A SIMPLE MODEL TO STUDY COMFORT AND ROAD-HOLDING ASPECTS OF AN OFF-ROAD VEHICLE

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Abstract. *A central problem faced in the development phase of an off-road vehicle is the selection and tuning of the suspension elements. A correct selection of the suspension system adjustment is fundamental to achieve an optimum ride and road-holding, and also to preserve a passenger health and comfort minimal conditions. 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 is developed to study the behavior of the suspension and the influence of the main parameters. The model considers the coupling between the front and rear suspension system and also the influence of the tires unsprung mass and flexibility. Numerical simulations are 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 model response is compared with experimental results obtained from accelerometers and load-cells in a simple test. 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

A central problem faced in the development phase of an off-road vehicle is the selection and tuning of the suspension elements considering the chassis, tires, road and driver characteristics. A correct selection of the suspension system adjustment is fundamental to achieve an optimum ride and road-holding, and also to preserve a passenger health and comfort minimal conditions. However, one general conclusion is that spring-damper adjustments for optimum ride and for road-holding are quite different.

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 is developed to study the behavior of the suspension and the influence of the main parameters, as well as to study the possibility of an optimum adjustment point considering the ride, the road-holding, and the passenger comfort and health. The model considers the coupling between the front and rear suspension system and also the influence of the tires unsprung mass and flexibility. Numerical simulations are 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 are analyzed in agreement with the procedures and limits supplied by the ISO 2631-1/97 standard (ISO, 1997), *Evaluation of Human Exposure to Whole-Body Vibration*, regarding the aspects related to the passengers health and comfort. The model response is compared with experimental results obtained from accelerometers and load-cells in a simple test. The study developed indicates that this methodology can be used as an effective tool for the design and improvement of vehicle suspensions.

Mini-Baja vehicle competition is an initiative coordinated by the worldwide *Society of Automotive Engineers* (SAE), and in the national level accomplished by *SAE BRASIL*. The objective of the competition is that each participant team build a prototype of a recreational vehicle, with characteristics predominantly of a mono-place "off-road" type of significant robustness. The project must also have an orientation for its eventual commercialization to an enthusiastic public and for no professional purpose. The teams should be composed by a maximum of 20 students of Engineering for various fields (Mechanics, Robotics, Metallurgy, Electronics, Automotive, Production, Industrial Automation, Aeronautics and Materials) and a Guiding Teacher.

Figure 1. *Mini-Baja* vehicle front suspension detail. (2002 "*B12C*" CEFET/RJ team)

2. COMFORT AND HEALTH LIMITS

In previous works (Buarque *et al.*, 2003*a*, 2003*b*; Pacheco *et al.*, 2001) the authors have addressed the subject of comfort/health risk criteria based in standards applied to a *Mini-Baja* vehicle. In this works the *weighted mean-square acceleration* or simply *weighted acceleration* (a_w) was used as a standard parameter to characterize vibration levels, according the ISO 2631-1/97 (ISO, 1997). 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.

ISO 2631-1/97 (ISO, 1997) establishes that for general applications the human threshold perception occurs around $a_w = 0.015$ m/s². However this value has little importance in the studies of the automotive applications, once the levels of tolerance and comfort in normal health are significantly superior of the perception threshold. This standard also observes that it is very important to consider high intensity peaks, when a_w does not produce an efficient limit reference. This standard suggests the *fourth power Vibration Dose Value method* (VDV) as more sensitive parameter to the quantification of large intensity peaks, that is defined as:

$$VDV = \left\{ \int_0^T [a_w(t)]^4 \, dt \right\}^{1/4} \quad (2)$$

whose units are approximately the same of the ones of acceleration (in the SI system is m/s$^{1.75}$).

3. GENERAL ASPECTS OF A SUSPENSION PROJECT

Due to the similar dimensions and characteristics for vehicles of the same category (*e.g.*, weight, length, distance of the bottom to the ground, etc), there is a small range for adjustments of the suspension operation parameters (*e.g.*, spring rate compression). In this respect, Mola (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 observes that, in spite of the small range for parameter variation, it is possible to adjust these parameters relatively to produce significantly reductions on the total vibration intensity (*e.g.*, making the front spring stiffness value equal to 80% of the rear spring stiffness value).

According to Milliken (Milliken, 1995), “damper rates for optimum ride and for road-holding are quite different, the latter being much harder”. The author observes that, in generally, for an optimum road-holding it is necessary to keep the tire in the maximum average pressure over the road to maintain a good adherence with the road. Therefore, an optimal suspension adjustment requires that both comfort and road-holding aspects be considered simultaneous in the analysis.

The analysis of the maximum adherence is not a simple subject involving non-linear aspects as the transition between static and dynamics friction forces, and contact phenomena between the tire and the road.

A simplified analysis of the adherence can be developed by studying the influence of suspension parameters in the contact forces between the tire and the road. The adherence can be improved by the maximization of these the contact forces. Supposing a vehicle in a continuous curve whose trajectory and speed drive produce centripetal force and radial friction force between the tire and the road. If there is a decrease in the contact force, produced for example by an obstacle, the vehicle can skid.

In this work, comfort is addressed considering the *Vibration Dose Value* (VDV) and the road-holding considering a variable γ denominated *Average Acceleration*, that is defined by:

$$\gamma = \int_0^T \frac{F_c}{m} dt \quad (3)$$

where F_c is the contact force and m the weight of the vehicle.

4. TWO-DEGREE OF FREEDOM MODEL

In previous works (Buarque *et al.*, 2003*a*, 2003*b*) a two-degree of freedom model (2DOF) was presented to study the dynamic behavior of a two coupled (rear and front) vehicle suspension, according Meirovitch (Meirovitch, 1975).

The full suspension is modeled considering a system with an equivalent mass (m), equal to half the total vehicle mass, uniformed distributed across a bar with a length equal to the shafts distance.

Spring and damper elements are used to represent the continuous connection between the ground and the vehicle frame of the two-degree of freedom model (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, where i = 1 for rear suspension and i = 2 for front suspension). The vertical displacement of the frame mass center (MC) is u as shown in Fig. 2.

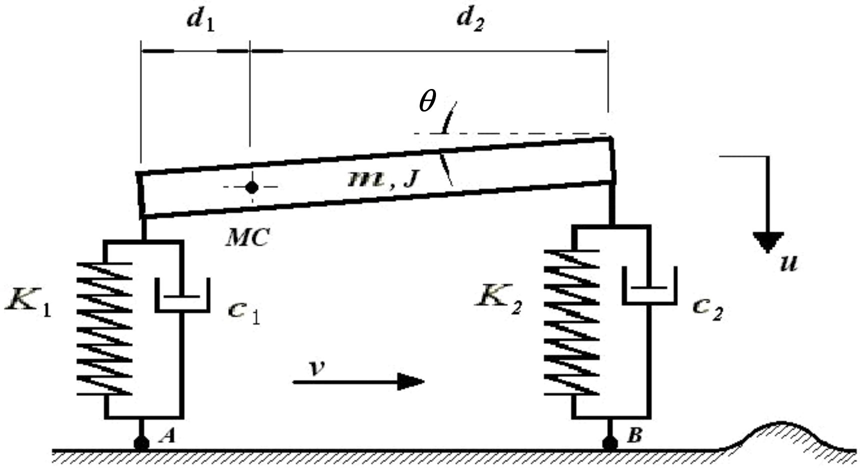


Figure 2. A two-degree of freedom model (2DOF) for the full suspension.

It is 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). Of course, besides the expected vertical displacement, this two-degree of freedom model has also the possibility to rotate around its mass center through an angular variation (θ). Therefore, this model is also under the influence of the value of the vehicle mass moment of inertia (J).

The *resultant suspension spring rate* (K) is calculated as a combination of the *suspension spring rate* (K_s) and *tire spring rate* (K_T), in series, that is:

$$K = \frac{K_S . K_T}{K_S + K_T} \tag{4}$$

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

$$\ddot{u} = (1/m)\left[-\dot{u}(c_2 + c_1) - u(K_2 + K_1) + \dot{\theta}\cos\theta(d_2c_2 - d_1c_1) + \mathrm{sen}\theta(d_2K_2 - d_1K_1) + mg - c_2\dot{f}_2(t) + K_2 f_2(t) + c_1\dot{f}_1(t) + K_1 f_1(t)\right] \tag{5}$$

$$\ddot{\theta} = (1/J)\left[\dot{u}(d_2c_2 - d_1c_1) + u(d_2K_2 - d_1K_1) - \dot{\theta}\cos\theta(d_2^2c_2 - d_1^2c_1) - \mathrm{sen}\theta(d_2K_2^2 - d_1K_1^2) + d_2[c_2\dot{f}_2(t) + K_2 f_2(t)] - d_1[c_1\dot{f}_1(t) + K_1 f_1(t)]\right] \tag{6}$$

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 shafts. 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:

$$f_2(t) = \begin{cases} (A/2)[1 + \sin(\pi t / T_{ob} - \pi/2)] \quad ; \quad 0 \le t \le 2T_{ob} \\ 0 \quad ; \quad T_{ob} < t \end{cases} \tag{7}$$

$$f_1(t) = \begin{cases} (A/2)[1 + \sin(\pi t / T_{ob} - \pi w / T_{ob} - \pi/2)] \quad ; \quad w \le t \le 2T_{ob} + w \\ 0 \quad ; \quad 0 \le t < w \quad and \quad T_{ob} + w < t \end{cases} \tag{8}$$

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 shafts.

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

5. FOUR-DEGREE OF FREEDOM MODEL

Figure 3 presents a four-degree of freedom model (4DOF) developed to study the dynamic behavior of a two coupled (rear and front) vehicle suspension. The suspension is modeled considering a system with an equivalent mass (m), equal to half the total vehicle mass. The unsprung mass (M_u) is equal to the full assembled car wheel (tire + frame). The damping coefficient of the both tires (front and rear) is not considered, and $c_T = 0$. Some authors (*e.g.*, Mola, 1969) points that the damping of the tires is small and is usually not considered in the analysis.

Different from the two-degree of freedom model presented in Fig. 4, in this model the stiffness of the tire (K_T) is not direct connected with the spring of the vehicle shock absorber system (K_i). Between them is the unsprung mass (M_u), as shown in Fig. 3.

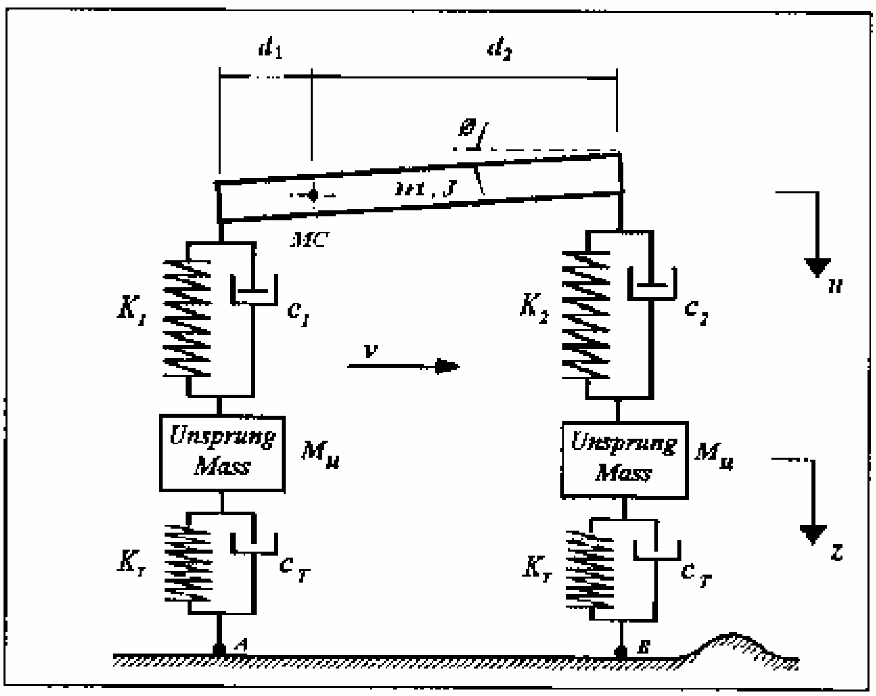


Figure 3. A four-degree of freedom model (4DOF) for the full suspension.

In the presented analysis, the front and rear tire stiffness has the same value. All other conditions and parameters values are considered equal to the ones used for the two-degree of freedom model, including the disturbance function.

For the sprung mass (m) the equations of motion are the following:

$$\ddot{u} = (1/m)\left[-\dot{u}(c_2 + c_1) - u(K_2 + K_1) + \dot{\theta}\cos\theta(d_2c_2 - d_1c_1) + \operatorname{sen}\theta(d_2K_2 - d_1K_1) + mg - (c_2\dot{z}_2 + K_2z_2) - (c_1\dot{z}_1 + K_1z_1) \right] \tag{9}$$

$$\ddot{\theta} = (1/J)\left[\dot{u}(d_2c_2 - d_1c_1) + u(d_2K_2 - d_1K_1) - \dot{\theta}\cos\theta(d_2^2c_2 - d_1^2c_1) - \operatorname{sen}\theta(d_2K_2^2 - d_1K_1^2) + d_2(c_2\dot{z}_2 + K_2z_2) - d_1(c_1\dot{z}_1 + K_1z_1) \right] \tag{10}$$

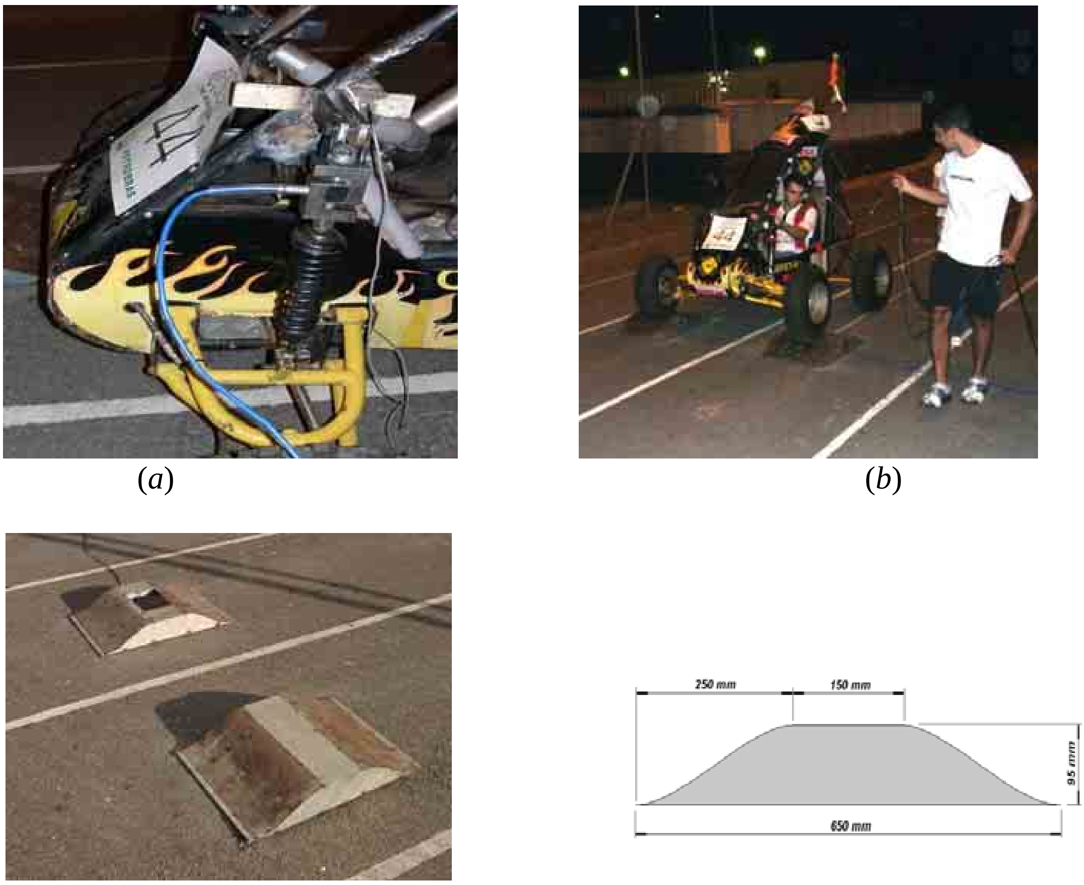
For the unsprung mass (M_u), to front and rear wheels, respectively, the equations of motion are the following:

$$M_u \ddot{z}_1 = M_u g - K_T[z_1 - f_1(t)] + K_1[(u_1 - d_1 sin\theta_1) - z_1] \tag{11}$$

$$M_u \ddot{z}_2 = M_u g - K_T[z_2 - f_2(t)] + K_2[(u_2 - d_2 sin\theta_2) - z_2] \tag{12}$$

6. EXPERIMENTAL ANALYSIS

In a previous work (Buarque *et al.*, 2003*a*) developed by the authors, experimental tests were developed to verify the proposed two-degree of freedom model behavior. Two accelerometers (strain gage type) bonded to the ends of the left frontal spring-damper are used to measure accelerations developed during the test. One load-cell (strain gage type) is fixed to the upper end of the spring-damper in order to measure the load transmitted to the structure at that point. Other load-cell of the same type was placed at the sinusoidal obstacle to measure the tire force reaction and the average speed. The signals from accelerometers and load-cell are processed by a data conditioned system. More details can be obtained in the reference Buarque *et al.*, 2003*a*. Figure 4 presents details of the experiment.



(*a*) (*b*)

(*c*) (*d*)

Figure 4. Experimental test. Front suspension accelerometers (*a*), test (*b*), obstacle (*c*) and obstacle cross section (*d*).

7. NUMERICAL SIMULATIONS

Numerical simulations considering the two models (two and four degree of freedom) are developed adopting parameters to reproduce the conditions associated with the experiment described in the previous section. Therefore, the following parameters are used in the numerical simulations: m = 130 kg, J = 48 kg.m^2, c_1 = 92 N·s/m, c_2 = 138 N·s/m, K_1 = 0.53 kN/m, K_2 = 0.72 kN/m, K_T = 5 kN/m, v = 2.0 m/s (7.2 km/h), d_1 = 0.470 m, d_2 = 0.851 m, L = 0.655 m, A = 0.095 m, M_u= 6.0 kg.

In the Fig. 5 it is possible to observe that the response of both models present a good agreement with the experimental result, indicating that both models can be used as an effective tool for the design and improvement of vehicle suspensions. The oscillations observed at the end (between 4 s and 5 s) are due to the lack of damping in the tires.

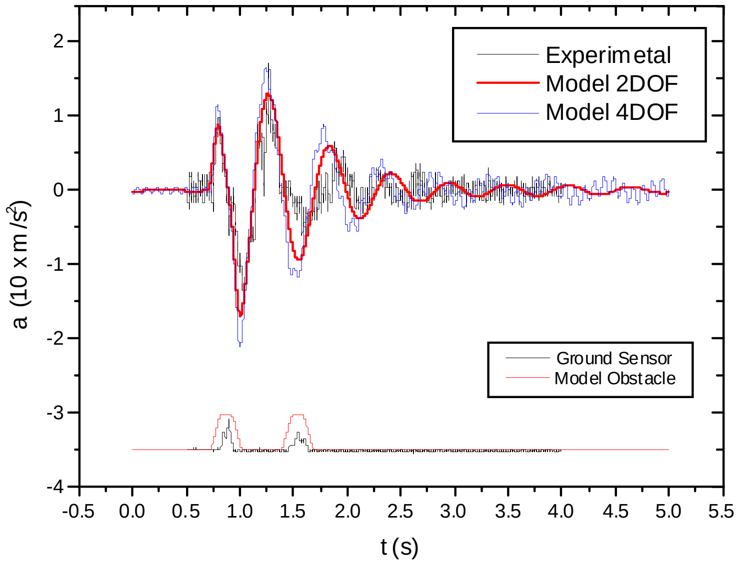
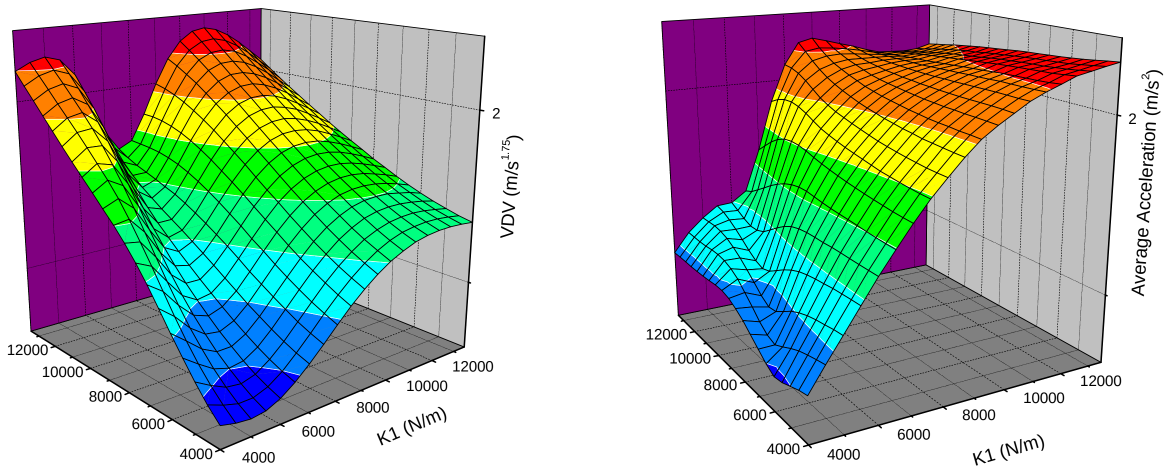


Figure 5. Results from the two models (2DOF and 4DOF) and the experiment.

To establish an optimum parameter configuration, the ride spring rates values for both suspensions were studied through an acceptable range, from 4 kN/m to 12 kN/m, keeping fixed all other values. The four-degree of freedom model (4DOF) is considered in the analysis. Figure 6 presents the *VDV* and the *Average Acceleration* parameters as function of the front and rear ride spring rates (K_1 and K_2, respectively). Figure 6 shows that the region for a good ride, or comfort, (lower values of Fig. 6*a*) and the region for a good road-holding (lower values of Fig. 6*b*) are not coincident. Therefore, to permit a simple analysis of the optimum region, an objective function ϕ is defined as:

$$\phi = \gamma - VDV \tag{13}$$

The two variables, *VDV* and γ, have different units, m/s$^{1.75}$ and m/s^2, but their values are of the same order of magnitude. Therefore Eq. (13) can be used to develop a simple analysis to investigate the contribution of comfort and road-holding for an optimal parameter configuration. Figure 7 presents the function γ through an acceptable range of ride spring rates values, from 4 kN/m to 12 kN/m, keeping fixed all other values.



(a) (b)

Figure 6. (*a*) *VDV* and (*b*) *Average Acceleration* γ.

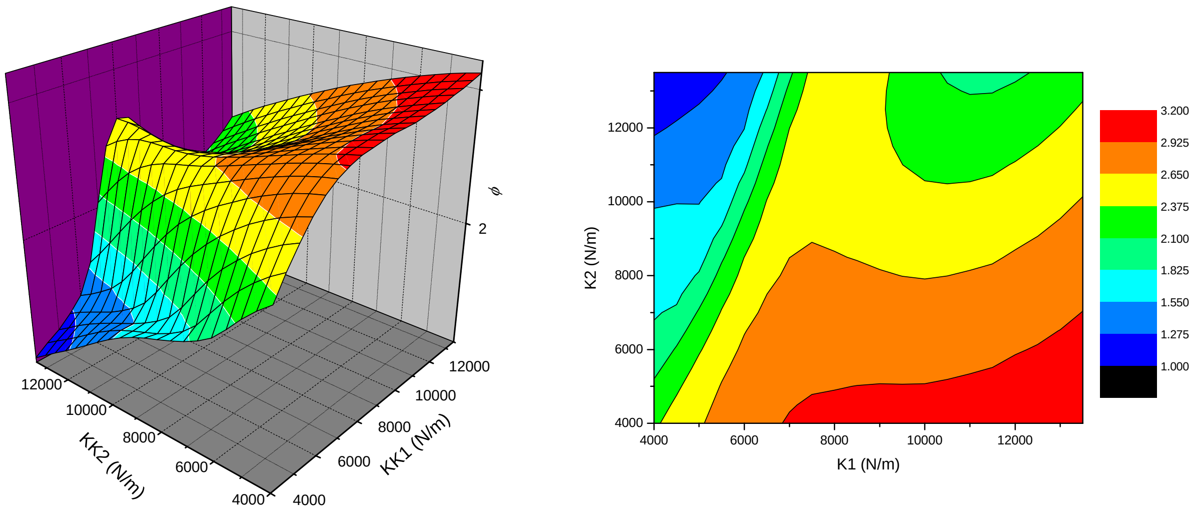


Figure 7. (*a*) Function ϕ

The function ϕ, is used for the evaluation of comfort and adherence levels simultaneously in a qualitative way. But it evidences about the existence of optimum adjustment areas, as well as also it makes possible the observation of the opposite tendencies between ride and road-holding.

Concerning the four-degree of freedom model (4DOF), it has produced very similar numerical results when it is compared with two-degree of freedom model. The influence of the unsprung mass (M_u) acts as a high frequency modulation in the main curve and it does not change the main shape of the function, as well as it has had a little influence in the VDV values.

While, in a general way, the *VDV* results showed in the Fig. 6 point to lower values for a soft calibration of the suspension system (by reducing the both values of the spring stiffness), the *Average Acceleration* results points in the opposite direction. This was already observed by Milliken (Milliken, 1995) regarding the difficult to chose a parameterization to optimize the ride and the road-holding simultaneously. However, it is possible to observe Fig. 7 that there are some regions with local optimal points.

It is worth to mention that the variable ϕ is the result of the subtraction of two variables with different units. The major proposal of this plot is to evidence the presence of local optimum regions and help in the tuning of the vehicle suspension parameters.

8. CONCLUSIONS

The methodology proposed in this work uses a simple numerical model combined with comfort/health risks standards and simple road-holding criteria to study the suspension system dynamical performance and the influence of the suspension main parameters considering the accelerations present during a *Mini-Baja* race. An estimate for an optimal suspension adjustment was obtained with this simple model. The model response was compared with experimental results obtained from accelerometers and load-cells in a simple test. The numerical and experimental results present a good agreement and it is possible to state that the model captures the main behaviors of the dynamic problem. 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 *Average acceleration* values. Such experimental program is now under development by the authors.

9. ACKNOWLEDGEMENTS

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