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THE IMPLEMENTATION OF AN EDUCATIONAL ODS TOOL USED TO VERIFY DYNAMIC VEHICULAR RESPONSES

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Abstract. *Operating Deflection Shapes is a technique used in structural dynamics to help understand how a structure deforms under operating conditions. ODS are used for visualization of the vibration pattern of the structure. Vibration measurements are performed at different points and directions (degrees of freedom, DOFs) on the structure and the vibration pattern can be shown as an animated geometry model. As ODS is related to the solution of forced vibration equation of motion, it is a result of the object properties and force characteristics. This paper presents an educational ODS toll development. This tool was developed using MatLab language and tested in an off-vehicle. The results were promising and proved the tool applicability for animating and visualizing chassis and component deflection modes at selected frequencies. The ODS software evaluates all spectra at the given frequency, determines the amplitude and phase of motion. The measurements are made; a computer program examines all the data and produces a series of animated 3-D pictures on the screen that shows the motion of the vehicle chassis at selected frequencies. The ODS analysis provides information to the designer about how to modify the structure to solve the vibration problem by pointing out the locations and directions in which the excessive motion is occurring.*

Keywords *Operating Deflection Shapes, dynamic vehicular response, educational tool, chassis motion.*

1. Introduction

The industry is today highly competitive, globalized, and characterized by continuous efforts to improve its products quality and integrity. Traditional design approaches improve the product quality and integrity by making extensive product in-use tests. But the adoption of this method for improving automotive industry products would carry on several prototypes on-road tests that would increase the production costs. This costs increment causes a paradox because today the manufactories are also 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 more precise and realistic of in-use conditions.

There are several commercial software packages developed to support the “road-to-lab-to-math”. If one 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. 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 that permit more complex analysis, using for example Finite Element Method (FEM) and Operating Deflection Shapes (ODS). Due to this, this paper focus the development of an educational software based on MatLab language that is dedicated to multi-body-based handling, ride and comfort analysis, and as well to FEM and ODS analysis (Saturnino, 2004).

The ODS tool is used for visualization of the vibration pattern of a structure under given operating conditions. Vibration measurements are performed at different points and directions (degrees of freedom - DOFs) on the structure and the vibration pattern can be shown as an animated geometry model of the structure or listed in a table. This article describes the part of the educational software developed to help mechanical engineering students to understand and visualize the vehicle chassis vibration under given operating conditions (for example, during ride analysis).

2. ODS - Operating Deflection Shapes

Vibration problems in structures, vehicles or operating machinery often involve the excitation of structural resonances, or modes of vibration. Many types of machinery and equipment can encounter severe resonance related vibration problems during operation. In order to diagnose these problems, an animated display of the machine, vehicle or structure's operating deflection shapes is often very useful. In most cases, structural responses at or near a resonant (modal) frequency are "dominated" by the mode, and the ODS closely approximates the mode shape. Due to this, Richardson (1997) affirms that ODS and modes of vibration are closely related.

Experimental modal testing (performing a modal survey) is usually done under controlled stationary (non-time varying) conditions, using one or more exciters. Furthermore, the excitation forces and their corresponding responses are simultaneously measured. In many cases, especially with vehicles and operating equipment, the measurement signals may be non-stationary (time varying) and the excitation forces cannot be measured. In these cases, the use of a controlled sine force to excite the structure is common. Due to this, different post-processing is required aiming to display ODS's from a set of measurements. In order to obtain the modes of vibration, the excitation force is applied on a specific place that is not a vibration mode node (the structure point that has no displacements) and the excitation frequency should be close to one of the resonance frequencies. In this situation, the ODS presents one vibration mode and shows the related structure displacement.

ODS applications are usually divided into three categories: visualization of the vibration pattern of a structure under given operating conditions (Pai and Young, 2001), structural fault detection (Waldron et al., 2002), and modal parameters acquisition without the need of an experimental modal analysis (Margolis and Shim, 2001, Parloo et al., 2002, and Han and Feeny, 2003).

ODS analysis can provide significant trouble-shooting and analysis advantages for vehicle manufacturers at much less expense than other methods, such as modal analysis. As advantages it is possible to cite:

- The measurement of force inputs are not required for ODS, but must be measured to perform modal analysis (an expensive additional task);
- ODS algorithms are straightforward and easy to execute, providing very quickly, an overall view of structural problem areas;
- ODS is not limited by a basic mathematical model.

Of course, modal analysis has its advantages as well, and one may elect to use one method to complement the other.

The experimentally acquired vibration data are frequently displayed on time or frequency domain charts. Unfortunately, these representations produce a poor visualization of the correlation that exists between all structure acquisition points. As a result of this, the use of 3-D animation procedures is very common in ODS studies.

In this work, the ODS tool was implemented together with a FE tool, following the static equilibrium condition for each time step t_i as it is written on Eq. (1):

$$[K]\{u(t_i)\} = \{r(t_i)\}\text{, for } i = 1,2,\ldots,T \qquad (1)$$

Where: K is the structural stiffness matrix;
$u(t_i)$ is the structural displacement vector;
$r(t_i)$ is the instantaneous force vector;
i is the i-th time step.

The instantaneous force and displacements vectors may be written as:

$$[R] = \begin{bmatrix} r_1(t_1) & r_1(t_2) & \cdots & r_1(t_T) \\ r_2(t_1) & r_2(t_2) & \cdots & r_2(t_T) \\ \vdots & \vdots & \vdots & \vdots \\ r_n(t_1) & r_n(t_2) & \cdots & r_n(t_T) \end{bmatrix} \qquad (2)$$

$$[U] = \begin{bmatrix} u_1(t_1) & u_1(t_2) & \cdots & u_1(t_T) \\ u_2(t_1) & u_2(t_2) & \cdots & u_2(t_T) \\ \vdots & \vdots & \vdots & \vdots \\ u_n(t_1) & u_n(t_2) & \cdots & u_n(t_T) \end{bmatrix} \qquad (3)$$

Where: n is the total acquisition point quantity.

Re-writing Eq. (1):

$$[K][U]=[R] \tag{4}$$

The degrees of freedom that matches the acquisition points present known displacements $[U]_b$, where: $[U]_b=\left[\{u(t_1)\}_b \quad \{u(t_2)\}_b \quad \cdots \quad \{u(t_T)\}_b\right]$ and T is the total quantity of time steps. The unknown displacements are represented by $[U]_a$, where $[U]_a=\left[\{u(t_1)\}_a \quad \{u(t_2)\}_a \quad \cdots \quad \{u(t_T)\}_a\right]$. The equilibrium equations can be divided as follows:

$$\begin{bmatrix} [K]_{aa} & [K]_{ab} \\ [K]_{ba} & [K]_{bb} \end{bmatrix} \begin{bmatrix} [U]_a \\ [U]_b \end{bmatrix} = \begin{bmatrix} [R]_a \\ [R]_b \end{bmatrix} \tag{5}$$

Considering that all external forces $[R]_a$ are null:

$$\begin{bmatrix} [K]_{aa} & [K]_{ab} \\ [K]_{ba} & [K]_{bb} \end{bmatrix} \begin{bmatrix} [U]_a \\ [U]_b \end{bmatrix} = \begin{bmatrix} [0] \\ [R]_b \end{bmatrix} \tag{6}$$

Taking the first equation set:

$$[K]_{aa}[U]_a+[K]_{ab}[U]_b=[0] \tag{7}$$

So:

$$[K]_{aa}[U]_a=-[K]_{ab}[U]_b=[\overline{R}]_a \tag{8}$$

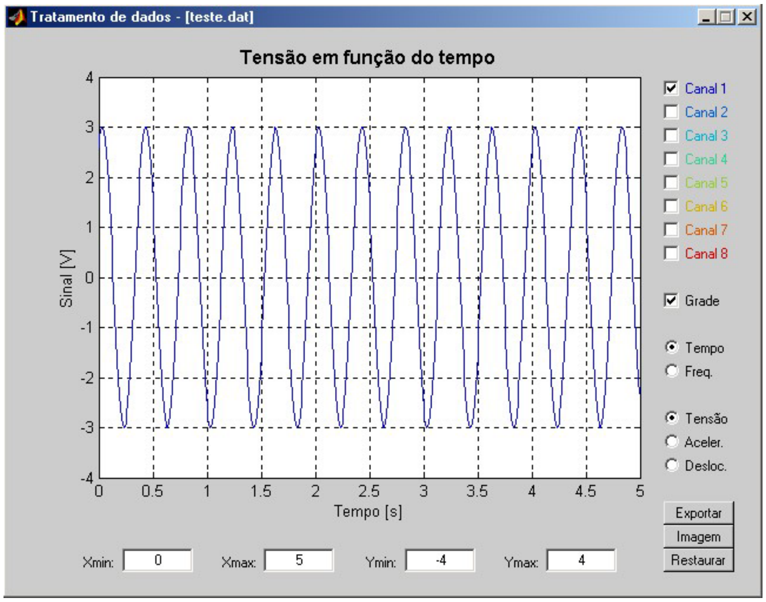
The displacement matrix $[U]_a$ is calculated by a proper algorithm for static solution case (Saturnino, 2004).

3. Software Development

The MatLab language was used to implement our educational tool due to its interface and graphic simplicity. All the previous equations were implemented. In order to facilitate the ODS results interpretation, a visualization tool was developed. The experimental data are acquired as a time function voltage signal and latter converted to frequency domain. This procedure is done using the MatLab Fast Fourier Transform function (*fft*). The visualization tool allows the user to obtain the acceleration and displacements as a frequency function and the displacements as a time function. Figure 1 shows the interface window. The data acquisition and treatment tool is able to use 8 data acquisition channels. Aiming to make the displacement responses analysis effortless, a finite elements graphic window was developed (Figs. 3 and 4). So, it is possible to use this tool to perform static and dynamic evaluations of mechanical structural elements.

4. Results

In order to show the software tool applicability, an instrumented off-road vehicle (Mini-Baja) was used. An on-board instrumentation / data acquisition system (DAS) was entirely developed. Eight accelerometers mounted on vehicle structure were used to acquire vehicle signals during maneuvers. The sensor positions were carefully chosen based on the *effective independence distribution vector* algorithm results (Atalla, 1996). Figure 1 shows the Data treatment tool interface window. Using this window, the software user may select the accelerometer channel and view the data that would be used for ODS procedure. It is possible to view the data as voltage signals, as displacements or as accelerations in time and frequency domain.



☑ Canal 1
☐ Canal 2
☐ Canal 3
☐ Canal 4
☐ Canal 5
☐ Canal 6
☐ Canal 7
☐ Canal 8

☑ Grade

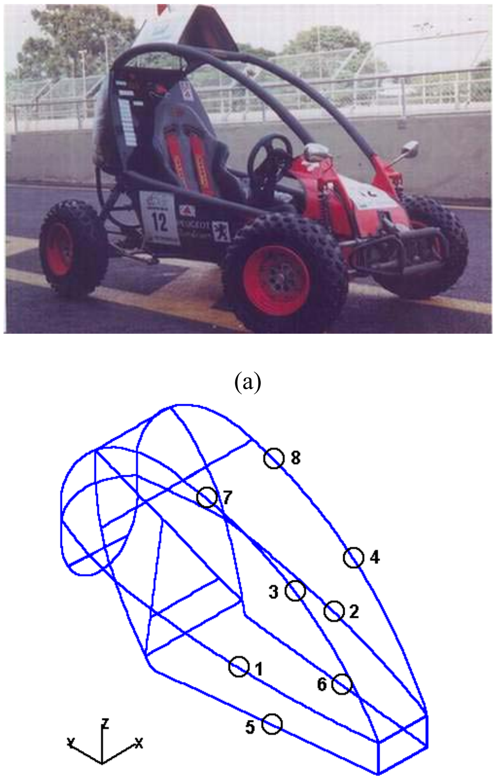
● Tempo
○ Freq.

● Tensão
○ Aceler.
○ Desloc.

Xmin: 0 Xmax: 5 Ymin: -4 Ymax: 4

Figure 1. Data treatment tool interface window.

Figure 2 shows the Mini Baja used to test our software tool (Fig. 2-a) and the accelerometer positions on vehicle frame (Fig. 2-b and Table 1). Eight DOFs were chosen between the 16 possible DOFs trying to keep the symmetry and obtain relevant results on X, Y and Z directions. Table 2 shows each DOF used.



(a)

(b)

Figure 2. Off-road vehicle photo (a) and the accelerometer positions on the vehicle structure (b).

Table 1. Accelerometer Positions.

Position	X [mm]	Y [mm]	Z [mm]
1	-155	773	302
2	477	773	302
3	-31	523	798
4	353	523	798
5	-68	639	0
6	390	639	0
7	-61	1076	1069
8	383	1076	1069

Table 2. DOF for each accelerometer Position.

Position	1	2	3	4	5	6	7	8
DOF	X	X	Y	Y	Z	Z	Z	Z

Firstly it was developed an entire vehicle model using the ALGOR software in order to determine its natural frequencies. The ALGOR results are presented on Table 3 and they are used to assist the experimental result analysis.

Table 3. Mini-Baja natural frequencies obtained using ALGOR software.

Natural Frequency	[Hz]
1st	30.807
2nd	44.975
3rd	51.668
4th	56.350
5th	56.804
6th	67.802
7th	68.326
8th	72.159
9th	88.356
10th	93.261
11th	93.261
12th	93.261
13th	93.261
14th	101.81
15th	102.50
16th	111.61
17th	114.06
18th	119.21
19th	137.14
20th	143.37

Then the data acquisition was carried out. Next figures show the Mini-Baja dynamic behavior for two different situations: disengaged gear and vehicle stopped (Figs. 3 and 4) and engaged gear and vehicle moving at constant speed (Figs. 5 and 6). These experimental results were used in order to produce the ODS analysis. It is possible to observe on Fig. 3 that the overall acceleration amplitude level for all accelerometer positions was close to ± 15 m/s^2, i.e. ± 1.5 g.

Analyzing the results on the frequency domain using the Fast Fourier Transform (Fig. 4), it is possible to observe that the vehicle acceleration amplitude peak value occur close to 29.3 Hz. This frequency corresponds to 1,758 rpm that is very close to 1,750 rpm (this rotation is the disengaged gear motor rotation according to the motor manufacture, Briggs & Stratton). The 14.65 Hz frequency corresponds to 29.3 Hz first sub-harmonic. The 29.3 Hz frequency presented significant acceleration amplitude because it was close to the vehicle first natural frequency (30.9 Hz), i.e. close to resonance.

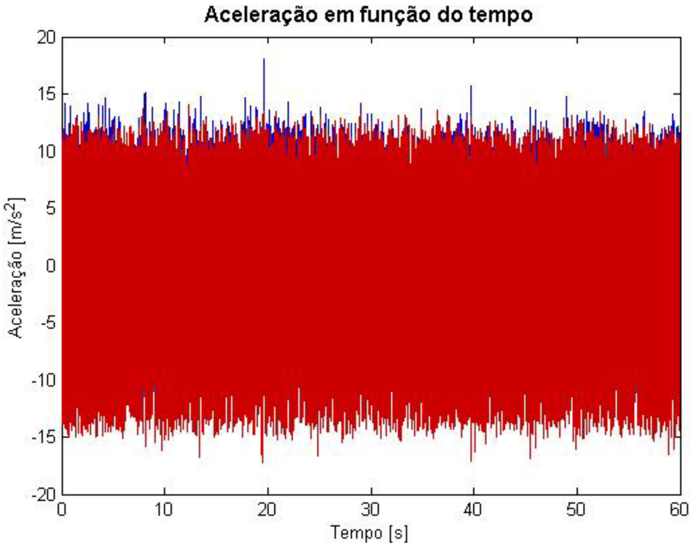


Figure 3. Vehicle acceleration in time domain – disengaged gear and vehicle stopped.

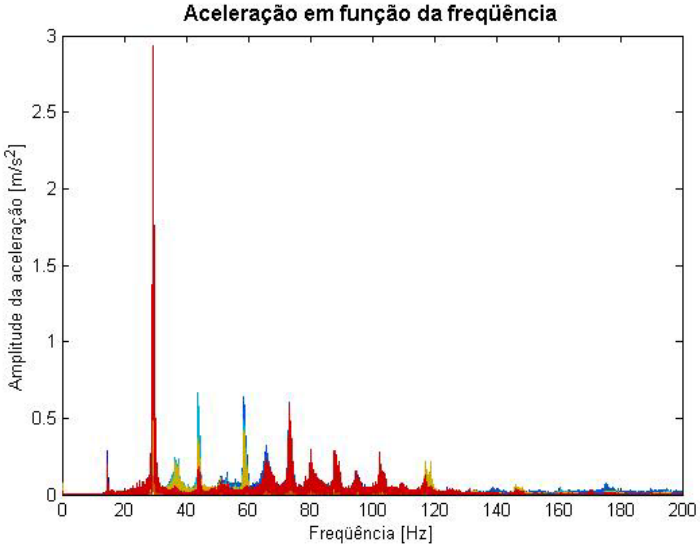


Figure 4. Vehicle amplitude acceleration in frequency domain – disengaged gear and vehicle stopped.

On the other hand, when the vehicle gear is engaged and it is moving at constant speed its dynamic behavior changes completely. It is possible to divide Fig. 5 into 3 different phases: during the first one, i.e. the first 15 sec., the vehicle was stopped; during the second phase, from 15 sec. until 52 sec., the vehicle was moving at constant speed; and the last phase, from 52 sec. until 60 sec., the vehicle was decelerating. The overall acceleration amplitude level for all accelerometer positions was less than ± 40 m/s^2, i.e. ± 4.0 g

Once again the frequency domain analysis provided more relevant results. One may observe acceleration amplitude peak values close to 30 Hz and 45 Hz that correspond to the two first vehicle structure natural frequencies. Another frequency peak is observed close to 90 Hz that is close to 9th and 10th natural frequencies (see Tab. 3).

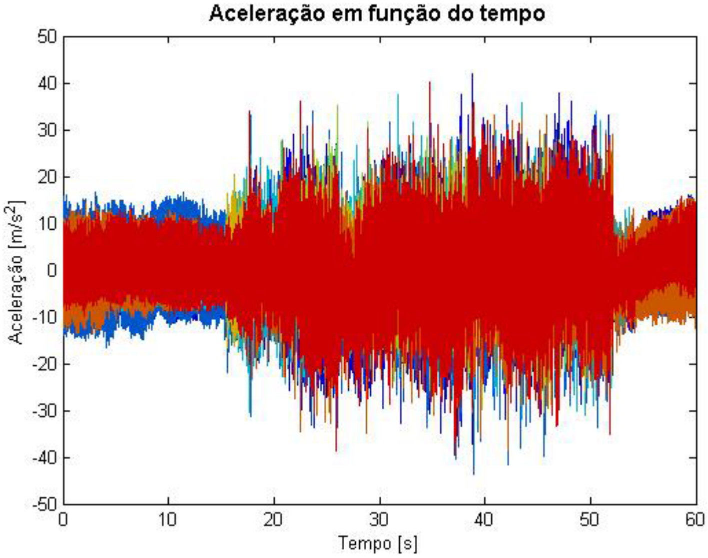


Figure 5. Vehicle acceleration in time domain – engaged gear and vehicle moving at constant speed.

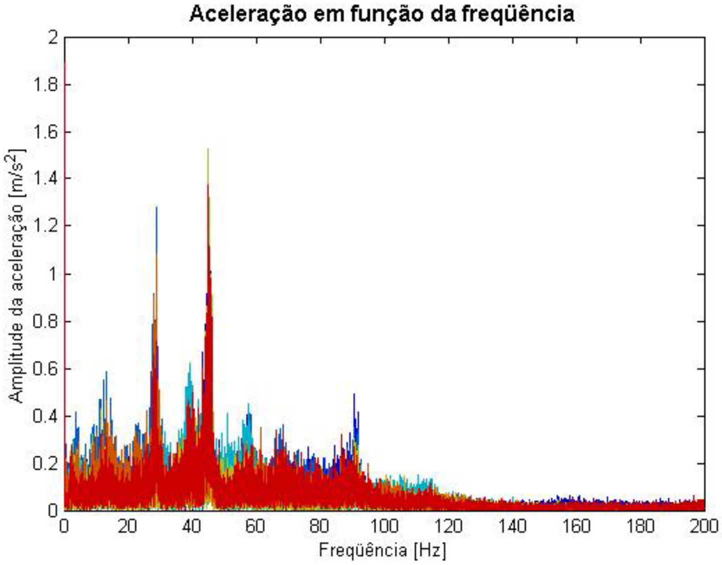
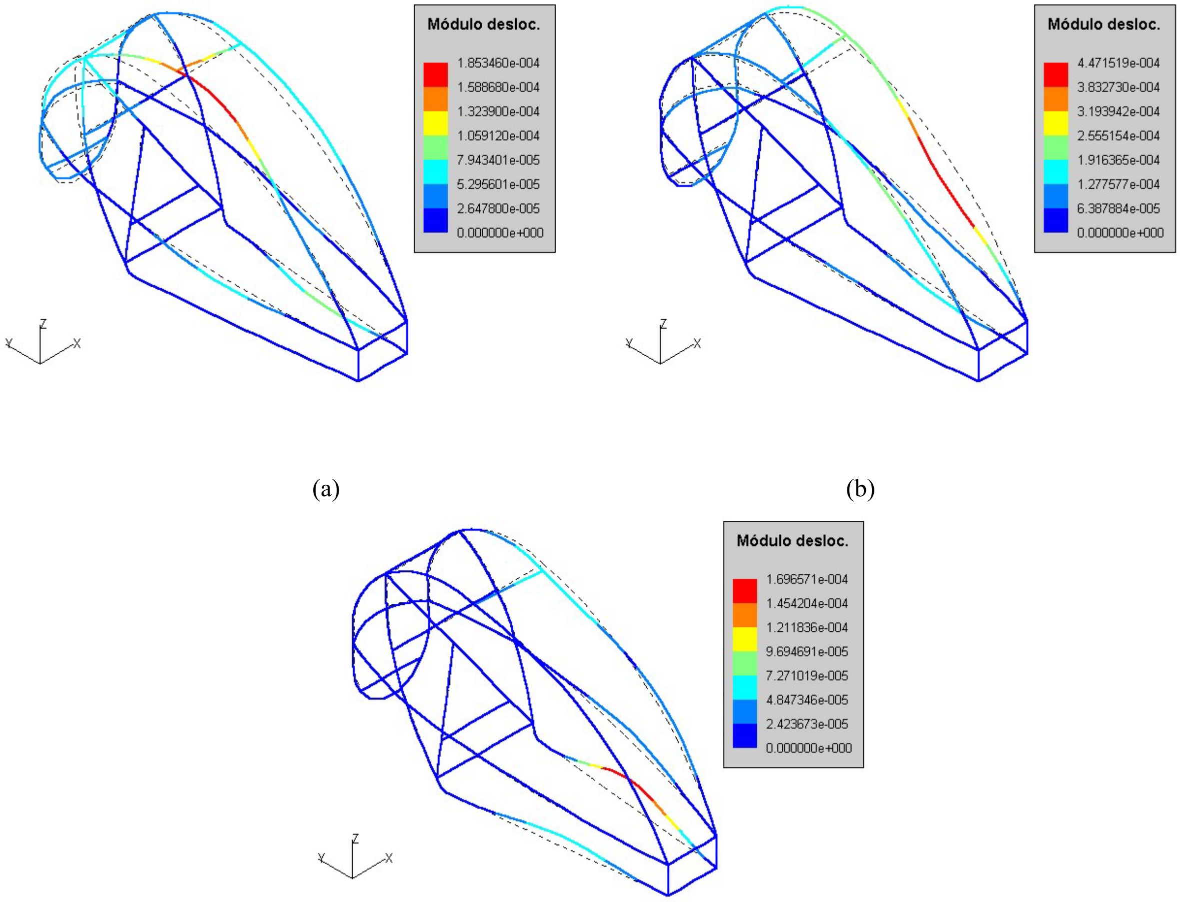


Figure 6. Vehicle amplitude acceleration in frequency domain – engaged gear and vehicle moving at constant speed.

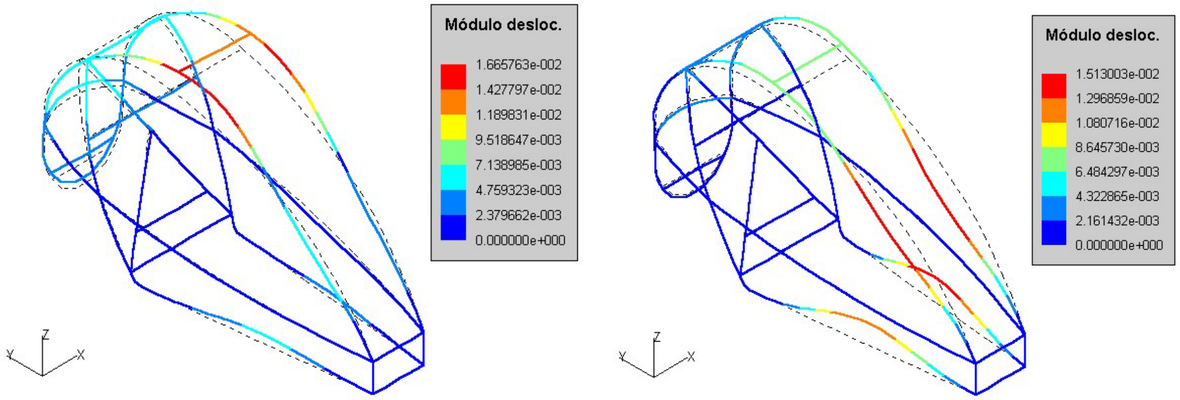
Figures 7 and 8 show respectively the vehicle structure response obtained using the ODS software tool for the disengaged gear and vehicle stopped and engaged gear and vehicle running at constant speed, at the following time steps: 29.5 sec., 30 sec., and 30.5 sec.



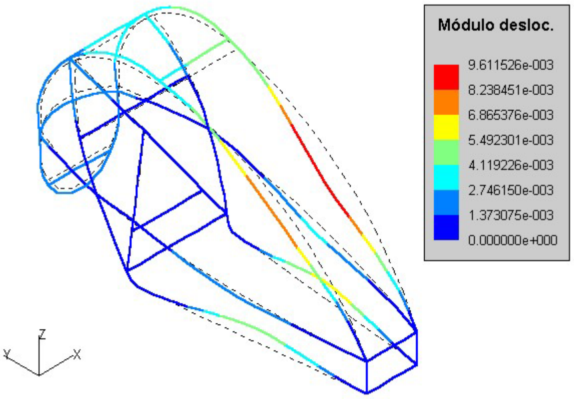
(a) (b)

(c)

Figure 7. Vehicle structure response obtained by the ODS software tool for the vehicle stopped and its motor running at the following time steps: 29.5 sec. (a), 30 sec. (b), and 30.5 sec. (c).



(a) (b)



(c)

Figure 8. Vehicle structure response obtained by the ODS software tool for the vehicle running at constant speed at the following time steps: 29.5 sec. (a), 30 sec. (b), and 30.5 sec. (c).

The ODS software tool evaluates all spectra at the given frequency, determines the amplitude and phase of motion, and maps these to the appropriate point on the vehicle frame. At this point, the animated image is immediately available. Each animation shows the deformation at a particular frequency. By scrolling through a series of animations, one can understand how the structure behaves across a broad frequency range. ODS software tool uses frequency domain data and compares spectra, rather than time traces, to determine the relative motion of points in a structure. This means that phenomena occurring at different frequencies can be separated for analysis. For example, a torsional resonance at one frequency can be independently analyzed from a bending resonance at another frequency.

5. Conclusions

This paper presented the development of an educational ODS software tool. As previously cited, the measurement of force inputs are not required for ODS, and the ODS are easy to execute, providing quickly an overall view of structural problem areas. Nevertheless it is necessary to acquire data from an adequate DOF quantity in order to obtain reliable results. The most relevant DOFs are related to the larger structure displacements. The software tool is able to produce a series of animated 3-D pictures on the screen that shows the motion of the vehicle chassis at selected frequencies. The ODS analysis provides information to the designer about how to modify the structure to solve the vibration problem by pointing out the locations and directions in which the excessive motion is occurring. The authors plan to incorporate an optimalization tool that uses ODS results as input data.

As the software tool was implemented using the MatLab language, many times the processing time expended was too large. When the pictures number was too large, MatLab started to use the virtual memory. Aiming the software improvement, the authors are also planning to translate the software code to C++ language and to use Open GL tool to improve the animated pictures 3-D view.

6. Acknowledgements

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7. References

Atalla, M. J., 1996, *Model Updating Using Neural Networks*, Ph.D. Thesis, Virginia Polytechnic Institute, State University, Blacksburg, Virginia, USA.

Han, S. and Feeny, B., 2003, "Application of proper orthogonal decomposition to structural vibration analysis", Mechanical Systems and Signal Processing, n.17(5), pp. 989-1001.

Margolis, D. and Shim, T., 2001, "A bond graph model incorporating sensors, actuators, and vehicle dynamics for developing controllers for vehicle safety", Journal of the Franklin Institute, no. 338, pp. 21-34.

Pai, P. F. and Young, L. G., 2001, "Damage detection of beams using operational deflection shapes", International Journal of Solids and Structures, n. 38, pp. 3161-3192.

Parloo, E., Verboven, P., Guillaume, P., and Van Overmeire, M., 2002, "Sensitivity-based operational mode shape normalization", Mechanical Systems and Signal Processing, n.16(5), pág 757-767.

Richardson, M. H., 1997, "Is It a Mode Shape, or an Operating Deflection Shape?", Sound & Vibration Magazine 30th Anniversary Issue.

Saturnino, L.J.M., 2004, *Desenvolvimento de Ferramentas para Definição, Análise e Avaliação de Desempenho de Veículos Automotivos*, MSc. Thesis, PUC Minas, Belo Horizonte – MG, Brazil (in Portuguese).

Waldron, K., Ghoshal, A., Schulz, M. J., Sundaresan, M. J., Ferguson, F., Pai, P. F., and Chung, J. H., 2002, "Damage detection using finite element and laser operational deflection shapes", Finite Elements in Analysis and Design, n. 38, pp. 193-226.

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