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Analysis of the Transmissibility of the Rear Suspension of a Mini-Baja Vehicle

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ABSTRACT

This work presents a dynamical analysis of the transmissibility of an off-road vehicle rear suspension, which was developed in *CEFET-RJ* for the *Mini-Baja / SAE-Brazil* competition. A finite element model was developed to identify the critical points of the structure. Afterwards, electric strain gages were bonded at the most critical points to measure the dynamic strains due to an impact load. Accelerometers were bonded before and after rear suspension system to measure the main transmissibility characteristics of the suspension. The data obtained through an A/D converter with instrumentation software was used to evaluate the transmissibility of the rear suspension and other important dynamic characteristics. Finally, a simple two-degree of freedom model was developed to study the behavior of the rear suspension and the influence of the main parameters in the transmissibility of accelerations and loads to the structure. An estimate for an optimal suspension adjustment was obtained with this simple model. 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, as the designer can work with more realistic loads.

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* competition since 1997. In the competition these vehicles are submit to several tests that exposed it to severe conditions, where should respect technical and safety SAE standards. These vehicles are highly competitive which demands an optimized project using advanced technologies. Figure 1 shows the *CEFET-RJ* vehicle that participated on the 1998 *SAE* 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 numerical and experimental 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.

Figure 1 –1998 - *Mini-Baja CEFET-RJ* vehicle.

In a previous work, an analysis of the front suspension loads was performed [1]. This work is a natural development of the previous study and consists in a dynamical analysis of the transmissibility of the rear suspension of the *Mini-Baja / SAE-Brazil* off-road vehicle developed at *CEFET-RJ*. A finite element model of the region of the frame structure near the suspension connection was developed to identify the critical points. Afterwards, electric strain gages were bonded at the most critical points to measure the dynamic strains due to an impact load. Accelerometers were bonded before and after rear suspension elements to measure the main transmissibility characteristics of the suspension. The data obtained through an A/D converter with instrumentation software was used to evaluate the transmissibility of the rear suspension and other important dynamic characteristics. Finally, a simple two-degree of freedom model was developed to study the behavior of the rear suspension and the influence of the main parameters in the transmissibility of accelerations and loads to the structure. An estimate for an optimal suspension adjustment was obtained with this simple model. 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, as the designer can work with more realistic loads.

This study was developed with the participation of several students and professors from *CEFET-RJ* and from University of Applied Sciences of Munich (*FHM*). These institutions have an exchange program in the mechanical engineering field that involves both professors and students. The presented analysis was developed under the project *Automotive Measurements Laboratory* sponsored by governmental agencies *CAPES* (Brazil) and *DAAD* (Germany) [2,3].

FINITE ELEMENT ANALYSIS

Numerical simulations were developed to identify the critical points in the vehicle frame, near the rear suspension connection, where the maximum strains occur. The numerical simulations were performed with commercial finite element code *ANSYS*, Release 5.7. Elements *PIPE16* and *BEAM4* (both with 2 nodes and 6 degree of freedom per node) were used [4]. The final mesh was defined after a convergence study and is shown in Figure 2 with the applied loads and boundary conditions.

A solid model of the frame region near the rear suspension connection was first developed with the 3D CAD software *MECHANICAL DESKTOP*, Release 4 [5], and then exported to the finite element software using the *IGES* format. This methodology is a current standard in the automotive industry and saves a lot of modeling time. It also permits simulate more realistic models with more precise geometry.

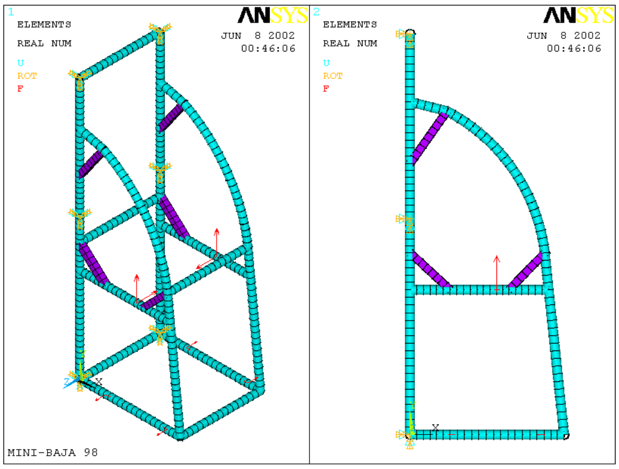


Figure 2 – Numerical analysis of the rear suspension. Finite element mesh with the applied loads and boundary conditions

Figure 3 shows the *von Mises* equivalent stress distribution of the rear suspension submitted to a static loading.

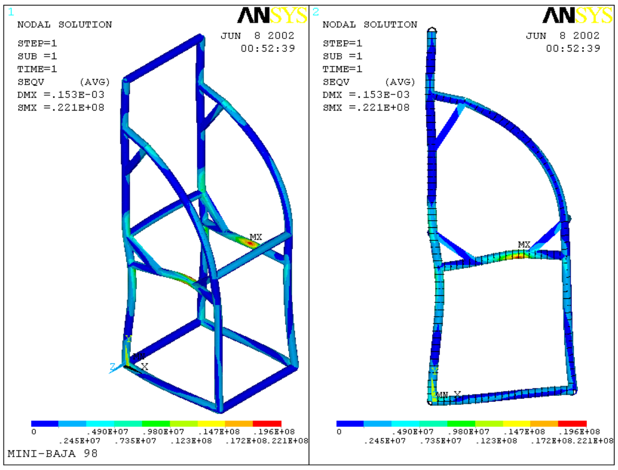


Figure 3 – Numerical analysis of the rear suspension. *von Mises* equivalent stress distribution.

EXPERIMENTAL ANALYSIS

Strain gages and accelerometers (strain gage type) were used to measure the strains and accelerations developed in some regions of the rear suspension during the dynamical loading used to simulate the impact loading on the vehicle. In this simplified analysis, to simulate this condition, the rear part of the vehicle was dropped from several heights (0.10, 0.20 and 0.30 m), with the regular loads present during the competition (engine, full fuel tank, etc.). This is a very severe condition that can be achieved after a jump during the rally test at an irregular ground. Also, a load-cell was used to record the load transmitted directly to the wheel/tire during the impact.

Three-wire technique was used to minimize the effects of wire electrical resistance and temperature [6]. Each strain gage was connected to the measurement circuit (Wheatstone Bridge) in a 1/4 bridge configuration. For the accelerometers and the load-cell a full bridge configuration was used.

The signals from the strain gages, accelerometers and load-cell were processed by a *Signal Conditioning Module LYNX AI-2160* [7]. This system has bridge completion circuits, voltage excitation, offset nulling circuit, amplifiers and filters. The conditioned analog signal was converted to a digital one by the *A/D Conversion Module LYNX AC-2120* [8]. This module has 16 channels with 12 bits resolution and a maximum sample rate of 50 kS/s (50,000 samples per second) and can be connected to a computer through a parallel port. Finally, the *AqDados – Lynx* software was used to initial zero balance and calibration, storing and plotting the measured signals from the strain gages and the accelerometers. During the measurements, 5 channels were used (2 for strain gages, 2 for accelerometers and 1 for load-cell), with a 1 kS/s sample rate per channel.

The uniaxial strain gages and accelerometers were bonded at four points in the rear suspension. These points were chosen using the information from the previous numerical analysis. Two strain gages were bonded at the critical points of the arm and the frame. One accelerometer was bonded in the arm, near the wheel and other in the frame, near the connection point of the spring-damper system. Figures 4 and 5 show the rear suspension with transducers bonded at the chosen points.

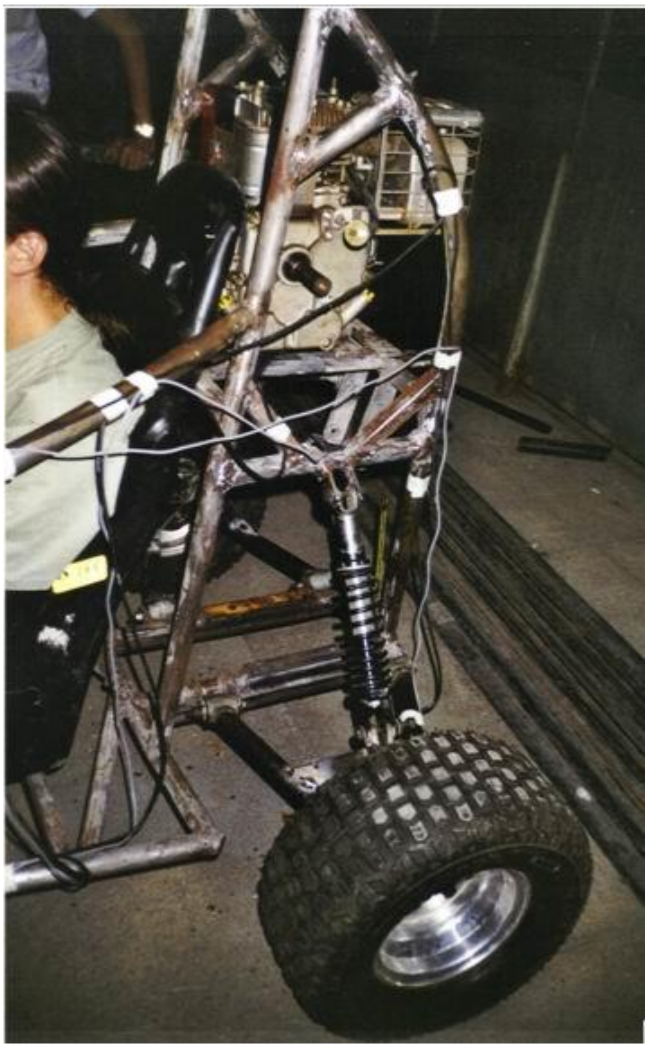


Figure 4 – Instrumentation of *Mini-Baja* rear suspension.

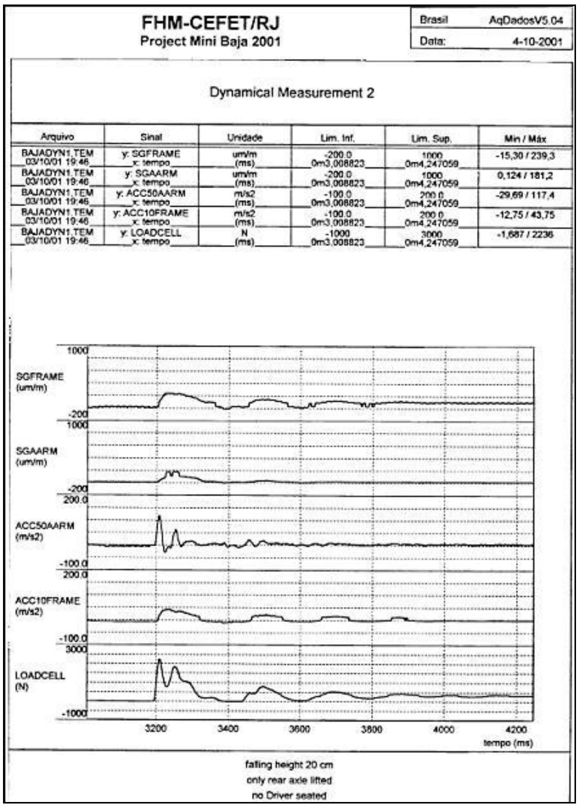
(*a*)

(*b*)

Figure 5 – Instrumentation of *Mini-Baja* rear suspension. Details in the arm (*a*) and frame (*b*).

Figure 6 shows the measured results of two dynamic loadings. This data presents the strain gages (arm and frame), accelerometers (arm and frame) and load-cell responses for the loads promoted by dropping the vehicle rear axle from the heights of 0.20 m and 0.30 m without the driver.

Maximum strain values of 200 μm/m in the arm and 277 μm/m in the frame can be observed from Figure 6, for a 0.30 m dropping height, resulting in a maximum stress of about 60 MPa. This number is four times lower than the yielding stress and about two times lower than the endurance limit of the mechanical components material (structural steel). Thus, for this loading, an infinite fatigue life is expected.



FHM-CEFET/RJ
Project Mini Baja 2001

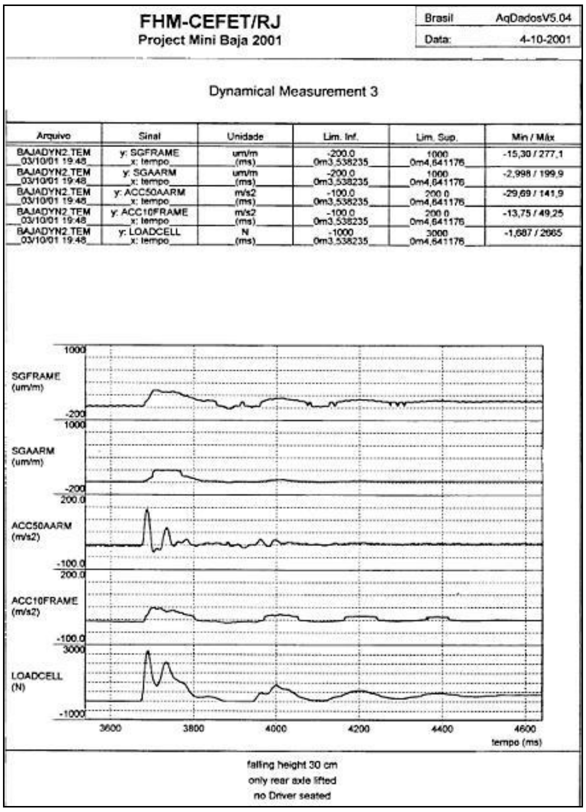
Brasil AqDadosV5.04
Data: 4-10-2001

Dynamical Measurement 2

Arquivo	Sinal	Unidade	Lim. Inf.	Lim. Sup.	Min / Máx
BAJADYN1.TEM 03/10/01 19:46	y: SGFRAME x: tempo	um/m (ms)	-200.0 0m3,008823	1000 0m4,247059	-15,30 / 239,3
BAJADYN1.TEM 03/10/01 19:46	y: SGAARM x: tempo	um/m (ms)	-200.0 0m3,008823	1000 0m4,247059	0,124 / 181,2
BAJADYN1.TEM 03/10/01 19:46	y: ACC50AARM x: tempo	m/s2 (ms)	-100.0 0m3,008823	200.0 0m4,247059	-29,69 / 117,4
BAJADYN1.TEM 03/10/01 19:46	y: ACC10FRAME x: tempo	m/s2 (ms)	-100.0 0m3,008823	200.0 0m4,247059	-12,75 / 43,75
BAJADYN1.TEM 03/10/01 19:46	y: LOADCELL x: tempo	N (ms)	-1000 0m3,008823	3000 0m4,247059	-1,687 / 2236

falling height 20 cm
only rear axle lifted
no Driver seated

(*a*)



FHM-CEFET/RJ
Project Mini Baja 2001

Brasil AqDadosV5.04
Data: 4-10-2001

Dynamical Measurement 3

Arquivo	Sinal	Unidade	Lim. Inf.	Lim. Sup.	Min / Máx
BAJADYN2.TEM 03/10/01 19:48	y: SGFRAME x: tempo	um/m (ms)	-200.0 0m3,538235	1000 0m4,641176	-15,30 / 277,1
BAJADYN2.TEM 03/10/01 19:48	y: SGAARM x: tempo	um/m (ms)	-200.0 0m3,538235	1000 0m4,641176	-2,998 / 199,9
BAJADYN2.TEM 03/10/01 19:48	y: ACC50AARM x: tempo	m/s2 (ms)	-100.0 0m3,538235	200.0 0m4,641176	-29,69 / 141,9
BAJADYN2.TEM 03/10/01 19:48	y: ACC10FRAME x: tempo	m/s2 (ms)	-100.0 0m3,538235	200.0 0m4,641176	-13,75 / 49,25
BAJADYN2.TEM 03/10/01 19:48	y: LOADCELL x: tempo	N (ms)	-1000 0m3,538235	3000 0m4,641176	-1,687 / 2665

falling height 30 cm
only rear axle lifted
no Driver seated

(*b*)

Figure 6 – Experimental results. Report generated for two dynamic loadings: (*a*) 0.20 and (*b*) 0.30 m dropping heights.

From the load-cell dynamic measurements is possible to establish an amplification factor α that represents the ratio between the maximum dynamic load, F_{din}, and the static load, F_{static}:

$$\alpha = F_{din} / F_{static} \tag{1}$$

where F_{static} is equal to the reaction on the wheel promoted by the vehicle weight and F_{din} is the impact load on the tire. An estimate of F_{din} based in a simple one-degree of freedom analytic model (spring-mass) can be obtained through an energy conservation analysis [9]:

$$F_{din} = F_{static} + \sqrt{(F_{static})^2 + 2KF_{static}h} \tag{2}$$

where K is the structure stiffness and h the dropping height. The structure stiffness can be represented by the equivalent stiffness of the wheel/tire stiffness and suspension stiffness in series. Experimental compression tests shown a wheel/tire stiffness of 55 kN/m and a spring stiffness of 37 kN/m, resulting in an equivalent stiffness of 22 kN/m. Figure 7 compares the factor α obtained from the Eq. (2) model and the one obtained from experimental data. Values up to 7.7 can be observed. This number is higher than 4, the factor usually used in the design of passenger vehicles [10-12]. But it is worth to mention that this value was obtained for an off-road vehicle, which must be designed for severe loadings.

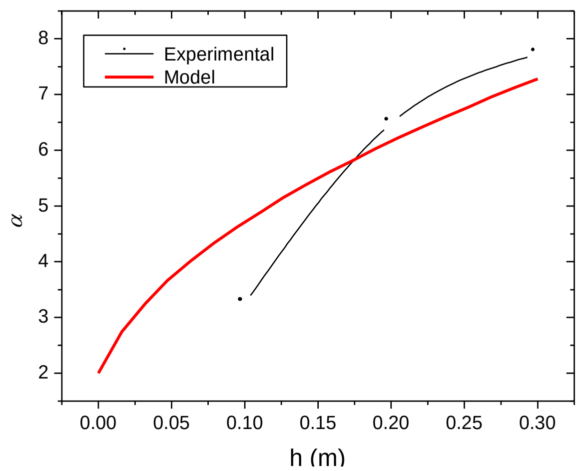


Figure 7 – Amplification factor α for several dropping heights.

From the accelerometers dynamic measurements is possible to establish a transmissibility factor β that represents the ratio between the maximum dynamic accelerations of the frame, a_{frame}, and the arm, a_{arm}:

$$\beta = a_{frame} / a_{arm} \tag{3}$$

This is an important parameter for the design of the components that are positioned after the suspension and for the driver comfort. Figure 8 presents the measured transmissibility factor β for several loading conditions.

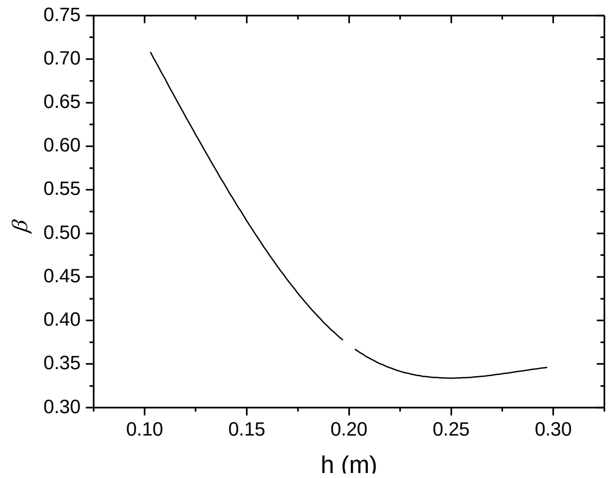


Figure 8 – Transmissibility factor β for several loading conditions.

Figure 8 shows that the acceleration transmissibility is higher for small dropping heights. This means that for small ground irregularities a major part of the tire/wheel acceleration is transmitted to the frame. In spite of lower acceleration intensities are expected, a long-term effect can occur on the frame stiffness, the bearings abrasion, or even on the drivers healthy. Therefore a complete study must also involve this “mild” loadings.

SIMPLE TWO-DEGREE OF FREEDOM MODEL

Figure 9 presents a simple two-degree of freedom model that was developed to study the dynamic behavior of the rear suspension [13]. The rear suspension was modeled considering a system with two lumped mass elements: the wheel/tire connected to the arm (m_1) and the frame (m_2). Spring and damper elements were used to represent the connections between the ground and the wheel (c_1 and K_1 – c is the coefficient of viscous damping and K is the stiffness) and between the wheel and the frame (c_2 and K_2 – the spring-damper system). The vertical displacements of the arm (or the wheel) and the frame are u_1 and u_2, respectively.

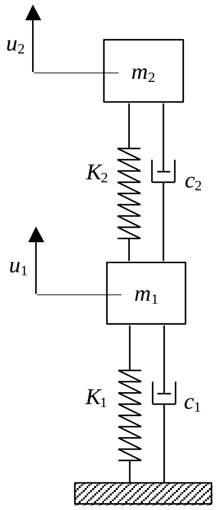


Figure 9 – A simple two-degree of freedom model for the rear suspension.

A free vibration analysis was considered with velocity initial conditions prescribed to both masses. The initial time, t = 0, of the analysis corresponds to the instant when the tire touches the ground. From an energy conservation analysis, at this time instant both masses have an initial velocity of $\sqrt{2gh}$, where g is the gravity acceleration and h the dropping height. By establishing the equilibrium of the system, equations of motion are written as follows:

$$\ddot{u}_1 = (1/m_1)\left[K_2(u_2 - u_1) + c_2(\dot{u}_2 - \dot{u}_1) - K_1 u_1 - c_1 \dot{u}_1\right] - g$$
$$\ddot{u}_2 = (-1/m_2)\left[K_2(u_2 - u_1) + c_2(\dot{u}_2 - \dot{u}_1)\right] - g \qquad (4)$$

where $(\dot{\ })$ represents the differentiation with respect to time. Numerical simulations were performed employing a fourth order Runge-Kutta method for numerical integration [14]. A convergence study was developed to chose the time step.

The four parameters used in the analysis are the following: m_1 = 2.6 kg, c_1 = 0, K_1 = 55 kN/m, m_2 = 32.6 kg, c_2 = 300 N·s/m, K_2 = 37 kN/m. The stiffness were measured through a compression test and the coefficient of viscous damping was estimated from the experimental dynamic data.

Figure 10 presents a comparison between the arm and frame measured accelerations and the ones obtained with the model for a dropping height of 0.20 m without driver.

It can be observed that the numerical response presents higher maximum values than those obtained in the measured data. However, the results present a good agreement and it is possible to state that the model captures the main behaviors of the dynamic problem.

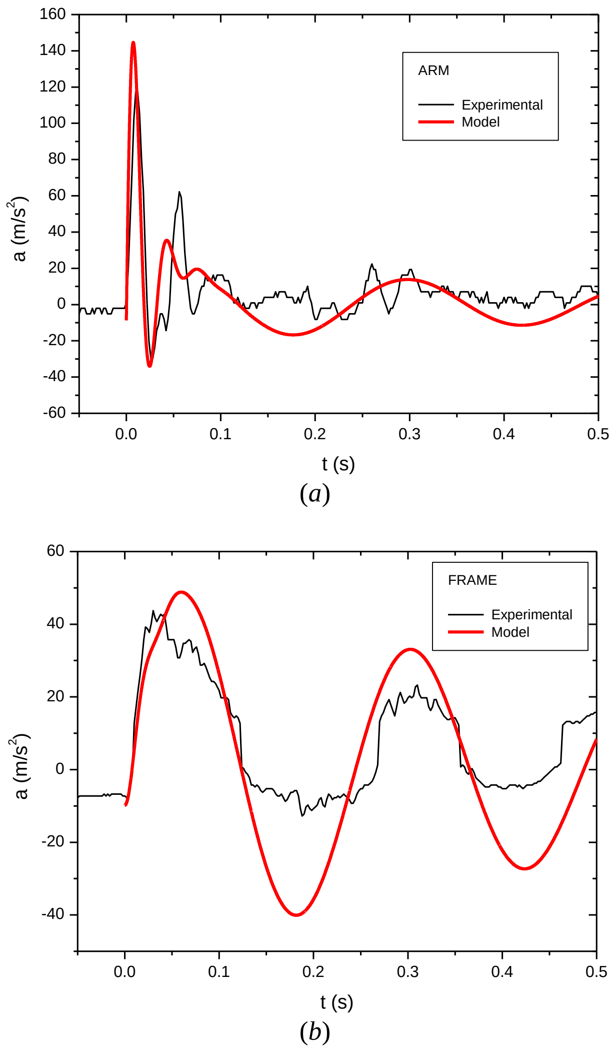


Figure 10 – Measured data and analytic model results for the rear suspension arm (a) and frame (b) for h = 0.20 m.

This simple model can be used to estimate an optimal suspension adjustment. Figure 11 shows the transmissibility factor, β, the load on the tire and the load transmitted to the frame as a function of c_2 and K_2, the two parameters that characterizes the dynamic behavior of the rear suspension.

It can be observed from Figure 11 that the coefficient of viscous damping has a very small influence on the loads transmitted to the arm and frame, and have a prejudicial effect on the transmissibility factor β. However, the stiffness has a major influence on the transmitted loads (in accordance with Eq. 2) and on the transmissibility factor in the way that a lower stiffness reduces simultaneously both variables. Therefore, a spring-damper system optimal adjustment requires the lowest possible stiffness value. Based in this analysis, the configuration c_2 = 300 N·s/m and K_2 = 10 kN/m was chosen as the optimal suspension adjustment.

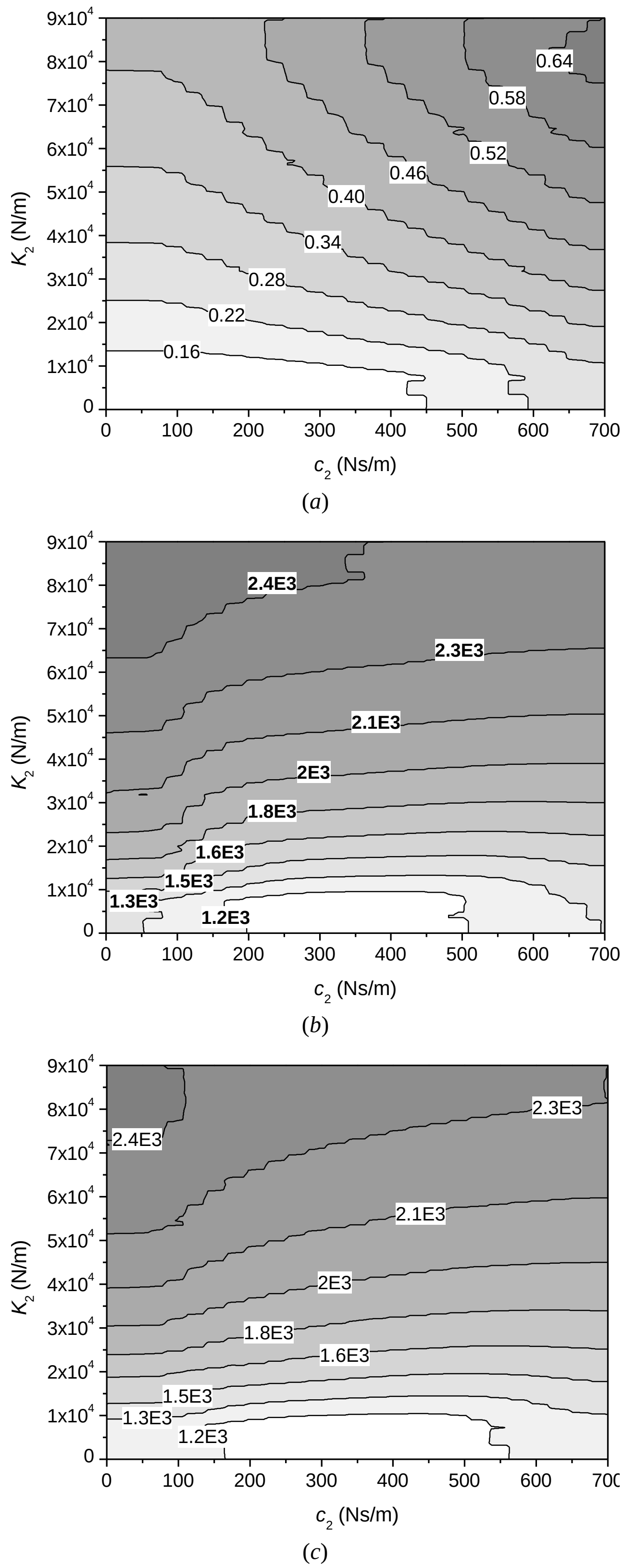


Figure 11 – Analytic model predictions for the transmissibility factor (*a*), the load on the tire (*b*) and the load transmitted to the frame (*c*), as a function of the rear suspension parameters. Loads in newtons and h = 0.20 m.

Figure 12 presents the predicted response for the optimal suspension adjustment. The measured and model responses with the original adjustment are also shown for comparison. The optimal adjustment results in a lower transmissibility factor (0.16 instead of 0.34) and a lower transmitted load to the frame (1,2 kN instead of 2 kN). This analysis indicates that the proposed adjustment can represent a considerable improvement in the original design. It is worth to mention that a softer suspension can affect other important factors as the vehicle driveability. An optimal condition must guarantee the best compromise among conflicting performance indices pertaining to the vehicle suspension system, i.e., comfort, road holding and working space [15]. Therefore, a complete study with a prototype vehicle must be done to verify the actual improvement of the optimal suspension adjustment in the vehicle overall performance.

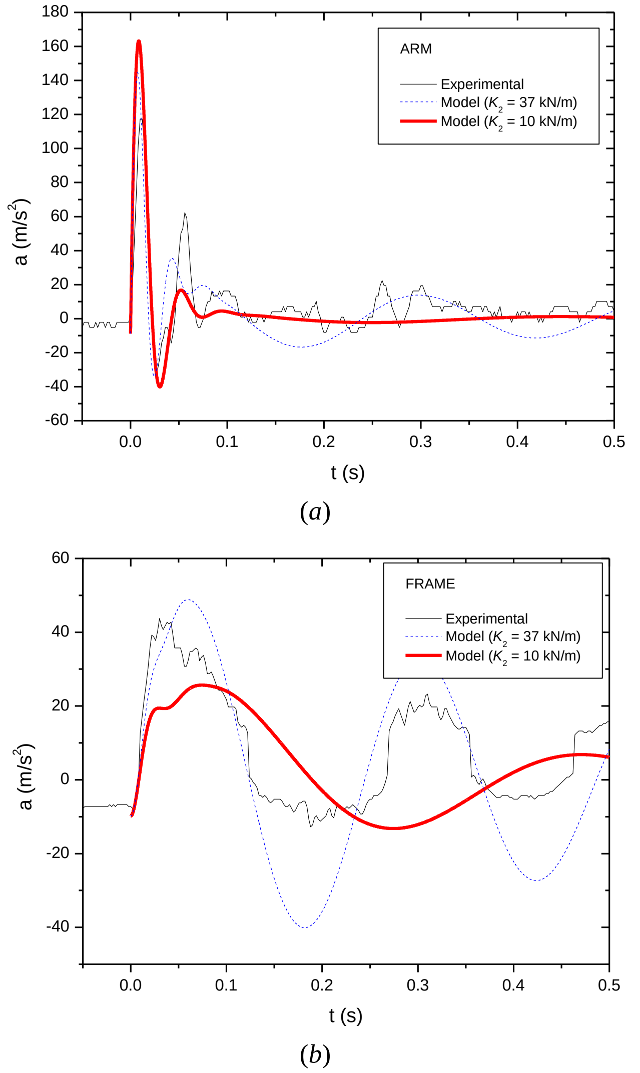


Figure 12 – Measured data and analytic model results for an optimal suspension adjustment. Rear suspension arm (*a*) and frame (*b*) accelerations for h = 0.20 m.

Figure 13 shows the loads as a function of suspension stiffness (K_2). As expected, both analytic models predict that the lower the suspension stiffness the lower is the load on the tire. A comparison between experimental an analytic results for this load shows that the analytic models predict values something lower (5 % lower for Eq. 2 model and 12% lower for Eq. 4 model).

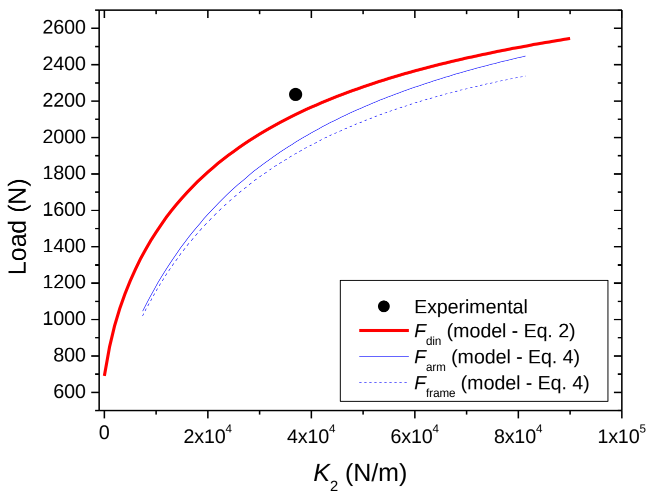


Figure 13 – Measured data and analytic models results for the loads as a function of suspension stiffness (K_2) for h = 0.20 m

CONCLUSIONS

The methodology adopted, using analytic, numerical and experimental techniques allowed the development of a simple methodology that can be used to study the suspension system performance and the influence of the main parameters. A simplified drop test was realized and the strain and acceleration measured in some critical points furnished data to estimate important dynamic parameters as the amplification factor and the transmissibility factor. A simple two-degree of freedom model was developed to study the behavior of the rear suspension and the influence of the main parameters in the transmissibility of the loads and accelerations to the structure. An estimate for an optimal suspension adjustment was obtained with this simple model. 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, as the designer can work with more realistic loads.

The presented analysis was developed under the project *Automotive Measurements Laboratory* sponsored by governmental agencies *CAPES* (Brazil) and *DAAD* (Germany) with the participation of several students and professors from *CEFET-RJ* and from University of Applied Sciences of Munich (*FHM*). During this project considerable amount of relevant knowledge in the automotive field was exchanged between the two institutions.

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