DYNAMIC TESTING DEVICE FOR OFF-ROAD VEHICLE SUSPENSIONS ANALYSIS

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Abstract. *This work is concerned with a laboratory device development for dynamic analysis of vehicle components behaviors. The concept of the device is to provide experimental data, mainly stiffness and damping coefficients, for full-scaled models of vehicle suspensions under realistic severe dynamic loading. The academic demand for these investigations is determined by current automobile all-terrain prototype SAE/Brazil-MINIBAJA project undertaken by CEFET/RJ students oriented by a professor board . 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, i.e., comfort, road holding and working space. Altogether with new-concept designed solutions during project steps, there is a natural demand for employing commercial parts for technical and economical reasons. The knowledge of the dynamic response of commercial dampers specially constitutes a throat for suspension design. The device in focus will allow the testing of real damper-spring-parts assemblies under controlled conditions in order to determine such parameters. The device is wholly modeled in computational environment through 3D-cad softwares (SOLIDWORKS) and dynamic simulation (ADAMS) according with analytical formulation expectations. The device operation includes data acquisition and interpretation through instrumentation resources. Besides the objectivity of technical procedures, designing steps and expected result, this work includes a major aspect: this is an academic experiment. The path from the initial definitions, reaching design solutions, mechanic production, instrumentation assembling, until the results interpretations are conducted by engineering under-graduated students. This paper presents the device - from conception to final mechanical model - the MINIBAJA-prototype dynamic behavior former results, a comparison with theoretical analytical formulations and an overview of the rich academic results obtained.*

Keywords: *Vehicle Suspension, Numerical Analysis, Dynamical Analysis, Testing Device.*

1. Introduction

CEFET/RJ is a Brazilian federal government institution dedicated to technological education that covers different teaching levels from technical high school, through Engineering Courses up to Master Course. There are permanent efforts to integrate technological research and development along these levels in harmony with the potential and maturity of students in each one. In this context this paper has two important guidelines: first one is to present a real learning-researching experience involving undergraduate engineering students and a practical engineering task. This activity is performed inside an institutional research line and represents an important educational project. The students are in contact with sophisticated dynamic modeling techniques applied to real (automobile) problems and are asked to contribute with prototype building and testing. These advanced professional skills are stimulated since the earlier semesters in order to prepare the students to further perspectives as a researcher or development engineer. The second guideline is to present the technical proceeding and the effective results available for practical applications and currently used in *SAE/Brazil MINIBAJA* development.

The *MINIBAJA* vehicle is completely developed and build by undergraduate engineering students under orientation of a professor board. During development, the students are exposed to a real engineering problem involving several areas of knowledge. *CEFET/RJ* participates on the *SAE/Brazil* competition since 1997. In the competition these vehicle are submit to several tests that exposes 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 in the *2001 SAE/Brazil* event.

Figure 1. *CEFET/RJ* 2001 *SAE/Brazil-MINIBAJA* vehicle.

Design optimization requires avoiding the use of highly conservative criteria, as well the risk of under-dimensioning the components, both consequences of a poor-refined model. Defining precisely design variables as loads, mechanical materials, physical properties, operating conditions and geometry is essential to generate refined models.

An important point about *MINIBAJA* suspension design is the deep comprehension of its behavior due dynamic loading, as the vehicle must resist all tests demanded in *MINIBAJA* competition. A profound comprehension of the theoretical model during design phase results in further development cost and weight reduction in the final assembly. Meanwhile, excessive highly refined modeling and fully instrumented real prototypes may lead benefits to a setback due to high-costly equipment and long time demanded.

Virtual modelers feed with data from real testing devices can be quite promising in technological developments. Computer simulations allow valuable natural cost reduction during design phase, model optimizing and other shortcuts.

This paper presents simple analytical numerical results generated through virtual modeling using ADAMS package (MDI, 1997). ADAMS package is a complete virtual prototyping environment that provide fully integrated modeling, solution, and visualization of the dynamics of rigid bodies. It permits to build and test virtual prototypes on computers and presents, both visually and mathematically, the full-motion behavior of complex mechanical systems designs.

Virtual modeling task sequence consists on building the model for the system , running it through a battery of tests, validating test results, refining the design, and optimizing system performance through iterations as shown in Fig.(2). The main characteristic is that all these steps are done on the computer — quickly, easily, and cost-efficiently. ADAMS package solves the equations of motion of rigid multi-bodies and yield complete system motion, forces, velocities and accelerations. This data provide a correspondent physical insight of mechanical systems behavior, as would a fully instrumented prototype with accelerometers, load-cells, etc. This is called functional virtual prototyping.

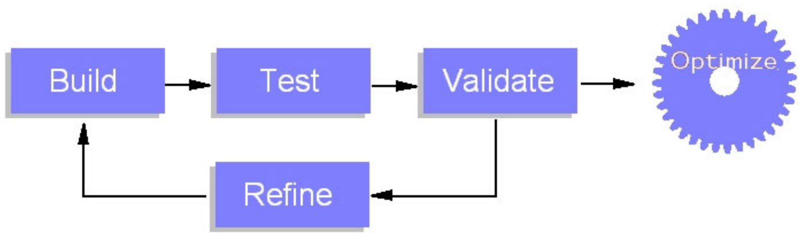

Figure 2. Iterative sequence in virtual modeling

Functional virtual prototyping supported with experimental data acquisition is a tendency among automotive industries and appears in many *MINIBAJA* teams as well. That seems to be the state-of-the-art concept for design optimization, and one of the main aims of *CEFET/RJ* researching group at this moment.

2. Objectives

This paper describes the experience using computational prototyping in *CEFET/RJ*, with potential application in *MINIBAJA* and many others projects. It offers an overview of the learning experience on science and technology environment embedded in one institutional project. Two correlated cases are studied: - *i*) a computational simulation of a test device developed to determinate the damping coefficient of commercial spring-dampers used in *MINIBAJA CEFET/RJ* project and, *ii*) a computational simulation of *MINIBAJA* vehicle rear suspension. The dynamic behavior is analyzed using a numerical model developed to study a dropping test carried out with real model at *CEFET/RJ* in previous works (Kenedi *et al.*, 2001; Pacheco *et al.*, 2002).

3. Proceedings

The design phase was preceded by a study group organization. It was intended to provide the students directly involved with theoretical reinforcement. Fundamental aspects such experimental stress analysis, analytical and computational vibration analysis for linear systems with one and two degree of freedom (DOF), finite element analysis were discussed. The group was later subdivided according these main areas and respective projects.

The group involved with vibration analysis had as main activity to learn ADAMS package. The task was to develop virtual prototypes of *MINIBAJA* subsystems and some testing devices. Experimental data were used to obtain reliable boundaries conditions and loadings for further finite element analysis. The spring-damper test device and the *MINIBAJA* rear suspension-dropping test were modeled and analyzed through numerical simulations, and the results are presented in this paper.

4. Spring-Damper Test Device

The spring-damper dynamic test device is used to determinate the damping coefficient and the stiffness of a spring-damper component. A tridimensional solid model of the test device developed with a SOLIDWORKS (Solidworks, 2002) solid modeler package is shown in Fig. (3). The device has a block of mass directly connected to the moving rod of the element being tested. The other side the element is connected to the device base in series with a load-cell. The block of mass is conducted by a positioning system (a hydraulic piston) from the static equilibrium position to a non-equilibrium position, distending the spring of the element. The block is then locked in this position and the positioning system is removed. As the mass is liberated from its non-equilibrium position, the system experiments a free vibration motion. Obviously the mass of the block must be much greater than the mass of the spring-damper element moving parts. During the test two signals are registered over time in a computer using a data acquisition system: vertical displacement of the block and force at the device base. The first one is obtained by placing a resistive linear motion transducer between the block and the device base, whereas the second one is obtained through the load-cell positioned at the device base.

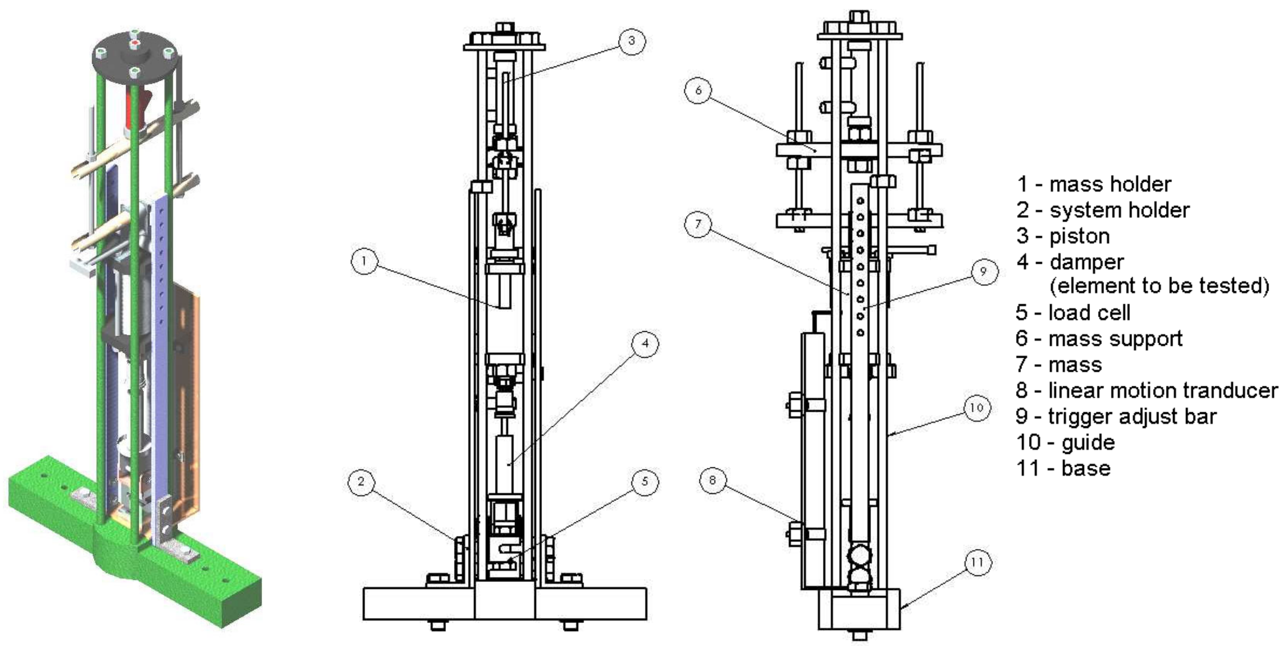

Figure 3. Dynamical spring-damping test device (spring is not shown).

The stiffness coefficient, *k,* is obtained by a simple static test. The spring becomes distended due static load promoted by the positioning system. The resulting displacement is used to calculate stiffness coefficient by the following expression:

$$k = \frac{F}{\Delta x} \tag{1}$$

where *F* is the static load applied by the positioning system, that can be measured by the load-cell, and Δx is the spring deflection due the load, that can be measured by the linear motion transducer.

The damping coefficient *c*, or alternatively, the damping ratio ζ, requires a more complex approach. While both mass and stiffness can be determined by static tests, damping determination requires a dynamic test.

The system composed by the mass and the spring-damper can be modeled through a simple single degree of freedom mass-spring-damper system whose behavior is described by the classic equation of motion for a single degree of freedom mass-spring-damper (Meirovitch, 1975)

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = 0 \tag{2}$$

where *m* is the mass of oscillating body, *c* is the damping coefficient and *t* is the time. Alternatively, Eq. (1) can be written as:

$$\ddot{x}^2 + 2\zeta\omega_n\,\dot{x} + \omega_n^2 = 0 \tag{3}$$

$$\zeta = \frac{c}{2m\omega_n} \tag{4}$$

where ζ is the damping ratio and ω_n the circular natural frequency given by $\omega_n = \sqrt{k/m}$. The analytic solution for Eq. (3) for an under-damped system can be easily obtained (Meirovitch, 1975):

$$x(t) = \left(e^{-\zeta\cdot\omega\cdot t}\,x_0\right)\left(\left(\zeta\frac{\omega_n}{\omega_d}\right)\sin(\omega_d t) + \cos(\omega_d t)\right) \tag{5}$$

The exponential term of Eq. (5) controls the amplitude decrement. By a clever manipulation of the first part of Eq. (5) it is possible to determine the damping ratio by using the concept of *logarithmic decrement* denoted by δ and defined by:

$$\delta = \frac{1}{n}\ln\frac{x(t)}{x(t+nT)} \tag{6}$$

where *x(t)* and *x(t + nT)* are two displacements amplitudes *n* periods apart and *T* is the oscillation period. Greater accuracy of the result can be obtained if $n > 1$ (Timoshenko *et al.*, 1974). The damping ratio, given the value of the logarithmic decrement, is determined by the following expression:

$$\zeta = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}} \tag{7}$$

Finally, by rearranging Eq.(4), the damping coefficient can be determined by:

$$c = 2\zeta\, m\omega_n \tag{8}$$

The damped circular natural frequency ω_d, which represents the frequency of the damped motion, is given by

$$\omega_d = \omega_n\sqrt{1-\zeta^2} \tag{9}$$

The damping ratio can be determined from a record of the displacement response of an under-damping system (Inman, 1996). Figure (4) shows a characteristic plot of a damping test and illustrates how to obtain the logarithmic decrement.

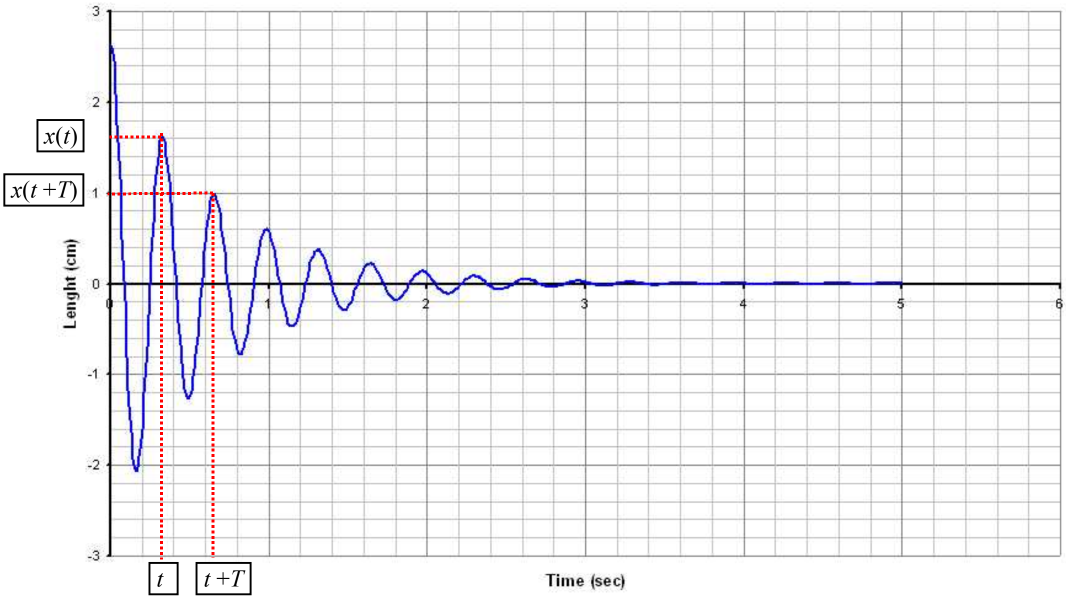

Figure 4. Determination of the damping ratio from a displacement record.

5. Numerical Simulations

5.1. Spring-Damper Device Virtual Prototype Simulation

The proposed virtual prototype methodology is applied to study the performance of the spring-damper test device presented in the last section before its construction. As pointed before, the total moving mass of the spring-damper must be much smaller than the mass of the block. With this requirement attended, a simple one-degree of freedom mass-spring-damper model furnishes a good representation of the dynamical behavior of the device and the equations developed in the last section can be applied to determine the damping ratio. Therefore, it is important to estimate the minimum mass value of the block and two models are considered: *i*) *2-DOF ADAMS model*, where the moving mass associated with the moving parts of the spring-damper is considered, and *ii*) 1-*DOF ADAMS model*, a single degree of freedom mass-spring-damper model, whose response is similar to the one given by Eq. (5) assuming a mass equal to the mass of the block (other moving masses are not considered).

Figure (5*a*) shows a tridimensional solid model of the spring-damper used in the analysis and Fig. (5*b*) shows the moving mass of the spring-damper. The solid modeler package used to develop the solid model (Solidworks, 2002) allows the estimate of several important parameters as the baricenter of the components or the total moving mass. Figure (6) shows the model developed with ADAMS package where M is the mass of the block, m is the mass of the moving mass and k and c are the stiffness and damping coefficients of the spring-damper being tested. In the analysis a moving mass of 1.0 kg, that represents the moving parts of the spring-damper, is used and lumped at the middle of the element. Moreover, the following parameters are adopted: k = 37 kN/m, c = 300 Ns/m (Pacheco *et al.*, 2002).

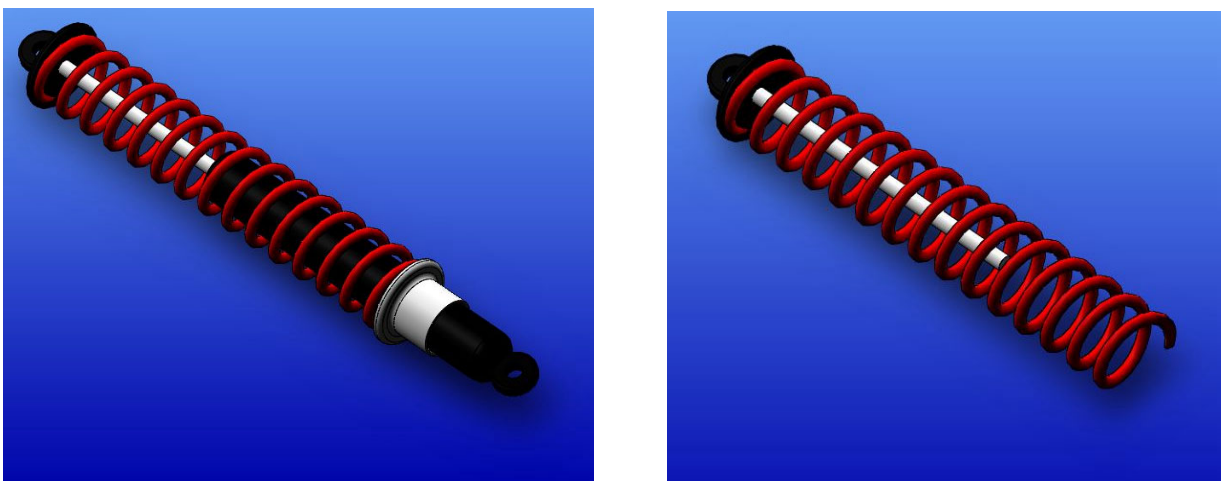
(a) (b)

Figure 5. Tridimensional solid model of (*a*) spring-damper and (*b*) moving mass

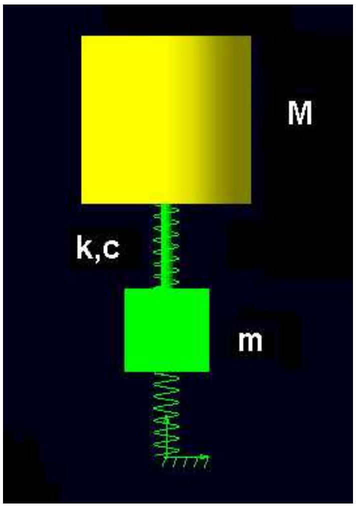

Figure 6. Spring-damper device virtual prototype model (*2-DOF ADAMS*).

Figure (7-8) shows the displacement of the block of mass *M* over time obtained with *2-DOF ADAMS* (curve in red) and *1-DOF ADAMS* (curve in blue) models for several values of mass. It can be observed that both curves converge for larger values of *M*. Moreover, the logarithmic decrement δ is difficult to measure for values of *M* smaller than 7 kg, as the ratios between the second and the first peaks are smaller than 10%. Finally, as pointed in Section 4, greater accuracy for the experimental determination of δ can be achieved using Eq. (6) with more than one oscillation period ($n > 1$). Meanwhile, results show that for mass values up to 7 kg amplitude peaks following the second one are difficult to identify and measure. Therefore, a mass *M* of 10 kg should be adopted. Table (1) presents the logarithmic decrement δ for the analytical and both ADAMS models, the amplitude ratio between the second and the first peaks (x_1/x_0) and damping coefficient *c* obtained from the methodology.

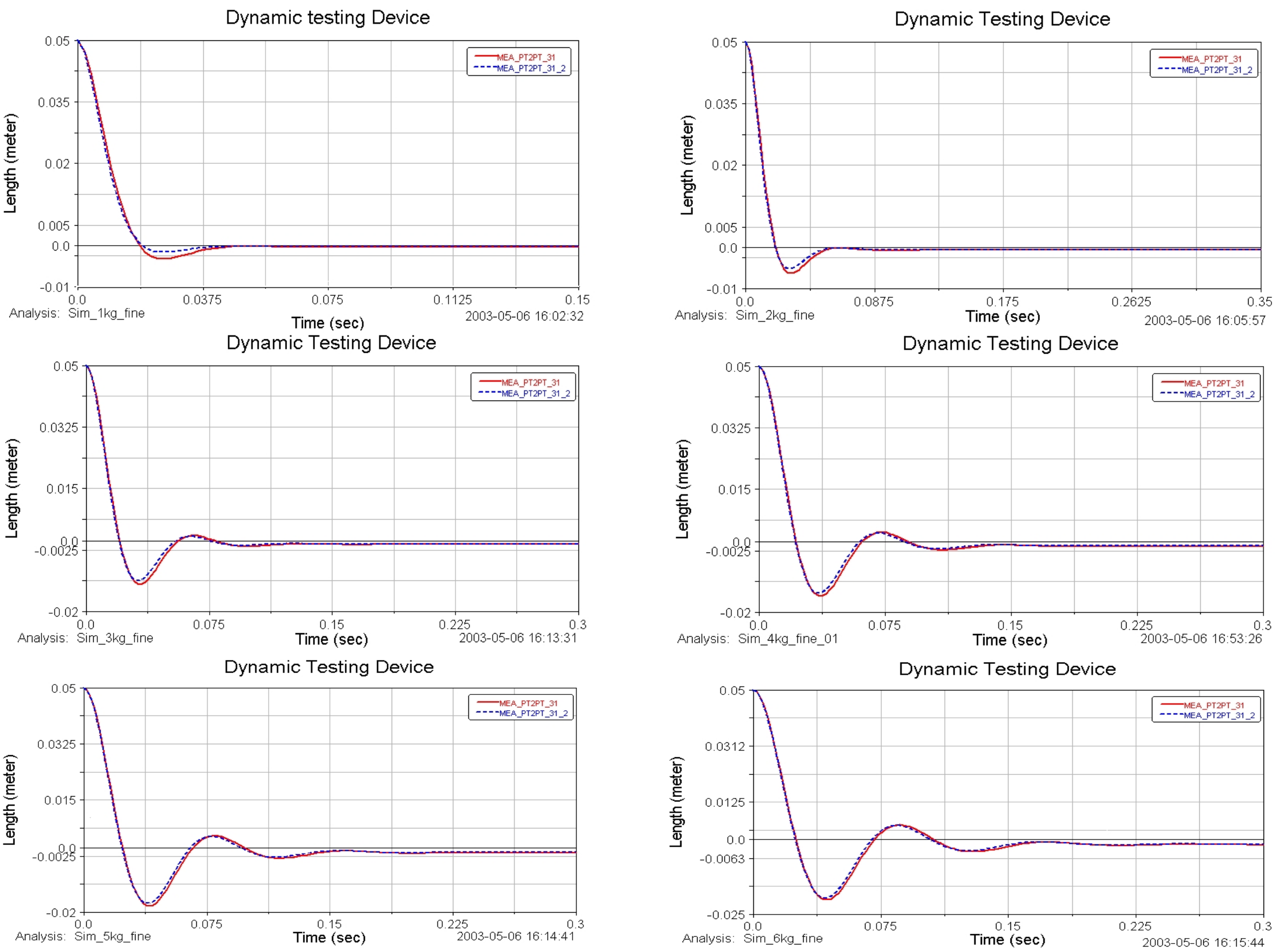

Figure 7. Displacement of the block of mass for the spring-damper device virtual prototype. Block mass from 1 kg to 6 kg in increments of 1 kg.

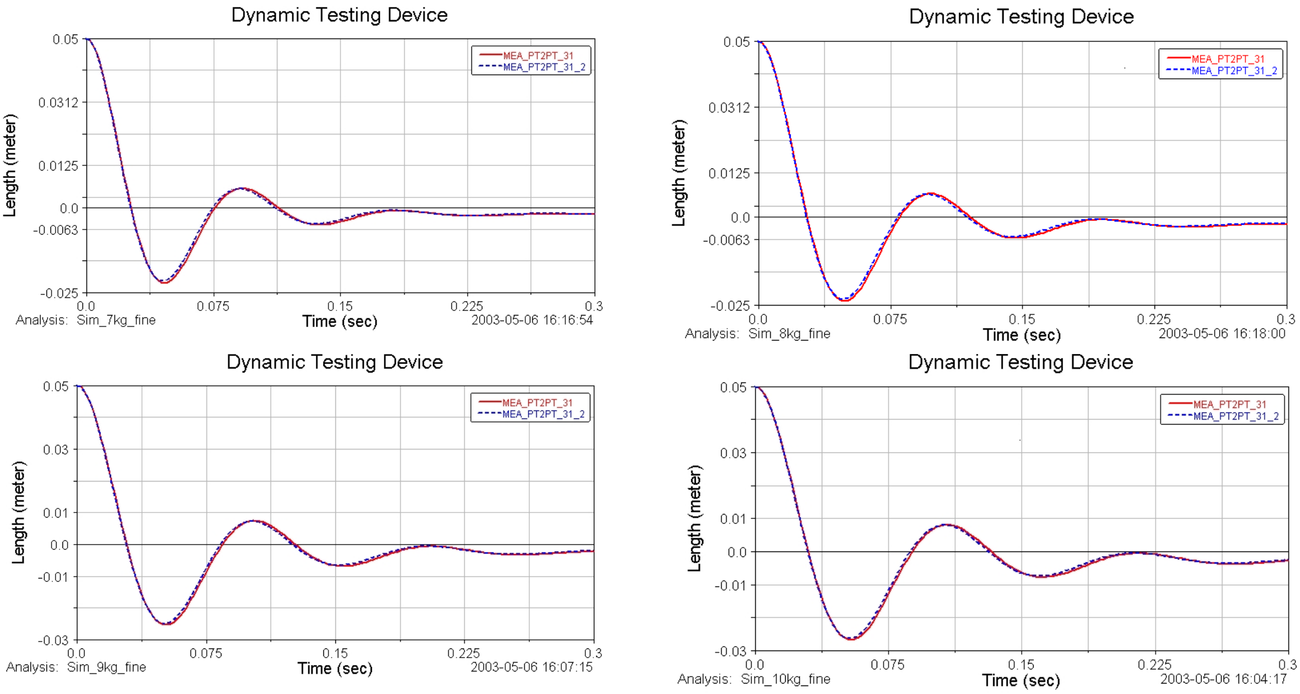

Figure 8. Displacement of the block of mass for the spring-damper device virtual prototype. Block mass from 7 kg to 10 kg in increments of 1 kg.

Table 1. Logarithmic decrement δ, amplitude ratio (x_1/x_0) and damping coefficient c.

Mass of the Block (kg)	δ ***Analytic***	***ADAMS 1-DOF model***		***ADAMS 2-DOF model***		$\mathbf{x_1/x_0}$ (%)	***c*** (Ns/m)
		δ	Error (%)	δ	Error (%)		
1	7.83	7.17	8.3	5.03	35.7	0.4	240.5
2	4.15	4.94	19.0	4.32	4.0	0.2	308.2
3	3.17	3.13	1.3	2.96	6.5	3.2	284.1
4	2.66	2.66	0.2	2.53	4.9	5.5	287.4
5	2.34	2.33	0.3	2.25	3.9	7.7	289.6
6	2.11	2.11	0.2	2.04	3.3	9.8	291.1
7	1.94	1.93	0.2	1.88	2.8	11.6	292.2
8	1.80	1.80	0.2	1.76	2.5	13.3	293.1
9	1.69	1.69	0.1	1.65	2.2	14.8	293.9
10	1.60	1.60	0.2	1.57	2.0	16.3	294.2

The results obtained with numerical simulation lead to a range of mass values in which further experimental procedures can be successfully conducted. This means that amplitudes (and so logarithmic decrement) and damping coefficients measurements will be feasible, considering normal experimental accuracy ranges. The model itself provides consistent results in both situations (1DOF and 2DOF) in comparison with analytical similar model. The damping coefficient obtained is close to the nominal known value indicating the proposed methodology can be used to estimate this mechanical property.

5.2. Suspension Virtual Prototype Simulation

Figure (9) shows a virtual prototype of the *MINIBAJA* vehicle rear suspension. This suspension have been already instrumented with accelerometers, strain gages and load cells, according to a real prototype test carried at *Laboratory of Experimental Stress Analysis and Instrumentation* (*LAETI-CEFET/RJ*) where drop test were performed (Pacheco *et al.*, 2002). The main idea is to obtain, using a computational method over a refined model, some reliable numerical results using similar conditions observed in real conditions. For the dropping test simulation, a falling from a height of 0.20 m is considered. That condition represents a severe dynamical load condition. Figures (10-12) show the results obtained during simulations with ADAMS package: accelerations on frame and on arm, spring-damper reaction force and reaction force on tire, respectively. Results are in accordance with the previous experimental tests and reveals that they capture the main dynamical behavior of the rear suspension. Therefore, virtual drop tests can be used as a powerful tool during the suspension optimization process.

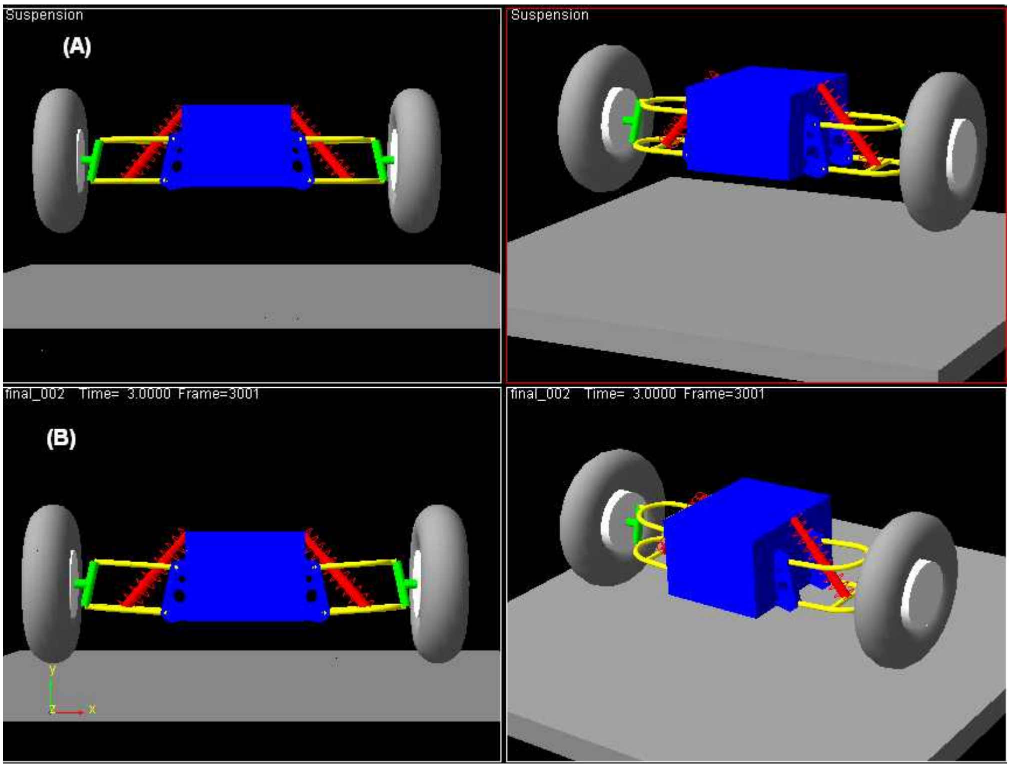

Figure 9. *MINIBAJA* rear suspension in virtual dropping test. (A) – Initial condition (0.20 m). (B) – Impact instant.

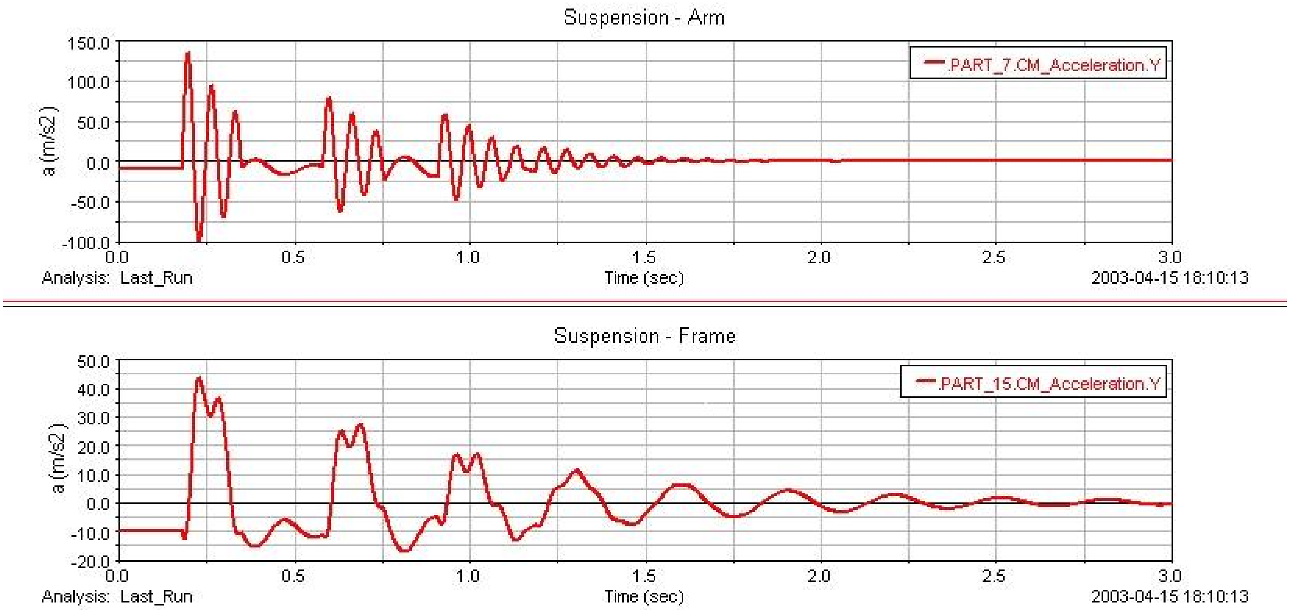

Figure 10. Acceleration on arm and frame. Virtual Dropping test, with h = 0.20 m.

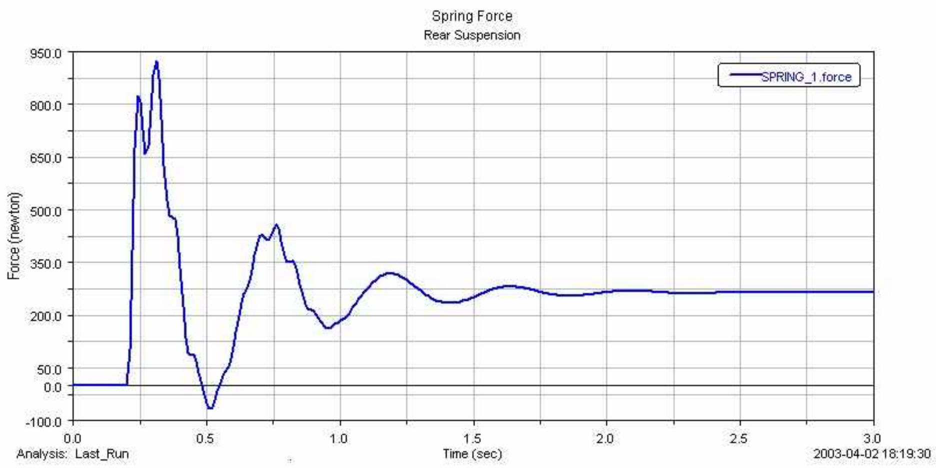

Figure 11. Spring force in virtual dropping test.

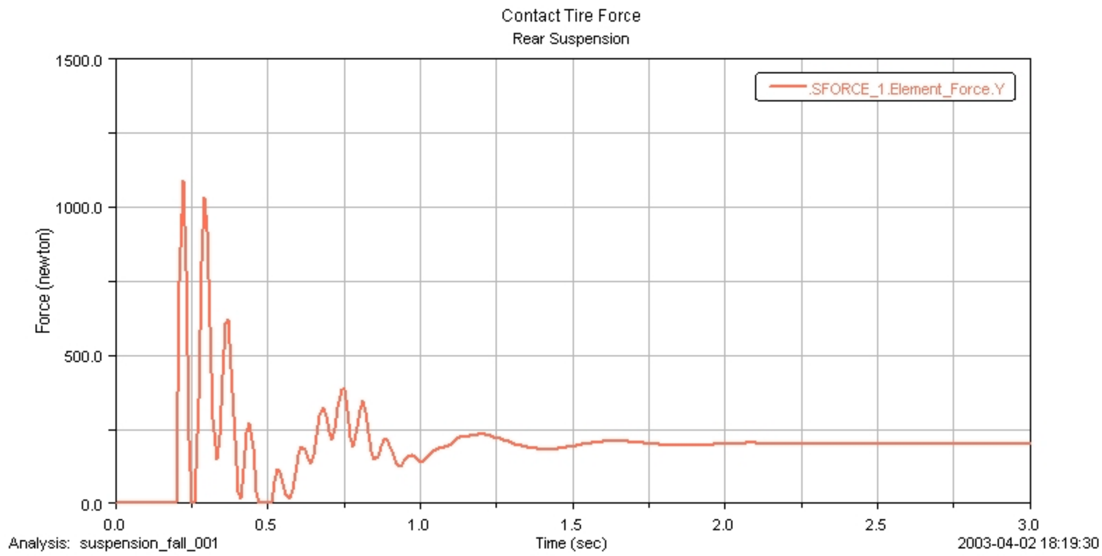

Figure 12. Contact force on tire in virtual dropping test.

6. Conclusion

The results obtained suggest not only the validity of the modeling adopted, but also, the maturity of the students to perform research and project work in engineering context. These results will guide the choice of parameters for experiments planning and final build-up of the testing device. They are also very important to guide the continuity of educational plans to disseminate this methodology in a wide universe of students and regular disciplines. On the other hand, the results contribute to optimize current and future projects through the reduction of expensive and complex experimental steps and the use of reliable numerical simulation. Once these simulations reach a level of high reliability, the "virtual device" can be used as a partial substitution for experimentation.

7. Acknowledgement

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8. References

Inman, D. J., 1996, *"Engineering Vibrations"*, Ed. Prentice Hall.

MDI, 1997, *"Using ADAMS/View"*, Release 11.

Kenedi, P.P., Pacheco, P.M.C.L., Jorge, J.C.F. Vieira, R.D., e Danninger, W.; "Dynamic Experimental Analysis of a Mini-Baja Vehicle Front Suspension", *SAE2001 - 10º Congresso e Exposição Internacionais de Tecnologia da Mobilidade - SAE*, São Paulo, Novembro de 2001.

Meirovitch, L., 1975, *"Elements of Vibration Analysis"*, McGraw-Hill.

Pacheco, P.M.C.L., Kenedi, P.P., Jorge, J.C.F. Vieira, R.D. and Danninger, W., 2002, "Analysis of the Transmissibility of the Rear Suspension of a Mini-Baja Vehicle", *SAE2002 - 11º Congresso e Exposição Internacionais de Tecnologia da Mobilidade - SAE*, São Paulo.

Prodonoff, V., 1990, *"Mechanics Vibrations – Simulation and Analysis"*, Ed. Maity.

Solidworks, 2002, *"User Guide"*.

Timoshenko, S., Young, D.H. and Wevar Jr., W., 1974, *"Vibration Problems in Engineering"*, John Wiley and Sons.

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