MODAL TESTING OF A TUBULAR VEHICLE CHASSIS

Frederico Mol Alvares da Silva, fredmol@task.com.br
Universidade Federal de Minas Gerais - UFMG

Luiz Felipe Adão Torre e Silva, luiz.felipe@llkengenharia.com.br
LLK Engenharia Ltda

Marcelo de Bittencourt Couto e Mello, marcelobcm@yahoo.com.br
Universidade Federal de Minas Gerais - UFMG

Abstract. *The operational deflection shape (ODS) of a structure is useful to investigate flexibility problems or regions of high relative displacements in the industrial field. Identifying the shape of the structure under real operating conditions can give a better understanding of the structural behaviour. In most cases the full numerical modelling of a complex system such a vehicle can be very difficult due to boundary conditions and joint modelling. In these cases the final result is inaccurate. The experimental ODS concerning a tubular structure chassis is discussed. The eigenvectors and eigenvalues from the modal analysis of a free-free finite element model are compared with the experimental operating modes. The frequencies of the first mode of several similar vehicles are compared in order to establish a minimum requirement for this dynamic parameter. Therefore, this parameter can be evaluated earlier, allowing modifications in the design to avoid dynamic problems. Failures in the chassis can be identified through deviations in the frequencies observed in the FFT or deformations observed in the ODS procedure. A software was developed to post-process and animate the measured ODS results, allowing the designer to better investigate the relative displacements of the structure.*

Keywords: *ODS, modal analysis, finite element, vehicle chassis.*

1. INTRODUCTION

This works presents the modal testing of a tubular structure vehicle chassis. A finite element model (FEM) of the chassis was constructed using beam elements and the results for the first frequency and mode shape were compared with the modal testing results. The tubular chassis was modelled as a space frame.

The vehicle is a monoplace SAE Baja vehicle, powered by a 10hp four stroke engine. The chassis is a tubular space frame made of thin walled steel tubes welded by TIG (tungsten inert gas) process.

A previous vibration measurement of velocity levels was conducted in a similar chassis (Silva 2004) and the results compared with FEM results. At that time the test equipment available was very simple, and the full ODS procedure was not performed. In this paper, the ODS method was performed for the modal testing using two accelerometers and the first mode shape was reconstructed using the data acquired.

The modal testing through ODS uses the FRF (frequency response functions) to estimate the relative displacements of the experimental mode shape. Since the structure must be measured at several points, one accelerometer measures the vibration signal at one fixed reference point and the second at the point of interest. The cross-correlation between these two signals are calculated and the phase and amplitude compared with the other points to reconstruct the 3D animated mode shape.

A comparison among FEM results of several similar chassis is presented in order to establish a stiffness or dynamic criteria for this type of vehicle. Improvements to the chassis were implemented and the results analysed by FEM model.

2. FINITE ELEMENT MODAL ANALYSIS

The structure was modelled with the rotational DOF constrained at the nodes, assuming that the real structure behaves like a 3D space frame. This assumption is based in the fact that the tube junctions (nodes) of the real chassis are welded all around in a way that the end of a structural member in a junction can not rotate. The opposite way is a space truss model where the nodes or junctions are free to rotate, which restricts the moment transfer between the structural members. In fact, the real welded junction is at some level between a space truss and a space frame. In several FEM books such as Logan (2002) and Weaver (1984) procedures to the solution of space truss and space frames are better explained.

Finite element model correlation and updating technology emerged in 1990s as a design tool in mechanical and aerospace engineering, Friswell (1995). The paper of Wu (2004) presents the modal testing to validate the FEM model of a crane structure, that was modified based on the experimental data to fit the numerical results. The main conclusion concerns the boundary conditions between the crane structure and the ground, assumed in the FEM model, as a significant parameter. In the work of Pavic (2003) a concrete floor is analysed with FEM and modal testing, and the material modulus of elasticity in the FEM model was adjusted to help fit the results.

The equations of motion of a multi DOF system are automaticaly generated by the FEM software employed, however, for an analytical approach, the equations of motion can be derived using the Lagrange's equations Eq. (1), where T is the kinetic energy, V the potential energy and Q the generalized degrees of freedom.

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial V}{\partial q_i} = Q_i \tag{1}$$

For the modal analysis the nonconservative (dissipative) forces are considered equal to zero and the system is conservative, the right hand side of Eq. (2) is a vector of zeros. The solution of Eq. (2) corresponds to the undamped free vibration of the structure considered. The characteristic determinant of the corresponding eigenvalue problem is given by Eq. (3), where [D] is the system dynamic matrix, λ the eigenvalue vector and [I] the identity matrix. Once the eigenvalues are known, the mode shapes or eigenvectors for each mode "i" can be calculated using Eq. (4).

$$[m]\cdot\ddot{\vec{q}} + [k]\cdot\vec{q} = \vec{0} \tag{2}$$

$$\Delta = |\lambda[I] - [D]| = 0 \tag{3}$$

$$[\lambda_i[I] - [D]]\cdot\vec{Q}_i = \vec{0} \tag{4}$$

The eigenvalues and eigenvectors were calculated using Ansys finite element software, Ansys (2007). The structural model of the vehicle chassis was modelled using a beam type element. Since the whole structure of the chassis is made of welded steel pipes, the PIPE16 finite element was considered in the model. This beam element is a two-node with six degrees of freedom at each node with bending, torsional and axial capabilities, Ansys (2007).

The first four modes extracted from the free-free FEM modal analysis are listed in Table 1. The mode shape of the first mode, which presents the highest amplitude among the frequencies measured is shown in Figure 1.

Table 1. Numerical results (FEM) for the first four modes.

Mode #	Frequency [Hz]	Mode Shape
1	39.7	Bending
2	43.7	Torsion
3	63.2	Torsion / Bending
4	67.3	Torsion / Bending

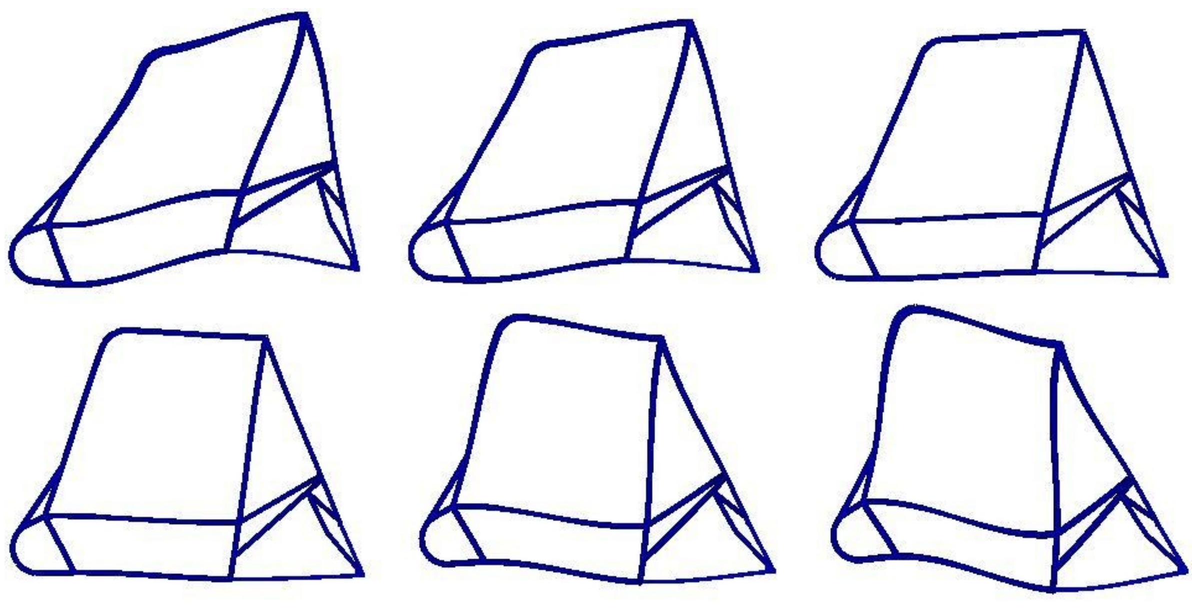

Figure 1. Frames of the first mode shape from the numerical analysis (from top left to right).

3. OPERATIONAL DEFLECTION SHAPE METHOD – ODS

The operational deflection shape is a powerful tool to analyze the behavior of structures when subjected to a vibrational source. In this methodology several measurements are made at arbitrary locations in the structure. A reference location is defined and the transfer functions are calculated for all points of interest (1,2,3 .. n) in relation to the fixed reference (point 0). The measurements are made using accelerometers, and the data can be acquired separately for each pair point 0 x 1..n. Since the excitation source and the measured data are time independent (not syncronized), this approach requires a minimal number of sensors. To acquire the data for the ODS process performed in this paper, only two accelerometers were used, which is a great advantage for the limited number os channels of some data acquisition systems. Several industrial applications of modal testing under operational conditions are well explained by Hermans (1999) using output-only data, similar to the approach adopted in this paper.

In the purpose of obtaining the correlation of phase between the vibrational signals acquired a Matlab function called cross power spectral density (*cpsd*) was used. This function estimates the cross power spectral density of the discrete-time using the Welch (1967) averaged modified periodogram method of spectral estimation. The cross power spectral (PSD) density is the distribution of power per unit frequency.

Welch's method (also called periodogram method) is carried out by dividing the time signal into successive blocks, and averaging squared-magnitude discrete Fourier transform of the signal blocks.

A computer program examines all the data and produces a series of animated 3-D pictures that shows the motion of the machine parts for a chosen mode.

During the measurement the LabView software was used to check the data being acquired on-line, avoiding wrong measurements due to noise sources or system malfunction, which might be discovered only after post-processing of all data. The diagram of the LabView on-line processing is shown in Figure 2.

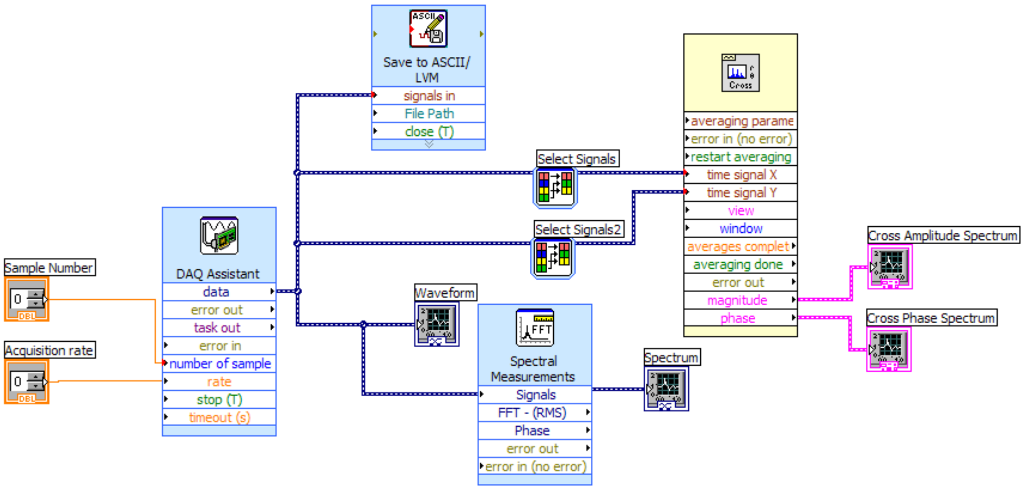

Figure 2. On-line check of the data acquisition process - software LabView.

The vehicle chassis was excited by the engine at idle speed and supported by the suspension system, see Figure 3. This assembly was considered a good approach, since the natural frequencies of the suspension system are very small (~1.5Hz – 2 Hz) when compared with the chassis (>40Hz) and engine modes, Gerges (2005). In this case, the chassis can be considered as isolated from the ground and is an approximation of the theoretical free-free modal extraction using FEM analysis.

Figure 3. Data acquisition to perform the ODS analysis.

In the case of the vehicle tested the engine was considered as the source of excitation for the ODS, Silva (2004). The engine was mounted in the chassis without silent blocks, which provides a better way for the excitation of the chassis. All measurements were made with the engine at idle (~1150rpm). Tests at other speeds were not considered, since was observed that the engine vibrates more at idle.

According to Rao (1995) the complete modal analysis is composed by three phases: test setup, frequency response measurements and modal parameter identification. Basically two modal parameters were identified on the approach presented in this paper, the mode frequencies and mode shapes. Other parameters such as damping and stiffness can be estimated by modal testing as suggested by Harris (2002) and Rao (1995).

The results of the FEM analysis of the system being tested can provide a method to validate the numerical modal model Harris (2002). The frequency and modal shape for the first mode were compared with the FEM results in order to validate the ODS methodology for tubular chassis, providing the designer with a useful numerical model to test new solutions to improve the dynamic behavior of the chassis as desired.

The estimation of the power spectral density (PSD) proposed by Welch (1967), based on the acquired time signal was implemented. The averaging of this method reduces the noise due to imperfect and finite data. A rectangular spectral window (Bendat, 1980) was adopted in the finite Fourier time-frequency transformation.

Afterwards the *circular cross-correlation* of the two modified periodograms (Welch PSD) were calculated, and the amplitude and phase of the signals properly animated. The whole process is presented in Figure 4.

Several commercial softwares are dedicated to ODS in the industrial field. However, these softwares are very expensive certainly not due to the complexity, but mainly due to the high benefits that the ODS results can provide to their costumers in the diagnosis of failures, free-plays and structural issues.

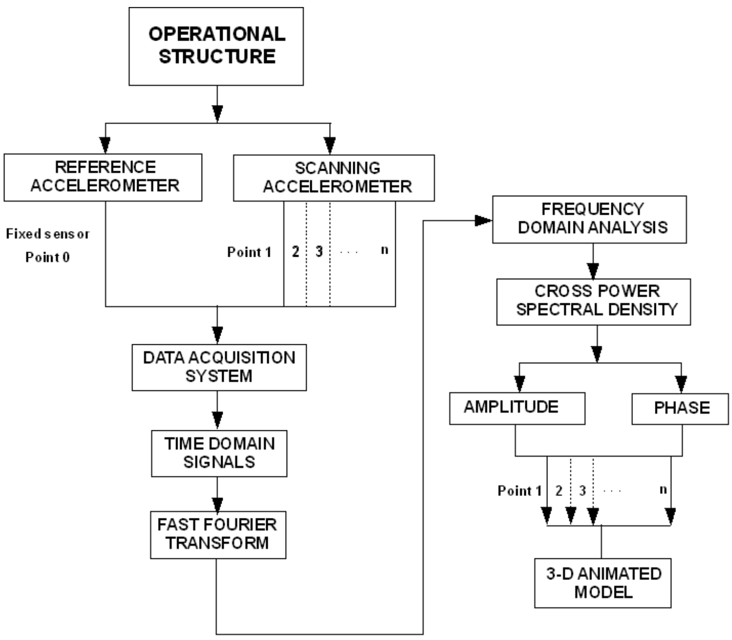

Figure 4. ODS process flowchart.

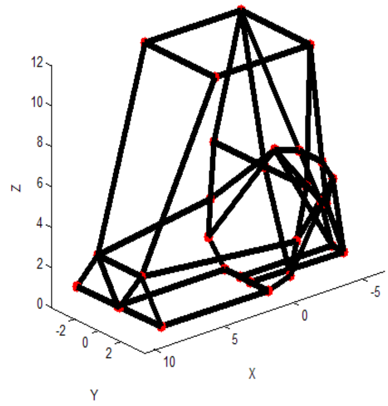

Figure 5. 3-D model of the chassis – LLK Operational Deflection Shape Program.

In Figure 5 the chassis model (undeformed) was plotted in the Operational Deflection Shape program developed by LLK Engineering. The red dots in the figure highlights the measured points. The Figure 6 shows the animation of the first mode of the structure (38.75Hz). The red arrows highlights the high displacements occurred in the cockpit, note the similarity with FEA results of Figure 1.

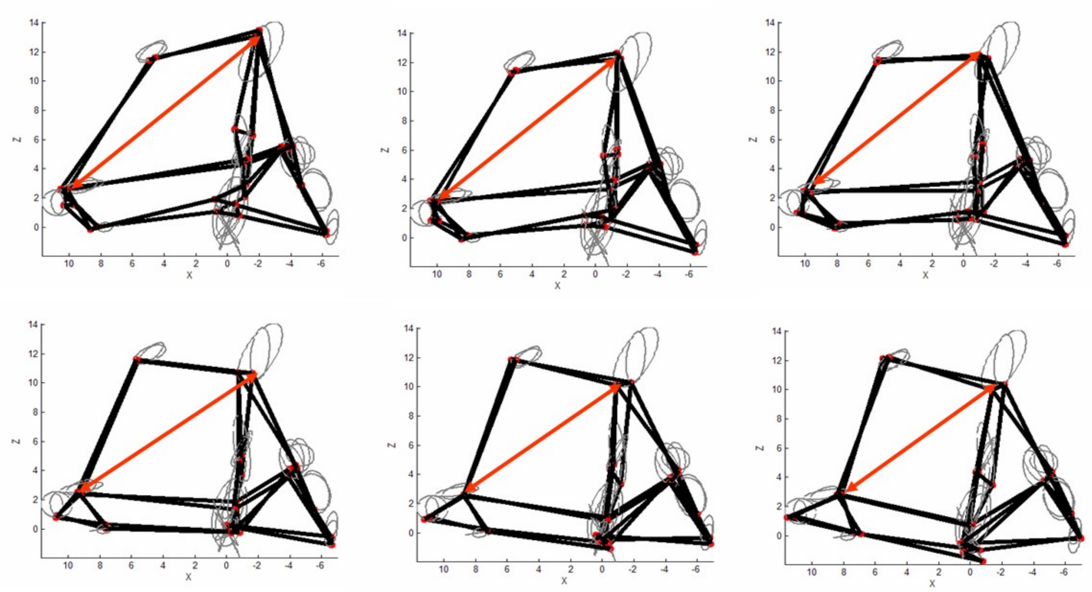

Figure 6. Frames of the first mode measured using ODS (from top left to right).

4. RESULTS ANALYSIS

The results presented in Table 2 show a comparison among different designs of similar tubular chassis. Observing the data of Table 2, the chassis PI2003 is stiffer than the chassis of newest vehicles, which in fact is due to the changes in the regulations that were conducted. However, with some reinforcements in the baseline design the chassis becomes stiffer than the PI2003 with minor weight penalty (+10%). It is noted for the SA2009 reinforced chassis that the first torsional mode frequency is the highest when compared with the others.

Typical chassis frequencies for normal use vehicles is in the range of 20 to 30Hz, Fenton (1996), which is approximately one half of the first frequency of a SAE Baja chassis.

The measured frequency spectrum show in Figure 7 shows the first peak at 38.75Hz, corresponding to a value 2.4% below the first frequency obtained in the FEM model, see Table 2.

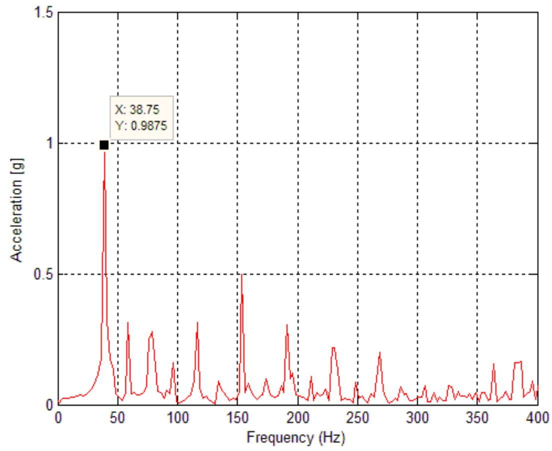

Figure 7. Frequency spectrum at reference point with the engine at idle.

Table 2. Comparison of frequencies among similar tubular chassis – FEM results.

Mode	#1	#2	#3	#4	Vehicle / Year
Freq. [Hz]	**39.7 (B)**	**43.7 (T)**	**63.2 (T)**	**67.3 (T)**	**SA2009 (baseline)**
	59.1 (B)	61.7 (T)	70.1 (T)	95.2 (T)	SA2009 (reinforced)
	40.2 (B)	44.1 (T)	63.6 (T)	67.8 (T)	SA2008
	43.2 (T)	45.5 (B)	54.4 (T)	59.1 (B)	DL2008
	56.4 (T)	59.5 (B)	67.8 (T)	69.3 (B)	PI2003 [(1)]

[(1)] : Silva et al. (2004); [(B)] : bending mode; [(T)] : torsional mode;

Comparing the mode shapes obtained by FEM (Figure 1) and modal testing (Figure 6) a good correlation is observed, mainly in the cockpit open frame deformations and in the bending of the horizontal central members.

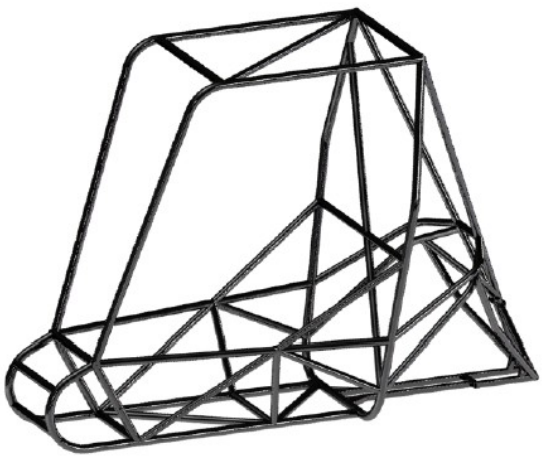

Figure 8. Tubular chassis after reinforcements. (SA2009 reinforced)

5. CONCLUSIONS

The FEA mode shape for the first frequency of a tubular chassis is very closer to the mode shape reconstructed using the ODS procedure with minimal equipment (only two accelerometers). A better precision for the first frequency (<2.4%) can be obtained by trial and error adjusting the parameters of the numerical model, such as material properties, not modelled masses or the inclusion of boundary conditions (suspension springs) instead of the free-free approach.

The FEM model of the chassis was considered satisfactory and the reinforcements were tested using only the numerical approach. The frequencies for the first four modes were significantly improved (stiffer chassis) with minimal weigth penalty (+10%).

The consideration adopted in the FEM model that assumes the structure as a space frame was considered acceptable, since the mode shape obtained from the numerical results for the first mode was closer to the experimental mode shape (ODS).

A minimum criterion for the first frequency of the Baja tubular chassis can be established as higher than 60Hz, this value can be used as reference for preliminary design studies, ensuring a properly stiffer chassis.

6. REFERENCES

ANSYS, "Software Documentation ANSYS V11", Ansys Inc, 2007.

Bendat, Julius S., "Engineering Applications of Correlation and Spectral Analysis", John Wiley & Sons, 1980.

Fenton, John, "Handbook of Vehicle Analysis", MEP, London, 1996.

Friswell MI, Mottershead JE., "Finite element model updating in structural dynamics", Dordrecht, The Netherlands: Kluwer Academic Publishers, 1995.

Gerges, Samir N. Y., "Ruídos e Vibrações Veiculares", 1rd Ed, NR Editora, 2005.

Harris, Cyril M., "Shock and Vibration Handbook", McGraw-Hill, 5rd Ed, 2002.

Hermans, L. and H. Van der Auweraer, "Modal Testing and Analysis of Structures Under Operational Conditions: Industrial Applications", Mechanical Systems and Signal Processing, Academic Press, 1999 13(2), pp. 193-216.

Logan, Daryl L., "A First Course in The Finite Element Method", 3rd Ed, 2002.

Pavic, A., P. Reynolds, "Modal testing and dynamic FE model correlation and updating of a prototype high-strength concrete floor", Cement & Concrete Composites 25 (2003) 787-799.

Rao, Singiresu, "Mechanical Vibrations", Wesley, 3rd Ed, 1995.

Silva, Frederico M., "Modal Analysis of a Tubular Structure Vehicle Chassis", SAE Technical Papers - Paper 000000298PE, ISSN: 0148-7191, 2004.

Weaver, W. and Johnston P. R., "Finite Element for Structural Analysis", Prentice-Hall, Englewood Cliffs, 1984, 403pp.

Welch, P.D., "The Use of Fast Fourier Transform for the Estimation of Power Spectra: A Method Based on Time Averaging Over Short, Modified Periodograms", IEEE Trans. Audio Electroacoust, Vol. AU-15 (June 1967), pp. 70-73.

Wu, Jia-Jang, "Finite Element Modelling and Experimental Modal Testing of a Three-dimensional Framework", International Journal of Mechanical Sciences 46 (2004) 1245-1266.

7. RESPONSIBILITY NOTICE

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