THE MULTI-BODY MODELING AS A DEVELOPMENT TOOL IN VEHICULAR PROJECTS

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Abstract. *This paper relates the importance of multi-body modeling as a development toll in vehicular projects. The automotive industry, due to the economy globalization process, is searching for time and cost reductions. This optimization of time and costs must be reached for all vehicle life cycle stages, from the vehicle design phase until its disposal. A particularly expensive process in Brazilian carmaker factories is the "Tropicalization" procedure - vehicle projects designed in the carmakers headquarters have to be carefully adapted to local conditions in order to attend comfort and ride requirements. We firstly present a common procedure used to model multi-body systems and some commercial software available. Then we described the Road roughness model used and focused on ADAMS® software and a passenger car case study. Finally the results obtained are presented and commented. The results obtained were encouraging and proved the importance of multi-body modeling as a development toll in vehicular projects.*

Keywords: *multi-body modeling, automaker industries, cost reduction, time reduction.*

1. Introduction

Today's advances in machine design require higher operation speeds and greater precision. To achieve these goals, a significant portion of mechanical engineering effort has been spent on developing more accurate, powerful and efficient analysis tools. During the last decade, although mathematical tools for analysis have experienced a tremendous growth, most research in multi-body dynamics was based on the assumption that systems are composed of rigid bodies or that ones whose elastic deformation is insignificant. In spite of the fact that all mechanical systems are actually more or less flexible, many multi-body systems have been modeled successfully by using the rigid body assumption.

An example of such models can be seen in the automotive industry recent increased emphasis on the control of a coil spring's load axis to reduce side force in a suspension system, resulting in improved ride and comfort (Costa *et al*., 1993, Yi, 2000, Nishizawa, *et al.*, 2002, and others). Another use of multi-body modeling analysis is the analysis of vehicle controllability and stability, where the behavior of the suspension determines the orientation and position of the tires and the deformation of the vehicle body that supports the suspension. In order to improve vehicle controllability and stability, clarification of the relationship between these components is indispensable, including the experimental measurement of body deformation. However, it is conventionally impossible to measure the diverse components of small body deformation in detail during road tests (Isomura, *et al.*, 2001).

The situation in Brazil is no different, but the fact that car projects are designed in the carmakers headquarters and only assembled in local factories calls for a process well-known in South America as "*Tropicalization*", where components and suspension parameters are carefully adjusted to the conditions of local highways and roads in order to improve comfort and handling. During many years this process had to depend on the opinion of expert pilots and engineers aiming for determining which parameters should be changed, and how. Recently, due to the great urge for time and cost optimizations, engineering teams involved in this optimization process have to adopt multi-body systems in order to reduce time and costs through automation.

For these reasons, the study of multi-body dynamics has become a challenging and most sought after research topics in several areas of manufacturing industry, including automotive, defense, aerospace, aircraft, off-highway equipment, mechatronics, biomechanics and many others.

2. Multi-body Modeling

The main reason for investments in mathematical models that represent, even if not in an entirely accurate form, the dynamic behavior of vehicles is that such models can predict this behavior without the need of real prototypes. These models help the project team from the beginning of a vehicle development until the last fine adjustments, even after the pre-series prototype construction (Biasizzo, 2001). In the case of dynamic simulations, vehicles are usually modeled by multi-body systems. The vehicle is divided into different subsystems, such as the vehicle framework and subsystems for the steering system and the drive train (Rill, 2005). For each of the vehicle subsystems there is the need to write the corresponding equations, observing free body motions, modal coordinates, generalized coordinates and applied forces and torques.

After the relevant parts of the vehicle are chosen and modeled, it is necessary to describe them schematically, specifying all the components and subsystems. Each subsystem is represented by chassis fixed points and by the geometric location of each of the components, with their respective values for mass, inertia momentum, stiffness, and other component features. Taking a suspension subsystem as an example, the basic mass-spring-damper model can be used to represent ¼ vehicle model, with two degrees of freedom - 2 DOF, see Fig. 1. In this case, the vehicle system input is the road roughness represented by h and located at the model's base. The vehicle sprung and chassis mass are represented by the M_s mass, while the spring and the damper elements represent stiffness (K_s) and the damping (C_s) of the suspension system. The tire is considered as an unsprung mass (m_u) and also as stiffness (K_p) and damping (C_t) elements (Biasizzo, 2001).

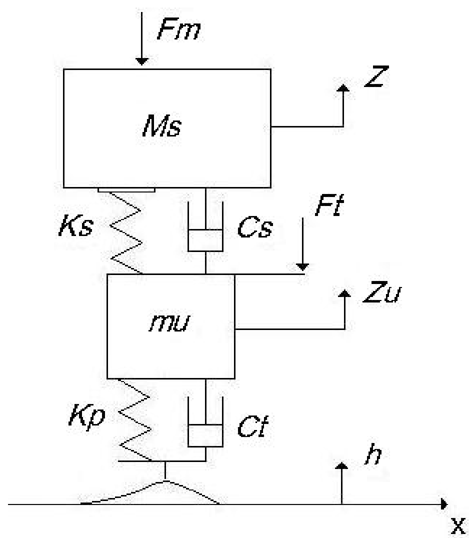


Figure 1. Mass-spring-damper system.

Applying the second Newton law for a dynamic system represented by the 2 DOF model shown in Figure 1, we obtained the following system of differential equations:

$$\begin{aligned} &Ms.\frac{dZ^2}{d^2t}+Cs.\left(\frac{dZ}{dt}-\frac{dZu}{dt}\right)+Ks.(Z-Zu)=Fm \\ &mu.\frac{dZu^2}{d^2t}+Cs.\left(\frac{dZu}{dt}-\frac{dZ}{dt}\right)+Ks.(Zu-Z)+Ct\left(\frac{dZu}{dt}-\frac{dh}{dt}\right)+Kp(Zu-h)=Ft \end{aligned} \tag{1}$$

Rewriting the second equation results in:

$$\begin{aligned} &Ms.\frac{dZ^2}{d^2t}+Cs.\left(\frac{dZ}{dt}-\frac{dZu}{dt}\right)+Ks.(Z-Zu)=Fm \\ &mu.\frac{dZu^2}{d^2t}+Cs.\left(\frac{dZu}{dt}-\frac{dZ}{dt}\right)+Ks.(Zu-Z)+Ct.\frac{dZu}{dt}+Kp.Zu=Ft+Ct\frac{dh}{dt}+Kp.h \end{aligned} \tag{2}$$

The equation system above can be rewritten in the matricial form as follows:

$$\begin{bmatrix} Ms & 0 \\ 0 & mu \end{bmatrix} \cdot \begin{bmatrix} \frac{dZ^2}{d^2t} \\ \frac{dZu^2}{d^2t} \end{bmatrix} + \begin{bmatrix} Cs & -Cs \\ -Cs & Cs+Ct \end{bmatrix} \cdot \begin{bmatrix} \frac{dZ}{dt} \\ \frac{dZu}{dt} \end{bmatrix} + \begin{bmatrix} Ks & -Ks \\ -Ks & Ks+Kp \end{bmatrix} \cdot \begin{bmatrix} Z \\ Zu \end{bmatrix} = \begin{bmatrix} Fm \\ Ft + Ct.\frac{dh}{dt} + Kp.h \end{bmatrix} \tag{3}$$

Where: M_s is the sprung mass;
m_u is the unsprung mass;
C_s is the suspension system damping;
C_t is the tire damping;
K_s is the suspension system stiffness;
K_p is the tire stiffness;
F_m is the reaction force from the sprung mass due to the road roughness;
F_t is the reaction force from the unsprung mass due to the road roughness;
Z is the sprung mass displacement;
Z_u is the unsprung mass displacement;
h is the road roughness.

Observing the 2 DOF Model used (Fig. 1) and Eq. (3) it is possible to conclude that the system model becomes more and more complex as the model DOF grows. For this reason, it is interesting to use some commercial software capable of analyzing all of these components and DOFs in a simple way, in order to obtain a more accurate behavior of a real system. This may be achieved through commercial simulation packages that implement multi-body models composed of both rigid and flexible parts, such as ADAMS (MSC – USA), AutoSim (Mechanical Simulation Corp. – USA), DynaFlex (Waterloo – Canada), MECANO (Samtech- Belgium), RecurDym (Function Bay Inc. – Korea) and many others.

3. Road Roughness Model

Road roughness includes everything from potholes resulting from pavement failures to the ever-present random deviations reflecting the practical limits of precision to which the road surface can be constructed and maintained. Roughness is described by the elevation profile along the wheel tracks over which the vehicle passes. As road profiles fit the broadband signals general category, they can be described either by the profile itself or by its statistical proprieties. In automotive engineering literature, the most applied statistical propriety is the Power Spectral Density function - PSD (Gillespie, 1992). The PSD describes how the power (or variance) of a time series is distributed in frequency domain. Mathematically, it is defined as the Fourier Transform of the autocorrelation sequence of the time series. The road roughness was simulated combining a finite number of harmonic components and a random phase generator, whose parameters (amplitudes and frequencies) were defined by the road PSD (Cardoso and Marczak, 1995). Equation (4) shows the road elevation profile temporal signal (h(t)):

$$h(t) = \sum S_m.\sin(\varpi_m.t + \theta_m) \tag{4}$$

Where: θ_m: m-th frequency random phase angle ($0 \sim 2\pi$);
ω_m: m-th signal frequency [rd/sec];
S_m: road elevation profile amplitude [mm].

$$S_m = \sqrt{2.S(\varpi_m).\Delta\varpi} \tag{5}$$

Where: $\Delta\omega$: frequency step [rd/sec];
$S(\omega_m)$: PSD value @ ω_m.

The $S(\omega_m)$ value was considered constant during $\Delta\omega$. According to Neto and Prado (1998), the $\Delta\omega$ value was obtained as a function of the simulation time (Eq. (6)). The series number of terms was defined by the Equation (7) based on the highest desired frequency (ω_H) to build the signal. According to Fenton (1996), an automobile first body flexible mode is close to 20 Hz. Due to this the 20 Hz frequency was considered as the highest desired frequency because the model did not consider body flexible modes. Consequently, the model cannot represent any 20 Hz higher values. The PSD values were obtained based on MIRA (*Motor Industry Research Association*) data. Table 1 shows the ω_1, ω_2 and G_0 values according to MIRA for various road types. The Equations (8) and (9) present an analytical model proposed by MIRA to determine typical road PSDs:

$$\Delta\varpi = \frac{2\pi}{T} \tag{6}$$

$$N = \frac{\varpi_H}{\Delta\varpi} \tag{7}$$

$$S_\gamma = G_0 \cdot \left(\frac{\gamma}{\gamma_0}\right)^{-\varpi_1} \quad \text{for } \gamma \le \gamma_0 \tag{8}$$

$$S_\gamma = G_0 \cdot \left(\frac{\gamma}{\gamma_0}\right)^{-\varpi_2} \quad \text{for } \gamma > \gamma_0 \tag{9}$$

Where: S_γ: PSD Amplitude [m^2/cycles/m];
γ: Wave number [cycles/m];
G_0: Road Roughness Level [m^2/cycles/m];
$\gamma_0 = 1/2\pi$: Cutoff Wave number that determines the model inflexion point [cycles/m];
ω_1: Dimensionless Exponent for Wave Numbers less than γ_0;
ω_2: Dimensionless Exponent for Wave Numbers greater than γ_0;

Table 1 – MIRA parameters.

Road Classification		G_0 (10^{-6} m^{-3}/ cycle)	ω_1		ω_2	
			Mean	Standard Deviation	Mean	Standard Deviation
Railway	Excellent	2 – 8	1.945	0.464	1.360	0.221
	Good	8 – 32				
Main Streets	Excellent	2 – 8	2.05	0.487	1.440	0.266
	Good	8 – 32				
	Average	32 – 128				
	Inferior	128 – 512				
Secondary Streets	Average	32 – 128	2.28	0.534	1.428	0.263
	Inferior	128 – 512				
	Rough rod	512 – 2048				

4. ADAMS®

The ADAMS® (*Automatic Dynamic Analysis of Mechanical Systems*) package was developed to test and analyze the behavior of some kinds of manufacturing industry, including automotive, defense, aerospace, aircraft, off-highway equipment, mechatronics, biomechanics and many others. It works as a virtual prototype environment using model simulation and integrates modeling, analysis and visualization. It utilizes the multi-body system theory, allowing these bodies to be rigid or flexible, in order to create and analyze mathematical models. It also has the ability of importing and converting CAD files, which enable the use of geometries created in design oriented software packages, and the use of flexibility, through the Finite Elements Method (FEM). The main objective of this software is to enable the engineering team to develop and test complete virtual prototypes in a much shorter time and with a much shorter budget, enhancing the quality of the development process and reducing the need for actual physical prototypes. The ADAMS® package has two basic modules: ADAMS/Solver, which is the module responsible for solving the differential equations that compose the model, and ADAMS/View, pre and post processor, which has a graphic user interface and facilitates the creation of models and the interpretation of results obtained through ADAMS/Solver. There are also some more specific modules, like ADAMS/Car, for analyses of car models, ADAMS/Aircraft, for aircraft models and components, among a number of others.

5. Example of Application

A nice example of the application of multi-body tools is the previous work developed by Biasizzo (2001). He analyzed the suspension system of a real passenger car, developing a mathematical model. The modeled passenger car has a MacPherson front suspension, frontal anti-roll bar and a trailing arm rear suspension. Its model was implemented using the ADAMS package (version 10.0) and the entire model illustration is presented in Figure 2. The model consists of 25 parts, 31 restrictions (27 joints and 4 forced movements), 23 flexible connections (12 springs, 8 bars, and 3 bushed pivot links).

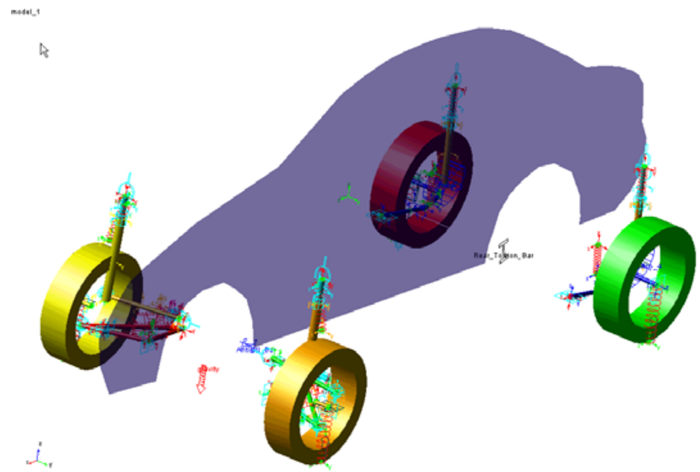
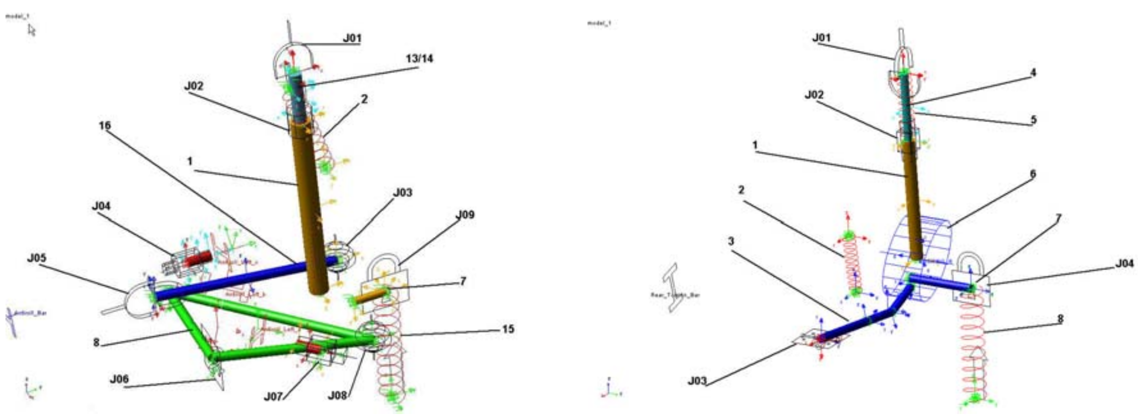


Figure 2. ADAMS vehicle model view

The MacPherson front suspension was modeled as 18 parts, 18 joint elements, 8 spring elements, and 7 bar elements. The anti-roll bar is modeled as 8 parts and 7 bar elements. On Figure 3-a each bar element is illustrated as an I-bar section icon and two of the parts are represented as red cylinders close to J04 and J08 joints (these joints show respectively the linkages between the anti-roll bar, the car body, and the A-arm). The trailing arm rear suspension was modeled as 8 parts, 8 joint elements, 8 spring elements, 2 bushing elements, and 1 bar element. Figure 3-b presents the trailing arm rear suspension model (the vertical tire stiffness is connecting the set A-arm, wheel, and tire to the ground (road surface).



(a) (b)

Figure 3. Front suspension model (a) and Rear suspension model (b).

Two road models were developed using the MIRA parameters. The first model was applied to simulate the vehicle in a smooth road (Good Main Street) at 100 Km/h and the other one, in a rough road (Inferior Main Street) at 40 Km/h. The road roughness inputs to rear wheels were the same ones to front wheels (delayed by a certain amount of time that represents the relation between wheelbases and vehicle speed). Each ADAMS® spring element used to model the 4-tire stiffness (see Figs. 2-b and 4) has an ADAMS® motion restriction. This ADAMS® restriction imposes predetermined translation or rotation motions. Vertical displacements were applied to these motion restrictions in order to introduce the road roughness in the model.

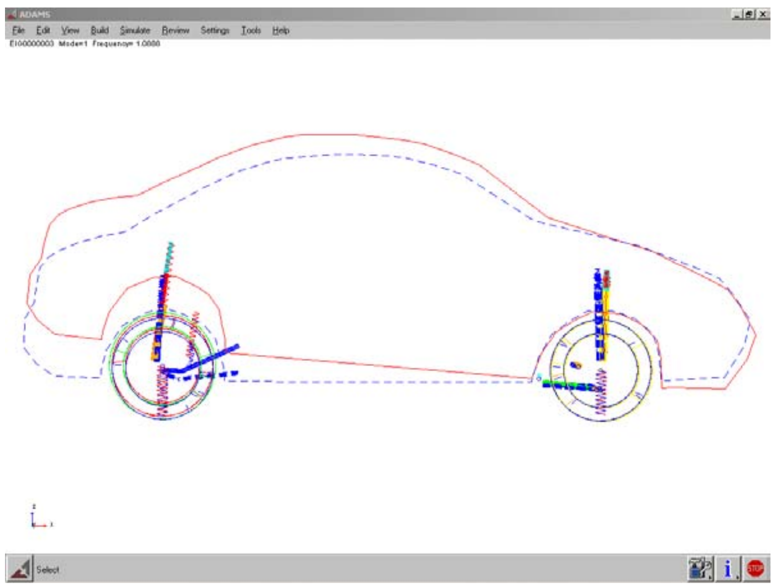
And, based on this model, some dynamic analysis tests were performed, giving the temporal solution of the values of translation, speeds, accelerations and internal forces caused by the road roughness where the vehicle runs.

6. Results

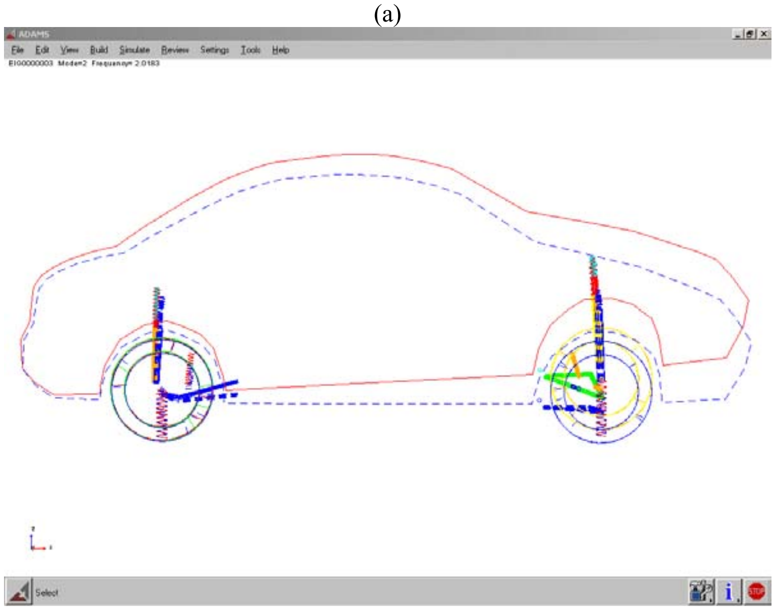
The vehicle dynamic behavior can be explained by the correlation between inputs and outputs, where inputs are any kind of external forces (or combination of forces) or displacements caused by the road roughness and the outputs are obtained through the chassis acceleration (Biasizzo, 2001). In order to analyze the results obtained using the ADAMS® vehicle model, we decided to analyze the chassis vertical acceleration frequencies. This is a common procedure because these frequencies are the most important and largely used parameters in the study of the vehicle comfort and ride behaviors.

6.1. Natural Modes of Vibration – Mode Shapes

The obtainment of natural modes is of great utility in order to adjust and correct some of the model parameters, for example, constraint positions and types. These mode validations prove that the vehicle model is capable of representing the real vehicle. Figure 4 shows the first mode of vibration, whose Pitch parameter is predominant, and the second one, where there is a predominance of Bounce.



(a)



(b)

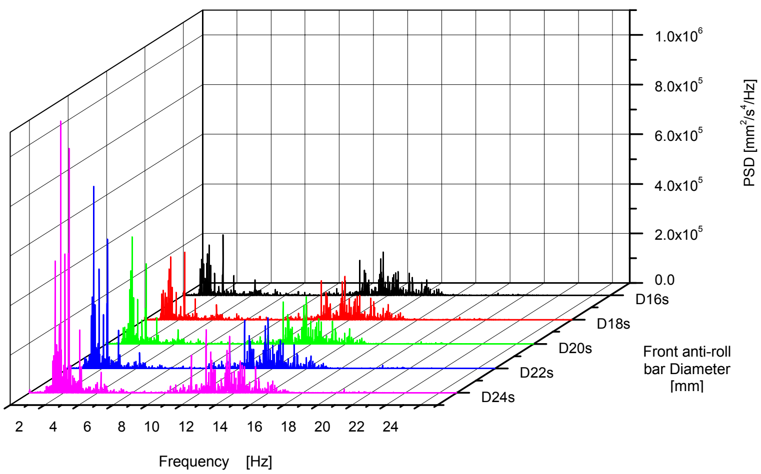
Figure 4. Vehicle body mode shapes

The chassis modes shown in Figure 4 show that Pitch and Bounce are coupled. It happened because the vehicle dynamic index is greater than the unity (Gillespie, 1992). It is possible to notice in Figure 4-a the predominance of the Pitch mode with the repercussion center positioned near the front suspension. On the other hand in Figure 4-b there is the presence of both Pitch and Bounce modes, with a predominance of Bounce with the repercussion center behind the rear suspension

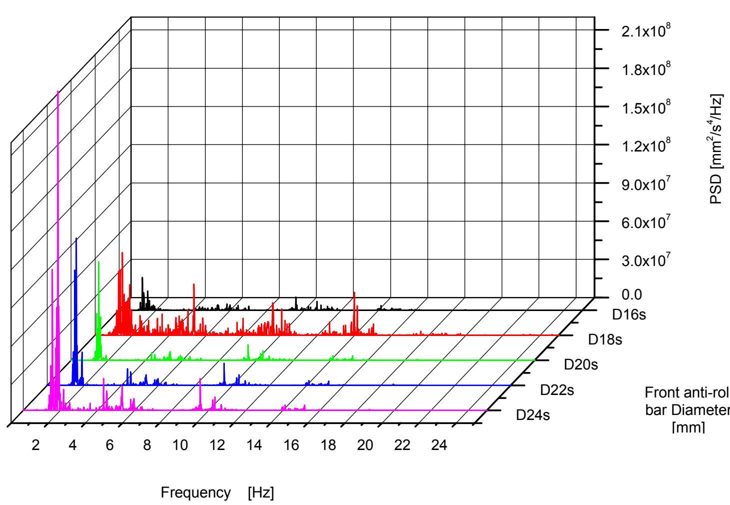
5.2. Chassis Frequency Response

Figure 5 shows the PSDs charts of vertical chassis acceleration for five different anti-roll bar diameters, using smooth and rough roads and different vehicle speeds as input signals. This procedure was used in order to analyze the influence of the front anti-roll bar on vehicle vertical accelerations. The chart in Fig. 5-a shows the general behavior of chassis response to external forces coming from the irregularities of a smooth road, with a vehicle speed of 100km/h, and the one in Fig. 5-b shows the vehicle body behavior for a rough road and vehicle at 40 km/h.

One may observe in Fig. 5-a that as the anti-roll bar diameter increases, the vibration transmission to the chassis increases as well. This behavior has been predicted, as the analysis of the natural modes has shown the enhancing of resonance frequencies in the bounce modes of the chassis.



(a)



(b)

Figure 5. Vehicle body vertical acceleration PSD, on smooth road and vehicle @ 100 km/h (a) and on a rough road vehicle @ 40km/h (b).

The chart on Fig. 5-b shows a similar behavior when compared to Fig. 5-a. The chassis acceleration peak values occur between 1.3 Hz and 2.5 Hz. Another peak value due to unsprung mass is observed at 10 Hz. The great suspension system displacements caused by the road roughness provoke the springs to act more effectively, transmitting a greater load to the damping towers when compared to the vehicle driving on smooth roads.

6. Conclusion

This paper presented the importance of multi-body modeling procedure as a design tool of vehicular projects. The software packages used to achieve this propose, like the ADAMS® package, tend to make the dynamic response analysis of some vehicle mechanisms, specially the suspension set, effortless, after the vehicle model validation.

As an example of application, it was shown the study of suspension systems behavior, analyzing the influence of stiffness variation on the front anti-roll bar on the vertical vibration of the vehicle chassis, which is caused by the road roughness. Using the multi-body modeling procedure it was possible to obtain satisfactorily the vehicle natural modes and the frequency output of chassis accelerations. The results obtained were coherent with information found in literature and the experimental data provided by the Fiat automotive industry.

Concluding, the proposed procedure can reduce drastically the time-to-market and developing costs for the carmakers industries, in particular when adapting a vehicle to the local environmental and road conditions ("*tropicalization*").

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8. Responsibility notice

The authors are the only responsible for the printed material included in this paper.