Identification of Dynamics Modal Parameter for Car Chassis

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Identification of Dynamics Modal Parameter for Car Chassis

[1]M.S.M. Sani, [1,2]M.M. Rahman, [1,2]M.M. Noor, [1,2]K. Kadirgama and [1]M.H.N. Izham

[1]Faculty of Mechanical Engineering, Universiti Malaysia Pahang, 26600 UMP, Pekan, Pahang, Malaysia

[2]Automotive Excellence Center, Universiti Malaysia Pahang, 26300 UMP, Kuantan, Pahang, Malaysia

Abstract This paper explores and investigates the dynamic characteristics of car chassis structure by using experimental modal analysis (EMA) method and modal testing. Dynamic characteristics are divided into three parameters include natural frequency, damping factor and mode shape. In this study, modal testing was performed on the car chassis including the impact hammer and shaker test. Data analyzer was used to convert the response signal from the sensor, which was in the time domain to frequency domain. Result obtained from both methods, is compared on each axis (X, Y and Z axis). However, small discrepancy was observed in terms of natural frequency, which is within the range of 5%. Based on the results, interpretation and comparison were made for both methods.

1. Introduction

A major concern in analyzing practical mechanical structures is to reliably identify their dynamic characteristics, i.e., their natural frequencies and vibration mode shapes. These vibration characteristics are needed in order to achieve effective design and control of the vibrations of structural components. Due to the increasing design demands for quality and reliability of almost all engineering structures, it has become essential to be able to estimate accurately structural dynamic characteristics (natural frequencies and mode shapes). In practice, natural frequencies and mode shapes can be established by analytical approaches such as finite element analysis, or experimental approaches such as modal testing (Lin and

Lim, 1997, Hurlebaus et al., 2008). The limitation of this testing was studied by Ewin and Sidhu (1982). Accurate identification of structural dynamics is important to the understanding of a structure's behaviour. In order to design structures under dynamic load conditions, it is necessary to analyze the structure's dynamic response. In addition, dynamic investigation by monitoring natural frequencies or magnitudes of frequency response functions can be useful in detecting faults and mechanical failures (Farshidi et al. 2010; Sheen 2008; Benko et al. 2005). Analytical dynamic analysis does not always provide accurate results due to inaccurate boundary conditions and the inability to accurately predict damping parameters. Typically, experimental modal analysis (EMA) is carried out to identify the vibration characteristics of a mechanical system.

It is accepted now a days that, due to the advances made in instrumentation and measurement techniques, vibration properties identified from measured data are regarded as being closer to the true representation of a structure provided sufficient care is given to experimental and identification procedures. By measuring the response data of a structure and performing a subsequent modal analysis of these data, accurate modal parameters which are the natural frequencies and mode shapes of the structure can be identified. Comprehensive reviews on the research of modal identification methods can be found in the work of Ewins (1984) and Maia (1988). Depending on the domain where the identification process is carried out, modal analysis techniques can be categorized into the time-domain and frequency-domain methods. As one of the frequently used time-domain techniques, the complex exponential method (Brown et al., 1979) seeks to calculate the modal parameters by analysing the impulse response functions which are the inverse Fourier transforms of the measured frequency response function (FRF) data.

Car chassis is the main component in a vehicle system. Most of the problem exist in automobile industry is vibration on the chassis. The factors that bring to forced vibration are external loads, internally generated forces and many more. Resonance can be determined by the material properties (mass, stiffness and damping properties) and the boundary conditions of a chassis or structure (Schwarz and Richardson, 1999). Vibration of chassis can be formed due to dynamic forces such as the engines, unsmooth road and many more (Avitabile, 1998). Schedlinski et al. (2004) were investigated the real behavior of the body in white with impact hammer and shaker at different level of excitations. The accomplished modal analysis provides a sophisticated basis to improve the modeling of the investigated structure for control purpose (Popprath et al. 2006). Filho et al. (2003) have carried out a research for a commercial off-road vehicle chassis. The work consists in obtaining an optimized chassis design for an off-road vehicle with the appropriate dynamic and structural behavior, taking into account the aspects relative to the economical viability of an initial small scale production. In this paper, EMA was carried out to identify eigenvalues and eigenvectors by impact hammer and shaker method.

2. Materials And Methods

Modal analysis on car chassis structures has to be done very carefully. Since modal analysis consists in exciting a structure with a controlled signal and measuring its mechanical response, the problems on structures, due to the masses of the used sensors, occur when considering traditional contact methods. Modal analysis is often used to optimize structures or measurement instrumentation of structures. Eigen frequencies, mode shapes and damping ratios are often computed in order to eliminate parasitical vibrations on mechanical systems, but also to predict aging of tested structures. Modal parameters of a car chassis was determined by EMA (Ewins, 1995; Schwarz and Richardson, 1999). EMA is a process to describe Wira car chassis/structure in terms of modal parameters. Modal analysis is a process to determine the dynamic characteristic of Wira car chassis. Types of dynamic characteristics are natural frequency, damping ratio and mode shape (Emory and Zhu, 2006). Eigen frequencies and eigenvectors were employed and identified from vibration tests by EMA. There are two method is used to analyze the structure of used Proton Wira. Those two methods are likely to be shaker test and impact hammer test and the chassis of Wira is to be analyzed under free-free boundary condition (Elliot and Richardson, 1998; Haapaniemi et al., 2003). Experimental study of structural vibration has made significant contributions for better understanding in vibration phenomenon. Model parameters such as natural frequency, mode shape and damping ratio were extracted from the structure experimentally. This experiment has done to determine dynamics characteristics of go-kart chassis structure by impact hammer and shaker method (Lin and Lim, 1997). Experimental is done by using impact hammer to excite the crankshaft and data recorded using data acquisition system (DAS) connected to sensor located on the crankshaft (Sani et al. 2010).

In order to performed analysis on the chassis, used Proton Wira is dismantled. Chassis of Proton Wira is drawn using ME' Scope. Then chassis is hanged on the test rig using supporting belt or elastic cord and identify the points or degree of freedoms (DOFs) on the chassis. The desired points are selected and marked using wax and 18 DOFs are marked to define the modal parameters of the structure (Fig. 1). Figure 2 shows the test rig with car chassis. The 8 channel FFT analyzer was used to collect time data. The time data is the ratio of output response due to an input. The time data is converted into frequency domain (Frequency Response Function, FRF). FRF is easier to evaluate and using the Pulse-Lite software to select signal response. For impact hammer excitation, each accelerometer response DOF is usually fixed and reflects a reference DOF. The hammer is then moved around the structure and used to excite every 18 DOF needed in the model.

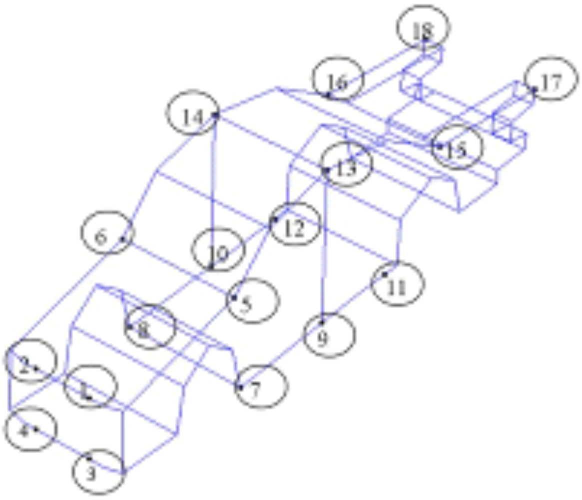


Figure 1. The structure of Wira car chassis with 18 DOF's

The relationship between input (force excitation) and output (vibration response) of a linear system is expressed as Eq. (1):

$$Y_i = \sum_j H_{ij} X_j \quad (1)$$

where Y_i is the output spectrum at DOF i, X_j is the input spectrum at DOF j, and H_{ij} is the frequency response function (FRF) between DOF j and DOF i. The output is the sum of the individual output caused by each of the inputs. The FRF's are estimated from the measured auto and cross-spectra of and between input and output. Only one response DOF is needed by one accelerometer position (point and direction) in impact hammer test. Equation (2) is expressed the single-input, single-output test configuration.

$$\begin{Bmatrix} Y_1 \\ Y_2 \\ Y_3 \\ \vdots \\ Y_n \end{Bmatrix} = \begin{bmatrix} H_{11} & H_{12} & H_{13} & \cdots & H_{1n} \\ H_{21} & H_{22} & H_{23} & \cdots & H_{2n} \\ H_{31} & H_{32} & H_{33} & \cdots & H_{3n} \\ \vdots & \vdots & \vdots & \cdots & \vdots \\ H_{n1} & H_{n2} & H_{n3} & \cdots & H_{nn} \end{bmatrix} \begin{Bmatrix} X_1 \\ X_2 \\ X_3 \\ \vdots \\ X_n \end{Bmatrix} \quad (2)$$

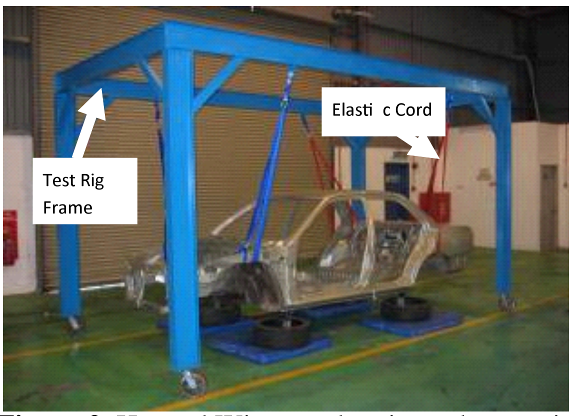
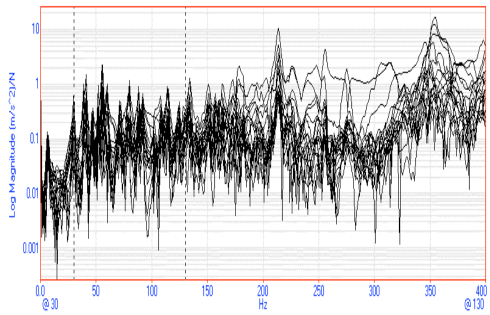


Figure 2. Hanged Wira car chassis on the test rig

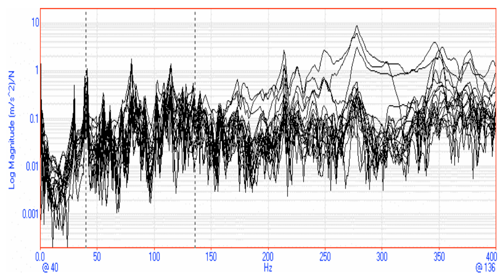
3. Results and Discussion

Two methods of modal testing are tested on the Wira car chassis, and 18 DOFs is selected. Signal response is captured and the data collected is imported to the ME' Scope VES software. Then, chassis is simulated using ME' Scope VES software. Frequency range of interest is set between 30 Hz to 130 Hz for both methods for X and Z axis, however for Y axis, frequency range of interest for impact hammer test is set between 40 Hz to 136 Hz and for shaker test is set between 37 Hz to 136 Hz. After the desired range of frequency is selected, 4 mode shapes is obtained from the simulation.

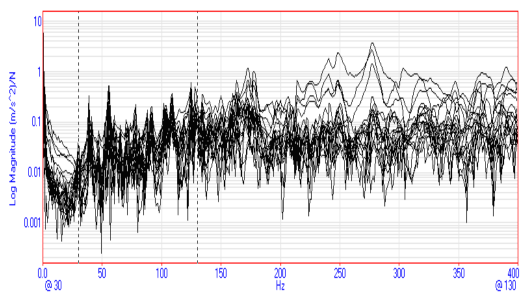
Figure 3 shows a superimposed FRF for all excited points in X, Y and Z axis for hammer testing. It can be observed in X axis that there are two types of good coherence, peaks close to the value of 1 and the possessed to uniform shape. Even though the analysis produces the coherence close to the value of 1 however there is a big or narrow drop on the coherence, it can be considered as a bad coherence. Frequency range of interest is set between 30 Hz to 130 Hz. Four mode shapes is selected within the frequency range of interest. Same frequency range of interest is used on Y axis. Frequency range of interest is set between 40 Hz to 136 Hz. The frequency range of interest for Z axis is set between 30 Hz to 130 Hz. To differentiate between all three axes is to assign the direction of the data based on the response direction.



(a) X axis



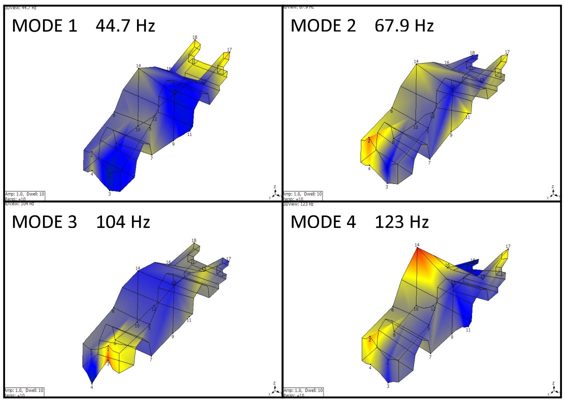
(b) Y axis



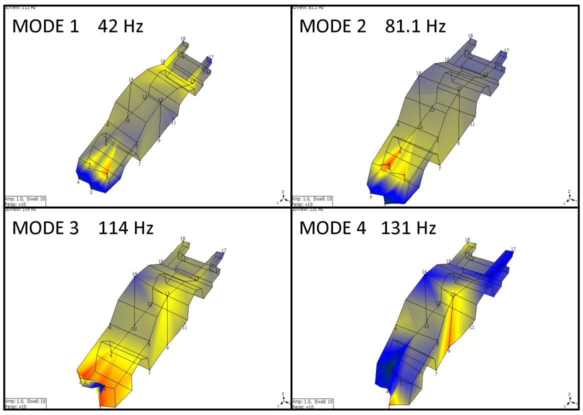
(c) Z axis

Figure 3. Superimposed FRF for Z axis by hammer test

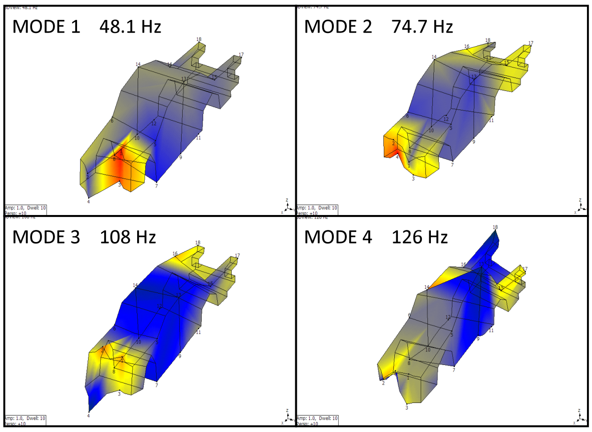
Figure 4 shows the mode shapes obtained for different axis using the hammer method. The bending occurs on the chassis as in mode 1 when dwelling at the first natural frequency. Then, the twisting occurs on the chassis as in mode 2 when dwelling at the second natural frequency. While in mode 3, dwelling in the third natural frequency produced second bending on the chassis. Lastly, on mode 4, dwelling in the fourth natural frequency displayed second twisting on the chassis.



(a) X axis



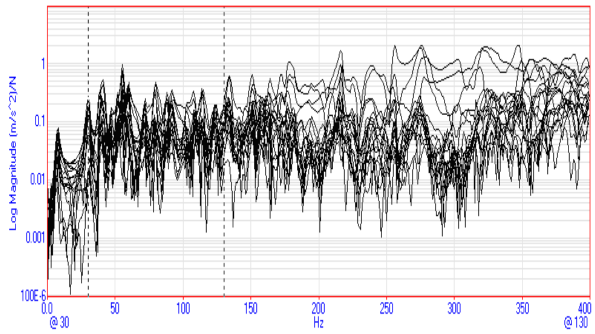
(b) Y axis



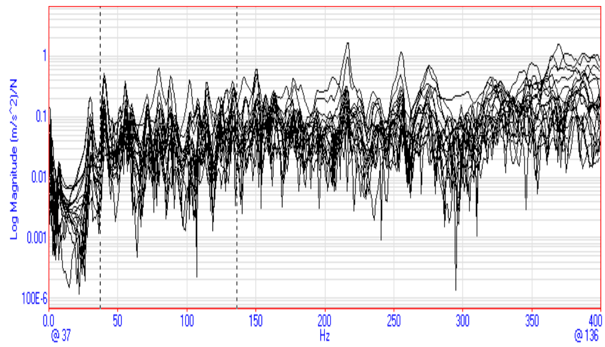
(c) Z axis

Figure 4. Modes shape for different axis by hammer test

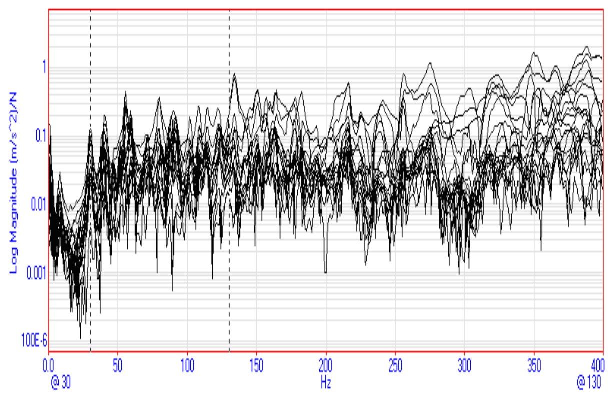
Results obtain from the analysis performed using ME' Scope are shown in Figure 5 and Figure 6 for shaker test. The deflection shape on the chassis for X axis by shaker test is similar to the impact hammer test. The position of shaker is maintained at the same axis. Frequency range of interest for X, Y and Z axis are set same as hammer test.



(a) X axis



(b) Y axis



(c) Z axis

Figure 5. Superimposed FRF for different axis by shaker test

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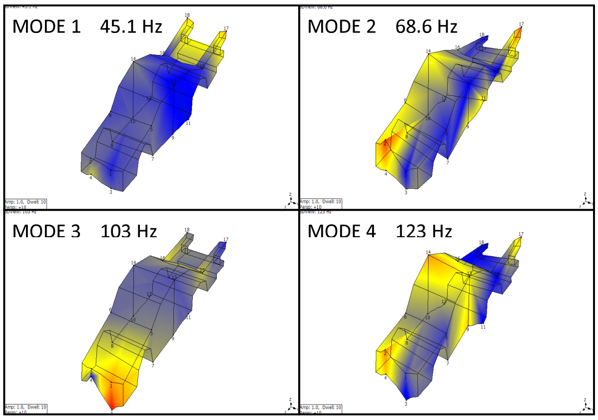
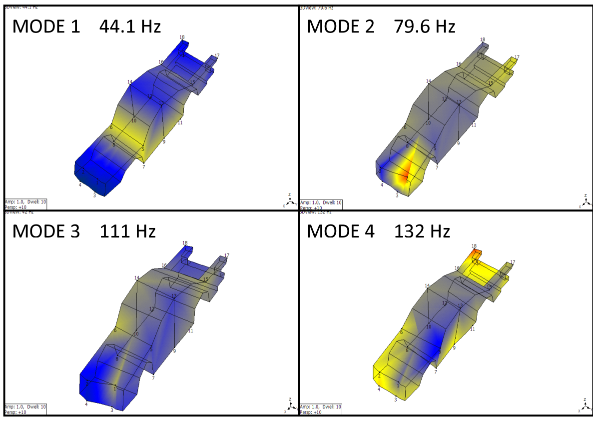
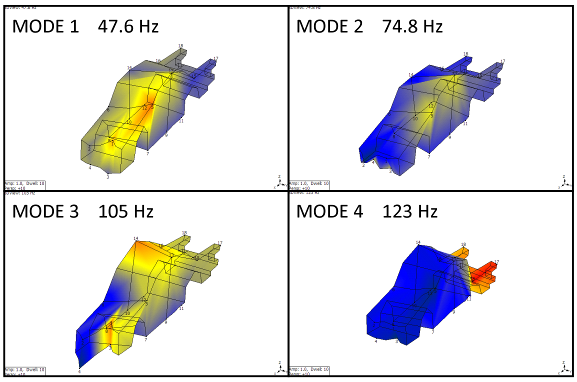


(a) X axis



(b) Y axis



(c) Z axis

Figure 6. Modes shape for different axis by shaker test

The deflection shape for shaker test in Z axis is not similar with the deflection shape of impact method. When dwelling at first natural frequency, twisting occurs as in mode 1. Second twisting occurs when dwelling at second natural frequency as in mode 2. While for mode 3, first bending exist on the chassis and lastly second bending occurs on the chassis as in mode 4. Comparison is made between the two methods on each axis. Comparison of result between two methods on X, Y and Z axis are listed in Table 1, 2 and 3 respectively. The percentage difference on each axis between roving impact hammer test and shaker test is less than 5% which is tolerable range. Table 4 shows the discrepancies percentage of natural frequency between two methods in three directions X, Y and Z.

Table 1. Comparison of mode shape for X axis

Mode Shape	Impact Hammer Test		Shaker Test	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
1	44.7	0.684	45.1	1.44
2	67.9	0.656	68.6	1.15
3	104	0.713	103	0.914
4	123	0.499	123	0.579

Table 2. Comparison of mode shape for Y axis

Mode Shape	Impact Hammer Test		Shaker Test	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
1	42	0.435	44.1	0.911
2	81.1	0.694	79.6	0.622
3	114	0.655	111	0.702
4	131	0.854	132	0.303

Table 3. Comparison of mode shape for Z axis

Mode Shape	Impact Hammer Test		Shaker Test	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
1	44.7	0.684	45.1	1.44
2	67.9	0.656	68.6	1.15
3	104	0.713	103	0.914
4	123	0.499	123	0.579

Table 4. Discrepancies percentage of natural frequency between impact hammer test and shaker test

Mode	X Axis	Y Axis	Z Axis
1	0.89%	4.76%	1.04%
2	1.02%	1.85%	0.13%
3	0.96%	2.63%	2.78%
4	0.00%	0.76%	2.38%

There are varieties of frequency range of interest, and the frequency range of interest is selected based on the deflection pattern of the chassis and the magnitude of natural frequency. Frequency range of interest for Y axis is different from X and Z axis. This is to ensure the natural frequency for both methods is closed to each other. The results of natural frequency obtain from the simulation in the frequency range of interest produce a small difference between the two methods. The deflection shapes of the chassis in depending on the axis selected since the frequency range of interest is been set up depending on the axis. For X and Y axis, the pattern of the mode shapes for both methods is almost similar. However, for Z axis, the pattern of the mode shapes is not similar, as in mode 4 for impact test is twisting but for mode 4 by shaker test is bending. The natural frequency obtain for all the three axes is below than 5%, but the pattern of the mode shape for Z axis does not get as desired. This can happens as the tri-axial accelerometer is not accurate compare to the single accelerometer. Tri-axial accelerometer is used on the chassis as the single accelerometer is not suitable for some of the selected DOFs.

4. Conclusions

Problems of very close modes often arise in engineering practice due to structural symmetries with little damping and the accurate determination of the modal parameters of such modes is very important in response and stability calculations such as those of large flexible automotive structures. EMA methods including the impact hammer and shaker test were performed on the chassis structure. The chassis has been impacted on Y axis for both methods; however, signal response is obtained for all three axes. The percentage difference on each axis between roving impact hammer and shaker test is within 5%. Natural frequency is obtained from hammer and shaker test and small percentage discrepancies makes both of the methods can be applied to obtain the dynamic characteristic of the structure. It is very important to study and predict the dynamic characteristic of the car chassis, so that resonance does not occur on the chassis. Further research work in this direction is recommended.

Acknowledgment

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