2)\theta+(-(c_1+c_2)\alpha_I^2-(c_3+c_4)\alpha_{II}^2)\dot{\theta}-k_1\alpha_I z_1-k_2\alpha_I z_2+k_3\alpha_{II}z_3+k_4\alpha_{II}z_4-c_1\alpha_I\dot{z}_1-c_2\alpha_I\dot{z}_2+c_3\alpha_{II}\dot{z}_3+c_4\alpha_{II}\dot{z}_4 \quad (8)$$

where I_y is the inertia moment of mass along the y axis. For each wheel, the equations are:

$$m_1\ddot{z}_1= k_1 z - k_1\alpha_I\theta - k_1\frac{t_I}{2}\Phi + (-k_1-k_1^p)z_1 + k_1^t z_1^s + c_1\dot{z} - c_1\alpha_I\dot{\theta} - c_1\frac{t_I}{2}\dot{\Phi} - c_1\dot{z}_1 \quad (9)$$

$$m_2\ddot{z}_2= k_2 z - k_2\alpha_I\theta + k_2\frac{t_I}{2}\Phi + (-k_2-k_2^p)z_2 + k_2^t z_2^s + c_2\dot{z}-c_2\alpha_I\dot{\theta} + c_2\frac{t_I}{2}\dot{\Phi} + -c_2\dot{z}_2 \quad (10)$$

$$m_3\ddot{z}_3 = k_3 z - k_3\alpha_{II}\theta + k_3\frac{t_{II}}{2}\Phi + (-k_3-k_3^p)z_3 + k_3^t z_3^s + c_3\dot{z} - c_3\alpha_{II}\dot{\theta} + c_3\frac{t_{II}}{2}\dot{\Phi} - c_3\dot{z}_3 \quad (11)$$

$$m_4\ddot{z}_4 = k_4 z + k_4\alpha_{II}\theta - k_4\frac{t_{II}}{2}\Phi + \left(-k_4-k_4^t\right)z_4 + k_4^t z_4^g + c_4\dot{z} + c_4\alpha_{II}\dot{\theta} - c_4\frac{t_{II}}{2}\dot{\Phi} - c_4\dot{z}_4 \quad (12)$$

where m_i is the mass of each wheel, z_i^g is the ground displacement in each wheel and k_i^t is the stiffness of each tire. With all the equations in hand, it is possible to write the equation of motion of the system, in the matricidal form, that results in:

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}= \{\overline{F}\} \quad (13)$$

where [M] is the mass matrix, [C] is the damping matrix, [K] is the stiffness matrix and $\{\overline{F}\}$ is the force vector. In the case of this analysis, when only the ground unevenness excites the model, the force vector is defined as:

$$\{\overline{F}\}= z^S(t) = \left\{0 \quad 0 \quad 0 \quad k_1^p z_1^s(t) \quad k_2^p z_2^s(t) \quad k_3^p z_3^s(t) \quad k_4^p z_4^s(t)\right\}^T \quad (14)$$

where $z^S(t)$ is the vector of the ground excitations.

3.2 State Space Technique

To apply the state space technique, the system must be described with first order equations. As long as the full car system is described with second order equations, the procedure suggested by Thorby (2008) is adopted, in which it renames the variables of displacements and velocities as:

$$\{x_d\}= \begin{Bmatrix}\{x\}\\ \{\dot{x}\}\end{Bmatrix} \quad (15)$$

Replacing the new nomenclature in Eq. 13, the state space equations can be modeled, for an open loop control system, as suggested by Thorby (2008), in the following way:

$$\{\dot{x_d}\}= [A]\{x_d\}+ [B]\{F\} \quad (15)$$

$$\{y\}= [C_{ss}]\{x_d\}+ [D]\{F\} \quad (15)$$

where A is the state matrix, given by:

$$[A]=\begin{bmatrix}[0]_{7x7} & [I]_{7x7}\\ -[M]^{-1}[K] & -[M]^{-1}[C]\end{bmatrix} \quad (15)$$

and

$$[B]=\begin{bmatrix}[0]_{7x7}\\ [M]^{-1}\end{bmatrix} \quad (15)$$

The *y* vector, also called the observation vector, will not be used in this work, since none control module is included at the present stage of the work. With the first order equations, the Runge-Kutta method can be performed, obtaining the output data of the model. Numerically, the method is performed by the MATLAB command “ode45”, which applies the method to the resolution of first order equations.

3.3 Ground Conditions

In the case of a Baja SAE, the conditions considered are very common obstacles in the endurance tracks. First, the obstacle *bumptrack,* which can be modeled as a sine wave with short period, as Fig. 5 illustrates. The frequency of the wave depends of the distance between the peaks, and the velocity of the vehicle. There will be considered two different velocities for each obstacle: 4 m/s and 12 m/s, which are the lowest and highest average of velocity in the endurance tracks.

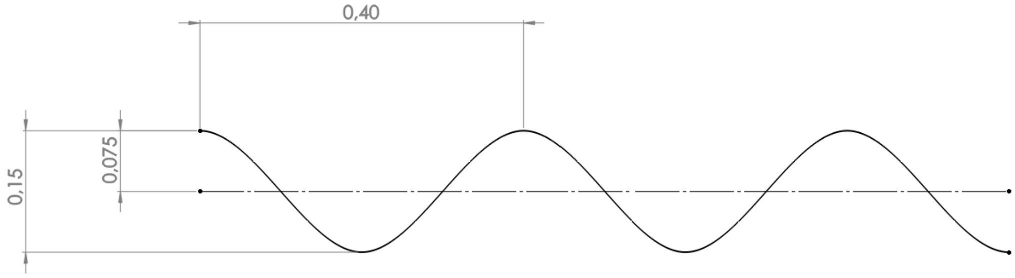

Figure 5 - *Bumptrack* profile

The next obstacle consists in several cavities on the ground, alternately displaced. The obstacle is very similar to the *bumptrack,* except that excites roll movements because of the cavities positions. A top view of the obstacle is seen in Fig. 6.

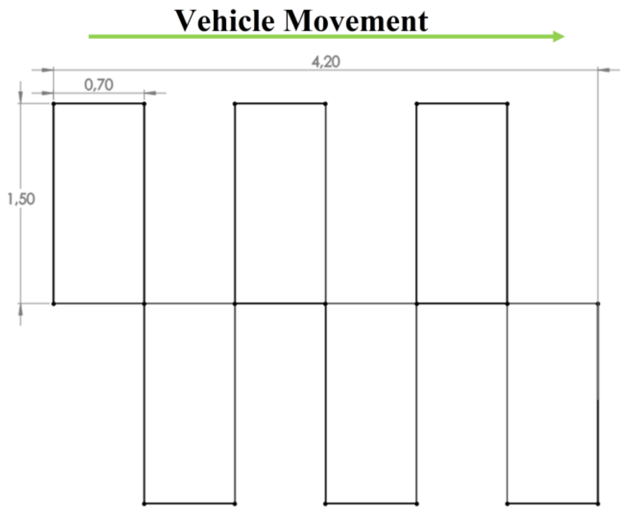

Figure 6 - Cavities positions and sizes

3.4 Vehicle Information

The vehicle has some fixed parameters, as the masses, moments of inertia and dimensions. These fixed parameters are in Tab. 1. The only change in the parameters consists in the shock fluid regulation, by turning two buttons in the shock, choosing between a softer and a stiffer damping value. The only remaining data is the shock damping value.

Table 1 - Fixed vehicle parameters

Property	Unit	Symbol	Value	Property	Unit	Symbol	Value
Sprung Mass	Kg	m_s	240	Front Tire Stiffness	N/m	k_1^t,k_2^t	29000
Longitudinal Inertia Moment	$\frac{Kg}{m^4}$	I_{xx}	57	Rear Tire Stiffness	N/m	k_3^t,k_4^t	32000
Transversal Inertia Moment	$\frac{Kg}{m^4}$	I_{yy}	41	Distance Between Rear Axle and Center of Mass	m	α_I	0,58
Front Unsprung mass	Kg	m_1,m_2	6,6	Distance Between Front Axle and Center of Mass	m	α_{II}	0,87
Rear Unsprung mass	Kg	m_3,m_4	7	Front Gauge	m	t_I	1,24
Front Spring Stiffness	N/m	k_1,k_2	6145,63	Rear Gauge	m	t_{II}	1,24
Rear Spring Stiffness	N/m	k_3,k_4	8149,17				

3.5.1 Damping Value

The shock absorber is regulated in three different conditions: one keeps all regulations to the softest damping value, other keeps all regulations to the stiffest damping value, and the last one is an intermediate stage. The linearized values are in the Fig. 7. The graph below gives that the damping constants of each regulation are: 850,25 N.s/m, 2331,11 N.s/m and 4009,03 N.s/m for the regulations 1, 2 and 3, respectively.

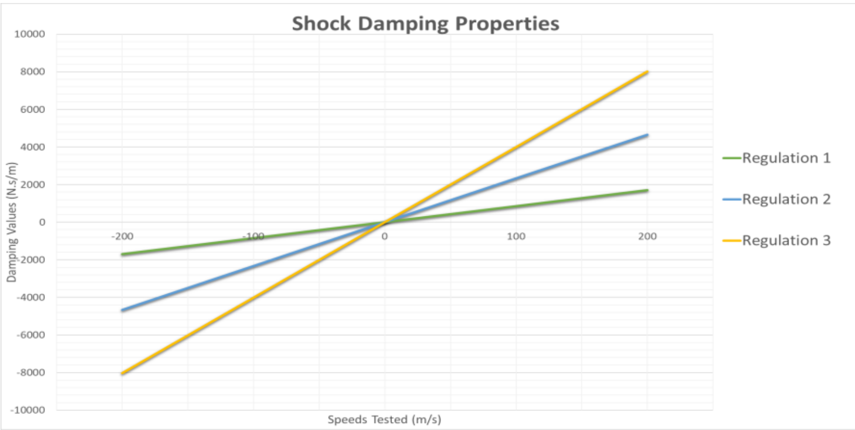

Figure 7 - Shock Damping Properties

4. RESULTS AND DISCUSSION

The results for each motion of the *bumptrack* obstacle are in Figs. 8 and 9; as long as the movements in the cavities obstacle are in Figs. 10, 11 and 12. The roll motion is null in the *bumptrack* because the tires of the same axle hit the obstacle simultaneity.

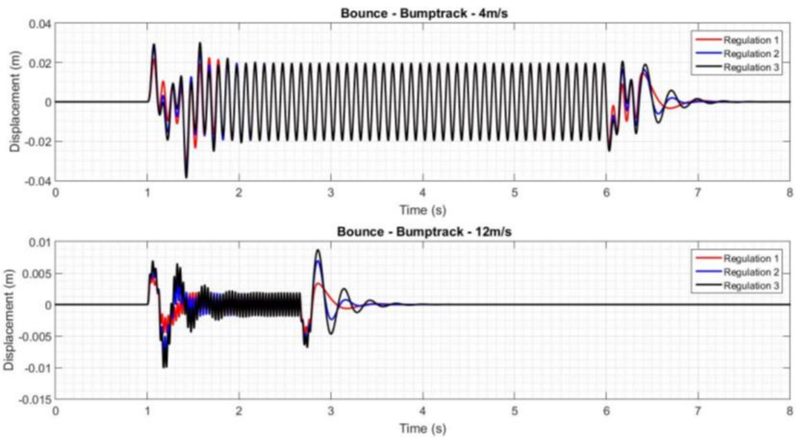

Figure 8 - Bounce in the *bumptrack* obstacle

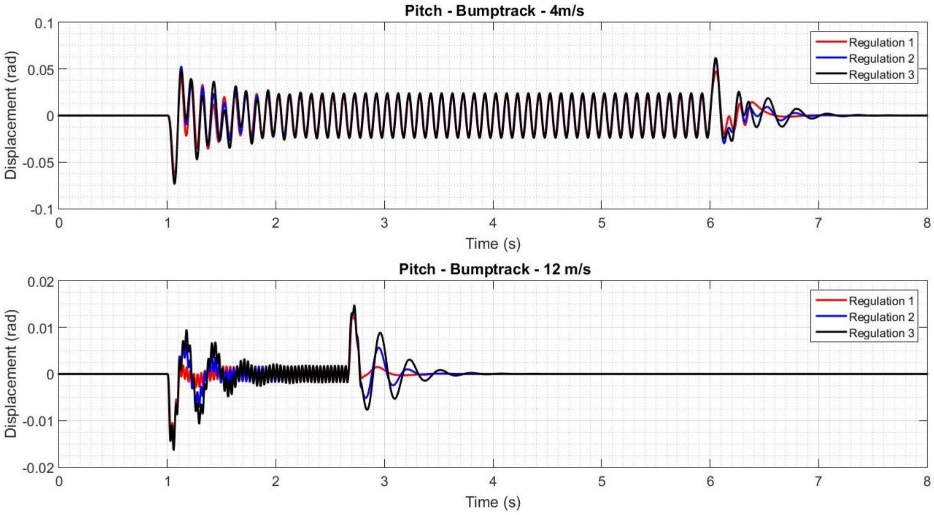

Figure 9 - Pitch motion on the *bumptrack* obstacle

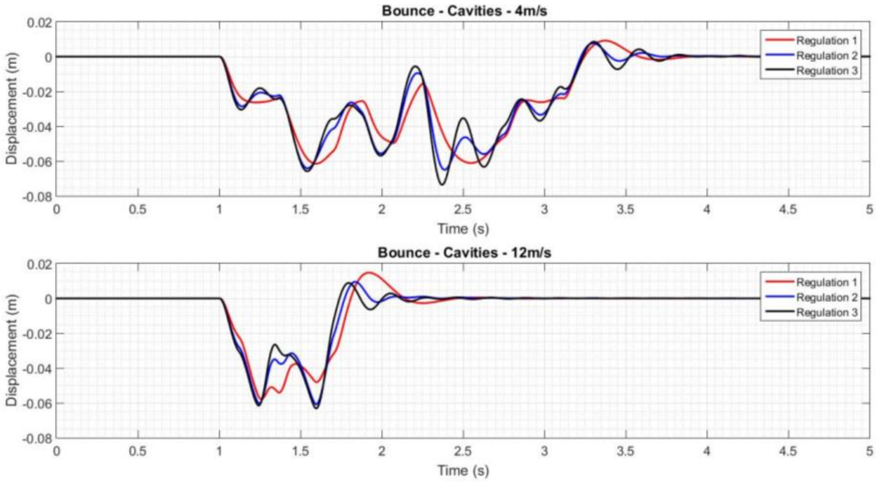

Figure 10 - Bounce on the cavities obstacle

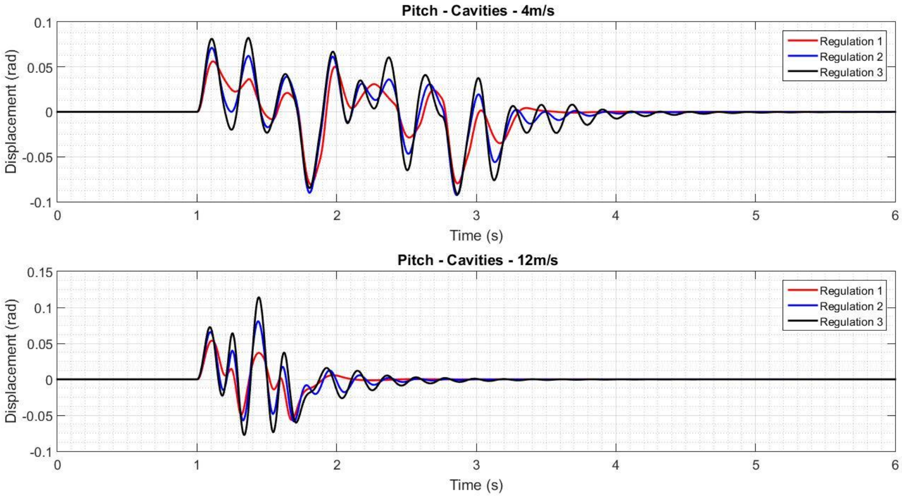

Figure 11 - Pitch in the cavities obstacle

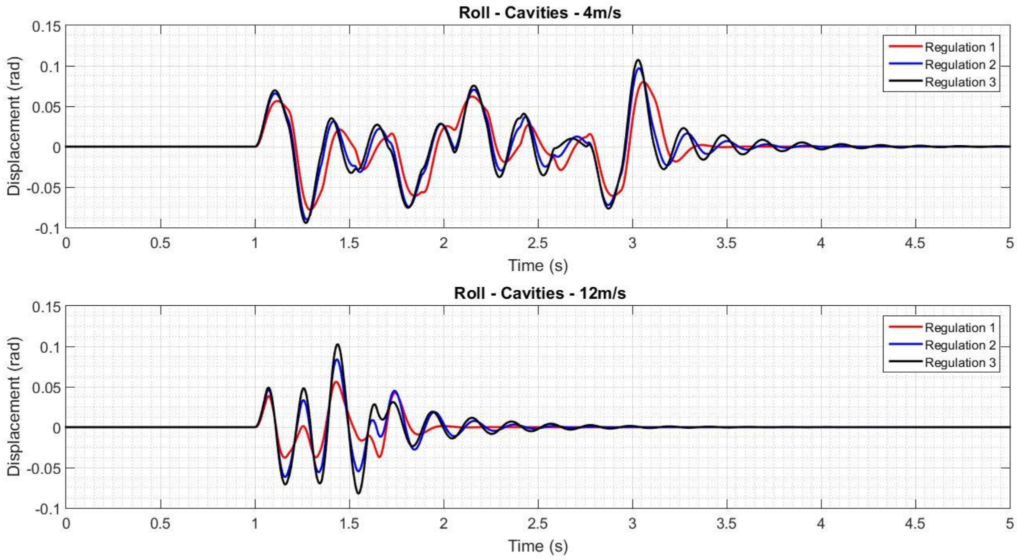

Figure 12 - Roll in the cavities obstacle

It can be observed that, the softer the regulation is, the less displacements it allows to the sprung mass. Besides that, the softest regulation takes a smaller amount of time for the system to return to the dynamic equilibrium, what demonstrates that the shock absorbers are not so soft that will transmit too many oscillations to the chassis of the vehicle. So far, the softest regulation seems to be more appropriate to the baja suspension. To confirm that, the data are processed along the ISO 2631-1 (1997) methodology, what gives the values on Tab. 2.

Table 2 - Final value of accelerations

Regulation	**Regulation 1**		**Regulation 2**		**Regulation 3**	
Vehicle Velocity	**4 m/s**	**12 m/s**	**4 m/s**	**12 m/s**	**4 m/s**	**12 m/s**
$a_{w,e}$ **(m/s²)**	0,749	0,701	0,994	0,886	0,995	0,993
a_v **(m/s²)**	0,772	0,714	1,005	0,891	1,011	0,922

As can be observed, the softest regulation indeed reduces the accelerations felt by the driver, resulting in a higher level of comfort during the operation of the vehicle. It can also be observed that with the higher velocity, the accelerations transmitted are smaller, except in the $a_{w,e}$ values of regulation 3. This indicates that with a smaller velocity, the sprung mass will be dislocated with the wheel movement, as long as in a higher velocity, the tires copy the ground more efficiently, transmitting a lower level of vibrations to the sprung mass.

Analyzing the values of Tab. 2 in relation to the health exposure, all of them are in the dashed zone of the graphic in Fig. 2, what will require an intervention for daily exposure, but without emergency. Still, there are two points to de discussed: several accelerations felt by the driver are not being considerate in this model, and the real full level of vibration could be dangerous for the driver, in a daily exposure. However, the use of a Baja SAE vehicle is a sporadic use, in the case of competitions and testing, what does not exceed three utilizations in a month, indicating that all the regulations tested can be used for the vehicle configuration.

For comfort issues, the analysis in through the Tab. 3, taken from ISO-2631(1997).

Table 3 - Accelerations values and its levels of comfort

Acceleration Range (m/s²)	**Situation**
Less than 0,315	Not uncomfortable
0,315 to 0,63	A little uncomfortable
0,5 – 1	Fairly uncomfortable
0,8 – 1,6	Uncomfortable
1,25 – 2,5	Very uncomfortable
Greater than 2	Extremely uncomfortable

Only the regulation 1 can be included in the fairly uncomfortable level, as long as all the other are classified as uncomfortable. However, the activity being performed, and the expectation of vibration of the driver, are very decisive in the comfort classification. As long as the standard values were determined in a public transportation, it is important to highlight that the Baja driver already expect a higher level of vibrations.

5. CONCLUSION

The suspension system of a Baja SAE vehicle was analyzed with three linear damping parameters, in order to search for the better regulation to be used on the shock absorbers. The system had its seven equations of motion determined, allowing to construct the structural matrices and insert them in the state space equations. Using the Runge-Kutta method, it was possible to retrieve the response of the vehicle in front of common obstacles in the Baja SAE competitions, at two different speeds.

Analyzing the retrieved data, it can be concluded that the softer regulation of the damping value of the shocks is not so small that will transmit too many oscillations to the driver, being able to reduce the damping of the fluid as long as it can, for vertical dynamics purposes. The lateral dynamics effects were not considered in this work.

Also, it can be concluded, among the possible regulations that the shock offers, the softer the damping value is, the lowest are the amplitudes of the movements felt by the driver. This is confirmed analyzing the values of *frequency weighted acceleration* and *total vibration value*, which show growing values of accelerations, that result in a reduction in the comfort level, as long as the damping value increases.

As for future works, some improvements are taking place. Firstly, the non-linearity of the shocks will be included in the simulation, since its damping values differs on high speed compression and low speed compression. Also, a validation using linear potentiometers along the shocks is being developed, in order to validate the data returned by the simulation.

6. ACKNOWLEDGEMENTS

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