DYNAMICAL ANALYSIS OF AN OFF-ROAD VEHICLE SUSPENSION

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Abstract. *The suspension behavior affects significantly the performance of a vehicle. An optimal condition must guarantee the best compromise among conflicting performance indices pertaining to the vehicle suspension system, that is, comfort, road holding and working space. Therefore, in the development phase of a vehicle design a complete study should be done to verify the actual improvement of optimal suspension characteristics and adjustments in the vehicle overall performance. This work presents a dynamic analysis of an off-road conventional automotive suspension, which was developed for an off-road vehicle type Mini-Baja. A four-degree of freedom dynamic model was developed to study the behavior of the suspension and the influence of the main parameters in the transmissibility of accelerations to the passengers. The model considers the coupling between the front and rear suspension system. Numerical simulations were developed to study the behavior of the suspension submitted to a sequential disturbance, just as it happens when a vehicle in movement comes across a road obstacle like a hole or an elevation. The results obtained with this methodology were analyzed in agreement with the procedures and limits supplied by the ISO 2631-1/97 standard, Evaluation of Human Exposure to Whole-Body Vibration, regarding the aspects related to the passengers health and comfort. The study developed indicates that this methodology can be used as an effective tool for the design and improvement of vehicle suspensions.*

Keywords. *Vehicle Suspension, Modeling, Numerical Analysis, Dynamical Analysis*

1. Introduction

The *Mini-Baja* vehicle is completely developed and built by undergraduate engineering students with the orientation of a professor board. During the development, the students are exposed to a real engineering problem involving several areas of knowledge. *CEFET/RJ* participates on the *SAE-Brasil* competition since 1997. In the competition these vehicles, which should respect technical and safety *SAE* standards, are submitted to several tests that expose it to severe conditions. *Mini-Baja* vehicles are highly competitive which demands an optimized project using advanced technologies. Figure (1) shows the *CEFET/RJ* vehicle that participated on the *VI Baja Cross São Carlos 2002 SAE-Brasil* event.

During the design process of the *Mini-Baja* structure it is necessary to quantify the maximum loads in the suspension and the accelerations and loads transmitted to the structure by the suspension. Usually, in the design of a vehicle, a static analysis is developed considering a static load that is equivalent to the maximum dynamic load. The equivalent static load is estimated using factors obtained in literature. These factors are generally quite conservative and they strongly depend on the suspension type. It is well know that the use of these factors can lead to a heavy vehicle. In that way, this work presents results from a project that is under development at *CEFET/RJ* that contemplates the use of experimental and numerical analysis to gain insight and improve an off-road vehicle, which is developed every year in *CEFET/RJ* to the *Mini-Baja / SAE-Brazil* competition.

(a)

(b)

Figure 1 - *Mini-Baja CEFET/RJ* vehicle (2002 *B12C* team). (*a*) Overall view and (*b*) front suspension detail.

This type of vehicle experiment heavy loads during the race that is performed in a very rough off-road track. One of the main functions of the suspension is to isolate the pilot from acceleration peaks promoted by the several obstacles present in the track. It is well know that human beings have a finite tolerance to mechanical vibrations. Depending of the magnitude and duration of the vibration the pilot can experience from discomfort to health damage. Therefore, in the development phase of a vehicle design a complete study should be done to establish an optimal suspension characteristics and adjustments in the vehicle so that an improved overall performance can be achieved.

In previous work, experimental and numerical analysis of both suspension loads and acceleration transmissibility were performed (Kenedi *et al.*, 2001; Pacheco *et al.*, 2002). This work is a natural development of the previous study and consists in a dynamical analysis of the transmissibility of the coupled front and rear suspension of the *Mini-Baja / SAE-Brazil* off-road vehicle developed at *CEFET/RJ* considering the passage of the vehicle through an idealized road obstacle. A simple four-degree of freedom model was developed to study the behavior of the suspension dynamic behavior and the influence of the main parameters in the suspension performance. A methodology to estimate optimal suspension adjustment was developed using this simple numerical model combined with comfort/health risk criteria furnished by standards. The results obtained with this methodology indicates that it can be used as an effective tool for the design and improvement for *Mini-Baja* vehicle suspension, as the designer can work with more realistic loads.

2. Comfort and Health Limits

The evaluation of the human being tolerance to mechanical vibrations is a complex problem comprising various areas of knowledge. Meanwhile, there are some relevant studies and standards that present important subsidies for the development and evaluation of equipments from the standpoint of the human being tolerance to mechanical vibrations. The evaluation of this tolerance requires knowledge of three aspects separately, although they are interrelated: the vibration perception threshold, the comfort levels promoted by this perception and the health damage promoted by the submission of human body to various types of vibrations.

To quantify a vibration motion, at least two parameters are necessary: the amplitude and the frequency of the motion. These two parameters act together on the human perception and tolerance. The ISO 2631-1/97 (ISO, 1997) establishes a parameter called *weighted mean-square acceleration* or simply *weighted acceleration* (a_w) to characterize the vibration. This parameter represents an averaged acceleration (translational or rotational) over a measurement time T and is defined as:

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

where t is the time and $a_w(t)$ is the acceleration as a function of time history.

For the threshold perception, the ISO 2631-1/97 standard establishes that people in residential buildings are likely to complain if the vibration magnitudes are only slightly above the perception threshold. The average experimental value established by this standard for the perception threshold for general applications is 0.015 m/s². Other specific studies, as the ones developed by Misael (2000), shows that this value depends strongly on the frequency. However, these subjects have little importance in the studies of the automotive applications once the levels of tolerance and comfort of persons in normal health comfort are significantly superior of the perception threshold. Indeed, the acceptable comfort levels depend on the people predisposition to the vibrations. While any perceptible vibration is intolerable to the occupants of a building, a ship crew sustains for months without complain much larger frequencies and amplitudes. Likewise, there are special situations where larger amplitudes with smaller frequencies could be extremely pleasurable, as in the case of a person on a swing. Meanwhile, it is intuitive that there are limits for the average amplitude of a vibration in that larger values could be considered tolerable but not comfortable. The ISO 2631-1/97 furnishes some absolute reference values for these limits. Table (1) presents some values for the public transports area:

Table 1 - Approximated indications of likely reactions to various magnitudes of overall vibration total values in public transports (ISO, 1997).

Vibration Level (m/s²)	**Comfort Level**
≤ 0.315	Not uncomfortable
0.32 - 0.63	A little uncomfortable
0.50 - 1.00	Fairly uncomfortable
0.80 - 1.60	Uncomfortable
1.25 - 2.50	Very uncomfortable
2.00 ≥	Extremely uncomfortable

Although the ISO 2631-1/97 standard points for the difficulty to establish general correlations between the vibrations characteristics and the comfort levels, the earlier edition of ISO 2631 standard (ISO, 1985) was based mainly in studies on aircraft pilots and drivers and furnishes more general suggestions for the limits. Other works developed within the same approach presents similar results. Figure (2) presents data from different studies, including the values suggested by the ISO 2631-1/85 standard for periods of one hour and one minute (Neto, 1990). The curves correlate the average acceleration values, his frequencies and the acceptable comfort levels, represented by the region below the curves.

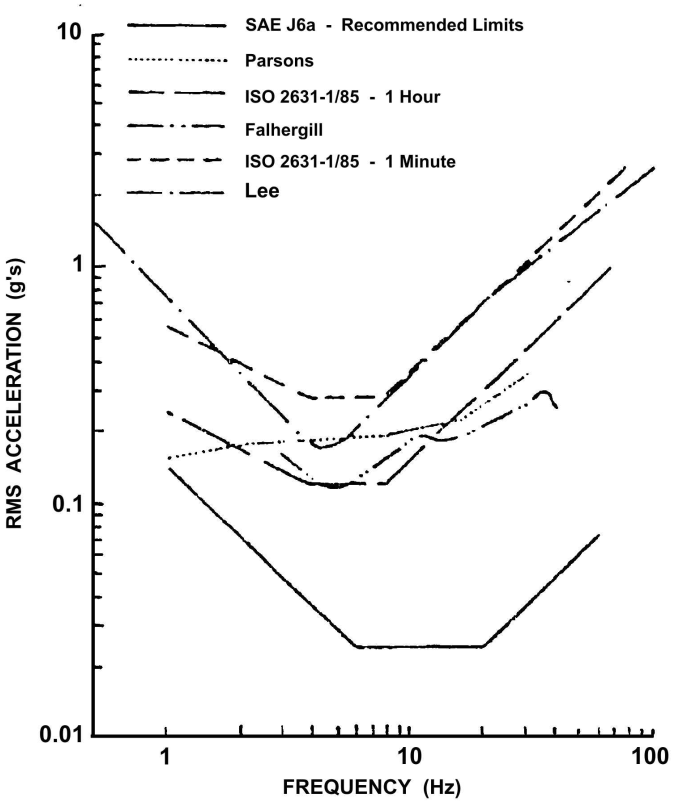

Figure 2 - Comfort curves x frequency x intensity (Neto, 1990).

It is important to note that all the experiments shown in Fig. (2) present minimum values in the range of 4 Hz to 8 Hz. Some important observations need to be made about the eventual health damage promoted by the submission of the human body to vibrations with diverse intensity and duration. At first, it is important to note that as the majority of mechanical materials the human body is also subjected to fatigue. Meanwhile, the human body has the capability of regeneration when it is allowed to recovery (rest) during a considerable period of time. Therefore, it is difficult to establish absolute limits for the damage promoted by the exposition to vibrations. Some research studies points for the existence of dangerous permanent damages even after the execution of complete regeneration cycles. Sandover (1998), for example, points to the permanent injuries in the bones submitted to a vibration fatigue process, which are reconstituted by the formation of micro callus that are less permeable than the original bone constitution and so may reduces the nutrient supply causing an eventual degeneration. Sandover (1998) uses available data set from a variety of heavy vehicles in practical situations for predictions of spinal stress and fatigue life and concludes that long term weightings approaches, as the BS 6841 (BS, 1987) and ISO 2631-1/85 (ISO, 1985), have shortcomings when peak values are important and probably under-estimate the risk of injury.

In accordance with ISO 2631-1/85, the curves that represent the safety limits for the human health associated to the effects of vibrations have a format similar to the comfort curves. Therefore comfort curves, like the one in the Fig. (2), can be used as reference for the health damage limits by applying a multiplying factor $k \cong 2.0$. However, in spite of the critics accomplished by Sandover (1998) regarding the first edition of the ISO 2631-1/85, the second edition, ISO 2631-1/97, includes news specifics criterions for peak values.

For continue vibrations, the ISO 2631-1/97 establishes that the health limits are described by a specific graphic with operation regions disposed below limiting curves, which is presented in Fig. (3).

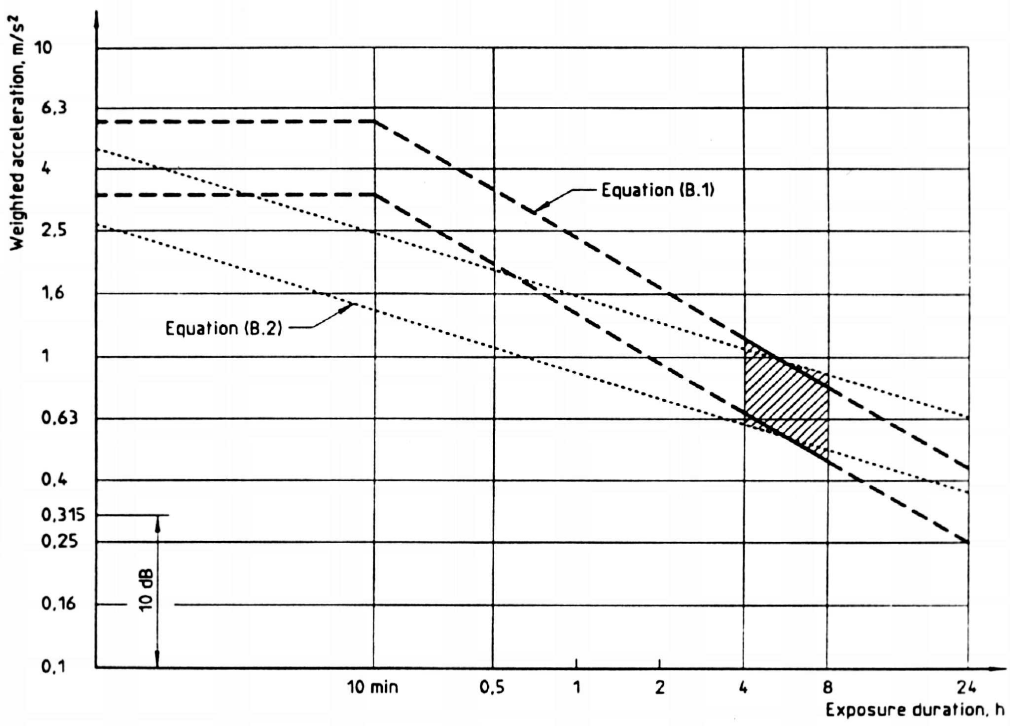

Figure 3 - Health guidance caution zones (ISO, 1997).

The curves from Fig. (3) were developed from studies mainly focused on exposition in the range of 4 h to 8 h, which corresponds to working periods of regular professional activities. These curves are based in the hypothesis that the responses are related to energy and then, two different daily vibration exposures, a_{w1} and a_{w2}, are equivalent when (curve B.1):

$$a_{w1}.T_1^{1/2} = a_{w2}.T_2^{1/2} \tag{2}$$

Other equivalent studies pointed for the following relationship (curve B.2):

$$a_{w1}.T_1^{1/4} = a_{w2}.T_2^{1/4} \tag{3}$$

ISO 2631-1/97 standard points that for exposures below the region between the two parallels curves (B.1), health effects have not been clearly documented or objectively observed; in the region between the two curves, caution with respect to the potential health risks is indicated and above this region the health risk are likely to occur.

When the vibration exposure consists of n periods of exposure involving different magnitudes and durations, the energy-equivalent vibration magnitude, a_w^e, corresponding to the total duration of exposure, can be evaluated by using a weighted average as the following ISO 2631-1/97 formula suggestion:

$$a_w^e = \left[\frac{\sum_{i=1}^{n} a_{wi}^2 \; T_i}{\sum_{i=1}^{n} T_i} \right]^{1/2} \tag{4}$$

where a_{wi} is the weighted acceleration vibration magnitude for exposure duration T_i.

Both methods produces reliable references that are, nevertheless, very conservatives because of the concern with the danger of a generalization, once there are many effects and circumstances of difficult evaluation like long term damage, individual health and physical condition and occurrence of simultaneous accelerations em various directions.

There is also the problem of high intensity peaks. The standard ISO 2631-1/97 considers a parameter called *crest factor*, defined by the ratio between the peak amplitude and the root-mean-square value of the incident vibration. This parameter only points the presence of extreme values. Two criteria are used to quantify the vibration peak intensity: the *running root-mean-square* method and the *fourth power vibration dose* method. The fourth power vibration dose method is pointed by the standard as the more sensitive to the quantification of large intensity peaks, and a *VDV* (vibration dose value) value is defined as:

$$VDV = \left\{ \int_0^T [a_w(t)]^4 \, dt \right\}^{1/4} \tag{5}$$

When the vibration exposure consists of n periods, of different magnitudes, the vibration dose value for the total exposure should be calculated from the fourth root of the sum of the fourth power of individual vibration dose values:

$$VDV_{total} = \left[\sum_{i=1}^{n} VDV_i \right]^{1/4} \tag{6}$$

ISO 2631-1/97 standard establishes that the vibration dose values corresponding to the lower and upper bounds of the region given by Eq. (2) and its equivalent curve (B.1) in the Fig. (3) are:

$$8.5 \le VDV \text{ or } eVDV \le 17 \tag{7}$$

where $eVDV$ is an estimated vibration dose given by the formula:

$$eVDV = 1.4 \, a_w \, T^{1/4} \tag{8}$$

and a_w and T are the same parameter of VDV calculation. According the same standard, the experience suggest that the use of the additional evaluation method, like the VDV method, will be important for the judgment of the effects of vibration on human beings when the following ratio is exceed for evaluation health or comfort:

$$\frac{VDV}{a_w \, T^{1/4}} = 1.75 \tag{9}$$

3. General Aspects of a Suspension Project

Nowadays there are a great number of vehicle suspension models with geometries and various operation principles. In spite of this diversity, the models are essentially based on the operation characteristics of the group spring-shock absorber in parallel, as pointed by Paula (2002). While different type of devices can be used for the spring, as helical springs, sprain bars, pneumatic cylinders, or even the combination of these, for the shock absorber the use of the hydraulic cylinders prevails.

Due to the similar dimensions for vehicles of the same category (ex: weight, length, distance of the bottom to the ground, etc) there is a small range to adjust the suspension operation parameters (ex: the spring rate compression). Neto (1990) observes that the main limitation to a wider adjustment of a suspension is the workspace (essentially given by the course of the suspension). Em general, as smaller is the natural frequency of a suspension, smaller will be the transmissibility, but a great workspace is needed. Neto (1990) also informs that for American passengers cars the values for natural frequency are in the range of 1 Hz to 1.5 Hz, while for sportive vehicles they are in the range of 2.0 Hz to 2.5 Hz.

Another important aspect is that, in spite of the relatively small range of adjustment that can be made to a suspension, it is possible to promote some proportional tunings to improve its operation. In this respect, Mola (1969) describe the *Flat Ride Turning* feature, which consists in optimizing a suspension submitted to a sequential disturbance, just as it happens when a vehicle in movement comes across a road obstacle like a hole or an elevation. Mola (1969) observes that in spite of the small parameters variation range, it is possible to adjust these parameters relatively to produce significantly reductions on the total vibration intensity (for example, making the front spring stiffness value equal to 80% of the rear spring stiffness value).

4. Four-Degree of Freedom Model

Figure (4) presents a simple four-degree of freedom model that was developed to study the dynamic behavior of a two coupled (rear and front) vehicle suspension (Meirovitch, 1975). The full suspension was modeled considering a system with a lumped mass element (m) equivalent to half the total vehicle mass.

The damping coefficient of the tires was set equal to zero, because in generally it is very small (Mola, 1969). Also the *suspension spring rate* (K_s - spring stiffness) can be replaced by a *ride spring rate* (K) as a combination of the *suspension* and *tire spring rate* (K_T) in series, that is (Mola, 1969):

$$K = \frac{K_S . K_T}{K_S + K_T} \tag{10}$$

where K is the equivalent ride spring rate of the front and rear suspension, K_T and K_S are the spring rate of the respective tire and suspension spring.

Spring and damper elements were used to represent the continuous connection between the ground and the vehicle frame ($K_{i,}$ and c_i , where c_i is the coefficient of viscous damping and K_i is the stiffness of the vehicle shock absorber system – i = 1 for rear suspension and i = 2 for front suspension) of the four-degree of freedom model. The vertical displacement of the frame mass center (*MC*) is u as shown in Fig. (4).

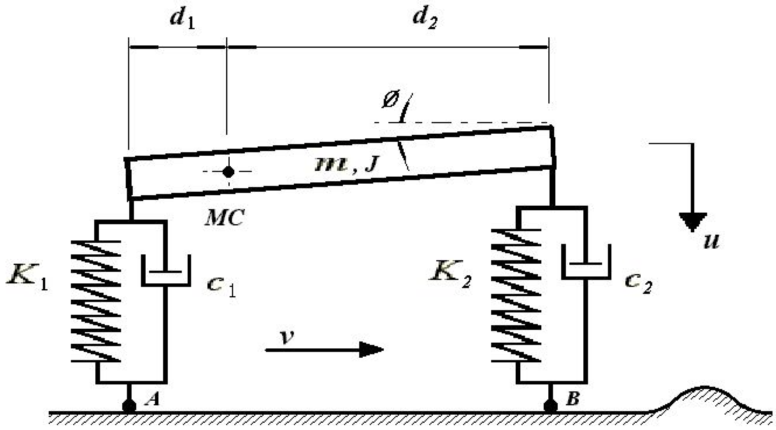

Figure 4 - A four-degree of freedom model for the full suspension.

It was considered that the vehicle is running on a leveled road, with a horizontal speed (v) and then is subjected to an idealized temporary sinusoidal external disturbance promoted by the road. This disturbance is experimented first by the front suspension and after a time delay by the rear suspension.

The adopted initial conditions consider the vehicle in static equilibrium with a constant horizontal speed v. The initial time of the analysis, $t = 0$, corresponds to the exactly instant when the front tire (represented by the point B in Fig. 4) reaches the edge of the road sinusoidal disturbance. Of course, besides the expected vertical displacement, this four-degree of freedom model has also the possibility to rotate around its mass center through an angular variation (Ø). Therefore, this model is also under the influence of the value of the vehicle mass moment of inertia (J).

Starting from the balance of force and momentum equations of each suspension element, it is possible to construct the equations of the movement:

$$\ddot{u} = (1/m)\left[-\dot{u}(c_2 + c_1) - u(K_2 + K_1) + \cos\theta\,(d_2 c_2 - d_1 c_1) + sen\theta\,(d_2 K_2 - d_1 K_1) + mg - h_1(t)\right] \tag{11}$$

$$\ddot{\theta} = (1/J)\left[\dot{u}(d_2 c_2 - d_1 c_1) + u(d_2 K_2 - d_1 K_1) - \cos\theta\,(d_2^2 c_2 - d_1^2 c_1) - sen\theta\,(d_2 K_2^2 - d_1 K_1^2) + h_2(t)\right] \tag{12}$$

where:

$$h_1(t) = [c_2 \dot{f}_2(t) + K_2 f_2(t)] + [c_1 \dot{f}_1(t) + K_1 f_1(t)] \quad ; \quad h_2(t) = d_2[c_2 \dot{f}_2(t) + K_2 f_2(t)] - d_1[c_1 \dot{f}_1(t) + K_1 f_1(t)] \tag{13}$$

Function (f_2) represents the idealized sinusoidal disturbance (obstacle) with amplitude A and a period T_{ob}. The function (f_1) has the same format, however it presents a time delay w, due to the distance between the wheels axes. Obviously, the period of the sine function (T_{ob}) and the delay of the impact (w) depend both of the vehicle speed. The disturbance functions are described as following:

$$f_2(t) = \begin{cases} (A/2)[1 + \sin(\pi t / T_{ob} - \pi/2)] \quad ; \quad 0 \le t \le 2T_{ob} \\ 0 \quad ; \quad T_{ob} < t \end{cases} \tag{14}$$

$$f_1(t) = \begin{cases} (A/2)[1 + \sin(\pi t / T_{ob} - \pi w / T_{ob} - \pi/2)] \quad ; \quad w \le t \le 2T_{ob} + w \\ 0 \quad ; \quad 0 \le t < w \quad and \quad T_{ob} + w < t \end{cases} \tag{15}$$

In the same way, the sinusoidal period $T_{ob} = 2L/v$ and the delay time $w = (d_1 + d_2)/v$, where L is the length of the obstacle and $(d_1 + d_2)$ is the distance between the wheels axes.

Numerical simulations were performed employing a fourth order Runge-Kutta method for numerical integration (Nakamura, 1993). A convergence study was developed to chose the time step.

4. Numerical Simulations

Numerical simulations were developed to study the dynamical behavior of the four-degree of freedom model subjected to an idealized obstacle represented by a sinusoidal disturbance. The parameters used in the analysis are the following: m = 115 kg , J = 38.8 kg.m^2 , $c_1 = c_2 =$ 300 N·s/m , K_T = 37 kN/m, K_S = 55 kN/m, v = 10.0 m/s (36 km/h), d_1 = 0.422 m, d_2 = 0.908 m, L = 0.655 m, A = 0.095 m . The stiffness was measured through a compression test and the coefficient of viscous damping was estimated from the experimental dynamic data.

Figure (5) shows the results from the numerical simulations, considering all parameters described above, presenting the acceleration and vertical displacement of the mass center (*MC*) as the vehicle pass the obstacle.

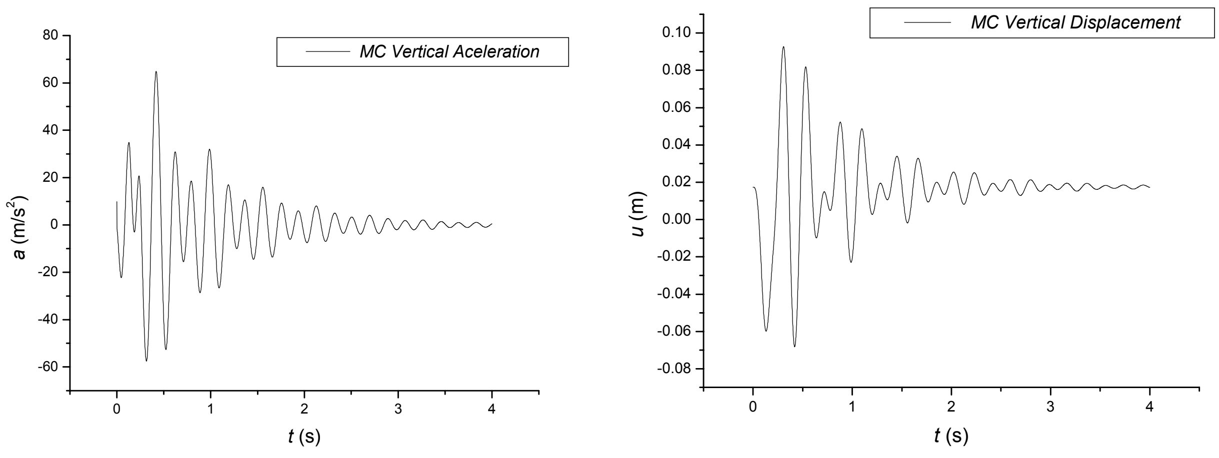

Figure 5. Numerical results. (*a*) Acceleration and (*b*) displacement.

The result presented in Fig.(5*a*) shows peaks of acceleration in the vertical direction of the order of 60 m/s^2 ($\cong 6g$), suggesting the necessity to evaluate the health effects.

The application of Eqs. (4) and (6) during the total simulation time, T = 4.0 s, revels a *weighted acceleration* of 19.15 m/s² and a *vibration dose value* of 32.4 m/s$^{1.75}$. Eq. 7 and Fig. (3) shows both that values overcome the daily limits prescribed by the ISO 2631-1/97. It is important to note that the *Mini-Baja* pilot will not be submitted to this acceleration doses daily, as race events occurs two or of three times a year. Therefore, the following analysis uses this data to develop a qualitative study to optimize the suspension performance focusing the pilot comfort. In order to analyze the suspension performance considering the eventual influences in the passengers comfort limits, the ride spring rates values for both suspensions were studied through an acceptable range, from 10 kN/m to 90 kN/m, keeping fixed all other values. Figures (6) and (7) presents the *VDV* and a_w values for this range.

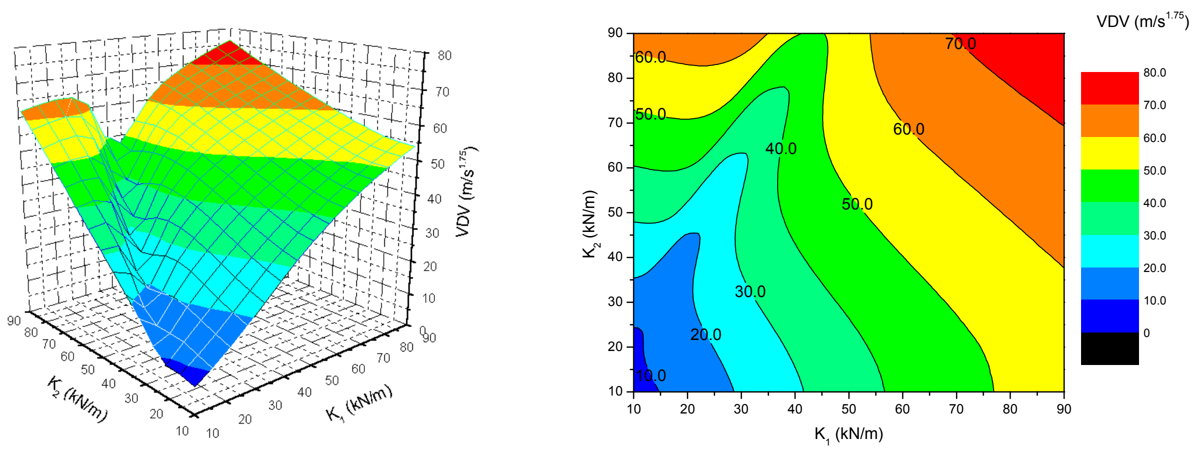

Figure 6. *Vibration dose value*, *VDV*, as a function of K_1 and K_2.

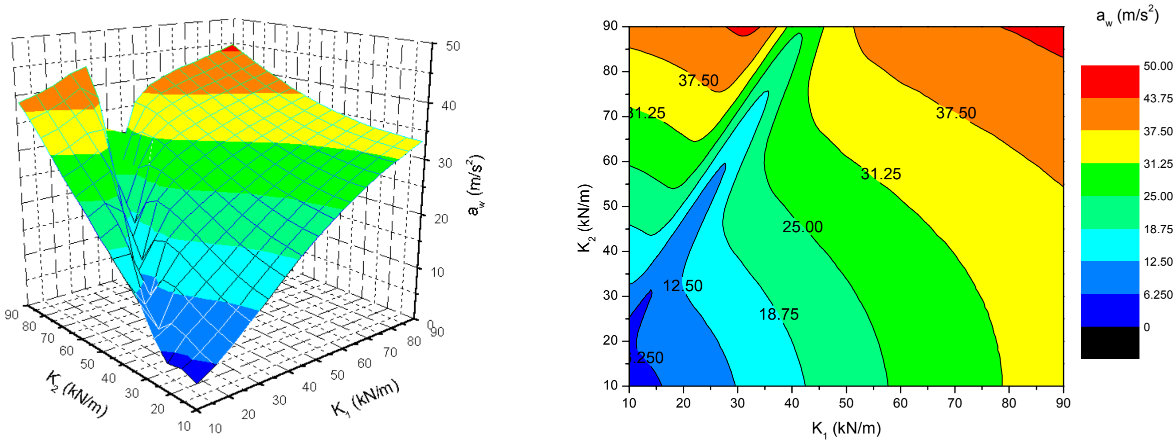

Figure 7. *Weighted acceleration* , $a_{w,}$ as a function of K_1 and K_2.

The results presented above show that the values of *VDV* and a_w tends to decrease concomitantly with the reduction of the ride spring rates values. Low suspension ride spring rates needs considerable workspace and can compromise the vehicle handling. However, intermediary minimums regions are observed, as pointed by Mola (1969), and these regions can be used to achieve a suspension condition/adjustment that results in a good overall suspension performance with a reasonable workspace. Results also show that no combination of ride spring rates values is able to produce an acceptable level of vibration in agreement with the daily limits suggested by the standard ISO 2631-1/97. In spite of the fact that the *Mini-Baja* pilot is not submitted to these doses daily, but few times a year, the results indicate that complementary studies should be developed to evaluate the health risks to the *Mini-Baja SAE* pilot. Moreover, the proposed methodology can be use to improve the suspension system in order to maximize the pilot comfort as it strongly contributes to the pilot performance during the race.

5. Conclusions

The methodology proposed in this work uses a simple numerical model combined with comfort/health risks standards criteria to study the suspension system dynamical performance and the influence of the suspension main parameters in the pilot comfort/health risks considering the accelerations present during a *Mini-Baja* race. An estimate for an optimal suspension adjustment was obtained with this simple model. The results obtained with this methodology suggest that it can be used as an effective tool for the design and improvement for *Mini-Baja* vehicle. Signals obtained from experimental measurements can be used as input signal for the numerical model in order to compute real *VDV* (*vibration dose value*) and a_w (*weighted acceleration*) values. Such experimental program is now under development by the authors.

6. Acknowledgement

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