AN EDUCATIONAL TOOL TO STUDY THE VEHICLE DYNAMIC RESPONSE DURING RIDE ANALYSIS

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Abstract. *An automotive vehicle has thousands of components, but when performing most elementary analysis one may consider that all components move together. During acceleration, breaking, and change direction maneuvers the vehicle size can be ignored and only its mass center behavior need be considered when studying its motion. However, it is a common procedure to model separately the vehicle body and its wheels when performing ride analysis. Special attention must be taken concerning the road model. This model should be able to represent reliably the road real features. This paper presents a 2-D didactical toll development for vehicle dynamic response study. It is possible to study the influence of the sprung and unsprung mass and front and rear suspensions on the vehicle behavior. The softwareis still under development and at this moment it was developed using MatLab language. Tthe results obtained were promising.*

Keywords*: multi-body modeling, vehicle dynamic response, educational tool, ride analysis.*

1. Introduction

The automotive industry is today highly competitive, globalized, and characterized by continuous efforts to improve its products quality and integrity. It is well known that traditional design approaches improve the product quality and integrity by making extensive product in-use tests. Some people believe that the only way to get a proper adjusts for the performance, handling, ride and comfort of a new car is to drive it. But if we adopt this method for improving automotive industry products, we will carry on several prototypes on-road tests. On-road testing of prototype vehicles can be very expensive, not only due to the vehicles themselves, but also because they are composed of many prototype subsystems. If one of these subsystems fails, the other subsystems cannot be tested without it. On the other hand, the manufactories are under pressure to reduce their product costs and time-to-market. Solutions to this paradox do not permit trial-and-error, necessitating instead the adoption of a new, more complex developmental paradigm. In this scenario the term “road-to-lab-to-math” describes the effort to reduce the quantity of on-road testing and replaces it with laboratory testing components and subsystems, and to so efficiently by using complex mathematical models that make evaluation of in-use conditions more precise and realistic.

By bringing the vehicle subsystems and components to the laboratory and using some virtual prototyping tools, the interdependence on the others subsystems is eliminated. For example, it is possible to evaluate a prototype transmission without the need of a prototype engine attached to it, so engine problems would not affect the transmission test schedule. It is also possible to evaluate which suspension parameters should be changed in order to adjust the vehicle ride behavior.

There are several commercial software packages developed to support the “road-to-lab-to-math”. If we focus on virtual prototyping tolls applied to vehicle dynamic responses, all of the commercial simulation packages implement multi-body models composed of both rigid and flexible parts. It is possible to find on web several examples of these packages, such as ADAMS (MSC – USA), AutoSim (Mechanical Simulation Corp. – USA), DynaFlex (Waterloo – Canada), MECANO (Samtech- Belgium), RecurDym (Function Bay Inc. – Korea) and many others. Frequently these packages use a multi-body system approach to obtain the vehicle dynamic responses during maneuvers.

Most of the commercial software packages are prohibitively expensive for mechanical engineering students to buy them. In addition to this, usually these software packages do not have tools to permit more complex analysis, using for example Finite Element Method (FEM) and Operating Deflection Shapes (ODS). Due to this, we decided to develop educational software based on MatLab software that is dedicated to multi-body-based handling, ride and comfort analysis, and as well to FEM and ODS analysis. This article describes the part of our educational software developed to help mechanical engineering students to understand the vehicle dynamic responses during ride analysis.

2. Ride

According to Gillespie (1992), vehicles are exposed to a wide vibration spectrum when moving at high speeds. The vibrations are transmitted to the passengers by three main paths: tactile, visual, and aural. The term "ride" is popularly used in reference to tactile and visual vibrations. On the other hand, the aural vibrations are well known as "noise". The vibration spectrum may be divided up according to frequency and classified as ride (0-25 Hz) and noise (25-20 kHz). Nevertheless, the different vibration types are so interrelated that many times it is difficult to consider each one separately, i.e., noise is normally present when lower-frequency vibrations are excited.

In spite of the fact that vehicle ride vibrations may be excited by multiple sources, these sources frequently fall into two classes: road roughness and on-board sources. The on-board sources originate from the vehicle rotating components (e.g., tires, wheels, engine, and driveline). Otherwise, 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 (it is are usually measured by profilometers). 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). Figure 1 shows a typical PSDs of road elevation profiles for two different pavements. The green line represents a common Portland Cement Concrete road surface (rigid pavement) and the blue line, a bituminous asphalt surface (flexible pavement).

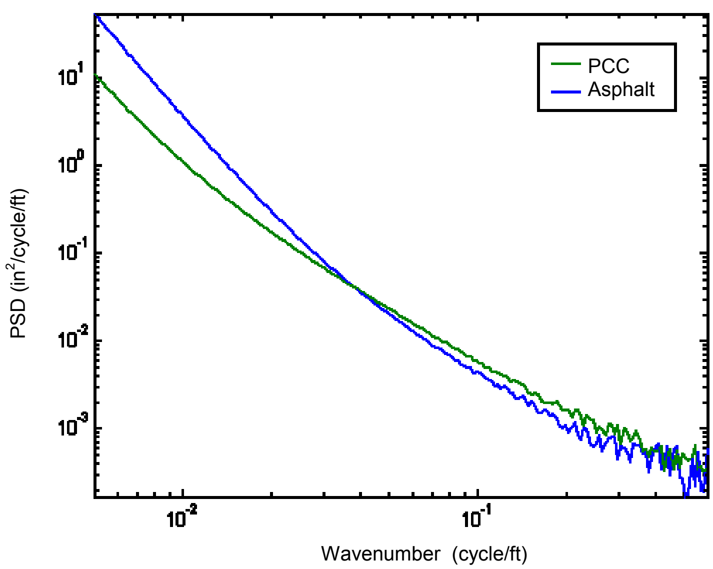

Figure 1. Typical PSDs of road elevation profiles for two different pavements.

Although the PSD of each road section is singular, all roads present a characteristic amplitude fall as the wavenumber increases. It makes apparent the fact that small wavenumber oscillations have small amplitude levels, while on the contrary, large wavenumber oscillations have large amplitude levels. The general PSD amplitude level indicates the road quality: as high is the amplitude, as rough is the road, and consequently, as worst is the road quality.

The average road proprieties PSD can be represented by the following equation (Gillespie, 1992):

$$G_z(\nu) = G_0 \frac{1 + \left(\nu_0 / \nu \right)^2}{(2\pi\nu)^2} \tag{1}$$

Where:

$G_z(\nu)$:PSD amplitude
ν: wavenumber

G_0: roughness magnitude parameter
ν_0: cutoff wavenumber

As previously mentioned, the roughness acts as a wheel vertical displacement input, thus exciting ride vibrations. If we consider that the vehicle is moving at a constant speed, it is possible to convert the road elevation profile to a time function displacement. The spatial frequency transformation to temporal frequency is carried out by multiplying the wavenumber by the vehicle speed.

3. Vehicle Modeling

The vehicle dynamic behavior is determined by its input road loads (aerodynamic force, rolling resistance, etc.) and the road roughness during maneuvers. During its design stage all vehicle components are developed in order to obtain a good compromise between its cost and performance (acceleration, braking, ride, handling, comfort, safety, etc.). Aiming to reach this goal, it is necessary to determine the vehicle exciting forces and reactions during maneuvers without ambiguities. As earlier cited on the introduction, the use of reliable vehicle models makes the vehicle evaluation of in-use conditions more precise and realistic, reducing the vehicle on-road test quantity, replacing it with laboratory testing components and subsystems, and reducing time-to-market and costs.

An automotive vehicle has thousands of components, but when performing most elementary analysis one may consider that all components move together. During acceleration, breaking, and change direction maneuvers the vehicle size can be ignored and only its mass center behavior need be considered when studying its motion (one should not to forget considering vehicle's mass and inertia moments). However, it is a common procedure to model separately the vehicle body and its wheels when performing ride analysis. In this case the vehicle's wheel mass is named unsprung mass and vehicle's body, sprung mass. The vehicle's suspension stiffness and damping proprieties are represented by a spring - shock absorber pair that connects both sprung and unsprung masses. Each tire is represented by a simple spring. The suspension mass is added to the unsprung mass and the engine and powertrain masses, to the sprung mass.

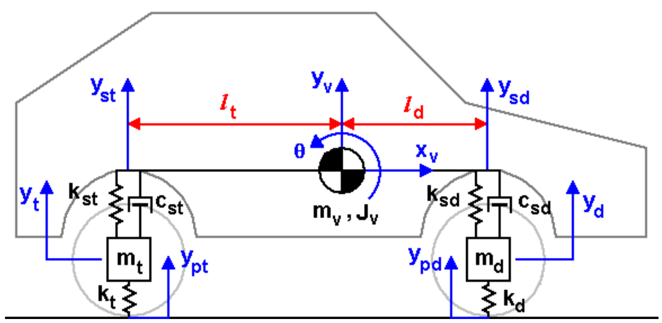

Figure 2. Vehicle model.

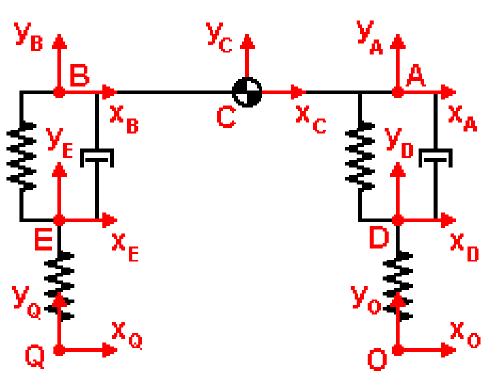

Figure 3. Vehicle model coordinate system.

Where:

l_d: longitudinal distance from front axle to center of gravity
l_t: longitudinal distance from rear axle to center of gravity
y_{pd}, y_{pt}: road vertical displacements with respect to O and Q, respectively
y_{rd}, y_{rt}: wheel vertical displacements with respect to D and E, respectively
y_{sd}, y_{st}: upper suspension vertical displacements with respect to A and B, respectively
y_v: vehicle vertical displacements with respect to C
θ: vehicle pitch angle around C

The vehicle sum of vertical forces is represented by:

$$\begin{aligned} &\sum F_y = m_v \ddot{y}_v \\ &m_v \ddot{y}_v = -k_{st}\left(y_{st} - y_{rt}\right) - c_{st}\left(\dot{y}_{st} - \dot{y}_{rt}\right) - k_{sd}\left(y_{sd} - y_{rd}\right) - c_{sd}\left(\dot{y}_{sd} - \dot{y}_{rd}\right) \end{aligned} \tag{2}$$

And the sum of torques around vehicle's mass center:

$$\begin{aligned} &\sum M_{CG} = J_v \ddot{\theta} \\ &J_v \ddot{\theta} = l_t k_{st}\left(y_{st} - y_{rt}\right) + l_t c_{st}\left(\dot{y}_{st} - \dot{y}_{rt}\right) - l_d k_{sd}\left(y_{sd} - y_{rd}\right) - l_d c_{sd}\left(\dot{y}_{sd} - \dot{y}_{rd}\right) \end{aligned} \tag{3}$$

The sum of vertical forces on front axle:

$$\begin{aligned} &\sum F_y = m_d \ddot{y}_{rd} \\ &m_d \ddot{y}_{rd} = -k_d\left(y_{rd} - y_{pd}\right) + k_{sd}\left(y_{sd} - y_{rd}\right) + c_{sd}\left(\dot{y}_{sd} - \dot{y}_{rd}\right) \end{aligned} \tag{4}$$

The sum of vertical forces on rear axle:

$$\begin{aligned} &\sum F_y = m_t \ddot{y}_{rt} \\ &m_t \ddot{y}_{rt} = -k_t\left(y_{rt} - y_{pt}\right) + k_{st}\left(y_{st} - y_{rt}\right) + c_{st}\left(\dot{y}_{st} - \dot{y}_{rt}\right) \end{aligned} \tag{5}$$

4. Software Development

All the software tool routines were developed using the MatLab language. For the purpose of facilitating the user interface and vehicle dynamic responses comprehension, a numerical model was implemented from the vehicle mathematical model. This numerical model identifies the vehicle modal responses from its mechanical proprieties. Figure 4 shows the interface initial window. This initial window allows the use to enter the vehicle's data.

Where:

l_d: longitudinal distance from front axle to center of gravity (m)
l_t: longitudinal distance from rear axle to center of gravity (m)
m_v: vehicle sprung mass (kg)
J_v: vehicle *pitch* moment of inertia (kg × m^2)
m_d: vehicle front axle unsprung mass (kg)
m_t: vehicle rear axle unsprung mass (kg)
k_d: front tire stiffness (kN/m)
k_t: rear tire stiffness (kN/m)
k_{sd}: front suspension stiffness (kN/m)
k_{st}: rear suspension stiffness (kN/m)
c_{sd}: front suspension damping coefficient (N × s/m)
c_{st}: rear suspension damping coefficient (N × s/m)

The software also allows the use to enter the road parameters and view its profile (Fig. 5). Using the numerical procedure the software allows the user the view the vehicle time and frequency responses for the specified road (respectively, Figs. 6 and 7).

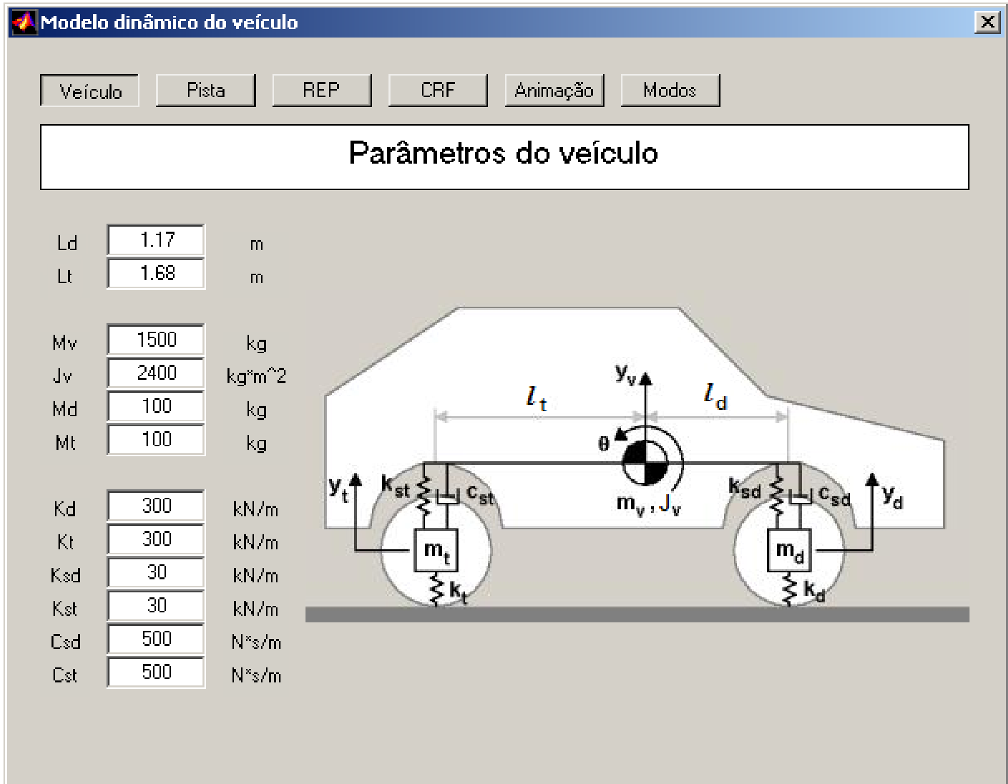

Figure 4. User interface initial window to enter vehicle parameters.

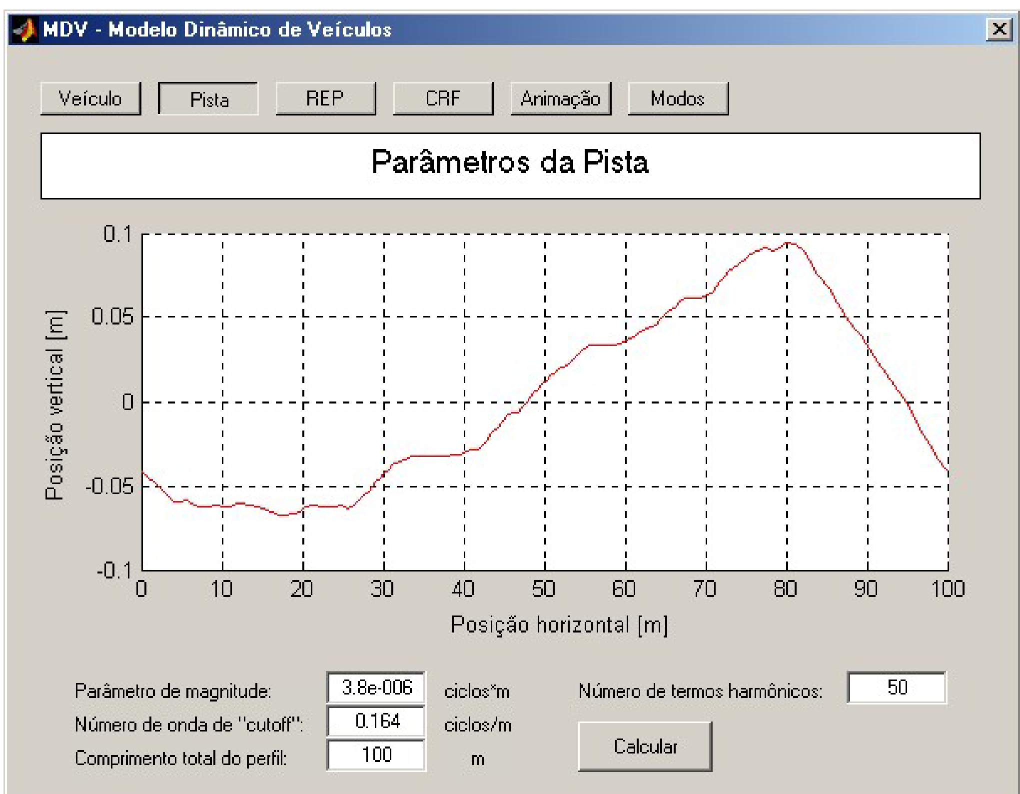

Figure 5. User interface window to enter road parameters.

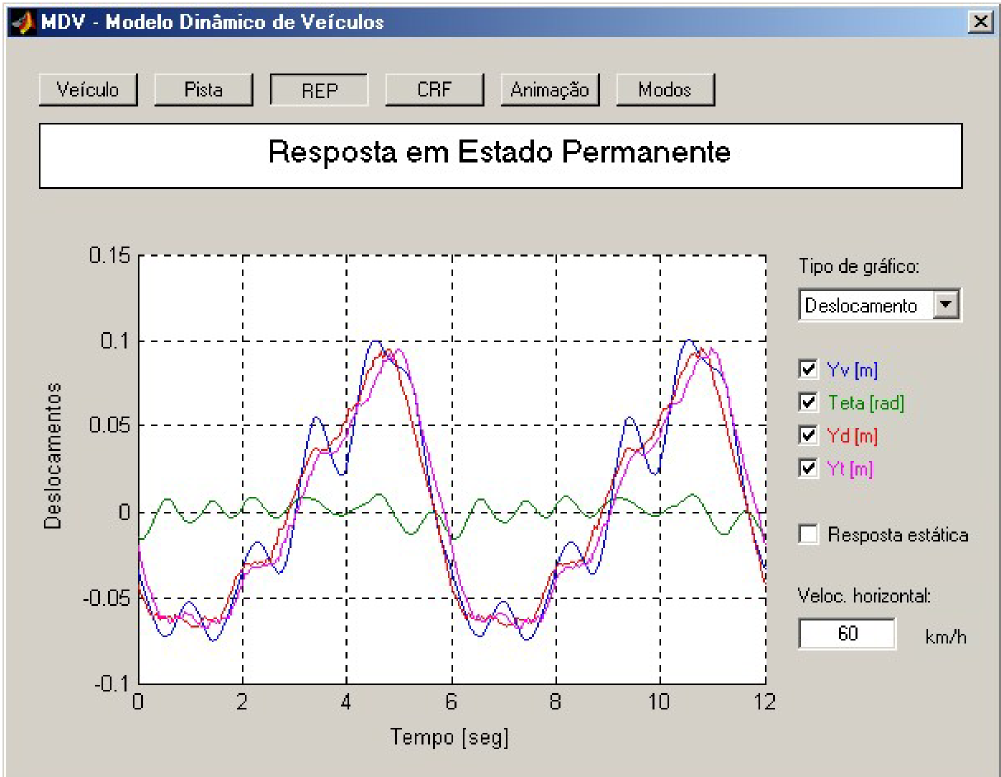

Figure 6. Vehicle displacement as a function of time.

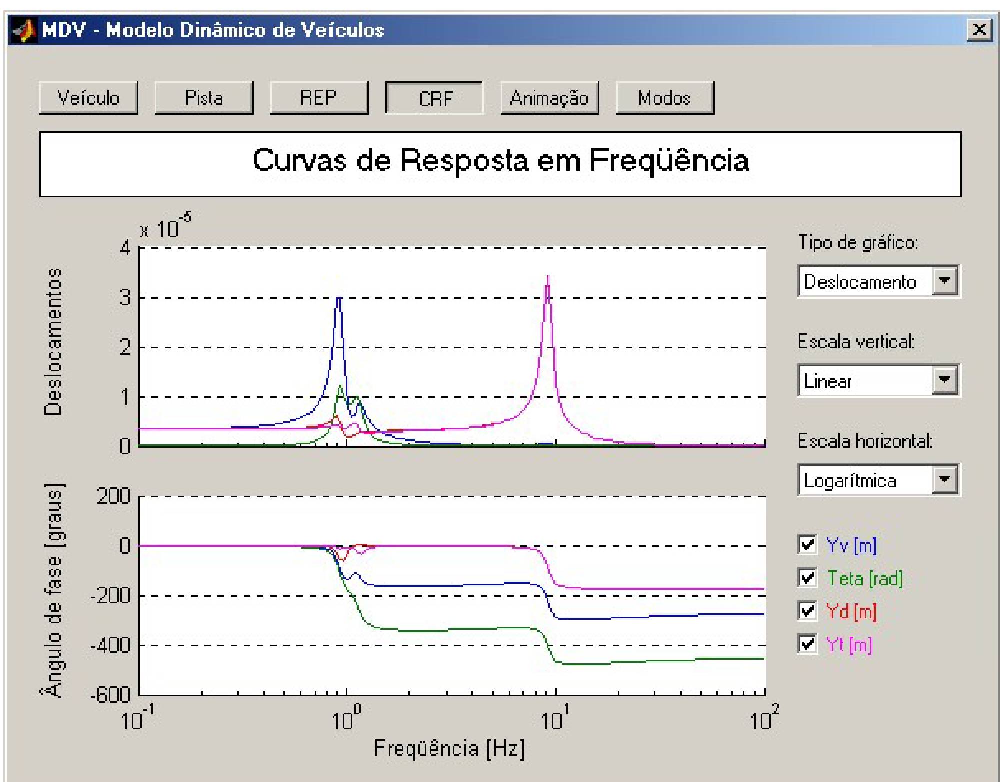

Figure 7. Vehicle displacement and phase angle as a function of frequency.

Where:

Yv: vehicle vertical displacements with respect to C (Fig. 3)
Teta (θ): vehicle pitch angle around C (Fig. 3)
Yd: front wheel vertical displacement
Yt: rear wheel vertical displacement

5. Results

Consulting the literature, we used the following vehicle data (Margolis and Shim, 2001) to simulate the vehicle behavior during ride analysis (Table 1):

Table 1. Vehicle parameters.

Parameter	Value
l_d	1.16 m
l_t	1.68 m
m_v	1500 kg
J_v	2400 kg × m^2
m_d	100 kg
m_t	100 kg
k_d	300 kN/m
k_t	300 kN/m
k_{sd}	30 kN/m
k_{st}	30 kN/m
c_{sd}	500 N × s/m
c_{st}	500 N × s/m

The road parameters used were based on Gillespie (1992). Table 2 shows the road parameters used to simulate the vehicle behavior.

Table 2. Road Parameters.

Parameter	Value
$G_z(\nu)$	3.8×10^{-6} cycles × m
ν_0:	0.164 cycles/m
Road length	100 m
harmonics	50

The software tool provides not only visual results on windows, but some charts as well, as represented on Tables 3 to 5, respectively, vehicle natural frequencies, vehicle modal damping, and vehicle static displacements.

Table 3. Vehicle natural frequencies.

1st	0.91072 Hz
2nd	1.1391 Hz
3rd	9.1438 Hz
4th	9.1443 Hz

Table 4. Vehicle modal damping.

1st	4.326 %
2nd	5.4042 %
3rd	4.4431 %
4th	4.4955 %

Table 5. Vehicle static displacements.

Front Springs	289 mm
Rear Springs	201 mm
Front Tires	32 mm
Rear Tires	23 mm

One may observe in Fig. 7 that the wheel displacements present a high-frequency component (Yd and Yt). Nevertheless, these components are relatively inexpressive if we consider the whole vehicle displacement (Yv and Teta). It happened because the vehicle suspension filtered the high frequencies components. Comparing Tab. 3 with Fig. 7, it is possible to verify that the graph peaks are in agreement with the obtained natural frequencies.

6. Conclusions

This paper presented the development of a vehicle dynamic modeling educational tool. This tool proved to be very useful to perform a preliminary analysis of the suspension stiffness and damping characteristics in order to improve the vehicle behavior on different road roughness. As we implemented the software tool using the MatLab language, many times the processing time expended was too large. Other software toll drawback was the fact that at this moment the vehicle model did not consider roll and yaw motions (we used a planar model). We are working on a new spatial model that would solve this drawback and allow the simulation of some more complex ride testing maneuvers.

Aiming the software improvement, we are also planning to translate the software code to C++ language and to use Open GL tool in order to improve the simulation view, making easier to view 3-D illustrations.

7. Acknowledgements

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8. References

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9. Responsibility notice

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